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THE SENSITIVITY OF DIESEL ENGINE PERFORMANCE TO FUEL INJECTION PARAMETERS AT VARIOUS OPERATING POINTS

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Thesis Submitted to the University of Nottingham for the degree of Doctor of Philosophy

September 2004
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This thesis describes research undertaken to establish the advantages and disadvantages of using high pressure common rail fuel injection systems with multiple injection capabilities. The areas covered are detailed as follows.

Oscillations in the rail pressure due to the opening of the injector can affect the quantity of fuel injected in subsequent injection events. The source of these oscillations has been investigated. A method of damping or reducing the oscillations has been defined and was applied. This successfully reduced the level of unpredictability of the quantity of injected fuel in subsequent injection events. A relationship between needle lift, injection pressure and the quantity of fuel injected was established.

The effects of fuel injection parameters (main injection timing, split main separation and ratio) and engine operating parameters (boost pressure and EGR level) on emissions formations and fuel economy have been investigated at five operating points. Design of Experiments techniques were applied to investigate the effect of variables on pollutant emissions and fuel consumption. The sensitivity and linearity of responses to parameter changes have been analysed to assess the extent to which linear extrapolations will describe changes in...
smoke number (FSN) and oxides of nitrogen (NO\textsubscript{x}); and which parameters are the least constricting when it comes to adjustments of parameter settings on the FSN-NO\textsubscript{x} map.

Comparing results for split main and single injection strategies at the five operating conditions shows that split main injection can be exploited to reduce NO\textsubscript{x} or FSN values at all conditions and both NO\textsubscript{x} and FSN simultaneously at high load conditions. The influence of changing engine speed and brake mean effective pressure (BMEP) on FSN and NO\textsubscript{x} emissions with given fixed values of parameter settings has been investigated. This established how much of the operating map could be covered by discrete calibration settings. Finally the variation in parameter settings required to maintain fixed FSN and NO\textsubscript{x} values across the operating map, near the optimum trade-off on the FSN-NO\textsubscript{x} map, was analysed. Combining the information gained from the individual investigations carried out highlighted some techniques that can be used to simplify the calibration task across the operating map, while also reducing the amount of experimental testing required.
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Throughout my time at the University of Nottingham, I have received assistance and support from a number of individuals. Paul Shayler, Ford Professor of Mechanical Engineering and Head of the Engines Research Group, is acknowledged and thanked for his support and guidance during my research and my writing of this thesis.

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All the research described within this thesis was undertaken with the generous financial support and collaboration of the Ford Motor Company, in particular Mike Watts is thanked for his assistance. I would like to express my sincere thanks to my single cylinder colleagues, Tom Brooks and Gareth Pugh, who both did so much work in helping to commission the test facility. I would also like to thank all my other friends and colleagues in Engines Group, past and present, for their support and good humour throughout my time there.

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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_n )</td>
<td>Nozzle minimum area</td>
<td>( [m^2] )</td>
</tr>
<tr>
<td>( B_{FUEL} )</td>
<td>Bulk modulus of fuel</td>
<td>( [N/m^2] )</td>
</tr>
<tr>
<td>( c )</td>
<td>Speed of pressure wave</td>
<td>( [m/sec] )</td>
</tr>
<tr>
<td>( C_D )</td>
<td>Discharge coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>( \frac{dP}{d\theta} )</td>
<td>In-cylinder pressure derivative</td>
<td>( [\text{bar} / \circ \text{CA}] )</td>
</tr>
<tr>
<td>( l )</td>
<td>Length</td>
<td>( [m] )</td>
</tr>
<tr>
<td>( f_{\text{CLOSED-OPEN}} )</td>
<td>Frequency of closed-open system</td>
<td>( [\text{Hz}] )</td>
</tr>
<tr>
<td>( f_{\text{CLOSED-CLOSED}} )</td>
<td>Frequency of closed-closed system</td>
<td>( [\text{Hz}] )</td>
</tr>
<tr>
<td>( m_{\text{FUEL}} )</td>
<td>Mass of fuel injected</td>
<td>( [\text{kg}] )</td>
</tr>
<tr>
<td>( m_i )</td>
<td>Mass of component</td>
<td>( [\text{kg}] )</td>
</tr>
<tr>
<td>( m_{\text{TOTAL}} )</td>
<td>Total mass</td>
<td>( [\text{kg}] )</td>
</tr>
<tr>
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<td>Mass flow rate of air intake</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( m_{\text{CO2AIR}} )</td>
<td>Mass flow rate of ( CO_2 ) in air intake</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( m_{\text{CO2EGR}} )</td>
<td>Mass flow rate of ( CO_2 ) in EGR</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( m_{\text{CO2MAN}} )</td>
<td>Mass flow rate of ( CO_2 ) in manifold</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( m_{\text{EGR}} )</td>
<td>Mass flow rate of EGR</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( m_{\text{FUEL}} )</td>
<td>Mass flow rate of fuel</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( \dot{m}_i )</td>
<td>Mass flow rate of component</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( \dot{m}_{\text{MAN}} )</td>
<td>Mass flow rate of manifold gases</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( \dot{m}_{\text{TOTAL}} )</td>
<td>Total mass flow rate</td>
<td>( [\text{kg/hr}] )</td>
</tr>
<tr>
<td>( M_{\text{AIR}} )</td>
<td>Molecular weight of air intake</td>
<td>[-]</td>
</tr>
<tr>
<td>( M_{\text{CO2}} )</td>
<td>Molecular weight of ( CO_2 )</td>
<td>[-]</td>
</tr>
<tr>
<td>( M_{\text{EGR}} )</td>
<td>Molecular weight of EGR</td>
<td>[-]</td>
</tr>
<tr>
<td>( M_i )</td>
<td>Molecular weight of component</td>
<td>[-]</td>
</tr>
<tr>
<td>( M_{\text{MAN}} )</td>
<td>Molecular weight of manifold gas</td>
<td>[-]</td>
</tr>
</tbody>
</table>
\begin{tabular}{ll}
\textbf{M$_{\text{TOTAL}}$} & Total molecular weight [-] \\
n$_{\text{EXH}}$ & Number of moles in exhaust gas [-] \\
n$_{\text{H2O}}$ & Number of moles of H$_2$O removed [-] \\
n$_i$ & Number of moles of component [-] \\
n$_{\text{TOTAL}}$ & Total number of moles [-] \\
NL & Needle lift [\mu m] \\
P$_{\text{inj}}$ & Injection pressure [bar] \\
P$_{\text{cyl}}$ & In-cylinder pressure [bar] \\
w & Humidity ratio [%] \\
x$_{0,..n}$ & Input parameter values [-] \\
x$_i$ & Mass fraction of component [-] \\
$\tilde{x}$ CO$_2$AIR & Mole fraction of CO$_2$ in air intake [-] \\
$\tilde{x}$ CO$_2$EGR & Mole fraction of CO$_2$ in EGR [-] \\
$\tilde{x}$ CO$_2$MAN & Mole fraction of CO$_2$ in manifold [-] \\
$\tilde{x}_i$ & Mole fraction of component [-] \\
$\tilde{x}_i$DRY & Mole fraction of component dry analysis [-] \\
$\tilde{x}_i$ WET & Mole fraction of component wet analysis [-] \\
y & DoE modelled response [-] \\
% CO$_2$MAN & CO$_2$ in intake manifold [%] \\
% CO$_2$EGR & CO$_2$ in EGR [%] \\
% CO$_2$AIR & CO$_2$ in ambient air [%] \\
\end{tabular}

**Greek Symbols**

$\beta_{0,...n}$ Regression coefficients

$\varepsilon$ Error

$\phi$ Fuel/air equivalence ratio
\[ \lambda \quad \text{Relative air/fuel ratio} \]
\[ \theta \quad \text{Crank angle} \quad [^\circ \text{CA}] \]
\[ \rho_{\text{FUEL}} \quad \text{Fuel density} \quad [\text{kg/m}^3] \]
\[ \Delta p \quad \text{Pressure drop across injector} \quad [\text{bar}] \]

**Acronyms and Abbreviations**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>Alternating current</td>
</tr>
<tr>
<td>ACEA</td>
<td>European Automobile Manufacturers Association</td>
</tr>
<tr>
<td>ATDC</td>
<td>After top dead centre</td>
</tr>
<tr>
<td>AFR</td>
<td>Air/fuel ratio</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom dead centre</td>
</tr>
<tr>
<td>BMEP</td>
<td>Brake mean effective pressure</td>
</tr>
<tr>
<td>BO</td>
<td>Boost pressure</td>
</tr>
<tr>
<td>BS</td>
<td>Brake specific</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before top dead centre</td>
</tr>
<tr>
<td>C</td>
<td>Carbon</td>
</tr>
<tr>
<td>C_3H_8</td>
<td>Propane</td>
</tr>
<tr>
<td>CA</td>
<td>Crank angle</td>
</tr>
<tr>
<td>CI</td>
<td>Compression ignition</td>
</tr>
<tr>
<td>CCP</td>
<td>Central composite plan</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
</tr>
<tr>
<td>CO_2</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>DC</td>
<td>Direct current</td>
</tr>
<tr>
<td>DI</td>
<td>Direct injection</td>
</tr>
<tr>
<td>DoE</td>
<td>Design of Experiments</td>
</tr>
<tr>
<td>DOHC</td>
<td>Double overhead camshaft</td>
</tr>
<tr>
<td>ECE15</td>
<td>Elementary Urban Cycle</td>
</tr>
<tr>
<td>EG</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>EOI</td>
<td>End of injection</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>EUDC</td>
<td>Extra-Urban Drive Cycle</td>
</tr>
<tr>
<td>EUI</td>
<td>Electronic unit injector</td>
</tr>
<tr>
<td>FC</td>
<td>Fuel consumption</td>
</tr>
<tr>
<td>FID</td>
<td>Flame ionisation detector</td>
</tr>
<tr>
<td>FIE</td>
<td>Fuel injection equipment</td>
</tr>
<tr>
<td>FSN</td>
<td>Filter smoke number</td>
</tr>
<tr>
<td>H₂</td>
<td>Hydrogen</td>
</tr>
<tr>
<td>H₂O</td>
<td>Water</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>HEUI</td>
<td>Hydraulically actuated EUI</td>
</tr>
<tr>
<td>HP</td>
<td>High pressure</td>
</tr>
<tr>
<td>HPCR</td>
<td>High pressure common rail</td>
</tr>
<tr>
<td>IDI</td>
<td>Indirect injection</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated mean effective pressure</td>
</tr>
<tr>
<td>LP</td>
<td>Low pressure</td>
</tr>
<tr>
<td>MAF</td>
<td>Mass airflow</td>
</tr>
<tr>
<td>MCC</td>
<td>Mixing controlled combustion</td>
</tr>
<tr>
<td>MEUI</td>
<td>Mechanically actuated EUI</td>
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<tr>
<td>MR</td>
<td>Split main ratio</td>
</tr>
<tr>
<td>MS</td>
<td>Split main separation</td>
</tr>
<tr>
<td>MT</td>
<td>Main injection timing</td>
</tr>
<tr>
<td>NEDC</td>
<td>New European Drive Cycle</td>
</tr>
<tr>
<td>N₂</td>
<td>Nitrogen</td>
</tr>
<tr>
<td>NO</td>
<td>Nitric oxide</td>
</tr>
<tr>
<td>NO₂</td>
<td>Nitrogen dioxide</td>
</tr>
<tr>
<td>NOₓ</td>
<td>Oxides of nitrogen</td>
</tr>
<tr>
<td>NVH</td>
<td>Noise, vibration and harshness</td>
</tr>
<tr>
<td>O₂</td>
<td>Oxygen</td>
</tr>
<tr>
<td>PAH</td>
<td>Polycyclic aromatic hydrocarbons</td>
</tr>
<tr>
<td>PM</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>PMEP</td>
<td>Pumping mean effective pressure</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts per million</td>
</tr>
<tr>
<td>ppr</td>
<td>Pulses per revolution</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Definition</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------------------------------------</td>
</tr>
<tr>
<td>RP</td>
<td>Rail pressure</td>
</tr>
<tr>
<td>SI</td>
<td>Spark ignition</td>
</tr>
<tr>
<td>SOI</td>
<td>Start of injection</td>
</tr>
<tr>
<td>TCU</td>
<td>Trigger control unit</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead centre</td>
</tr>
<tr>
<td>TEOM</td>
<td>Tapered element oscillating micro-balance</td>
</tr>
<tr>
<td>VCO</td>
<td>Valve covered orifice</td>
</tr>
</tbody>
</table>
1.1 Background

The investigations described in this thesis are concerned with gaining a better understanding of the effects that fuel injection and engine operating parameters have on the exhaust emissions and fuel consumption (FC) of a modern design of diesel engine. The experimental work was carried out on a direct injection (DI) diesel engine with a high pressure common rail (HPCR) fuel injection system typically used for light duty passenger and commercial vehicles. A number of individual investigations were undertaken and are linked by the requirement to establish ways of selecting the fuel injection and engine operating parameter settings across the operating map. The influence of individual parameters on engine out responses has been highlighted at a number of operating conditions and the selection of these parameter settings as the operating map is traversed has been shown. The benefits of using split main injection strategies compared to single injection strategies have been highlighted and the influence of speed and load on fixed parameter settings was investigated. The large number of controllable parameters associated with these HPCR diesel engines should allow better optimisation of the combustion process, but this also increases the workload involved in defining calibration details [1].

HPCR fuel injection systems supply fuel to the injectors via a common fuel rail maintained at a regulated mean pressure, which can be varied independently of operating condition. At constant pressure the injected fuel quantity is directly proportional to injector opening time, making injected fuel quantity independent of engine speed [2]. The fuel pressure in this type of system can, however, exhibit high frequency pressure oscillations produced by the operation of the injectors, which can in turn cause variations in the quantity of fuel delivered in subsequent injection events [3]. The problem of pressure wave oscillations in HPCR systems and the associated variation in injected fuel
quantity has been investigated and methods to address this issue are presented in this thesis.

The many positive features of diesel engines include good fuel economy, low greenhouse gas emissions, durability, reliability and fuel safety. On the negative side, diesels are noisy, have high oxides of nitrogen and particulate emissions and are expensive to produce. Diesel engine technology has made significant advances over the past 10 years and as a result diesel cars are faster, more efficient, drive better and are quieter than ever before [4]. It is not surprising therefore that an analysis of sales and production trends by Ricardo Consulting [5] reveals the increasing interest in diesel engine vehicles. European sales by country shows that in France the diesel car market increased to a record 1.35 million cars sold in 2002 and that diesel penetration accounted for 63% of the cars sold; at the same time sales of gasoline cars fell by 19%. In Germany diesel car sales reached 1.24 million and the penetration increased to 38%, while gasoline car sales fell by 8%. While in Belgium diesel represents 64% of car sales and in Spain 59%. Furthermore, as shown in Figure 1.1, sales of diesel cars in UK approached the 700,000 mark. In 2002 both the VW Audi Group and Daimler Chrysler sold more diesel than gasoline powered cars for the first time, while Peugeot SA came close to 50% diesel sales. Overall in Europe, diesel penetration has reached 41% up from 28% in 1998 and remains on an upward trend gaining an average three percentage points of penetration each year. There is also a small but significant diesel passenger car market starting to emerge in North America; this is in addition to the established market for diesel sports utility vehicles and pick-ups.

This trend, of an increasing proportion of diesel car sales, can result in a number of environmental benefits including reduced fleet fuel consumption. Other benefits are low levels of carbon dioxide (CO₂), reduced levels of gaseous exhaust carbon monoxide (CO) and hydrocarbons (HC) and very low levels of evaporative hydrocarbons [6]. Diesel engines have a downside in the form of relatively high emissions of oxides of nitrogen (NOₓ) and particulate matter (PM). These pollutants continue to attract concern and the limits on diesel emissions specified in European regulations and those in the USA have
become more stringent at each revision, with the result that technology for reducing emissions continues to advance [6]. The pressure to reduce emissions whilst maintaining competitive fuel economy, specific power characteristics and acceptable levels of noise, vibration and harshness (NVH) is a major challenge [7]. HPCR fuel injection equipment is a technology which is being exploited to meet these challenges [8, 9, 10, 11, 12]. The development of a fuel injection system that can vary the injection pressure, the quantity of fuel delivered and the number of injection events per stroke independently and precisely is therefore most desirable.

1.1.1 Emissions Legislation

Exhaust emissions regulations, setting limits on levels of pollutants which can be emitted into the atmosphere, were first introduced in California in 1959 to control CO and HC emissions from gasoline engines [13]. Today standards of this type are spread across the world. The diesel exhaust emissions, which are regulated in many areas of the developed world, are CO, NOx, HC and PM. Carbon monoxide is a colourless and odourless gas, making it even more dangerous, which reduces the capacity of the blood to carry oxygen (O2) to vital organs in the body. High concentrations of CO can be fatal and even low concentrations pose a health risk, especially to those suffering from heart disease [14, 15]. Exposure to high levels of nitrogen dioxide (NO2), part of the NOx grouping, has been linked with respiratory problems and long term exposure may affect lung function and increase the response to allergens [14, 15]. HC’s contribute to ground level ozone formation, which can lead to damage of the respiratory system [14]. The fine particles of PM also have an adverse effect on the respiratory system and have been associated with bringing forward the deaths of those suffering from respiratory illness [14]. Although CO2 is not directly harmful to human health it is the most significant of the greenhouse gases contributing to climate change. In response to these concerns, at the Kyoto Conference on Climate Change in December 1997, many developed countries agreed to legally binding targets to reduce greenhouse gas emissions [16, 17]. Following this the European Commission and the European Automobile Manufacturers Association (ACEA) came to an agreement in July 1998 to reduce the CO2 emissions from new passenger cars.
by over 25% to an average of 140 g/km by 2008 [14]. A number of steps have been introduced in the UK as incentives for the purchase and use of more efficient vehicles to lower CO₂ emissions. Since March 2001 a system of graduated vehicle excise duty has been in operation for new cars based on the level of CO₂ emissions and since April 2002 company car tax has been based on the CO₂ emissions of the vehicle provided to an employee for their private use [14].

Table 1.1 shows the permissible limits for past, present and future exhaust emissions limits for diesel passenger cars. Conformity tests entail driving the test vehicle through a standard pattern of vehicle speeds and recording the average mass per kilometre of the pollutant emitted. In Europe the standard test cycle is the New European Drive Cycle (NEDC). A graphical representation of the vehicle speed trace for the NEDC is shown in Figure 1.2. This lasts for 1180 seconds, approximately 20 minutes, and is comprised of four runs of the Elementary Urban Cycle (ECE15) followed directly by an Extra-Urban Drive Cycle (EUDC) to simulate higher speed driving conditions up to 120 km/h [15]. European regulations for passenger cars were originally conceived in 1970 with a European Union directive, 70/220/EEC [13]. Amendments to this original regulation have been numerous from the introduction of Euro I in 1992 and Euro II, to the more recent Euro III and future Euro IV standards [13, 14, 15]. Euro IV, planned for 2005, also includes the requirement that vehicles must meet the required standards after 100,000 kilometres or five years, whichever is sooner, and also incorporates more stringent fuel quality rules to significantly reduce the sulphur content of diesel fuel to 50 ppm (parts per million). Euro V regulations [18] are due for introduction in 2010 and the forecast pollutant levels are also shown.

1.1.2 History and Development of Diesel Fuel Injection Systems

Proposed further lowering of the limits on emissions levels for Euro V require improved methods of aftertreatment or reduced levels of formation of both soot and NOₓ emissions particularly [19]. Combustion in diesel engines is a complex heterogeneous spray process, which is highly dependent on fuel injection parameters [12]. Precise control over fuel injection, and thus spray
formation, is essential for the control of the combustion process. The low pressure side of the fuel injection system consist of the fuel tank, fuel filter, supply pump, overflow valve and delivery lines. The fuel pressure required for injection is generated on the high pressure side of the system, where the fuel is pumped through the high pressure fuel lines and nozzle holder assembly to the injector nozzle [20], which has a large pressure differential across it. The pressure differential is required so that the injected liquid fuel jet will enter the combustion chamber at a sufficiently high velocity to atomise the fuel into small droplets in order to enable rapid evaporation and mixing [21]. This pressure differential also enables the injected fuel to transverse the combustion chamber in the time available in order to fully utilise the charge air.

Diesel engines have been the subject of continued developed since the original patent by Rudolf Diesel in 1893 [22]. The main types of fuel injection systems include pump-line-nozzle, electronic unit injector (EUI), mechanically actuated EUI (MEUI), hydraulically actuated EUI (HEUI) and HPCR. Light duty automotive applications are now dominated by EUI and HPCR systems.

Pump-line-nozzle systems use in-line or distributor pumps connected via high pressure fuel lines to the injection nozzle. In-line fuel injection pumps have the same number of plungers, driven by the camshaft, as cylinders in the engine and they can have either mechanical governors or electronic actuators [12, 23]. Distributor fuel injection pumps, with axial or radial plungers, were designed to be smaller and lighter than in-line types and work by injecting fuel into each cylinder with the rotation of a single plunger. A solenoid valve controls the injection timing and meters the fuel [12, 23]. The unit injection system combines the pump and the injector nozzle in one unit and one of these unit injectors is installed in the cylinder head for each cylinder [12]. Bosch introduced an in-line pump in 1987, a unit injector in 1994 and a common rail system for passenger cars in 1997 [2, 22]. Caterpillar introduced their MEUI system in 1989 and a HEUI system in 1995 [24]. Denso [23] introduced their in-line pump in 1981, an electronic distribution pump in 1985 and a common rail system for trucks in 1995; a common rail system for passenger cars followed in 1999. With the advance of electronically controlled systems, the
diesel fuel injection system has become one of the critical emissions control technologies in recent years [10]. Electronically controlled common rail fuel injection systems are attracting considerable interest and have been investigated by a number of authors [2, 10, 11, 25, 26, 27, 28, 29, 30]. Parallel developments in the design of the injector nozzle have led to a reduction in sac volume, with a related improvement in engine HC emissions, with the introduction of the mini-sac, micro-sac and valve covered orifice (VCO) nozzles; this has also stabilised the fuel spray behaviour for the low fuel masses typically seen with pilot injections [2, 31].

1.1.3 High Pressure Common Rail Fuel Injection System
HPCR systems separate fuel pressurisation and injection from each other. The fuel rail pressure is generated using a high pressure pump and controlled by a regulator valve located either in the pump or the rail itself. The injector is essentially made up of a nozzle and a solenoid valve or piezo-electric actuator, which is energised and controlled by an electronic control unit or driver unit, connected to the rail by a short high pressure fuel line. Changes in engine speed and load, hence fuel injection requirements, have no effect on the generation of injection pressure [2, 12]. The quantity of fuel delivered is dependent on the opening period of the injector and the fuel pressure [15]. The injector can be energised several times during a single cycle of the engine and in this way pilot, split main and post injections are feasible [32]. These systems are designed to operate with flexible electronic control of fuel delivery, injection timing, injection pressure and rate of injection. By considering these parameters, the HPCR system is capable of achieving a level of performance and driving comfort for diesel cars similar to that for gasoline powered models with significant fuel economy and low exhaust emissions [33]. A diagram depicting the component parts of an HPCR fuel injection system from Denso is shown in Figure 1.3. Piezo-electric injectors respond rapidly with switching times of less than 100 µsec and as a consequence injected fuel quantities below 1 mm³ per stroke can be attained [27]. The piezo-electric system used during the studies detailed in this thesis provided a great deal of flexibility over the injection timing and the shaping of the injection profiles; it also allowed the fuel quantity and the fuel injection pressure to be varied independently of
engine speed. The system was designed to deliver up to 5 injection events per cycle with separations as low as 0° CA and minimum fuel delivery quantities down to 0.5 mg per injection event.

The latest designs of HPCR systems have fast acting solenoids or piezo-electric injectors capable of providing two or more separate injection events in a single engine cycle, rather than the traditional single spray just prior to or during the combustion stroke of a diesel engine [12]. The terminology and definitions of the injection profiles considered are described graphically in Figure 1.4 and are used throughout this thesis. This includes split main separation and ratio, main injection timing and pilot injection separation in terms of needle lift measurements and crank angles in the engine cycle. Where used, the pilot separation was kept at 25° CA from end of pilot to start of main injection event and the pilot quantity was kept to approximately 0.5 mg per injection event.

1.2 Aims and Objectives of Thesis

The investigations reported in this thesis were undertaken to evaluate HPCR fuel injection strategies through performance studies and to identify how independent of operating conditions these strategies are. This included the examination of the effects of fuel injection and engine operating parameters on engine performance. The fuel injection parameters were rail pressure, main injection timing, split main injection separation and ratio. The engine operating parameters were boost pressure and exhaust gas recirculation (EGR) rate. These parameters are highly inter-dependant on a conventional multi-cylinder engine but could be set to any combination as required with this test facility. The investigations were carried out across the speed and load operating map in order to establish ways of reducing exhaust emissions and fuel consumption.

The parameter settings in different regions of the speed and load operating map have been explored to understand the effects of the fuel injection and engine operating parameters on engine out emissions and fuel consumption. Comparisons of results obtained using split main and single injection strategies provide an indication of the relative performance, benefits and penalties. The
The adverse effects of pressure waves in HPCR fuel injection equipment (FIE) are well documented [2, 3, 34, 35, 36, 37, 38, 39] and an investigation was undertaken to show the impact of these pressure variations on fuel delivery; a relationship between needle lift, injection pressure and the quantity of fuel injected was developed. Methods to eliminate the pressure waves, and the associated variations in the quantity of fuel injected, were developed and applied. A patent was developed from this work which has been filed with the Patent Office in association with the Ford Motor Company [40].

1.3 Layout of the Thesis

The previous sections have outlined the investigations that have been presented in this thesis and described the background to these investigations. A literature review is presented in Chapter 2, which provides background information on emissions characteristics and the effects that the fuel injection and engine operating parameters have on engine out emissions and fuel consumption.

The initial part of this investigation was taken up largely by installation, commissioning and trouble shooting of the test facility and this is detailed in Chapter 3. This included the acquisition and assembly of the engine hardware and HPCR fuel injection system, the setting up of the data acquisition and
control systems, along with the instrumentation and data processing requirements.

Chapter 4 was concerned with gaining an understanding of the fuel delivery characteristics of the injector. Needle lift traces were interrogated to develop relationships between needle lift, injection pressure and quantity of fuel injected. An investigation into pressure oscillations in the FIE was undertaken and the successful damping of these oscillations was demonstrated. This resulted in a more consistent and predictable quantity of fuel being delivered in the second part of a split main injection.

The individual effects and sensitivities of the fuel injection and engine operating parameters on the engine-out responses were shown in Chapter 5. The DoE methods used in the experimental work undertaken in this thesis are also introduced here.

Chapter 6 detailed studies undertaken to show the effects of engine speed and load on various engine-out responses. This also showed how far a particular calibration, or combination of parameter settings, could be moved across the operating map before exhaust emissions became problematic. This gave an indication of how much of the operating map could be covered by a limited number of fixed calibrations. It was also possible to show here how the parameter settings need to be adjusted in order to maintain values near the optimum FSN-NO_x trade-off point at different points on the operating map.

Split main injection strategies were compared to single injection strategies in Chapter 7. These comparisons and studies from Chapter 5 and Chapter 6 were brought together to highlight techniques that could be used across the speed and load operating map to simplify the calibration process and reduce the amount of experimental testing needed.

Chapter 8 gives an overall appraisal of the work undertaken, highlighting the features, the difficulties and the findings, while summarizing the significant conclusions that were drawn.
2.1 Introduction

This Chapter provides an overview of the literature on the influence that fuel injection and engine operating parameters have on pollutant emissions and fuel economy. The quality of combustion in a diesel engine is strongly influenced by how air and fuel are introduced into and mixed in the combustion chamber. The overall objective of a fuel injection system is to generate fuel droplets of the right size and place them in the right location, in regions of high turbulence to improve mixing. It is also important to inject the fuel at the right time and at a sufficient rate and pressure to ensure complete combustion in the time available. In general combustion in diesel engines goes through four stages: ignition delay, pre-mixed combustion, mixing controlled combustion (MCC) and late combustion [32]. These phases are shown in Figure 2.1 and can be summarised as follows:

- Ignition delay is the period between the start of fuel injection and the start of combustion. Experimental results have shown that the ignition delay is primarily a chemical effect, with a value of about 5° to 9° CA.
- Pre-mixed combustion is the rapid combustion of the air and fuel mixture prepared during ignition delay, producing a high rate of pressure rise and heat release.
- Mixing controlled combustion occurs when the pre-mixed air and fuel mixture has been consumed. The combustion rate is then determined by the rate at which new readily combustible mixture is being formed, which is mainly determined by the rate at which the air and fuel are mixed.
- Late combustion has a lower rate of heat release and continues well into the expansion stroke, until the full utilisation of the air and fuel has occurred. As with MCC, late combustion is controlled by diffusion [32].
The phases listed above are strongly influenced by the behaviour of a DI diesel engine fuel spray. As the fuel jet leaves the nozzle, the liquid core penetrates the combustion chamber, becomes turbulent and spreads out as it entrains and mixes with the surrounding air. The initial velocity of the fuel jet is greater than $10^2 \text{ m/sec}$ [32]. As this develops, fuel droplets along the edge and at the tip are atomised into drops, of order $10 \mu m$ in diameter, and mix with the air to form a fuel and air cloud surrounding the liquid core [32]. It is this near homogeneous cloud that ignites first and creates the pre-mixed combustion. As the injection progresses and the flame propagates, more of the liquid core is atomised and vaporised by the rapidly increasing in-cylinder temperatures. This fuel then burns at a rate governed by the rate at which the fuel and air mix together, which is the MCC phase.

2.2 Emissions Formations and Fuel Economy

DI diesel engines are more efficient than spark ignition (SI) gasoline engines producing the same power. For this reason diesel engines are widely used in heavy-duty transport applications [19]. However, diesel engines suffer from relatively high emissions of NO$_x$ and particulate emissions [41] and stringent emission standards have been imposed on diesel engine emissions because of this. In the DI diesel engine the fuel is not evenly distributed within the combustion chamber. Within regions where the fuel concentration is close to stoichiometric the combustion temperature is high and hence more NO$_x$ emissions are produced, whereas, in regions where the fuel concentration is rich the lack of oxygen results in more smoke production [42]. Therefore, in order to simultaneously decrease NO$_x$ and smoke emissions from a DI diesel engine, it is necessary to create a proper spatial distribution of the injected fuel and to reduce as much as possible the regions of fuel concentration where NO$_x$ and smoke emissions are generated.

The objectionable constituents present in the exhaust of diesel engines include smoke or soot from carbon in the fuel (C), NO$_x$, HC’s, CO and PM, which is principally carbon [43]. The quality of diesel fuel combustion is strongly influenced by the mixing of the fuel and air and how they are introduced into
the combustion chamber. The diesel engine combustion process is predominately an unsteady turbulent diffusion flame and the fuel is initially in the liquid phase [32]. Diesel engines are superior to gasoline engines in terms of fuel economy and CO₂ emissions. However, they have problems awaiting solution relating to NOₓ, smoke and particulate emissions as well as combustion noise. Fuel has to be mixed with the air intake charge thoroughly in order to achieve maximum chemical energy release and controlled combustion duration. Shorter combustion duration is required to reduce the time that combustion gases are exposed to higher temperatures, reducing NOₓ emissions. Whereas, longer combustion periods may help burn off remaining HC and PM emissions [44]. The exhaust from a diesel engine is a complex mixture of organic and inorganic compounds in solid, liquid and gaseous phases. These components have been identified in Table 2.1 [15].

Traditionally the reduction of NOₓ, smoke and PM emissions from diesel engines has been challenging. Strategies employed to reduce the amount of NOₓ formed tend to raise smoke and PM emissions and vice versa [45]. Also strategies which reduce NOₓ emissions are likely to incur a fuel consumption penalty [12]. Various techniques have been employed by the industry to tackle the problem of balancing the reduction in NOₓ, smoke and PM exhaust emissions as well as improving fuel consumption. Techniques such as increasing injection pressure and boost pressure have been used [8, 9, 10, 11, 12, 46, 47]. Turbocharging increases the specific power of these engines to achieve values comparable to gasoline engines [15]. Retarding injection timing [32] and the use of electronically controlled fuel injection systems [24, 27, 30, 48, 49, 50, 51, 52] are also techniques that have been adopted. By employing DI combustion systems fuel efficiency can be improved by about 15% compared to indirect injection (IDI) swirl chamber engines [15].

2.2.1 NOₓ Emissions
Nitric oxide (NO) is the predominant oxide of nitrogen formed inside the engine cylinder, but it is usually grouped together with nitrogen dioxide (NO₂) as oxides of nitrogen (NOₓ). NO principally arises in the cylinder but it oxidises to form NO₂ in the exhaust pipe or when entering the air. The
principal source of NO\(_x\) is from the oxidation of atmospheric nitrogen, thus NO and NO\(_2\) are produced when nitrogen and oxygen react in-cylinder at high temperatures. NO\(_2\) is a poisonous gas, which destroys lung tissue and damages resistance to viral infection.

The rate of NO\(_x\) formation depends on temperature and pressure and the availability of O\(_2\) in the combustion chamber. Heisler [53] states that the amount of NO\(_x\) created is an exponential function of combustion temperature, so that even a small decrease in the combustion temperature will produce a significant reduction in NO\(_x\) production. NO\(_x\) forms in the high temperature burned gas behind the flame through chemical reactions involving nitrogen and oxygen atoms and molecules. The fuel and air distributions within the burned gases are non-uniform and NO\(_x\) formation rates are highest in the close-to-stoichiometric regions [32], as maximum combustion temperature occurs when the fuel/air equivalence ratio, \(\phi\), is close to a value of 1. In addition, the length of time that combustion occurs at higher temperatures is important as this increased dwell time leads to increased NO\(_x\) levels. The critical time period is when the burned gas temperatures are at a maximum, which is between the start of combustion and shortly after the occurrence of peak cylinder pressure. Therefore, a mixture which burns early in the combustion process, before top dead centre (TDC), is especially important since it is further compressed to give even higher temperatures, increasing the NO\(_x\) formation rate. The bulk gas temperature decreases as the cylinder gases expand and as the high temperature gases mix with cooler burned gas, this freezes the NO chemistry and stops the decomposition of NO [32]. Therefore, limiting the amount of locally available O\(_2\) and reducing the peak in-cylinder temperature will limit NO\(_x\) formation.

The most important parameters for peak temperature reduction are retarding the injection timing, reducing the inlet air temperature by using an intercooler, combustion chamber design and compression ratio [54]; the combustion chamber design and compression ratio can not be readily adjusted however. Turbocharging increases in-cylinder pressures which in turn leads to increased in-cylinder temperatures, resulting in higher NO\(_x\) emissions. The addition of boost pressure also results in increased NO\(_x\) emissions due to increased levels of O\(_2\) which are locally available. The adverse effects of turbocharging can be
addressed by increasing the amount of EGR used, which reduces the amount of O₂ available [55]. A reduction in O₂ concentration reduces the flame temperature. Reductions in flame temperature result in a reduction in NOₓ formation rate and levels of NOₓ in the exhaust [56]. It is important to cool the EGR before reintroducing it into the cylinder in order to avoid raising the intake temperatures. Controlling NOₓ emissions has a number of fundamental drawbacks however. The high thermal efficiency of diesel engines inherently results in higher peak temperatures and reducing these will erode the fuel economy advantage of the diesel engine. Furthermore, there is generally a trade off between NOₓ and smoke and particulates and a reduction in NOₓ tends to cause an increase in smoke and particulates.

2.2.2 Soot, Filter Smoke Number and Particulate Matter

Compression ignition (CI) engines tend to emit smoke, which has the characteristically grey or black colour of soot, or carbon, particles. Here this does not include the bluish smoke that signifies lubricating oil being burned, or the white smoke that is characteristic of unburned fuel; these types of smoke occur with malfunctioning engines [57]. The black smoke from diesel engines is generated by high temperatures in the fuel-rich zone during diffusion (or mixing) controlled combustion, MCC, and by the low values of the local air/fuel ratio, AFR, and is derived from the incomplete combustion of the fuel [15]. After the rapid combustion at the end of the delay period, the subsequent combustion of the fuel is controlled by the rates of diffusion of air into the fuel vapour and vice versa, and the diffusion of the combustion products away from the reaction zone; this is the diffusion controlled combustion phase. The final rate of soot release depends on the difference between the rate of formation and the rate of oxidation. The smoke emissions can be reduced by shortening the diffusion controlled combustion phase, since this gives less time for soot formation and more time for soot oxidation. The diffusion phase can be shortened by increasing swirl, having more rapid fuel injection, which is achieved with higher injection pressures, and utilising a finer fuel spray. Advancing injection timing also reduces smoke emissions [57] by allowing more time for better mixing of the fuel and air and also by allowing more time during the expansion stroke for oxidation of the soot formed [15]. Diesels
always run lean of stoichiometric but an increased amount of injected fuel can result in reduced amounts of locally available $O_2$ and consequently more fuel-rich areas that lead to increased levels of smoke production. Even though the overall equivalence ratio may remain lean, locally over-rich fuel conditions may exist through the expansion stroke and into the exhaust process [32]. Fuel flow rates are limited by this appearance of soot in the exhaust that did not burn to $CO_2$ or CO and this occurs even though the engine is running lean [21].

Carbon particles are formed by the cracking of large hydrocarbon molecules on the fuel-rich side of the reaction zone during the MCC phase, where air mixing with the outer edges of the fuel jet sustains the combustion. Soot forms in the high temperature unburned fuel containing core of the fuel sprays, within the flame region, where the fuel vapour is heated by mixing with hot burned gases, but is too rich to be oxidised. Soot then oxidises in the flame zone when it contacts oxygen, giving rise to the yellow luminous character of the flame [32]. The soot formations occur in fuel-rich regions [15, 58] and soot growth occurs in overly-lean region as unburned HC’s become attached to soot particles [32]. These diesel soot particles consists of collections of primary particles or spherules agglomerated into aggregates called particles, which range in appearance from chains of spherules to clusters of spherules containing as many as 4000 spherules. These spherules range in diameter of 10 nm to 80 nm, while the particles have mean diameters in the range 50 nm to 220 nm [32]. Both these types of particles have absorbed or condensed hydrocarbons associated with them, which are partly burned HC’s from fuel-rich regions in combustion mixture and also, to a lesser extent, from fuel lean regions where mixture is too lean to combust [32]. Most of the soot is oxidized to $CO_2$ during the combustion process and further oxidation occurs during the expansion stroke, after the end of the MCC phase. However, complete oxidation is difficult to achieve because combustion is limited by the rate of mixing toward the end of combustion and in-cylinder temperatures are falling due to expansion. The end of injection should be sharp to minimise soot formation due to fuel entering into cylinder late in the cycle [54].
The AVL 415S Variable Sampling Smoke Meter uses the filter paper method to provide a measure of the carbon content in the exhaust gases, referred to as soot [59]. The measurement value, Filter Smoke Number (FSN), corresponds to the soot content, or soot concentration in mg/m³, of the exhaust gases. An exhaust sample is passed through clean filter paper in the instrument and the soot content causes the blackening on the filter paper, which is detected by a photoelectric measuring head and evaluated in the microprocessor to produce the result in FSN [59].

Particulates are defined as any matter collected on a filter paper at a temperature of 325 K [15, 60, 61] through which diluted exhaust gases have been drawn; the bulk of this matter is either unburned hydrocarbons or soot. This simple definition hides the immensely complicated nature of PM emissions. The material collected on the filter is generally classified into two parts: a solid carbon material or soot and an organic fraction that consists of hydrocarbons (unburned fuel and lubricating oil) and their partial oxidation products, whether condensed onto the filter or absorbed to the soot [62]. The soot is visible as smoke, therefore any measure that reduces either the exhaust smoke or HC emissions will also reduce particulate levels [57]. PM can be classified into five distinct composition groups: carbonaceous, inorganics, organics, sulphates and nitrates. The typical composition of diesel exhaust particulate matter is 31% carbon, 40% unburned oil, 14% sulphate and water and 7% unburned fuel, and the remaining 8% is unknown [15]. Soot is the non-soluble fraction of particulate matter [12].

2.2.3 HC Emissions

HC emissions originate from three main sources in a properly functioning engine. Firstly, in the regions where the flame is quenched on the cylinder walls. Secondly, around the perimeter of the reaction zone where the mixture is too lean to burn as excessive dilution with the charge air prevents the combustion process from either starting or going to completion. And thirdly, from fuel that vaporises from the nozzle sac volume into the combustion chamber during the later stages of combustion [32]; the end of injection should be sharp to minimise this fuel entering the cylinder late [54]. Lubricating oil is
also a source of HC emissions. HC emissions increase at part load where there is an increase in ignition delay and the quantity of mixture at the perimeter of the reaction zone that is too lean to burn increases. Advancing injection timing, to reduce ignition delay, reduces HC emissions, but this leads to increased NO\textsubscript{x} and noise [57]. Furthermore, HC emissions or unburned fuel in the exhaust may condense to form white smoke during engine starting and warm up.

HC emissions are composed of many different organic compounds, such as aldehydes, ketones, alcohols, ethers, alkanes, alkenes, aromatics and carboxylic acids. Specifically the aromatic HC compounds in the exhaust gases are the source of diesel odour. No discrimination is made between these different compounds during detection and measurement. Some HC compounds have a narcotic effect; others irritate the mucous membranes, while polycyclic aromatic hydrocarbons (PAH) are known to be carcinogenic. In addition, HC’s can contribute to acid rain and react in the presence of ultra violet light to cause photochemical smogs [53].

2.2.4 Noise

Combustion noise is an unattractive feature of performance. Combustion-generated noise is produced by the high rate of heat release immediately following the ignition delay period [32]. The in-cylinder pressure derivative, \( dP/d\theta \), is closely related to combustion noise [48]. A pilot injection significantly reduces the noise generated [3, 63] especially when the engine is not fully warm [3], by limiting the amount of pre-mixed combustion [48]. Without pilot injection, in-cylinder pressure rises very steeply at the start of combustion and features a sharp peak in maximum pressure and temperature [27]. A pilot injection can be used throughout the whole speed range to reduce the combustion noise from high speed DI diesel engines [26]. Furthermore, turbocharging is beneficial in the reduction of combustion noise due to the shortened ignition delay period created by the increased in-cylinder pressures [64] and improved mixing of the air fuel charge. Smoke, HC and PM emissions and fuel consumption penalties are generally seen when combustion noise is reduced.
2.2.5 CO and CO₂

CO₂ is a “greenhouse gas” which contributes to global warming. Improved overall fuel economy limits the production of CO₂ [15] and another practical way of reducing CO₂ is to use fuels with lower carbon content [15]. CO₂ is a complete product of combustion of a carbon based fuel. Some carbon from the fuel turns to CO and represents chemical energy that is not exploited in the combustion process. CO is generated during combustion in the fuel-rich regions where insufficient oxygen is available for complete combustion, or oxidisation, to convert CO to CO₂ and due to dissociation from CO₂ [65]. These fuel-rich regions can occur in a diesel engine even though there is an overall surplus of oxygen. Formation of CO also occurs where the flame is quenched by cold surfaces such as cylinder walls, which explains why CO is significantly higher in a cold engine as can be seen in the NEDC where the majority of CO formation is seen in the early part of the cycle [65]. The production of CO should not be a problem in a well calibrated diesel engine as the AFR is always lean, ensuring that there is enough oxygen for complete combustion of the fuel [64]. CO₂ emissions for a diesel engine are better than for a gasoline engine as CO₂ formations are directly related to the quantity of fuel used and diesel engines are significantly more fuel efficient than their gasoline counterparts [15].

2.3 The Effects of Fuel Injection and Engine Operating Parameters on Emissions and Fuel Economy

2.3.1 Pilot Injection

In addition to the effect on noise levels described in Section 2.2.4, a pilot injection has the effect of lowering the levels of O₂ available for the main injection fuel spray to mix with. The main injection entrains the burned gas of the pilot injection which results in the combustion progressing more gradually. Flaig et al [27] suggest that a pilot injection of 1 to 3 mg per stroke occurring anywhere up to 90° CA BTDC pre-conditions the combustion chamber and improves the efficiency of the combustion. Pilot injections with a separation of between 3° and 15° CA are known to provide substantial improvements in combustion noise [66], while at higher speeds there is less potential for noise
reduction by pilot injection [49]. If the pilot is very close to the main injection, the pilot has little time to mix before the main injection, so the noise reduction is less effective, but less soot is formed in the diffusion burn [49]. The pilot quantity has to be controlled precisely and must take place at the right time interval before the main injection. A pilot injection too small and too early can raise the combustion noise, while a pilot too large can increase the particulate emissions. According to Stumpp and Ricco, the pilot fuel quantity must decrease with increasing engine speed and its separation to the main injection must increase with rising engine speed; such a variable pilot injection is feasible with the common rail system [25].

Diesel engines with high pressure injection systems can misfire sometimes directly after start, because of poor combustion conditions. A pilot injection can reduce the risk of misfire by reducing the ignition delay of the main combustion; which results in a decrease in white smoke [27] and HC emissions. With a pilot injection combustion occurs earlier due to the shorter ignition delay thereby reducing fuel consumption. If the pilot injection occurs too early however this can produce negative torque, which increases fuel consumption [27]. Pilot injection is effective at reducing NO\textsubscript{x} and HC emissions at the lower load conditions considered, but it increases smoke to some degree. When main injection timing is retarded to reduce NO\textsubscript{x} emissions the pilot is effective to keep particulates approximately constant. At higher loads, the pilot injection has little impact [67].

2.3.2 Split Main Injection Strategies
Advances in diesel engine FIE have allowed high pressure and multiple injections to be used to reduce particulate emissions without a significant penalty in NO\textsubscript{x} emissions [68]. Multi-stage fuel injection is used to reduce engine noise and NO\textsubscript{x} emissions. This reduces the proportion of pre-mixed combustion and increases the proportion of MCC [69]. Reduced pre-mixed combustion lowers the peak in-cylinder pressure and temperature and hence reduces NO\textsubscript{x} emissions. Whereas, the increased duration of the MCC means that there is less time for soot oxidation during the expansion stroke, resulting
in higher levels of FSN. Therefore with multi-stage fuel injection, a lower peak in-cylinder pressure is expected when compared to single injection.

Modelling work undertaken by Han et al [8] indicates that soot reduction, without NOx penalty, achieved through the use of multiple injections is due to the reduction in the quantity of fuel contained in the fuel-rich region near the tip of the spray [12], as shown in Figure 2.2. With a split main injection strategy the fuel is injected over a longer period meaning that more thorough mixing occurs with the available charge air and the second part of the injection is entering a more turbulent region. There is always excess air overall in diesel combustion, but stratification can result in fuel-rich regions which give rise to soot and particulates. Locally over-rich conditions may exist throughout the expansion stroke and into the exhaust process [32]. Engine tests carried out on a small DI diesel engine by Corcione et al [2] showed that splitting the injection process into many steps improves control over the in-cylinder mixture process. Tow et al [70] found that at both high and low loads, double and triple split injection strategies reduced both soot and NOx emissions. Furthermore, Corcione et al [71, 72] demonstrated that the modulation of the injection can simultaneously control NOx and soot emissions in light duty diesel engines as the fuel required for a particular operating condition can be injected in many steps according to the amount of oxygen available for mixing and combustion.

Chan et al [19] compared multiple injection strategies with two and three injections with a baseline single injection and demonstrated the possibility that soot emissions could be reduced due to the delay between injections. Initial injections generate turbulence, which encourages air entrainment in the fuel spray of subsequent injections, resulting in better overall mixing of the air and fuel charge. This reduces the fuel-rich zones which are areas of soot production. Even though the overall equivalence ratio may remain lean, locally over-rich fuel conditions may exist through the expansion stroke and into the exhaust process [32]. Furthermore, Bower and Foster [73] concluded that a split main injection changed the spray mixing, vaporisation and fuel distribution, which had a significant impact on peak combustion pressure and rate of pressure rise. While, Nehmer and Reitz [74] showed that split main
injections could reduce NO\textsubscript{x} without a significant increase in soot and particulate matter. This was attributed to the better utilisation of the charge air to reduce soot and the relatively late combustion compared to a standard single injection strategy to reduce NO\textsubscript{x} emissions.

Dwell time for split injections, or split main separation, is important to levels of soot production as the timing of the second injection dictates if it will strike a soot rich region left in the bowl from the first injection. Optimised dwell timing can result in significant reductions in NO\textsubscript{x} and soot. The reduction in soot with split main injection strategies stems from the reduced quantity of fuel in the first injection event; it is this fuel that primarily gives rise to soot production even when the two injections are close coupled [75]. For NO\textsubscript{x} production the first injection is again important. When the split main is close coupled, the second injection extends the period over the cycle during which high temperatures are seen, which increases NO\textsubscript{x} production. When the separations are large, the second injection comes too late in the expansion stroke to produce more of the high temperature gas associated with NO\textsubscript{x} production [75].

Furthermore, Li et al [76] have speculated that using varying injection pressures in a single cycle can be beneficial. The theory states that a combination of relatively low fuel pressure for the 1\textsuperscript{st} injection, for reducing NO\textsubscript{x}, and high fuel pressure for the 2\textsuperscript{nd} injection, for increasing the oxidation of soot, is perhaps the optimum split main injection scheme.

2.3.3 Post Main Injection
Post main injection during the expansion stroke can be used to manage conditions in aftertreatment components that promote regeneration [3]. Specifically, post injections can serve as a reducing agent for a lean-NO\textsubscript{x} catalyst [25] by providing the HC’s to activate the catalyst [2]. Post injections can also improve the efficiency of a DeNO\textsubscript{x} catalyst [3]. Furthermore, a post injection can oxidise, or reduce, the soot generated earlier in the combustion process [77].
2.3.4 Injection Pressure

The key feature of HPCR fuel injection systems is the high fuel injection pressures that are possible. High injection pressures affect both atomisation and penetration which in turn affect engine output, fuel economy and emissions formation.

Injection pressures have steadily risen over the last 30 years and research has shown that there are still benefits to be realised by further increasing the injection pressure and rate [31]. These developments may proceed at a slower rate due to the technical, cost and reliability challenges. Increasing the maximum injection pressure will allow the use of smaller nozzle holes in the injector without reducing maximum power. The use of smaller injection nozzles, which reduces smoke, particulates and soot, gives better atomisation of the spray and faster rates of air entrainment and mixing in the fuel vapour jet [48, 49]. Flexible electronic control of injection timing and pressure at each engine operating condition allows a good emissions/performance trade-off to be achieved. Higher injection pressures should be used for the part load range as this enables particulate emissions to be lowered due to better penetration, atomisation and mixing of the fuel in the combustion chamber. In turn, higher EGR rates can be used to reduce NOx emissions without particulates penalty [27]. Higher injection pressures result in low particulate emissions and a large maximum engine torque, even at low speeds [25]. Higher injection pressure allows the adoption of smaller nozzle holes without any penalty to the maximum power and reduces smoke and particulates, which are either soot or unburned HC, by allowing the use of smaller injection nozzles to give better atomisation of the spray and faster rates of air entrainment and mixing in the fuel vapour jet [49]. However, increasing rail pressure can increase combustion noise, fuel consumption and NOx emissions.

It was shown by Badami et al [48] that the principal effect of an increase in injection pressure is the reduction of the combustion angle. This is the combustion period in crank angles degrees. When the combustion angle is reduced higher temperatures and pressures are reached during combustion and the thermodynamic efficiency increases. However an increase in combustion
noise is seen. Furthermore, Badami et al [48] found that with increased injection pressure the FMEP increased by 0.2 bar, due to the increased power required by the high pressure fuel pump. The same increase in injection pressure, however, gave a rise in IMEP of about 0.5 bar, so the positive effect is slightly greater than the negative effect. In order to reduce the power absorbed by the high pressure pump it is recommended to use high injection pressures only at those operating conditions where particulate emissions need to be reduced, which would improve fuel economy [25]. An increase in injection pressure results in an increase of the fuel mass taking part in the pre-mixed combustion phase. This results in FSN reduction as the pre-mixed phase is almost smoke free as it takes place after good air fuel mixing has occurred [48]. In terms of improved combustion, Lapuerta et al [28] reported a 1 bar increase in IMEP for a 600 bar increase in rail pressure with start of injection at TDC on a single cylinder DI diesel engine with 572cc displacement. Badami et al [48] report a 5 to 7 bar increase in peak cylinder pressure with a 200 bar increase in rail pressure from 1300 to 1500 bar and injection at 17.4° BTDC on a 2.4l I5 configuration DI Fiat engine with an HPCR fuel system. This equates to a 2.3% increase in maximum power and a corresponding 2.7% improvement in brake specific fuel consumption.

Overall using higher injection pressures results in better fuel atomisation resulting in improved mixing of fuel and air. Thus increasing the pre-mixed combustion, reducing the combustion duration and increasing the rate of the mass of fuel burned. Releasing the fuel energy faster during the period where cylinder volume is increasing slowly increases temperature and pressure and therefore increases thermal efficiency of the engine. These higher in-cylinder pressures near TDC have a detrimental effect on mechanical stress and engine friction however and the increase in pre-mixed combustion and therefore peak in-cylinder temperature adversely affect NO\textsubscript{x} production.

2.3.5 Main Injection Timing
Pre-mixed combustion is an important source of NO\textsubscript{x} emissions and with more advanced timing there is more thorough mixing occurring, greater levels of pre-mixed combustion and hence higher levels of NO\textsubscript{x} emissions and generally
lower FSN production. Retarding injection timing reduces NO\textsubscript{x} as peak in-
cylinder temperatures and pressures are lower later in the cycle after TDC [32, 49], however, this increases fuel consumption as the thermodynamic efficiency
is reduced. This also has the effect of increasing FSN and HC emissions as less
complete combustion occurs. Timing of the start of injection can be varied to
reduce NO\textsubscript{x} emissions by retarding or particulate and smoke emissions by
advancing, but only with a corresponding increase in the other pollutant [41].
Advanced injection timing improves fuel consumption while the addition of
EGR, to slow combustion rates, helps to stop the NO\textsubscript{x} increases usually
associated with advancing injection timing [12].

2.3.6 EGR
EGR is a method by which a percentage of exhaust gas is routed from the
exhaust back into the intake manifold of the engine. This was found to reduce
the flame temperature and speed, which gives a significant reduction in NO\textsubscript{x}
emissions at the detriment of soot levels [57]. After air and therefore nitrogen,
the principal constituents of EGR are CO\textsubscript{2} and H\textsubscript{2}O [78]. These diluents cause
an increase in ignition delay and hence retard the start of combustion. The
whole combustion process is therefore shifted towards the expansion stroke.
This results in the combustion products spending shorter periods at high
temperatures, which reduces the NO\textsubscript{x} formation rate [78].

The use of EGR is a widespread practice, primarily to limit NO\textsubscript{x} emissions, on
both diesel and spark ignition engines [55]. The composition of the exhaust gas
of a diesel varies with load. At idle, there is little CO\textsubscript{2} and H\textsubscript{2}O and the
composition does not differ much from that of air. At higher load the heat
capacity of the exhaust gas increases as the concentrations of CO\textsubscript{2} and H\textsubscript{2}O are
substantially higher [32]. Cooling the re-circulated exhaust gas enhances the
effects of using EGR. The use of EGR is associated with increased exhaust
smoke [79]. If the level of EGR is too high, soot, CO and HC emissions and
fuel consumption increase due to insufficient O\textsubscript{2} being available for good
combustion to occur [27]. The addition of EGR increases smoke emissions in a
number of ways. Firstly, there is less O\textsubscript{2} available during combustion to enable
complete burning and there is less O\textsubscript{2} available for soot oxidation [32]. And the
ignition delay, associated with addition of EGR, reduces in-cylinder pressures and temperatures, which reduces the soot oxidation rate [80].

2.3.7 Boost Pressure
Variable boost pressure is another technology used to gain more control over the combustion process in diesel engines. A variable boost system allows flexible control and optimisation of boost pressure for different speed and load conditions. In addition to power and efficiency goals, optimisation of boost pressure has also been shown to improve exhaust emissions [12]. The particulate emissions reductions seen by Tanin et al [45] were due to the dilution effect of increased boost pressure, which allowed more advanced injection timing for constant NOx levels. This more advanced timing gave more time and higher in-cylinder temperatures to oxidise soot late in the cycle. Turbocharging is beneficial to the reduction of both PM and HC emissions due to the shortened ignition delay period created by the increased in-cylinder pressures [64]. Furthermore, the engine is more efficient when turbocharged because the energy that would otherwise be wasted is recovered from the exhaust gas [21], which gives the additional benefit of improved fuel consumption. Increasing boost pressure increases NOx [66] with increasing AFR as a result of an increased concentration of O2 in the combustion chamber [81]. Furthermore, NOx emissions increase with the increased peak pressures associated with increased boost pressure [32].

2.4 Concluding Remarks
The aim in this Chapter has been to provide a background to the work described in the following chapters and to gain an understanding of the formation processes of the pollutants. These are essential to the understanding of the effects of fuel injection and engine operating parameter adjustments. It is clear that the trade-off between the exhaust emissions of different pollutants and fuel consumption is a complicated balancing act. Any process employed to reduce one problem by adjusting a particular parameter generally has a detrimental effect on another. Diesel engine design and technology is going through rapid development. Most advances are being driven by the need to
meet emissions regulations, but the perception of the consumer is also important. They do not seem to like their diesel cars sounding or driving like old taxis; diesel clatter is unattractive while increased power output is desirable. The literature on the parameter effects helped in the process of investigating the effects of the parameters considered here and with the planning and execution of the experimental work, especially with the selection of the parameter setting ranges for the DoE work. The literature available on the influence of split main injection on engine responses highlighted how complicated these effects are and emphasised how methods that can be used to simplify the adjustment of these parameters, and the calibration process in general, would be most valuable.
CHAPTER 3

SPECIFICATIONS OF HPCR SINGLE CYLINDER TEST ENGINE FACILITY, INSTRUMENTATION AND CONTROL CAPABILITIES

3.1 Single Cylinder Engine and Test Bed Facility

The test facility used during the experimental work was built from scratch and several people contributed to its development. The author's involvement and responsibilities were primarily in the areas of installation and commissioning. On the engine this included EGR, air intake, exhaust and cooling tower systems, the encoder and engine control panel, which involved control for the oil and water heaters and pumps and the emergency shut off for the engine. The installation of the dynamometer and its controller, along with the calibration of the load cell. The commissioning of the HPCR fuel injection system, including the hardware and the control software, whilst liaising with Bosch. The installation and initial calibration of the Signals exhaust emissions equipment and the TEOM (tapered element oscillating micro-balance) particulate measuring system. The installation and calibration of pressure transducers including the 150 bar in-cylinder transducer. The author also set up the initial Amplicon data acquisition system, which was later replaced with a more powerful dSPACE data acquisition system.

A schematic of the test facility is shown in Figure 3.1 and photographs of test cell and control room are shown in Figure 3.2 and Figure 3.3. While schematics of the facility instrumentation, the control set up and data acquisition system are shown in Figure 3.4, Figure 3.5 and Figure 3.6 respectively. This test facility has a great deal of independent control of the fuel injection and engine operating parameters and as such is complicated to start up, run and even shut down. As such a full start up and shut down procedure for the test facility hardware, instrumentation and software was written and is given in Appendix A – Operating Procedure for Test Facility.
3.1.1 Engine Specifications

The single cylinder engine was a Hydra, designed and manufactured by Ricardo Consulting Engineers, with a Ford 2.0 litre Puma DI combustion system. The cylinder bore of 86 mm and stroke of 86 mm gave a swept volume of 0.4996 litres. The engine had a compression ratio of 18.2 and a swirl ratio of 1.4. There was a central vertical injector and four valves per cylinder (two intake and two exhaust) with a double overhead camshaft (DOHC). The engine was mounted on to the short test bed by G. Cussons and was connected, via a flexibly mounted prop-shaft, to a David McClure swinging frame DC dynamometer rated 60 kW at 4500 rpm. For control purposes a Control Techniques Mentor II controller was installed, which allowed the engine to be held at required constant speeds.

Upper speed and load limits were observed to avoid putting the engine and test facility under unnecessary strain and risk of failure. The peak in-cylinder pressures were limited to a maximum of 165 bar as recommended by Ricardo for the Hydra design [82]. The engine speed was limited to 3800 rpm, which is less that the safe limit of 5500 rpm specified by Ricardo [82], due to the limits of the data acquisition rate of the dSPACE system.

There are certain advantages in using single cylinder engines for research and development purposes:

- The problems associated with rejecting the heat produced during combustion are greatly reduced as there is only one cylinder producing combustion heat that needs dissipating.
- There are no inter-cylinder variations. The manufacturing and assembly tolerances in multi-cylinder engines can cause performance differences between cylinders [57].
- There is no mixture variation. It is difficult to calibrate fuel pumps and injectors to give even fuel distribution for each cylinder [57].
- For a given cylinder size the fuel consumption will be less and a smaller capacity, and cheaper, dynamometer can be used [57].
3.1.2 High Pressure Pump, Common Rail and Piezo-Electric Injector

Initially, for commissioning tests, a York rotary distribution fuel pump and a standard Puma injector were fitted to the engine. A "bomb" was used to house the three remaining injectors from this standard set up and to collect and recirculated the unused fuel. For the main investigation, this system was replaced with an HPCR Bosch fuel pump with the capability of achieving rail pressures of 1350 bar. A low pressure fuel lift pump and standard Puma fuel filters were included in the fuelling system. A heat exchanger was included between the lift pump and the high pressure fuel pump to reduce the temperature of the fuel entering the high pressure pump and a plate cooler was included to reduce the temperature of the fuel returning from the high pressure pump and spilling from the common rail.

The existing standard Puma engine head was modified to accommodate the piezo-electric injector and give the same injector tip protrusion (3.25 mm) from the flame face as the standard injector for this Puma design. The injector nozzle was based on a standard common rail Puma injector [83]. This was a VCO injector with 6 holes and a spray cone angle of 154°. The body of the injector contained a needle lift sensor, which output a signal via an amplifier to give a 0 to 10 Volt DC output. This allowed the position of the needle to be logged by the data acquisition system.

The fuel pressure in the rail was regulated using a pressure regulator operating with a closed loop control through dSPACE. Feedback on rail pressure was provided by a 2000 bar pressure transducer mounted in the rail. Excess fuel from the rail, the injector and the high pressure pump is returned to the fuel meter via the plate cooler. The rail has the dual role of fuel accumulator and pressure pulsation damper, although it clearly does not fully dampen the pressure fluctuations seen. The working fuel pressure range can be adjusted by adding shims to the injector internally. Initially the upper pressure limit was set to 950 bar and the majority of the testing was carried out with rail pressures set, with a safety margin, at 850 bar.
The injection profile and the injection timing are defined by the injector control system. The control system produces signals, which are sent to the injector via an amplifier and define the start and finish of each injection event per engine cycle. The injection profile shape is defined using the ASCET software. The start of injection is defined in crank angle degrees using an Etas Trigger Control Unit (TCU). The TCU uses the TDC signal and the 900 pulses per revolution (ppr) signal from the shaft encoder, along with the signal from the in-cylinder pressure transducer, in order to monitor the engine cycle and make certain that it outputs a signal only around TDC on the compression stroke and not on the exhaust stroke. The TCU system required a signal of 1800 ppr, however the shaft encoder was only able to output 900 ppr; this meant that there was a 0.2° CA resolution for injection timing. The ASCET Box combines the signals from the TCU and the ASCET software and outputs the injector control voltage signal in the range 0 to 4 Volt. This signal is then amplified by the Booster to give an injector driver voltage of 0 to 200 Volt. The body of the injector contains a needle lift sensor, which provides a 0 to 10 Volt DC signal to give needle position. A target variation of needle position with time can be defined with the ASCET software. Inputs to this are the required needle lift step, in the range 0 to 200 µm, and the time period for each needle lift step, in the range 0 to 2000 µsec. A spreadsheet was developed to help design the injection profiles in the crank angle degree domain. This allows the user to input the required needle lift step, in the range 0 to 200 µm, along with the desired crank angle degree for each needle lift step. The engine speed can then be altered as required and the spreadsheet displays the value of each needle lift step and the associated time period. These values then need to be input in to the ASCET software on the laptop. There is a maximum permissible rate of needle lift for this injector of 1 µm/µsec and the spreadsheet checks whether the injection profiles fall within this limit for the selected engine speeds. A schematic of Bosch piezo-electric injector system is shown in Figure 3.7.

3.1.3 Shaft Encoder

Crank position was derived from a Hohner Automation incremental optical shaft encoder mounted to the crankshaft. Three output channels were used on
the encoder: 360 ppr used as a data logging trigger and in the calculation of engine speed, 900 ppr for the control of the piezo-electric injector and 1 ppr used to identify the TDC position.

3.1.4 Oil and Coolant Systems
In order to avoid cold starts and to help maintain the running temperatures of the single cylinder engine, the facility had two Eltron Chromalox oil heaters mounted in the engine sump and one Watlow Industries 3 kW domestic immersion water heater mounted within the coolant circuit. These had dedicated pumps and so could be activated without the need to have the engine running. Both the coolant and oil circuits contained thermostatically controlled valves, which allowed the fluids to be circulated only in the engine until pre-set temperatures of 90° C were attained. When the oil or coolant temperatures surpassed this limit the respective thermostats opened and the fluids were diverted through two separate heat exchangers, where heat generated by the engine was rejected to a constant water supply from a Carter M/3 series cooling tower. This system replaced the standard vehicle radiator. The engine coolant was a 50:50 mixture of water and ethylene glycol.

3.1.5 Boost Pressure System
An ABAC screw compressor with a 2000 litre receiver was installed. The compressor was rated to deliver a working pressure of 8 bar and a free air delivery of 3 m³/minute to a receiver. A by-pass valve was fitted to allow ambient air to be used to simulate naturally aspirated operating conditions. An 80 litre plenum chamber was included in the air intake arrangement in order to minimise any oscillations of the intake air caused by the unbalanced nature of the single cylinder engine. The level of boost pressure was controlled using a Norgren Series 11-808-980 Pilot Operated Regulator with a Norgren Pneu-Stat Electronic Pressure Regulator. Closed-loop control was carried out through dSPACE and allowed the pressure to be set to the required value from the control room. Test data from Ricardo [66] and others [12, 42, 41] along with data from the Puma engine being used for a separate research project at the University of Nottingham [84] indicated that air intake pressures up to 1.9 bar
would be required for the planned experimental work and it was therefore clear that this system is capable of delivering the required levels of boost pressure.

3.1.6 EGR System

The EGR hardware fitted to the engine was a standard Ford Puma system including an EGR cooler, which was fed by the cooling tower for this facility. A number of modifications were made to give the required independent control of the level of EGR delivered. The EGR valve was fitted with an electronically controlled stepper motor and a brass ball valve with ceramic packing was fitted in the exhaust system, also with independent electronic control, in order to simulate the back pressures typically seen with a turbocharger, in terms of the increase in the pumping losses, or pumping mean effective pressure (PMEP), and to ensure that higher EGR values were attainable when running with boosted air intake. Data from the Nottingham Puma engine [84] indicated that the differential between the air intake and the exhaust pressures should be kept within the region of 0.2 to 0.5 bar, depending on the operating condition, in order to simulate the turbocharger at work and that this would allow the required EGR rates to be realised.

3.2 TDC Error, Accuracy and Position

Lancaster et al [85] describe how the TDC output signal from the shaft encoder can be accurately set to match the TDC location on the engine using a simple experimental procedure. The method is summarised below and reportedly leads to a TDC error of less than 0.2° CA.

An endoscope was set up to view the crank angle markings on the flywheel so that the position could be accurately set and the injector was replaced with an extended dial gauge to measure the displacement of the piston. The dial gauge was zeroed at 50° CA BTDC and the flywheel was then rotated through TDC and past 50° CA ATDC. It was then rotated back to 50° CA ATDC and the displacement measured. This was repeated rotating back through TDC, past the 50° CA BTDC marker to then approach it in a clockwise direction. This measured the clockwise and anti-clockwise measurements for 50° CA ATDC
and 50° CA BTDC and was repeated at 10° CA intervals between these limits. This approach is used to prevent any play in the system, meaning the movement or rocking that may be present due to the offset of the piston pin, affecting the measurement of TDC location as care should be taken to eliminate these clearance effects [86]. An error of 0.5° CA BTDC was established, which indicated that there was an error with the TDC marker on the flywheel. The corrected TDC was marked on the flywheel and this was aligned with the rising edge of the TDC signal from the encoder.

3.3 Instrumentation and Software

3.3.1 Exhaust Gas Analysis

Exhaust gas analysis was undertaken using a number of different analysers. For the analysis of CO, CO₂, O₂, NOₓ and HC emissions, a Signal Instrument MaxSys 900 Raw Test Bed Emissions Analysis System, with a Pre-Filter Module 352 was used. Heated sample lines were included in order to prevent the condensation of the exhaust gas sample. These analysers show the relative amount of each constituent in the exhaust gases in parts per million (ppm) or percent by volume. A schematic showing the set up of the Signals emissions equipment is shown in Figure 3.8. Specifically this system consisted of a 4000VM NOₓ Analyser, a 3000HM THC Analyser, three 7000FM CO/CO₂ Analysers and an 8000M O₂ Analyser. Two 7000FM CO₂ Analysers were required for the measurement of both the exhaust and the intake manifold CO₂ values in order to calculate the percentage of EGR being used. The 4000VM NOₓ and the 3000HM THC Analysers had accuracy and repeatability better than ± 1% range or ± 0.2ppm and the 7000FM Analysers had accuracy and repeatability better than ± 1% range or ± 0.5ppm; in each case the greater value needs to be considered. The 8000M O₂ Analyser had a repeatability of ± 0.01% O₂ and a zero stability of ± 0.002% O₂ per hour.

An AVL 415S Variable Sampling Smoke Meter was used for the measurement of the exhaust smoke emissions. The smoke meter uses a filter paper system to measure the carbon content, or soot, in the exhaust gas and outputs values in
terms of filter smoke number, FSN. Over the range of filter smoke number readings between 0.5 and 6.0 FSN the meter has repeatability of less than or equal to 0.05 FSN [59]. The smoke meter was operated through an AVL 4210 Instrument Controller and an RS232 Protocol interface with the dSPACE data acquisition system; this also enabled data logging of the output.

The measurement of exhaust particulate matter was undertaken using a Horiba Instruments MDT-905 TEOM 1105 Particulate Mass Monitor System. The TEOM system draws a set ratio of exhaust gases and ambient air through its exchangeable filter cartridge at a constant flow rate and continuously monitors the vibrations of the filter in order to calculate real time values for mass flow rate (g/sec), mass concentration (mg/m³) and total accumulation of mass (m). It provides output signals for the parameters being measured in the form of 0 to 5 Volt DC. The TEOM resolution or measurement step for mass flow rate is 2.5 x 10⁻⁷ g/sec and that for mass concentration is 2.5 mg/m³.

3.3.2 Fuel Consumption
Fuel consumption was measured and monitored using an AVL 733S Dynamic Fuel Meter, which works on the gravimetric measuring principle and has a measuring accuracy of 0.1%. The fuel meter was operated with an AVL 4210 Instrument Controller and an RS232 Protocol interface with the dSPACE data acquisition system; this also enabled data logging of the output.

3.3.3 Combustion Noise
Combustion noise was measured using the AVL 450 Combustion Noise Meter, which processes and filters the in-cylinder pressure signals to provide an output calibrated in decibels, with an error of ± 1dB. The signal processing executed by this instrument was determined empirically to represent the structure attenuation of an average engine. Thus the noise levels derived should not be considered as absolute measurements of combustion noise, but rather a relative figure to enable the comparison of combustion system variants.
3.3.4 Pressure Measurements

In-cylinder pressure measurements were taken using a Kistler 6123 piezo-electric transducer, calibrated to read a maximum pressure of 150 bar, which was mounted in the glow plug hole in the engine head. A Kistler 5011 charge amplifier was used with the transducer to give outputs of 0 to 10 Volts for the calibrated pressure range of 0 to 150 bar. Since the piezo-electric pressure measurement does not supply absolute values, it is AC coupled, the signal must be referenced to the intake manifold pressure in order to eliminate the signal drift. A mean in-cylinder pressure value was obtained by averaging over a range of 100° CA around BDC at the end of the air intake stroke. At this point it is safe to assume that the in-cylinder pressure is the same as the intake manifold pressure [86]. As the manifold pressure was measured in absolute terms it was used to calculate the offset for the in-cylinder pressure around BDC. This offset was then applied, during post processing of the data, across the whole cycle to give the absolute normalised in-cylinder pressure values.

Air intake pressure measurements were made using a Kulite Sensors pressure transducer, with a range of 0 to 4 bar, which was mounted in the air intake manifold. A Kulite Sensors pressure transducer, with a range of 0 to 2 bar, was mounted inside the test cell and was used to take atmospheric readings as a reference for the in-cylinder pressure readings. An amplifier was built in-house for these transducers and the arrangement was calibrated within the data acquisition system to give the required output pressure ranges.

Low pressure fuel pressure was monitored using a Kulite Sensors pressure transducer, with a range of 0 to 6 bar, mounted before the high pressure pump in order to make certain that the low pressure fuel supply was maintained to the high pressure pump. A second Kulite Sensors pressure transducer, with a range of 0 to 7 bar, was mounted in the oil circuit and was used to monitor oil pressure. An amplifier was built in-house for these transducers and the arrangement was calibrated within the dSPACE system to give the required output pressure ranges.
The high pressure fuel in the common rail and the fuel line pressure, near the injector, were measured using two Leafield Engineering 2000 bar pressure transducers. These were used along with a Vishay Measurements Group 2200 System Signal Conditioning Amplifier, which gave 0 to 10 Volt DC outputs with an accuracy of ± 0.5% [87].

3.3.5 Temperature Measurements
All temperature readings, including exhaust, air intake, water, oil and the fuel circuit, were measured using TC Limited K type Chromel-Alumel thermocouple probes. The micro-voltage outputs from these thermocouples were processed in the data acquisition software to give values in degrees centigrade for data logging and engine monitoring purposes. Three thermocouples were included in the fuel system. This was required between the intercooler and the input to the high pressure pump, between the return from the high pressure pump and the plate cooler and after the plate cooler to make certain that the fuel returning to the fuel meter was sufficiently reduced in temperature.

3.3.6 Shaft Encoder
Output signals from the Hohner Automation shaft encoder were processed to provide engine speed data using an electronic circuit, which was built in-house, to convert the output of 360 ppr into a DC voltage and calibration was performed using a signal generator and frequency meter to simulate the output from the shaft encoder. The TDC marker also generated a DC voltage and both voltages were logged by the data acquisition system. All three of the encoder output signals were passed through an in-house manufactured opto-isolator to reduce signal noise.

3.3.7 Engine Torque
The engine torque measurement was taken from the load cell mounted on the dynamometer and amplified by a Nobel Elektronik AST 3P analogue signal transmitter. The amplifier incorporated an analogue output filter with variable bandwidths, which was adjusted in order to reduce the unwanted naturally oscillating torque profile produced by the single cylinder engine.
3.3.8 Mass Air Flow
A standard Puma Ford mass airflow (MAF) sensor, part number 97BP-12B579-AA, was used to measure the mass airflow of the air intake, whether naturally aspirated or boosted.

3.3.9 Calibration of Sensors and Instrumentation
The following outlines the calibration procedures for the various sensors and instruments, which were performed through the data acquisition system. To achieve this, readings were taken by the data acquisition as the input signals were swept through their operating ranges, in order to calculate the gain and offset values for each individual signal.

All the thermocouples were calibrated using an ice bath to generate 0°C and a heated oil bath, which has an accuracy of within 1°C, to generate temperatures up 300°C. Calibration data were recorded at intervals of 50°C.

The Kistler in-cylinder pressure transducer was calibrated by applying a known pressure to the sensor and measuring the voltage from the charge amplifier. The dead weight system used had accuracy of 0.1%. These figures were then used to calculate the gain and offset values for the transducer. As the Kistler pressure transducer only generates a signal with a change in pressure, the cylinder pressure is referenced to atmospheric pressure during the exhaust stroke, with the resulting offset being applied automatically in the data acquisition software. All other pressure transducers were calibrated using a Druck model DP-601 portable pressure measurement instrument, capable of applying variable pressure to each sensor, with an accuracy of 0.1%. The pressure measurements are taken through the data acquisition system and the appropriate gain and offset values for each transducer calculated.

All emissions measurement equipment is calibrated daily prior to any experimental work according to the procedures detailed in the Appendix A – Operating Procedure for Test Facility.
3.4 Data Acquisition System

3.4.1 Initial Set Up – Amplicon System

Preliminary commissioning and testing of the facility was carried out using an Amplicon 200 Series High Performance Data Acquisition system. This was later replaced with a dSPACE system described below, but a brief record of the initial system is given here. The system briefly consisted of an Amplicon EX205 Signal Termination Panel, an Amplicon EX201 Analogue Input Extension Board and an Amplicon PC226 Data Acquisition Board, which was run from a host computer. The PC226 provided 16 analogue input channels with analogue to digital conversion, a counter or timer and the PC input/output interface [88], the EX205 board used with the PC226 to provide conditioning of the 16 analogue input channels, 4 analogue output channels, a cold junction reference temperatures for the thermocouples and a voltage reference [89], while the EX201 provided 32 extra analogue input channels [90]. The engine monitoring and control systems along with the data logging and the data processing software were written in the Turbo C++ and Pascal programming languages. These were initially conceived by Burrows [91] and modified by the author for this specific application.

3.4.2 dSPACE System

The Amplicon data acquisition system was replaced by a more powerful dSPACE system, which enabled more sophisticated data acquisition to be executed, with the capability of capturing a greater number of channels of real time data at the greater resolution of half crank angle degrees. This set up allowed for real time monitoring of engine data and greater control of the test facility to be realised.

The following dSPACE hardware was used: a DS1005 Processor Board with a 466 MHz processor used for all input/output and modelling requirements. Two DS2003 Multi A/D Boards, high-resolution multi-channel analogue/digital boards which were hardware triggered to collect 10 cycles of data sampled at \( \frac{1}{2} \degree \text{CA} \) resolution. The input voltage range for the channels is either ± 5 Volt or ± 10 Volt and they are used to collect temperatures, pressures, emissions,
needle lift signal, injector control voltage, TDC marker, rail and injection pressure data and in-cylinder pressure. A DS2201 Multi I/O Board with eight analogue ± 10 Volt output channels. This was used for actuator control of the EGR and exhaust back pressure valves and boost pressure control valve using closed loop control, with the desired value input by the operator through ControlDesk. A DS4002 Timing I/O Board, which detects the rising and falling, edges in the input signal. Time and polarity are stored for each edge and one channel is used to accurately calculate engine speed.

The dSPACE system was run from a PC. The required models were designed and edited using MATLAB/Simulink software and the facility control or interface was realised with Control Desk software. All data storage was undertaken using the same PC.

### 3.5 Exhaust Gas Analysis

Exhaust gas consists of water (H₂O), nitrogen (N₂), carbon monoxide (CO), CO₂, oxygen (O₂) and hydrogen (H₂) along with NOₓ, HC and soot emissions or solid carbon (C). The Signal Instrument Group MaxSys 900 Raw Test Bed Emissions Analysis System was used to determine the relative amount of each component as either a percentage, for CO, CO₂ and O₂, or as ppm, for NOₓ and HC emissions, on a volume basis. Each analyser outputs a 0 to 10 Volt signal proportional to the concentration of each species and calibration grade span gases were used to calibrate each analyser at the beginning of testing each day. The flame ionisation detector (FID), used for measuring unburned HC emissions, was spanned with propane (C₃H₈) and used a fuel mixture of 40% hydrogen and 60% helium for the flame. The CO₂ and CO concentrations were measured using non-dispersive infra-red technique and spanned with 10% CO₂ and 1% CO. NOₓ emissions were measured by a chemiluminescent light detector and spanned with 5000ppm nitric oxide (NO) and O₂ was measured using the paramagnetic method, the analyser was calibrated with zero grade air (20.9% O₂ by volume). All analysers were zeroed with N₂ [92].
The mass flow rate of NO\textsubscript{x} and HC emissions can then be found by post processing the data from the analysers and by considering the total mass flow rate of the exhaust; the steps for this are laid out in Appendix B. The CO, CO\textsubscript{2} and O\textsubscript{2} analysers give a dry analysis of the exhaust gas sample as it is passed though a cooler drier in the before entering the relevant analysers. This cools the samples to 5° C and means that much of the water present in the exhaust gas sample is removed before analysis is undertaken, which has a small but significant effect on these readings. The details of the steps undertaken to correct for this effect and give the mass flow rates of each species are presented in Appendix B – Exhaust Gas Analysis.

3.5.1 Percentage EGR Calculation

The definition of EGR rate used in this work is the ratio of cycle averaged mass flow rate of exhaust gas present in the manifold intake mixture and the mass flow rate of the total induced mixture in the manifold. As a percentage, this is:

\[
\text{EGR} \,(\%) = \left( \frac{m_{\text{EGR}}}{m_{\text{MAN}}} \right) \times 100 \quad (3.1)
\]

Where \( m_{\text{EGR}} \) is the mass flow rate of the EGR and \( m_{\text{MAN}} \) is the mass flow rate of the gases in the air intake manifold. The EGR rate can be determined from measurements of CO\textsubscript{2} concentrations, taken at the air intake manifold and the exhaust. This relationship is given in equation (3.2) and was derived in Appendix B with equation (3.1) as the starting point. Concentrations of gaseous emissions in the exhaust are measured in percent by volume by the CO\textsubscript{2} analysers used here [92], which corresponds to the mole fraction multiplied by 100 [32]. Therefore the EGR rates reported in this thesis are the volumetric percentage of intake charge that is exhaust products and can be written as [12]:

\[
\text{EGR} \,(\%) = \frac{\% \text{CO}_2\text{MAN} - \% \text{CO}_2\text{AIR}}{\% \text{CO}_2\text{EGR} - \% \text{CO}_2\text{AIR}} \times 100 \quad (3.2)
\]
Here $\% \text{CO}_2_{\text{MAN}}$ and $\% \text{CO}_2_{\text{EGR}}$ are the percentage readings from the Signal analysers of CO$_2$ in the intake manifold and the EGR (actually measured in the exhaust) respectively. The percentage of CO$_2$ in ambient air, $\% \text{CO}_2_{\text{AIR}}$, is taken from Rogers and Mayhew [93] to be approximately 0.03%.

3.5.2 Filter Smoke Number to Soot Concentration Conversion

The empirical correlation between filter smoke number (FSN) and soot concentration published by AVL [94, 95] is determined at AVL reference conditions of $10^5$ Pa and 25° C, which are consistent with ISO 10054. The conversion from FSN is therefore undertaken as follows:

$$\text{Soot} \ [\text{g/m}^3] = \frac{K}{10^3} \times 4.95 \times \frac{\text{FSN}}{0.405} \times e^{(0.38\times\text{FSN})}$$

(3.3)

where for FSN $\leq$ 8, $K=1$

and for FSN $> 8$, $K=1 + \left(\frac{\text{FSN} - 8}{2}\right)^{10}$

Figure 3.9 shows this relationship over a range of FSN values.

3.5.3 Particulate Matter, Soot and HC Emissions Relationship

Particulates are one of the exhaust gas species that must not exceed specified levels in terms of emissions legislation over the drive cycle. They are defined, in most countries, as any matter collected on a filter paper at a temperature of 325 K [15, 60, 61] through which diluted exhaust gases have been drawn; the bulk of this matter is either unburned hydrocarbons or soot. To measure particulate emissions in a way which is consistent with legislated test procedures typically requires a full dilution tunnel, or Constant Volume Sampling system, to be installed in a dedicated test cell [61]. These very expensive systems are designed to simulate the processes by which exhaust gases are transformed through their interaction with the ambient environment. In contrast smoke meters are relatively inexpensive, easy to operate and install, and because smoke and particulates are closely related it is common to use the
measurement of one as an indicator of the other, especially with regard to trends [61]. The soot produced during combustion is visible as smoke and measured as FSN values and any measures aimed at reducing FSN will also reduce particulates and as such in the work undertaken in this thesis FSN values have been used throughout.

Since the particulate matter consists mainly of soot and unburned hydrocarbons, it is possible to generate a correlation between soot, HC and PM concentrations. For small high speed, turbocharged diesel engines the following relationship was developed by Greeves and Wang [57, 60], values are concentrations in mg/m³:

\[
\text{PM} = 1.024 \times \text{soot} + 0.505 \times \text{HC}
\]  \hspace{1cm} (3.4)

This relationship comparing particulate data from the TEOM, soot data from the AVL correlation in equation (3.3) and HC experimental data, all taken from the single cylinder test facility, compares well to the graphical data presented by Greeves and Wang [60] and is shown in Figure 3.10. Also included in this figure is the data plotted on a log scale to highlight how the relationship works near the origin. The constant for soot in equation (3.4) is close to unity, which suggests that the soot concentration value derived from the smoke meter reading corresponds directly to the soot collected on the TEOM particulate filter paper. Whereas the constant for HC suggest that only 50% of the HC, which is measured by the HC analyser in the exhaust sample, condenses to liquid on the TEOM particulate filter paper and is thereby measured. The correlation shows that the origins of particulate emissions are a combination of black smoke, or soot, and HC emissions [60]. The importance here is the trends seen and it is clear that a change in FSN values will directly reflect a change in soot concentration, and when combined with HC data, a change in particulate emissions. Thus any measure that reduces FSN will be successful in reducing PM emissions over the drive cycle. With this in mind FSN has been used throughout this thesis to show what trends or improvements can be made with different parameter settings at different points on the operating map.
3.6 Concluding Remarks

The single cylinder test facility set up that was used in the research undertaken in this thesis has been outlined in this chapter and the level of work that was involved in developing such a sophisticated test facility has been highlighted. The general philosophy behind the test facility development was to have a high level of accuracy with sufficient robustness to allow day to day test repeatability. To ensure this repeatability, a standard test point was run at the start and end of each period of testing to ensure stability of readings, function of hardware and consistency of the instrument calibrations.

One particular point of importance concerns the use FSN in this thesis. According to emissions legislation, PM in g/km across the NEDC drive cycle should be investigated when considering whether a particular engine set up is acceptable in terms of particulates in the exhaust gases. However, it was shown in Section 3.5.2 and Section 3.5.3, and in Figure 3.9 and Figure 3.10, that FSN trends can be used to give a good indication of both soot and PM trends at various points on the operating map. Therefore, any measure that reduces either the exhaust smoke or HC emissions will also reduce PM levels [57].
CHAPTER 4

FACTORS AFFECTING MASS OF FUEL INJECTED AND MEASURES TO SUPRESS PRESSURE WAVE OSCILLATIONS

4.1 Introduction

This Chapter is concerned with the HPCR fuel injection system and the control of fuel injection into the combustion chamber. A relationship between needle lift trace, injection pressure and the quantity of injected fuel has been established. Pressure oscillations at the inlet to the injector due to the opening and closing of the injectors are an inherent problem in common rail systems [2, 3, 34, 35, 36, 37, 38, 39]. The pressure variations created by an initial injection can still be present when the injector is opened for a subsequent injection of a multiple injection strategy. This produces an unquantified effect on the quantity of fuel injected [3], spray atomisation, penetration and mixing of the injected fuel. These pressure variations also affect the function of the injector and modify the levels of needle lift.

Methods that reduce or eliminate these pressure waves are of particular interest here in connection with quantifying the mass of fuel delivered in each part of a split main injection. The experimental work undertaken to examine the problem and reported in this Chapter has concentrated on one operating point, engine speed 1600 rpm and load 6.76 bar BMEP, with a split main injection profile consisting of two main injections and no pilot injection, unless otherwise stated. The implications of the work for a wider range of strategies and operating conditions are discussed later. The split main ratio of 80:20 was established at a split main separation of 17° CA, while the main start of injection was held at 4.2° CA BTDC. The engine was naturally aspirated and 0% EGR was used. The definitions of the different parts of the split main injection profiles used here are illustrated in Figure 4.1. As shown, “1st_injection” refers to the quantity of fuel in the first part of the split main injection with the second injection event present. Similarly, “2nd_injection” is the quantity of fuel in the second part of the split main injection with the first
injection event present. While "1st_only" and "2nd_only" are cases where the other part of the split main profile has been removed and the remaining part of the split main injection has the original timing and duration in the crank angle domain. Each of the three profiles shown in Figure 4.1 was run separately to quantify the fuel delivered in each injection event and hence assess the influence of coupling between the different parts of the split main injection.

4.2 Needle Lift Profile, Injection Pressure and Fuel Delivery Relationship

Common rail systems are known as pressure/time systems and the injection rate varies according to the fuel pressure at the injector and the cross-sectional area of the nozzle that is exposed during the injection event [3]. The duration of fuel injection, the pressure drop across the injector and the level of needle lift each influence the quantity of fuel injected. These may not completely define the quantity of fuel injected and predicting fuel delivery characteristics or injection control is an area that many have investigated [2, 3, 39, 96]. Commonly, however, estimates are based upon the assumptions that flow through the nozzle is quasi steady, incompressible and one dimensional. The mass of fuel injected through the nozzle, \( m_{\text{FUEL}} \), is then given by [32]:

\[
m_{\text{FUEL}} = C_D A_n \sqrt{2\rho_{\text{FUEL}} \Delta p}
\]  

(4.1)

In equation (4.1), \( \Delta p \) is the pressure drop from the injector inlet pressure to the in-cylinder pressure and the density of the fuel is given by \( \rho_{\text{FUEL}} \). These were known for the test results in the current study, but values for the discharge coefficient, \( C_D \), and the nozzle minimum area, \( A_n \), were unknowns. However, assuming the product of these is proportional to the needle position suggests the fuel mass delivered during an injection will obey the relationship shown in equation (4.2):

\[
m_{\text{FUEL}} \propto \int \left( N L \sqrt{2\rho_{\text{fuel}} \Delta p} \right) dt
\]  

(4.2)
Where NL is the needle lift from its closed position and is a recorded parameter using the needle lift sensor in the injector body. All the experimental work carried out for this study was at an engine speed of 1600 rpm. As such it is possible to use the fuel consumption data from the fuel meter, in kg/hr, directly in relationship shown in equation (4.2).

Experimental tests were undertaken to determine the constant of proportionality for the relationship shown in equation (4.2). Data from a typical split main separation test run is shown in Figure 4.2, which is a graph of the relationship given in equation (4.2); fuel consumption against the product of area under the needle lift trace, injection pressure differential and fuel density. It can be seen in Figure 4.2 that there is an offset from the origin on the x-axis for the best-fit line for the data of the smaller injection events (2nd_injection), which indicates that needle lift is occurring but there is no fuel delivery. This lower needle lift limit was calculated by considering this offset and the duration of the injections used and was found to be 18.4 µm.

The data was corrected for this lower needle lift limit and was then re-plotted, as shown in Figure 4.3. This adjustment led to the best-fit line for the smaller injection events (2nd_injection) passing through the origin, but the expected linear relationship was still not apparent for the larger injection events (1st_injection) and again an offset is evident. This indicates that above a certain value of needle lift there is no extra fuel injected and by considering the offset in Figure 4.3 and the duration of the injections used, the upper limit on the needle lift was found to be 78 µm. For confirmation of these findings, a wide variety of other fuel injection data were processed. These data covered various split main separations and engine loads, but were taken at the same engine speed of 1600 rpm. The data were corrected for the upper and lower needle lift limits and plotted in Figure 4.4, along with the original data shown in previous figures. It is clear from this graph that there is a good correlation between the product of area under the needle lift trace and injection pressure differential and the quantity of fuel delivered, when taking into account the upper and lower needle lift limits, which confirms the relationship shown in equation (4.2).
Egnell [97] states that in the area between the needle and seat, the annulus area, varies with needle lift. It has been shown with this work that for recorded needle lift values below 18.4 µm there is no fuel delivery implying that the annulus area is zero. It cannot be determined here whether this 18.4 µm is a true needle lift or whether it is a calibration error in the needle lift sensor or a distortion of the needle itself as the piezo-electric stack is activated, but the needle lift sensor has been used throughout the work here and its output values have been taken as true needle lift values. At recorded needle lift values between 18.4 µm and 78 µm the fuel delivery is determined by the annulus area, which directly varies with changing needle lift. Above the 78 µm upper needle lift limit, the fuel delivery is determined by nozzle holes area and therefore any increase in needle lift has no effect on the nozzle minimum area and hence no increase in quantity of fuel delivered is seen. Figure 4.5 shows the upper and lower needle lift limits overlaid on a typical 80:20 split main injection profile which highlights the region where the quantity of fuel injected is proportional to the area under the needle lift profile.

4.3 Pressure Oscillations in FIE System and Damping

The relationship established in Section 4.2 allows the quantity of fuel delivered to be found from the needle lift trace and the injection and in-cylinder pressures, however, the problems of oscillations in the high pressure fuel system are still present. Investigations were carried out to examine these injection pressure variations. The frequencies of the pressure wave oscillations and their source were established. The possibility of damping or reducing the pressure oscillations, by modifying the injection profiles, was investigated. Furthermore, the effects of using such damping techniques were investigated to highlight the improvement that could be made in the variability of the fuel delivered per injection event.

Reducing or eliminating injection pressure oscillations to enable the positioning of multiple injection profiles without regard for the injection pressure oscillations is most desirable. Pressure wave dynamics in the fuel
injection system will present problems in terms of unpredictable values of injection pressure being present when the injector opens for subsequent injection events. From equation (4.1) it is clear that these pressure variations have the effect of modifying the amount of fuel delivered. This will have an effect on the engine load and exhaust emissions due to changing levels of atomisation, penetration and mixing of the injected fuel [27].

These pressure wave fluctuations are created in the high pressure fuel pipes, connecting the injector to the common rail, as the injector is opened in order to deliver fuel. The waves decay as a function of time, but remain sufficiently large in amplitude in the region of typical subsequent injections to affect the rate and quantity of fuel delivered in such injections. Figure 4.6 shows the needle lift trace, the injection pressure and rail pressure fluctuations, created by the normal operation of the injector, for a typical single injection event at an engine speed of 1600 rpm and a load of 5.25 bar BMEP. It is clear from this figure that the pressure wave fluctuations in the common rail have much lower amplitude than those that occur in the high pressure fuel lines connecting the rail and the injector, which is the injection pressure. The figure also illustrates that the fluctuation initially attains a maximum pressure of 980 bar and then reduces to a minimum value of 700 bar, which is an amplitude of 280 bar. Injection pressure amplitudes in the region of 250 bar are commonplace in common rail fuel injection systems [2, 3, 35].

4.3.1 The Frequency of the Oscillations in the High Pressure Fuel

Fourier Transforms were applied to the injection pressure data in order to highlight the individual frequencies and therefore identify the sources of the oscillations in the high pressure fuel. For each test run undertaken here, ten cycles of injection pressure data were captured and processed by taking Fourier Transforms to examine the frequency content of the pressure signal. Initially a motored test, at 1600 rpm, was carried out and the data processed to highlight those oscillations that were not caused by the operation of the injector. Other factors that caused disturbances in the fuel injection system were the high pressure fuel pump and the rail pressure regulator. The frequency spectrum for the disturbances caused by the high pressure fuel pump and by the rail pressure
regulator needed to be highlighted by running a motored test and are shown in Figure 4.7. There is good agreement between these experimental values and the calculated fundamental frequency and harmonics, which are summarised in Table 4.1. The 80 Hz fundamental frequency is from the high pressure pump, which has three pistons operating per engine revolution, while the 45 Hz fundamental frequency is the operating frequency of the rail pressure regulator.

A number of fired tests were undertaken with single injection profiles so that the injection pressure oscillations caused by the opening of the injector could be assessed. The frequency spectrum produced from the MATLAB Fourier Transform m-file for a typical single injection event is shown in Figure 4.8, this is at 1600 rpm and 4.39 bar BMEP. The dominant frequency of approximately 850 Hz has been highlighted in this figure and there is good agreement between the experimental values and the calculated fundamental frequencies and harmonics, which are shown in Table 4.2. This table also shows the equation definitions and the pipe lengths used in these calculations.

Two main types of standing wave are seen in the fuel injection system, which can be related to simple boundary conditions [34]. The first is due to a closed-open boundary system: in this case between the injector, along the high pressure fuel line and to the open end of the fuel line with its junction with the common rail. This has a total length of 0.4225 m and produces the 850 Hz fundamental frequency that is seen as the dominant frequency in the injection pressure wave variations. The fundamental frequency of a closed-open boundary system is calculated from the length of the pipe, $l$, and the speed of the pressure wave, $c$, and is given as [98, 99]:

$$f_{\text{CLOSED-OPEN}} = \frac{c}{4 \times l}$$  \hspace{1cm} (4.3)

The second type of standing wave is caused by a closed-closed boundary system: in this case between the injector, along the high pressure fuel line and to far wall of the common rail across its diameter. This has a total length of 0.4315 m and produces a fundamental frequency of 1669 Hz. The fundamental
frequency of a closed-closed boundary system is calculated from the length of the pipe, \( l \), and the speed of the pressure wave, \( c \), and is given as [98, 99]:

\[
f_{\text{CLOSED-CLOSED}} = \frac{c}{2 \times l}
\] (4.4)

The critical dimensions of the fuel injection system considered for these calculations are shown schematically in Figure 4.9. The speed of the pressure wave in the fuel was estimated from [99, 100, 101]:

\[
c = \sqrt{\frac{B_{\text{FUEL}}}{\rho_{\text{FUEL}}}}
\] (4.5)

Where the bulk modulus, \( B_{\text{FUEL}} \), is \( 1.7 \times 10^9 \) N/m\(^2\) [102, 103] and the diesel fuel density, \( \rho_{\text{FUEL}} \), is \( 850 \) kg/m\(^3\) [32, 104]. This gives a value of 1440 m/sec, which is in the range 1300 to 1500 m/sec given by other sources [100, 105].

Figure 4.10 and Figure 4.11 show the pressure waves produced by the action of the injection at five different speed and load points across the operating map. These graphs were produced from work undertaken in Chapter 5 and clearly show that the pressure oscillations in the fuel injection system cause similar problems across the operating map, even if they cannot be said to be fully independent of speed and load. There are some clear changes as engine speed and load are varied, and this is expanded upon below. The important point is that at each point on the operating map considered the disturbances are still present in the region where subsequent injection events are typically placed, up to 40\(^o\) CA after the end of the 1\(^{st}\) injection event, as shown in Figure 4.10 and Figure 4.11. The data shown in these two figures had a pilot injection present and this caused small oscillations in the injection pressure at the start of the main injection. These relatively small pressure waves did not have an adverse effect on this data and it is clear that there is a great amount of consistency in the pressure waves created by the main injection events shown in these figures at the different points on the operating map.
However, at the higher engine speed of 3400 rpm, shown in Figure 4.11, it can be seen that the pressure waves have a larger period on a crank angle basis compared to the lower engine speed of 1600 rpm, shown in Figure 4.10. In both instances the same base frequency of 850 Hz is present but at the higher engine speed the oscillation occurs over a longer period on a crank angle basis. The engine speed was increased from 1600 rpm to 3400 rpm, by a factor of 2.125, and this resulted in the peak to peak separation increasing by the same factor from approximately 11° CA to 24° CA.

Furthermore, increased engine load did not directly increase the initial amplitudes seen, although in these graphs these amplitudes range from approximately 165 bar to 250 bar. Relatively small injection events, like pilot injections, do result in lower initial amplitudes. However, once the injection has reached the size typically seen for main injections, the amplitudes produced are similar and large enough to cause variations in levels of fuel injected for subsequent injection events. The higher load points at the two engine speeds, shown in Figure 4.10 and Figure 4.11, both show that there is some reduction in the size of the initial amplitudes. This is due to interference of the pressure waves produced by the operation of the injector as the injections become longer and the injector is therefore open for longer. This interference helps to reduce the amplitude of the oscillations and is an idea that has been explored in the following section to find ways of reducing or eliminating these problematic pressure waves in the fuel injection system.

4.3.2 Methods of Damping
Previous investigations [36, 37, 39, 96, 106] undertaken to reduce the effects of pressure wave fluctuations in common rail fuel injection systems have concentrated on two main methods. The use of mechanical means, such as orifices and accumulators to reduce the amplitude of pressure variations, and the adjustment in the duration of an injection event, to correct for the varying injected fuel quantities caused by these varying pressures.

In order to undertake the investigations into the dampening of the injection pressure waves carried out here, only the first part of the previously defined
80:20 split main injection was considered. It was shown experimentally that this single injection resulted in an engine load of approximately 5.25 bar BMEP at these conditions. Damping optimisation work could then be executed on the single injection, maintaining the load at the required 5.25 bar BMEP. Once the damping of the pressure waves had been optimised it would then be possible to reintroduce the second part of the injection profile to complete the profile so that the effects of the damping could be fully investigated at the required speed and load point.

Here, the approach was to alter the initial injection profile to suppress or prevent the large amplitude oscillations. Three different methods were considered: increasing the duration of injection events while using a lower needle lift, the rate shaping of the falling edge of the injection profile and the addition of a second sub-injection close to the initial injection. In each instance, the aim was to produce a second pressure wave in the high pressure fuel pipes to interfere with the pressure wave resulting from the initial lifting of the needle from its seat. In each case the initial injection profile was modified in order to maintain the required engine load. Long injection durations, with a correspondingly lowered needle lift, did yield some improvement in the amplitude of the fluctuations. However, at lighter engine load conditions it was not possible to produce injection profiles with long enough duration to give sufficiently good reduction in the pressure waves. The rate shaping method also showed some improvements in the reduction of the fluctuations. These improvements were similar in magnitude to those found using longer duration injections. In many cases the result of rate shaping the falling edge of the injection profile merely served to extend the duration of the injection event.

The third method, the addition of a second sub-injection close to the initial injection, yielded significantly reduced injection pressure fluctuations. Different separations between the initial injection and the second sub-injection were investigated, as were different fuelling ratios between these two injections. The optimised injection profile, shown in Figure 4.12, demonstrates the significant reduction in the pressure fluctuations that are possible. Amplitudes of approximately 35 bar were demonstrated with this hydraulic
damping method, which is a great improvement on the typical values in the region of 250 bar. A clear illustration of the benefits of injection profiles with hydraulic damping over those without damping can be seen in Figure 4.13, which compares damped and undamped injection pressure patterns on the same graph. The dominant frequency of oscillation of the injection pressure wave was previously shown to be approximately 850 Hz, which at an engine speed of 1600 rpm was calculated to be 11° CA in the cycle, this peak to peak value is shown in Figure 4.6. Placing the second sub-injection, with a similar needle lift and fuel delivery to the initial injection, at a point half way through this 11° CA oscillation creates a similar fuel injection demand at the point which coincides with the first rising edge of the injection pressure wave. This creates another pressure wave in the system, which is out of phase with the initial pressure wave and which therefore creates a superposition interference resulting in the suppression of the fluctuations. The 5.5° CA separation between the two needle lift events, which results in the interference and hence hydraulic damping, is highlighted in Figure 4.14. This figure also shows the difference between the needle lift traces with and without hydraulic damping and that the end of injection with damping is extended by around 3° CA.

The work undertaken here was only carried out at one point on the operating map, but the strategy would be effective at all operating conditions, but some adjustments for speed would need to be made. As shown above, at the increased speed of 3400 rpm the peak to peak separation of the oscillation was calculated to be 24° CA in the cycle. Therefore the second sub-injection would need to be placed at a separation of 12° CA from the initial injection at a point which is half way through the 24° CA oscillation, in order to interfere with the first rising edge of the pressure wave. With engine load variations the size of the second sub-injection would need to be adjusted in order to be of similar size to the initial injection, which itself must be adjusted with load variations. As was shown above, a similar needle lift and fuel delivery to the initial injection is required to create another pressure wave in the system, which interferes with the initial pressure wave.
4.4 Injected Fuel Quantity Variation in Second Part of Split Main Injection

An investigation was carried out to determine the percentage variation of fuel consumption for the second split main injection with variation of the split main separation. For this purpose a split main injection ratio of 80:20 was set up at a split main separation of 17° CA and engine speed of 1600 rpm, as shown at the top of Figure 4.1, and this set up was tested with and without hydraulic damping.

This percentage variation of fuel consumption for the second split main injection was found in the following way. Initially, the injection profile was run using both the first and second parts of the split main injection profile, 1st_injection and 2nd_injection, to give the total fuel consumption. The second injection event was then removed and this profile, 1st_only, was run and the fuel consumption was noted. To attain the value of fuel consumption for the second split main injection with the first split main injection present, 2nd_injection, the value of fuel consumption for the first split main only was subtracted from the fuel consumption for whole profile as shown in the following relationship:

\[
FC (2nd\text{\_}injection) = FC (\text{total}) - FC (1st\text{\_}only) \tag{4.6}
\]

The first injection event was then removed from the initial whole profile and this profile, 2nd_only, was run and the fuel consumption was noted. Comparing the fuel consumption for the second split main injection in isolation, 2nd_only, with the fuel consumption for the second split main with the first split main present, 2nd_injection, demonstrates the effect that the first injection event has on the second injection event. The percentage variation of fuel consumption for the second split main injection can be defined as follows:

\[
\% \text{ FC} = \frac{FC (2nd\text{\_}injection) - FC (2nd\text{\_}only)}{FC (2nd\text{\_}injection)} \times 100 \tag{4.7}
\]
The split main separation was varied and the relationship shown in equation (4.7) was used to establish the percentage variation of fuel consumption for the second split main injection with varying split main separation.

4.4.1 Injected Fuel Quantity in Second Part of Split Main Injection With and Without Hydraulic Damping

Previous work undertaken by the Ford Motor Company, shown in Figure 4.15, demonstrates the variations of fuel consumption for the second split main injection with changes in split main separation for three different engine configurations. A similar series of test sweeps, considering split main separation, were carried out on the single cylinder engine to demonstrate the variation in fuel delivered. The split main separations ranged between 0° and 40° CA. The results show similar trends to the data supplied by Ford. The percentage variation of fuel consumption for the second split main injection varied between a maximum value of 55% and a minimum value of 8%, as shown in Figure 4.16.

A modified first split main injection with a second close sub-pulse was used to carry out a similar series of test sweeps with hydraulic damping. As before a standard 80:20 split main ratio profile was considered; this ratio was set up at a split main separation of 17° CA. The split main separations ranged between 0° and 40° CA. The percentage variation of fuel consumption for the second split main injection was reduced significantly, varying between a maximum value of 74% and a minimum value of 49%, as shown in Figure 4.17.

Figure 4.18 shows the 10-cycle averaged values of the needle lift traces for the second split main injection at each split main separation without hydraulic damping. The start of injection has been adjusted for each so that they overlay each other; hence the crank angle values on the x-axis therefore have no significance in absolute terms. This figure shows that there is a significant difference in the needle lift traces at each split main separation for test runs with no damping applied. Figure 4.19, which contains the same information as Figure 4.18 but for tests runs using hydraulic damping, demonstrates that there is good repeatability in the needle lift traces at each split main separation for
these damped test runs. The differences between these figures demonstrate that the fuel consumption of the second split main injection is varying with split main separation due to the variation of the needle lift traces and to the variation of the injection pressure. The inclusion of hydraulic damping led to a considerable reduction in the variability of the needle lift traces and hence the reduction in the variability of the amount of fuel delivered in the second split main injection.

4.5 Concluding Remarks

Pressure oscillations present in common rail fuel injection systems are generated by the normal operation of the injector. These disturbances affect the pressure differential across the injector and the needle lift response, which in turn affects the quantity of fuel injected. The pressure fluctuations seen for this fuel injection system had initial amplitudes in the region of 250 bar with the most problematic frequency at 850 Hz. The values of the frequencies and their harmonics were shown to be a function of the geometry of the fuel injection system and are excited by injection events. A method to successfully dampen these injection pressure oscillations to enable the positioning of subsequent injection events without restriction has been introduced in this chapter. This was achieved with the addition of a second sub-injection close to the initial injection. This produced a second pressure wave, which interfered with the first resulting in the significant reduction of the oscillations.

The application of this method of damping to the fuel pressure oscillations has been carried out. The reduction in the variability of the fuel delivered in the second part of a split main injection event, which is typically seen as split main separation is varied, was demonstrated. The timing and quantity of the sub-injection, to give maximum damping, can be tuned and mapped over the operating speed and load range and stored within the engine management system. This part of the work is the subject of a patent application [40] in collaboration with the Ford Motor Company. Each fuel injection system being considered will need to be investigated and tuned in order to establish where the second close sub-injection needs to be placed. This would ensure that
optimum damping occurred under all operating conditions for each fuel injection system. This damping method can however have a number of drawbacks. An increased number of injection events leads to greater injector wear and can shortened injector life on the engine. Also, with an increased number of injection events there are increased possibilities for late fuel sac leakage at the end of each injection event or needle closing event. This tends to result in increased HC emissions or FSN values depending on operating condition being used and the in-cylinder conditions at the time.

The relationship between injection pressure, needle lift and injected fuel quantity was established. The method outlined can be used to accurately determine the quantity of fuel injected from injection pressure and needle lift data. Greater needle lift directly results in more fuel being delivered and higher injection pressures also directly result in more fuel being delivered. Higher injection pressures also resulted in greater needle lift occurring for this particular injector, which again resulted in more fuel being delivered. Upper and lower needle lift limits were established where changes in the recorded needle position did not have an effect on the quantity of fuel delivered. The lower limit was either due to the needle not lifting sufficiently to allow flow of fuel to commence, or due to a calibration error in the needle lift sensor or due to a distortion of the needle itself as the piezo-electric stack is activated. The upper limit was due to the needle being raised above a point so that the fuel flow became restricted by the area presented by the injector nozzle holes. This method can be applied to any fuel injection system in order to establish the relationship between the quantity of fuel injected and the needle lift profile and the fuel injection pressure.
CHAPTER 5

THE SENSITIVITY OF RESPONSES TO FUEL INJECTION AND ENGINE OPERATING PARAMETERS AT FIXED SPEED AND LOAD OPERATING CONDITIONS

5.1 Introduction

The work described in this Chapter was undertaken to explore the influence of three fuel injection parameters and two engine operating parameters at a number of fixed engine speed and load operating conditions. The fuel injection parameters are split main separation (MS), split main ratio (MR) and main injection timing (MT), which were shown in Figure 1.4. The engine operating parameters are EGR rate (EG), which was defined in Section 3.5.1, and boost pressure (BO), which is defined here as the air intake pressure above atmospheric. Tests were carried out at five operating conditions covering light to medium loads and low to medium speeds. These points are labelled A to E and are shown in Figure 5.1. At 1600 rpm, BMEP values were 1.58, 5.50 and 8.45 bar and at 3400 rpm, BMEP values were 1.58 and 5.50 bar. The same speed or load was used at more than one point to allow trends over a region of the operating map to be explored. The experimental work was carried out using a pilot injection with a separation of 25° CA, from the end of the pilot to the start of the main injection, with a fixed quantity of 0.5 mg per event.

NO\textsubscript{x} and smoke, or more specifically PM, emissions are two of the pollutants that need to be controlled to certain levels in order to comply with emissions regulations and as such need to be considered when undertaking investigations into diesel engine performance. A reduction in one of these species tends to result in the increase in the other, resulting in a trade-off between these two pollutants. The optimum trade-off on the FSN-NO\textsubscript{x} map is defined here as the point where the lowest values for smoke and NO\textsubscript{x} emissions can be achieved simultaneously. A point near the optimum trade-off on the FSN-NO\textsubscript{x} map was considered as a starting point for the adjustment of the parameter settings for
the work undertaken in this Chapter at each of the operating conditions considered. Such a point was of interest because the values of NOx and FSN are close to the minimum for each species. Therefore, this point is in the region of the FSN-NOx map that would normally be used for the selection and adjustment of the parameter settings.

Design of Experiments (DoE) methods were used to design the test programme and to process the data generated. The data showed the overall effects that the parameters had on smoke, NOx and HC emissions and fuel consumption at each of the five operating points, while a sensitivity analysis highlighted the localised individual parameter effects on the measured responses. This made it possible to identify how parameter settings need to be adjusted in order to change particular responses as required and highlighted whether the benefits of adjusting a particular parameter outweighed the penalties. The adjustment of the parameters across their full range of values, highlighting the effect that this had on smoke and NOx emissions simultaneously, gave a measure of the linearity of the parameter effects on the FSN-NOx map. The linearity indicates the extent to which the effects of each parameter can be extrapolated, and therefore predicted, from a fixed point on the FSN-NOx map. This work shows which parameters behave more predictably, and are therefore less constricting, when it comes to large adjustments of the parameter settings.

5.2 Application of Design of Experiments in this Thesis

5.2.1 DoE Background

The statistical experimental design methods used in this study are introduced here and are detailed in Appendix C. The use of DoE methods helped to significantly reduce the amount of experimental testing required, without losing accuracy in the results [107], and to improve the analysis of test data. These statistical methods were used to develop mathematical models, in this case non-linear second order polynomial equation models, and to produce response curve fits and surface plots across the experimental space that predict how changes in the value of the input parameters change the responses or
outputs. Box, Hunter and Draper [107, 108] outline the mathematical principles of the DoE techniques applied here. Other important areas are regression analysis and the production of the regression coefficients, which produce the predictor equation models, and these are detailed by Groves, Davis, Montgomery, Box, Hunter and Draper [62, 107, 108, 109]. Details of these principles are also presented in Appendix C. The central composite plan (CCP) test matrix used in this study is detailed below and was designed by the author following methodology described by Montgomery [62] and summarised by Richardson [110].

Conducting a one-variable-at-a-time investigation by varying the five parameters independently at three levels, to generate non-linear models, would require $3^5 = 243$ test runs at each operating condition. This is clearly a great amount of testing, but by assuming the higher order, five-way, interactions (MR x MS x MT x BO x EG) are negligible and ignoring these interactions in a fractional factorial experimental design reduces the size of the test matrix significantly [110] and would require just $3^{(5-1)} = 81$ test runs. To further reduce the amount of testing needed while not compromising the accuracy of the models produced, a CCP test matrix was designed to produce models that are reliable and consistent over most of the design region [111]. The CCP is based on a two level linear model with additional axial and centre points included to help establish the curvature of the response surface [112] and give the required quadratic effects. The plan consists of three types of points: factorial, axial and centre points. The factorial corner points are used to determine the interactions, the axial points demonstrate the effect of varying each parameter while the others remain at nominal conditions and the centre point gives the magnitude of the quadratic effect [62, 109]. Furthermore, a half factorial CCP design reduces the number of the two level factorial corner points while leaving the axial and centre points unchanged.

The three level half factorial CCP test matrix for the five parameter used in the experimental work undertaken here required just $2^{(5-1)} + (5 \times 2) +1 = 27$ test runs at each operating point, which is a substantial reduction on the original
243 test runs required. The test matrix for this half factorial CCP design is shown in Table 5.1.

5.2.2 Initial Findings and Selection of Ranges for DoE Testing

Once a statistically designed experiment has been set up, the test data tends to yield very little in the way of intuitively useful information until all the testing is completed [111] and data analysis has been undertaken. This lack of feedback means that it is necessary to make initial assumptions about the likely levels of the responses and to perform some exploratory engine tests. These exploratory tests are needed to determine the possible combinations of parameter settings and parameter ranges that can be used at each of the speed and load operating points. The aim is to screen out combinations of parameter settings that lead to overly large exhaust gas emissions values. These need to be avoided as they can have a detrimental effect on the exhaust gas analysis equipment. Furthermore, very high response values are of little interest when undertaking investigations into diesel engine performance as they are in regions of the emissions and fuel consumption maps that would generally not be considered. The parameter ranges used for the DoE testing were established as described below and are given in Table 5.2.

The influence of split main ratio at three different injection pressures, 550, 700 and 850 bar, were investigated and the results are shown in Figure 5.2. The trends in FSN values, NO\textsubscript{x} emissions and fuel consumption demonstrate the trade-off issues that are present in diesel combustion. Larger ratios, in the region of 90:10, are clearly better for minimising smoke emissions and fuel consumption, but result in high levels of NO\textsubscript{x} emissions. Heisler [53] states that the amount of NO\textsubscript{x} created is an exponential function of combustion temperature, so even a small decrease in combustion temperature will produce a significant reduction in NO\textsubscript{x} emissions as the amount of fuel in first injection is reduced. A common way to control NO\textsubscript{x} emissions is to retard the main injection timing [54]; this is a similar effect to increasing the amount of fuel in the second injection as more fuel is being injected and combusted later in the cycle, which reduces the peak in-cylinder pressures and temperatures and hence reduces the NO\textsubscript{x} formation. FSN values increase with increased quantity
of fuel in the second injection, as shown in Figure 5.2. This is due to more fuel being injected later in the cycle allowing less time for mixing, combustion and oxidation of the soot before the end of MCC. This figure also shows that fuel consumption increases with an increased amount of fuel in the second injection. This is because the injected fuel is not able to produce as much useful work in the cycle around TDC as more fuel is injected later in the cycle. Increasing the amount of fuel in the second injection results in a fuel economy penalty in a similar way to the adverse effect on fuel economy seen with retarding injection timing [54]. This showed that split main ratios in the range 90:10 to 60:40 would be sensible for the DoE test work to be undertaken. Furthermore, work by Montgomery and Reitz [12, 41] and Takeda and Niimura [42] confirmed that that split main ratios down to 60:40 were as low as could be sensibly investigated without incurring large FSN and fuel consumption penalties and indicated that split main separations in the range 0° to 15° CA were reasonable for this type of experimentation. Split main separation was adjusted for increased engine speed as it was seen that separations of 0° CA at the higher engine speed gave no clear benefits in terms of NOx reductions. At 1600 rpm separation ranges of 0° CA to 10° CA were used, whilst at 3400 rpm the lower separation was increased to 5° CA, pushing the upper value to 15° CA.

The requirements for settings of boost pressure, EGR level and main injection timing at the different operating points were taken experimentally from a Ford Puma with common rail FIE being used for a separate research project at the University of Nottingham [84]. Contour plots for this standard production Puma engine calibration, shown in Figure 5.3, were used as the starting point for the setting of the parameter ranges at each of the operating conditions considered. Test data from Ricardo [66] along with other published data [12, 41, 42] also aided in the preliminary setting of these parameter ranges by defining previously tested or typical upper and lower values for EGR, boost pressure and main injection timing. It is generally useful to maintain the same range for each parameter at each of the operating points; however due to the influence of speed and load this was not always possible, particularly for EGR levels. At high loads only low levels of EGR, up to a maximum of 15%, were
possible before large FSN values were seen. While at low loads EGR levels up to 45% could be tolerated before similar FSN values were seen.

The data from the DoE testing were processed to develop the predictor equations for FSN, NO\textsubscript{x} and HC emissions and fuel consumption at each of the five operating points. The equations were used to build up a full picture of the responses as the parameters were varied over the full range of values and to give predictions of response values that were not found experimentally. Comparing measured and predicted values for fuel consumption, FSN, NO\textsubscript{x} and HC emissions validated the experimental matrix and the modelled predictor equations. The validation graphs for each of the five operating conditions are shown in Figure 5.4 to Figure 5.8 and these graphs show the reliability of the predictor equations as they are generally within an acceptable ±10% for a model of this nature.

5.3 Outline Graphs and Common Point on FSN-NO\textsubscript{x} Map

Values of NO\textsubscript{x} and FSN given by the predictor equations for the operating point at 1600 rpm 8.45 bar BMEP are shown as a scatter plot in Figure 5.9. The data covers all combinations of upper, lower and mid-value parameter settings. Figure 5.10 shows outline envelope plots for this modelled NO\textsubscript{x} and FSN data along with that for the other four operating conditions, which was developed by considering the outline envelopes of the scatter plots at each operating condition. These outline envelopes highlight the sensitivity of NO\textsubscript{x} and FSN at the different operating conditions. Test point A (1600 rpm 1.58 bar BMEP) has low sensitivity to smoke, reaching a maximum value of 1.25 FSN, but does produce high HC emissions instead. This occurs because at this low load point the fuel and air mixture is generally over-lean, leading to incomplete combustion, and at these conditions the in-cylinder temperatures are not sufficiently high to oxidise any unburned fuel later in the cycle. Figure 5.10 also shows that at the higher engine speed of 3400 rpm, at points D and E, NO\textsubscript{x} emissions are less sensitive because the lower residence times mean that less time is spent at higher temperatures to facilitate NO\textsubscript{x} production.
As shown in Figure 5.10, a wide range of values for the FSN and NO\textsubscript{x} are possible within the range of parameter settings that were used for the DoE testing. Within this range of values it was possible to select combinations of parameter settings at each of the five operating conditions to give similar levels of FSN and NO\textsubscript{x} emissions and hence a common NO\textsubscript{x} and FSN point was identified. This coincident point, shown in Figure 5.10, has values of smoke and NO\textsubscript{x} emissions which are close to the optimum trade-off on the FSN-NO\textsubscript{x} map. These values are 0.75 FSN for smoke and 4 g/kWhr for the BS NO\textsubscript{x} emissions. This point is of interest as it is within the region on the FSN-NO\textsubscript{x} map where parameter adjustments or calibrations typically occur. It was also of interest for the work in Chapter 6 as it allowed comparisons to be made between the parameter settings needed to maintain response levels on the FSN-NO\textsubscript{x} map at different operating conditions and gave a common starting point for the speed and load sweeps undertaken.

5.4 Individual Effects of Fuel Injection and Engine Operating Parameter on FSN-NO\textsubscript{x} Map

The intention of the work undertaken in this section was to show the influence of individual parameters on the responses, to investigate the sensitivity of the responses to parameter variations and the linearity of parameter effects on the FSN-NO\textsubscript{x} map. The individual parameter effects on the responses described here were taken from the hybrid half normal plots generated from the DoE data, as shown in Figure 5.11 to Figure 5.15, and show the effects of the individual parameters over their full range of values. These half normal plots highlight those parameters that have an effect on the responses [62, 109, 113], which are NO\textsubscript{x}, FSN, HC and fuel consumption in this case. Parameters with a positive sign show that as the parameter value is increased so does the response being measured, while those with a negative sign indicate that as the parameter value is increased the response being measured decreases. The generation of these plots is detailed in Appendix C. The following summary highlights these individual parameter effects from the normal plot data and tabulated summaries for each of the measured responses are given in Table 5.3 and Table 5.4.
Retarding main injection timing leads to reduced NO\textsubscript{x} emissions and increased FSN values. This adjustment also increased HC levels and increased FC.

Increased boost pressure results in increased NO\textsubscript{x} emissions and decreased FSN values. Decreased HC levels are generally seen but this depends on whether over-leaning occurs with increased boost, which tends to increase HC emissions. A decrease in FC is also generally seen.

Increased EGR levels result in decreased NO\textsubscript{x} emissions and increased FSN values. Increased HC levels and increased FC are also caused by increased EGR.

Changing split main ratio from 60:40 towards 90:10 results in increased NO\textsubscript{x} emissions and decreased FSN values. Generally decreased HC levels are seen but there are some complicated relationships occurring here with levels of mixing, over-leaning, partial and full burning, oxidation and smoke production. A decrease in FC is clearly seen.

Increasing split main separation results in decreased NO\textsubscript{x} emissions, increased HC levels and increased FC. FSN values are dependent upon engine speed over the ranges investigated. At low speeds increased separation increases FSN, but at the higher engine speed this increase in separation reduces FSN values. This is due to the relationship between engine speed and timing of the second injection event, if it is too early it enters a region that is too fuel-rich from the first injection and if it is too late it enters the combustion chamber too late to fully burn.

Due to the nature of the CCP test matrix certain combinations of parameters, which tend to lead to either high FSN or NO\textsubscript{x} emissions, need to be tested simultaneously at the corner points of the test matrix. A summary of these combinations is shown in Figure 5.16 and highlight that a combination of large separation, low ratio (towards 60:40), late injection timing, low boost and high EGR tended to produce low NO\textsubscript{x} but high FSN emissions. Conversely, small separation, high ratio (towards 90:10), early injection timing, high boost and low EGR combinations generally produced low FSN but with a NO\textsubscript{x} penalty.
5.4.1 Sensitivity Analysis of Responses to Parameter Variations

A sensitivity analysis of the fuel injection and engine operating parameters was executed and the effects on the responses at the different points on the operating map were considered. For this sensitivity analysis the coincident point close to optimum FSN-NO\textsubscript{x} trade-off, was used as the starting point at each operating condition. Figure 5.17 to Figure 5.21 show the plots of FSN against BS NO\textsubscript{x} at the five speed and load operating conditions as each parameter setting is independently varied.

The sensitivity analysis was undertaken using the data from these figures to highlight which parameters have the greatest effects on the responses at the coincident point and to identify those parameters that need to be treated with more care to avoid rapidly changing responses if the exact parameter settings are not maintained. For each operating condition considered it was not possible to maintain the same range of values for each parameter, therefore a unit change in each parameter setting from the starting point was considered. These unit changes are shown in Figure 5.22 and are as follows: split main separation increase of 1\degree CA, split main ratio increase of 5\% (for example from a ratio of 80:20 to 85:15), main injection timing retarded by 1\degree CA, boost pressure increase of 0.1 bar and EGR rate increase of 5\%. The effects of parameter adjustment seen in this sensitivity analysis are described below and generally agree with the data for the individual parameter effects over their full range of values seen in the previous section.

As shown in Figure 5.22, retarding main injection timing consistently reduces NO\textsubscript{x} emissions across the operating map as in-cylinder temperatures and pressures are reduced as the injection, and hence combustion, is moved later in the cycle. Increased boost pressure results in an increase NO\textsubscript{x} emissions at all operating conditions due to the associated increased in-cylinder pressures and temperatures. Increasing the levels of EGR consistently reduce NO\textsubscript{x} emissions across the operating map as the in-cylinder temperatures are reduced. The addition of EGR also dilutes the charge air by reducing the O\textsubscript{2} available for mixing and combustion, which also slows the rate of combustion and hence NO\textsubscript{x} production further. EGR also increases the heat capacity of the engine.
inlet charge, which reduces the flame temperature and hence reduces NO\textsubscript{x} formations by reducing the peak in-cylinder temperatures. A soot increase occurring for any reason will result in an increase in heat radiation from the hot soot particles [114], resulting in lower local temperatures. An increase in split main separation generally leads to decreased levels of NO\textsubscript{x} emissions, as shown in Figure 5.22, as there is less fuel burning earlier in the cycle which decreases in-cylinder pressures and temperatures. However, at test point C, 1600 rpm 8.45 bar BMEP and also shown in Figure 5.19, an increase in NO\textsubscript{x} emissions is seen with increased split main separation. This is because at this high load point in-cylinder pressures and temperatures are already high, leading to high NO\textsubscript{x} values, caused by the large first injection. The small second injection has the effect of sustaining these high values further into the cycle. Thus an increase in separation increases the period that these high pressures and temperatures are experienced, resulting in increased levels of NO\textsubscript{x} emissions. Increasing split main ratio generally shows a small increase in NO\textsubscript{x} emissions as more fuel is injected earlier in the cycle, which increases the peak pressures and temperatures. In Figure 5.22 it can be seen that increasing split main ratio at 1600 rpm 8.45 bar BMEP, test point C, has the effect of reducing NO\textsubscript{x} emissions slightly. It is also clear, however, from the individual parameter effects plot in Figure 5.19 that this effect is localised around the coincident point and the overall effect is an increase in NO\textsubscript{x} emissions.

The sensitivity of FSN values to each parameter at the different operating points is shown in Figure 5.23. This shows that the percentage changes in FSN values have a large degree of variation and as such are less uniform than the changes seen for NO\textsubscript{x} emissions. FSN levels generally increase with retarding main injection timing due to more fuel being injected later in the cycle, giving less time for mixing and combustion before the end of MCC. This is not the case at test point D, 3400 rpm 1.58 bar BMEP, where a significant HC penalty is seen, Figure 5.24, as FSN values reduce. This, along with a dramatic increase in fuel consumption, Figure 5.25, shows that the calibration here has become very poor and the combustion quality has reduced. Increased boost pressure generally reduces FSN values due to better levels of mixing and the shortened ignition delay created by the increased in-cylinder pressures. This
leads to higher in-cylinder temperatures, which enable the oxidation of the soot as the cycle proceeds. Increased boost pressure intensifies the turbulence level of the flow field caused by the increased pressure drop across the inlet valve [114]. This increased level of mixing due to the higher turbulence and the higher AFR, which increases the availability of O₂, reduces soot production as it not only improves the initial combustion, but also allows for soot oxidation later in the cycle. However, the FSN increase seen in Figure 5.23 at 3400 rpm 5.50 bar BMEP, test point E, with increased boost pressure is due to an increase in the over-lean regions leading to an increase in unburned and partially burned HC emissions. These then become attached to soot particles resulting in soot growth [32] and an increase in measured FSN values. There is also an increase in fuel consumption at this point, which indicates that the fuel is not being fully utilised. This contrasts with fuel consumption improvements generally seen with increased boost as the higher in-cylinder pressures and temperature tend to result in better utilisation of the injected fuel and furthermore, turbocharging tends to recover some of the energy that would otherwise be wasted in the exhaust gases. The addition of EGR generally increases FSN and HC emissions across the operating map, as there is less O₂ available to enable complete combustion of the injected fuel. High in-cylinder temperatures associated with higher loads and speeds tend to lead to FSN rather than the HC emissions seen at lower load points where in-cylinder temperatures are not as high and some fuel can remain unburnt. Furthermore, the ignition delay associated with the addition of EGR, reduces in-cylinder pressures and temperatures, as the pressure rise comes later in the cycle. These reduced temperatures reduce the oxidation of soot later in the cycle, which is further emphasised by the reduction of available O₂ needed for this oxidation process. As shown in Figure 5.23 increased FSN values are seen with increased split main separations at the lower engine speed of 1600 rpm. This is because the fuel in the second injection event is pushed later in the cycle and is therefore injected too late in the cycle to fully mix and combus before the end of MCC. However, at the higher speed of 3400 rpm a reduction in FSN values is seen with increased split main separations. This because a certain amount of time is needed to lean out the combustion zone sufficiently after the first injection, to reduce soot formation, before the second injection replenishes the
soot cloud produced from the first injection. Smaller separations at higher engine speeds do not allow full advantage to be made of the split main in terms of increasing the levels of mixing of the fuel with the charge air, more time is needed for the required levels of mixing with this increased engine speed. This is because the same period in terms of crank angles in the engine cycle occur over a shorter time period at these increased speeds. At the largest separation of 15° CA at 3400 rpm the in-cylinder temperatures remain high enough and the mixture is still turbulent enough to initiate rapid combustion of the second injection to avoid FSN penalties, but is far enough after TDC to avoid the large in-cylinder temperature and pressure rises that would otherwise result in increased NOx emissions. If split main separations were taken beyond 15° CA at 3400 rpm an increase in FSN would again be expected as fuel is injected too late in the cycle to fully mix and combust before the end of MCC. By the same token increasing split main ratio consistently results in reduced FSN values, for a fixed separation, as there is less fuel being injected late in the cycle, which would have less time to mix and fully combust before the end of MCC.

The sensitivity of HC emissions to each parameter is shown in Figure 5.24. It is clear that HC emissions are generally only problematic at the lower load points where the mixture can be over-lean or where the calibration is poor leading to incomplete combustion. Furthermore, at low load conditions the in-cylinder temperatures are not sufficiently high to oxidise any unburned fuel that may be present later in the cycle. At the higher load points there is generally less sensitivity to HC emissions as the in-cylinder temperatures are higher, which oxidises any fuel remaining unburned later in the cycle, but tends to result in high FSN values instead as was shown in Figure 5.23.

The sensitivity of fuel consumption to parameter variations is shown in Figure 5.25. As with HC emissions, fuel consumption is greatest at the low load points where incomplete combustion occurs, which means that the injected fuel is not being fully utilised, and the unburned fuel represents fuel that is wasted. Fuel consumption is also high at low loads because as loads are reduced engine friction becomes an increasingly large component of the power output of the engine. Fuel consumption would be expected to increase as the peak in-
cylinder temperatures and pressures are reduced, with the addition of EGR or the reduction of boost, and the injected fuel is not utilised as efficiently. Split main separation and ratio change the position of the fuel in the cycle and can therefore move it away from burning at the most efficient TDC position, but main injection timing has the greatest influence on fuel consumption in this case.

From this study the effects of the parameters at each operating condition can be clearly seen and these effects are quite consistent and uniform across the operating map. Knowledge of the effects that the individual parameters have on the responses would be useful when adjusting parameter calibrations close to the optimum point on the FSN-NOx map. This work has highlighted those parameters that need to be treated with care to avoid emissions penalties if calibration settings are not maintained.

It is clear that NOx emissions are most sensitive to EGR and injection timing at each operating condition, which highlights the problems involved in maintaining low and predictable NOx levels for the given FSN values near the optimum trade-off. A slight variation in EGR, which is quite possible due to sooting-up of the EGR system or due to EGR lag under transient operation, would result in large changes in NOx values at each of the operating points considered. Split main ratio and separation generally have the smallest effects on NOx emissions. This work also shows that the NOx penalties associated with increased boost pressure are outweighed by the reductions possible with the addition of EGR and the retarding of the main injection timing. Furthermore, it is clear that generally the addition of EGR reduces NOx emissions more than the associated increase in FSN values. Whereas, increasing boost pressure generally reduces FSN values more than the associated NOx penalty away from low load conditions. Split main separation and ratio have less influence on the calibration of the engine compared to EGR, boost and main injection timing.
5.4.2 Linearity of Parameter Effects on FSN-NO\textsubscript{x} Map

An investigation into the linearity of the parameter effects on the FSN-NO\textsubscript{x} map was undertaken by considering the changing sensitivities of both FSN and NO\textsubscript{x} emissions to parameter changes on the FSN-NO\textsubscript{x} map. If the rate of change of FSN values for a particular parameter adjustment is different to the rate of change of NO\textsubscript{x} emissions for the same parameter adjustment then it is clear that the effect of parameter changes on the FSN-NO\textsubscript{x} map will not be linear. The idea is to show which parameters have the most linear, and therefore predictable, behaviour when extrapolating from a fixed point on the FSN-NO\textsubscript{x} map at each of the operating points. This indicates which parameters have to be treated with more care to avoid unpredictable FSN or NO\textsubscript{x} emissions if accurate parameter settings are not maintained.

The data was taken by adjusting the parameters individually using the coincident point as the starting point at each operating condition on the FSN-NO\textsubscript{x} map as shown in Figure 5.17 to Figure 5.21. The zone of linearity is defined as a 36° change of the gradient for each parameter on the FSN-NO\textsubscript{x} map from the coincident point, which is a 10% change in gradient angle. A summary of the results is shown in Table 5.5.

The parameters can behave quite differently depending on the point on the operating map under consideration. It is clear from Table 5.5 however that overall main injection timing and EGR are the most linear on the FSN-NO\textsubscript{x} map and therefore the behaviour of these two parameters can be predicted from a fixed point on the FSN-NO\textsubscript{x} map. In contrast boost pressure followed by split main ratio are the least linear at each operating condition. This shows that it is not possible to extrapolate the behaviour of boost pressure and split main ratio variations with confidence any great distance from the starting point on the FSN-NO\textsubscript{x} map.

5.5 Concluding Remarks

DoE methods were exploited in the design of the test programs and the evaluation of the test results. This led to a substantial reduction in the number
of experimental runs required without adversely affecting the accuracy of the results. The half factorial CCP test matrix used in these studies reduced the required testing of the five parameters at each of the five operating points considered by 89%, from 243 to 27 experimental runs. The data were validated showing an accuracy of at least 10%.

This Chapter has highlighted the level of complexity involved in trying to determine a calibration strategy for an HPCR DI diesel engine with the capability of delivering multiple injections. The experimental work undertaken was based on global observations. These were combined with an understanding of how local in-cylinder conditions affect the mixing of the fuel and air and therefore combustion. Using the coincident point, near the FSN-NO\textsubscript{x} optimum trade-off, as the starting point each parameter was independently adjusted across the full range of values whilst holding the others fixed. This demonstrated the effect that each parameter had on FSN, NO\textsubscript{x} and HC emissions and fuel consumption at each speed and load operating point.

The individual parameter effects, taken from the normal plot data, can be summarised as follows:

- Increasing split main separation results in decreased NO\textsubscript{x} emissions, generally increased HC levels and increased FC, while FSN values are dependent upon engine speed over the ranges investigated. At low speeds increased separation increases FSN, but at higher engine speeds increased separation decreases FSN values.
- Changing split main ratio from 60:40 towards 90:10 results in increased NO\textsubscript{x} emissions, decreased FSN values, generally decreased HC levels and decreased FC.
- Retarding main injection timing leads to reduced NO\textsubscript{x} emissions, increased FSN values, generally increased HC levels and increased FC.
- Increased boost pressure results in increased NO\textsubscript{x} emissions, decreased FSN values, generally decreased HC levels and decreased FC.
- Increased EGR levels result in decreased NO\textsubscript{x} emissions, increased FSN values, generally increased HC levels and increased FC.
The sensitivity analysis confirmed the individual parameter effects noted above and demonstrated that the behaviour of the parameters is generally consistent at each speed and load operating point. This work shows how adjustments of the parameter settings can be made to alter the position on the FSN-NO\textsubscript{x} map at different operating conditions and therefore how an emissions and fuel consumption calibration can be improved. The penalties associated with the adjustment of each parameter are also clear. Overall the sensitivity of NO\textsubscript{x} emissions to the adjustment of the parameters is consistent across the operating map. The sensitivity of FSN to parameter adjustments is less systematic as more complicated effects are occurring due to levels of mixing, local AFR, in-cylinder temperatures and oxidation of soot later in the cycle. Engine speed has a clear effect on FSN with the variation of split main separation. This is driven by the levels of mixing and the amount of time available for combustion to occur and whether the second injection enters a soot-rich region left in the bowl from the first injection event. HC emissions are most sensitive at low loads where incomplete combustion can occur and the in-cylinder temperatures are not sufficiently high to oxidise any unburnt fuel later in the cycle. Fuel consumption is also most sensitive at low loads again due to incomplete combustion and because friction becomes an increasingly important factor as engine loads are reduced.

Generally EGR has the largest individual influence on NO\textsubscript{x} and generally has the second largest influence on FSN after boost pressure. This shows that increasing EGR levels reduces NO\textsubscript{x} more consistently than the associated FSN penalties seen. This means however that if EGR levels are not maintained at the required level then NO\textsubscript{x} can increase dramatically, which can cause problems for the calibration of production engines as manufacturing tolerances may lead to lower than expected EGR levels. This may also be problematic during transient operation where EGR lag will lead to lower than expected or required EGR levels temporarily. Furthermore, increasing boost pressure generally reduces FSN more than the associated NO\textsubscript{x} penalty, away from low load conditions. Again problems could be seen as boost pressure variations, due to turbo-lag, could cause temporarily lower than expected boost pressures resulting in higher than expected FSN values. This work also shows that the
NOx penalties associated with increased boost pressure are outweighed by the benefits seen with the addition of EGR and the retarding of the main injection timing. Overall both split main separation and ratio have less influence on the responses compared to EGR, boost and main injection timing.

High NOx emissions are generally caused by small separations, high ratios (around 90:10) and advanced main injection timing, which results in combustion of fuel earlier in the cycle and higher in-cylinder pressure and temperatures. High boost pressures and low EGR levels also result in increased in-cylinder pressures and temperatures and hence increased NOx emissions.

High FSN values are generally due to large separations, low ratios (around 60:40) and retarded main injection timing, which results in fuel entering the combustion chamber late in the cycle which does not fully mix and burn before end of MCC. Low boost pressures and high EGR rates result in reduced localised O2 concentrations, which reduces the mixing of the injected fuel with the available O2 to increase FSN levels.

High HC emissions are generally associated with particular speed and load operating conditions rather than the setting of the fuel injection and engine operating parameters. These occur at low loads where over penetration of the reduced amount of injected fuel can result in over-lean areas, which means the fuel cannot be fully combusted. Also poor levels of mixing of the injected fuel with the charge air can result in more locally fuel-rich zones and incomplete combustion before the end of MCC. This leads to HC emissions at low loads but FSN at higher loads, as in-cylinder temperatures are higher later in the cycle.

Fuel consumption penalties are generally due to large separations, low ratios (around 60:40) and retarded injection timing. This results in fuel entering the combustion chamber late in the cycle during the expansion stroke, which means the fuel does not fully burn around TDC where the greatest utilisation of the injected fuel would be seen. Low boost pressure and high EGR rates mean that the injected fuel may not be fully utilised due to the lower local O2
concentrations resulting in poorer mixing and combustion. Furthermore, at low loads the effects of the friction component becomes much more pronounced and this is reflected in the increased fuel consumption seen.

The linearity of parameter effects shows how much each parameter setting can be adjusted before the behaviour on the FSN-NO\textsubscript{x} map becomes unpredictable and it is no longer possible to extrapolate from a fixed point. The less linear the parameter effects are on the FSN-NO\textsubscript{x} map the more difficult it is to predict the parameter behaviour as settings are adjusted over larger ranges of values. This therefore shows which parameters are the most constricting when it comes to adjusting their settings as their effects on the responses become less predictable. Main injection timing and EGR have the most linear behaviour, while boost pressure is generally the least linear at each of the operating points.

In summary, it is possible with the experimental work undertaken in this Chapter to gain an understanding of the individual effects of the fuel injection and engine operating parameters at the different points considered on the speed and load operating map. High NO\textsubscript{x} is generally a result of high in-cylinder temperatures and pressures where fast and thorough mixing of the fuel and air has occurred; this also tends to lead to low FSN values and occurs with high boost, early main injection, low EGR and closely coupled injections with a 90:10 ratio. High FSN values generally occur where lower in-cylinder temperatures are present and the mixing is slower or poorer. This occurs with low boost, late main injection timing, high EGR and ratios of 60:40 with large separations. It was shown that the parameter effects on the responses at the coincident point are generally consistent at each operating condition, there are some exceptions and these have been explained. The individual plots at each operating point, Figure 5.17 to Figure 5.21, show what adjustments would need to be made to a calibration to move to a required point on the FSN-NO\textsubscript{x} map and therefore how to move closer to the optimum trade-off point.
CHAPTER 6

THE INFLUENCE OF SPEED AND LOAD ON ENGINE RESPONSES WITH FIXED PARAMETER SETTINGS

6.1 Introduction

Determining what values to set, say, injection timing to at a given operating condition is a calibration task. The task would be relatively straightforward if this were the only parameter that needed to be set. It becomes much more difficult when several parameters need to be set and each influences several important areas of performance and pollution emissions. This is the case for turbocharged diesel engines with high pressure common rail fuel injection systems and external exhaust gas recirculation. The constraints that speed and load impose on the calibrations, in terms of engine response variations, dictate how common a calibration strategy can be over the operating map.

Two major themes are covered in this Chapter. The first explores the differences and similarities in parameter settings giving the same NO\textsubscript{x} and FSN values at a set of speed and load points. The second concerns the area of the speed and load operating map which can be covered with a set of fixed local calibrations. These topics reflect the interest in minimising the amount of test work required to define a calibration strategy over the full operating map. The five operating points used in Chapter 5 have been used again here, and the experimental work was carried out with a fixed pilot injection of 0.5 mg occurring 25\textdegree CA before the start of main injection.

6.2 Parameter Settings at Common FSN and NO\textsubscript{x} Point

Moving between different speed and load points with very different fuel injection and engine operating parameter settings can cause difficulties in terms of defining a calibration strategy across the operating map. The common point close to the optimum trade-off on the FSN-NO\textsubscript{x} map, called the coincident point, was introduced in Chapter 5. It was considered as it could give insight
into how parameter settings need to be adjusted between the different speed and load conditions in order to maintain the NO\textsubscript{x} and FSN values, while remaining in the region of greatest interest on the FSN-NO\textsubscript{x} map. The responses at the coincident point were in the region of 0.75 FSN and 4 g/kWhr BS NO\textsubscript{x}.

NO\textsubscript{x} and FSN outputs are driven by in-cylinder conditions such as levels of turbulence to aid mixing, quantity of O\textsubscript{2} available to mix with the injected fuel, the amount of time available for mixing and combustion and the position in the cycle of the peak burn or in-cylinder pressure rise. These in-cylinder conditions are different at each of the speed and load points considered and therefore it follows that the parameter settings giving the same levels of NO\textsubscript{x} and FSN at the different operating conditions are likely to differ. The parameter settings giving the same pair of NO\textsubscript{x} and FSN values at the coincident point at each of the operating conditions considered are shown in Figure 6.1. This figure shows that required levels of NO\textsubscript{x} and FSN can be maintained as the operating map is traversed with a high degree of continuity in respect of the parameter settings. The interest here is not the specific parameter settings, but the trends in these across the operating map; an analysis of these trends is detailed below.

Boost pressure values needed to be increased with both increased speed and load, as shown in Figure 6.1. Increased loads, facilitated by increased mass of injected fuel, increases levels of black smoke in the exhaust, due to the inability of the increasing amount of fuel to mix with sufficient air, thus limiting the amount of fuel that can be burned efficiently [32]. The increased boost therefore improves mixing by increasing the amount of available O\textsubscript{2} and by creating higher levels of in-cylinder turbulence. Increased levels of boost were also required with increased engine speed in order to improve the rate of mixing and combustion. This is because the increased engine speed reduces the time available for combustion, which would result in increased FSN levels.

From Figure 6.1 it is clear that as both engine speed and BMEP are increased a reduction in EGR is required to maintain FSN and NO\textsubscript{x} levels. At low loads the fuel/air equivalence ratio, $\phi$, is low which means that relatively high
concentrations of $O_2$ are present. Therefore at low loads higher EGR levels can be used as there is already plenty of $O_2$ available in-cylinder for mixing and good combustion to occur [27]. The addition of EGR has the effect of reducing the $O_2$ trapped in the cylinders substantially, reducing the AFR, which leads to increased FSN emissions [80]. The impact of EGR on FSN at high loads is particularly detrimental because the AFR is already low and the EGR has a low $O_2$ concentration [80]. The reduction in available $O_2$ in the burning region impairs the soot oxidation process. Furthermore, this $O_2$ reduction reduces the local flame temperature, which further reduces the soot oxidation rate [80]. Therefore, as more fuel is injected with increased load, EGR must be reduced to avoid an increase in $\phi$, which would result in an FSN penalty. At lower engine speeds higher EGR rates are possible as there is more time available before the end of MCC for mixing of the injected fuel and the charge air to occur, leading to more complete combustion, and there is more time available for oxidation of soot to occur. As engine speeds are increased there is less time available for mixing of fuel and available oxygen to occur before start of combustion, resulting in poorer mixing and more locally fuel-rich zones. There is also less time available for combustion and oxidation of soot before end of MCC, which can lead to high FSN values.

Levels of FSN and $NO_x$ can be maintained by advancing main injection timing as either engine speed or load is raised, as shown in Figure 6.1. The advance of injection timing with increasing load avoids FSN penalties otherwise seen due to the increased amount of fuel injected. Increased fuelling means that the injection event stretches later into the cycle, towards the end of MCC, and that there is less time available for complete combustion to occur before this point if timing is not advanced. There will also be less time available for oxidation of soot to occur later in the cycle. With increased engine speeds there is less time available for mixing of fuel and available oxygen to occur before start of combustion, which results in more fuel-rich zones and high soot production. There is also less time available for combustion and oxidation of soot before end of MCC. Advancing main injection timing with increased speed effectively gives more time for these processes to occur.
Split main ratio needed to remain at 90:10, as the operating map was traversed while attempting to maintain FSN and NO\textsubscript{x} emissions, except at the low load and low speed point, test point A, where a ratio of 60:40 was used, as shown in Figure 6.1. It was shown in the outline plots from Chapter 5, Figure 5.10, that this test point generally produced low FSN values, with a maximum of about 1.25 FSN, while the coincident point has a value of 0.75 FSN. The coincident point FSN value is therefore close to the upper limit at this operating condition and this indicates that the ratio of 60:40 was required here to almost artificially raise the FSN values seen in order to hit the coincident point value.

As shown in Figure 6.1, moving between different load points did not require the adjustment of the split main separation. However, split main separation needed to be increased with increased engine speed as the increased speed effectively shortened the separations in terms of elapsed time or time available for mixing and combustion. If this separation were not increased levels of NO\textsubscript{x} would increase, as more fuel would be available early in the cycle, which would increase in-cylinder pressures early in the cycle. Also FSN levels would tend to decrease, as less fuel would be injected later in the cycle.

These findings allow predictions of the parameter settings needed to give certain NO\textsubscript{x} and FSN values to be made at untested points on the operating map. Considering this, it is possible to get a good initial indication of the required parameter settings at 2500 rpm 5.50 bar BMEP, for example, which sits between test points B and E (1600 rpm and 3400 rpm respectively at 5.50 bar BMEP). The parameter settings, if the parameter relationships with speed and load are linear or near linear, would be as follows: split main separation of 7.5° CA, split main ratio of 90:10, main injection timing of 6.1° CA BTDC, boost pressure of 0.53 bar and EGR of 11.5%.

### 6.3 Speed and Load Sweep Testing with Fixed Calibrations

Speed and load sweep testing was undertaken by traversing the speed and load map whilst leaving the parameter settings unchanged from a number of different starting points on the operating map. The idea was to establish the
region on the operating map over which a particular calibration can be used before FSN and NO\textsubscript{x} emissions reached unacceptably high levels due to the fundamental effects that speed and load have on combustion. FSN, NO\textsubscript{x}, HC, CO and CO\textsubscript{2} emissions along with fuel consumption were investigated.

A calibration can be considered to become unacceptable with changing speed or load by the amount that either NO\textsubscript{x} or FSN values increase; this level of deterioration in NO\textsubscript{x} or FSN was defined in order to ascertain the possible variations in speed and load from the starting points on the operating map. The coincident point, which was previously defined in Chapter 5 and is shown in Figure 5.10, and the associated parameter settings were used as the starting point for the sweeps. The coincident point value for FSN was in the region of 0.75 FSN at each of the test points on the operating map and for NO\textsubscript{x} emissions the starting point value was approximately 4 g/kWhr. An upper limit of 50% from these starting point values was considered here to be a reasonable, if not coarse, increase in responses, taking FSN from 0.75 to 1.125 FSN and NO\textsubscript{x} from 4.0 to 6.0 g/kWhr. In the following sections, the upper FSN and NO\textsubscript{x} limits and the coincident point starting values have been highlighted on the FSN and NO\textsubscript{x} graphs. This shows how far these upper limits allow the speed or BMEP to be changed before the calibration runs into trouble and results in poor combustion and high emissions.

The parameter settings for the coincident point at the five operating conditions were detailed in the previous section and have been used here as starting points, and were held constant, for the examination of trends in the engine responses as speed and load were adjusted. The parameter settings are given in Table 6.1. The key feature of the parameter settings at each of these operating points is that they give similar levels of brake specific NO\textsubscript{x} and FSN, which allows for direct comparisons to be made as speed and BMEP are adjusted from this common starting point. Four speed sweeps between 1600 rpm and 3400 rpm were undertaken, for increasing and decreasing speed, from test points A to D and B to E. Also four load sweeps were executed, for increasing and decreasing BMEP, from test points A to C and D to E. These were
executed between 1.58 bar and either 5.50 bar or 8.45 bar BMEP. Figure 6.2 shows a summary of the speed and load test sweeps undertaken here.

Maintaining the parameter settings while undertaking both speed and load sweeps is straightforward for main injection timing, boost and EGR levels, but is more involved when considering split main separation and ratio. With changes in engine speed the split main separation is maintained by simply making sure that the separation between the end of the first and the start of the second split main injection is maintained on a crank angle base, which means that this separation increases on a time scale base. To increase BMEP while maintaining the given starting ratio it is necessary to increase both the first and second injection fuel quantities simultaneously in order to increase load while maintaining the required ratio.

For the load sweeps the use of split main injections below the lower load of 1.58 bar BMEP was not investigated. This would only be of interest at idle where engine speed is in the region of 800 rpm. Furthermore, it was not possible to go above 8.45 bar BMEP at 1600 rpm, test point C, and not much higher than 5.50 bar BMEP at 3400 rpm, test point E, due to the peak in-cylinder pressures that were experienced above these points. The single cylinder engine is limited to a peak in-cylinder pressures of 160 bar and therefore this limited the levels of BMEP that could be attained, due in part to the relatively high levels of boost pressure that were used at some test points, without risking damage to the engine.

6.3.1 Load Sweeps

Figure 6.3 to Figure 6.8 show the responses for the load sweeps at the two speed levels. It is clear from Figure 6.3 and Figure 6.6, showing the effects on NOx and FSN responses at 1600 rpm and 3400 rpm respectively, that increased FSN values limit the calibration as load is increased. Whereas, increasing NOx emissions limit the range of the sweep as load is decreased.

An increase in the level of FSN is seen with increased load as shown in both Figure 6.3 and Figure 6.6. This occurs when there is not enough locally
available oxygen to convert the carbon in the injected fuel to CO₂ or CO, even though the engine is running lean [21] and takes place within the injected fuel spray; this solid carbon appears as soot or smoke. With high levels of injected fuel, even though the overall equivalence ratio may remain lean, locally over-rich fuel conditions may exist through the expansion stroke and into the exhaust process [32] resulting in smoke production [21]. To combat these lower mixing rates and increased fuel-rich zones the calibration could be adjusted by increasing the boost pressure to increase the available O₂ and to improve mixing due to increased turbulence. EGR could be reduced, which would increase the AFR and reduce smoke production. Furthermore, main injection timing could be advanced to give more time for mixing and combustion before the end of the MCC phase.

Increasing levels of NOₓ are seen with the reduction of load as shown in Figure 6.3 and Figure 6.6. The formation of NOₓ is dependent on temperature, local oxygen concentration and residence times [15]. As load is reduced, and hence fuelling is reduced, the relative local O₂ concentration is increased, which increases NOₓ formations. To combat the increased mixing and combustion rates boost pressure needs to be reduced, EGR levels need to be increased or main injection timing needs to be retarded. This would move the peak burn later in the cycle and reduce the amount of pre-mixed combustion, and the associated rapid burn rate, by allowing less time for mixing before start of combustion. In general, an increase in either FSN or NOₓ results in the decrease in the other response, as is typically seen with the FSN-NOₓ trade-off.

HC emissions, shown in Figure 6.4 and Figure 6.7, show similar trends and values. These figures show that HC emissions are higher at low loads for all the load sweeps undertaken, as the mixture becomes over-lean and combustion becomes poor. These figures also highlight that at the higher engine speed of 3400 rpm HC emissions are greater as BMEP is decreased, with a maximum of 2.5 g/kWhr being recorded, as there is less time available for complete combustion leading to higher levels of incomplete combustion.
Low values of brake specific fuel consumption are obviously desirable but the parameter settings required to give low BS FC values tend to result in high NO\textsubscript{x} emissions, which results in another trade-off with the selection of the parameter settings. The data for BS FC for the load sweeps, shown in Figure 6.4 and Figure 6.7, highlight that BS FC is significantly worse at lower loads with values of approximately 600 g/kWhr, while at higher loads this is halved to around 300 g/kWhr. This variation is due to the friction effects in the engine, which can be a sufficiently large fraction of the indicated work of an engine to be of great practical importance in engine design. This friction fraction typically varies from between 10% at full load and 100% at idle or no load [32]. It can also be said that at idle BS FC will become very large since the engine is producing little useful work while continuing to consume fuel.

Emissions of CO and CO\textsubscript{2} for the load sweeps are shown in Figure 6.5 and Figure 6.8 and show similar trends and values. These figures also indicate that there are increasing levels of O\textsubscript{2} available, as BMEP is reduced, to convert the carbon in the fuel to CO and CO\textsubscript{2} emissions, which rise steeply with decreasing BMEP.

A further point concerning combustion quality can be made here. For the increasing load sweep from point A to C, shown in Figure 6.4, there is initially a steady and expected reduction in HC emissions and BS FC. However, as the fuelling of this low load calibration is increased to reach a value of 5.50 bar BMEP the combustion become very poor, resulting in an increase in BS FC and HC emissions. This poor calibration is emphasised by the steep rise in FSN values shown in Figure 6.3, which at over 9 FSN is beyond acceptable levels. The poor combustion quality is again shown by the extremely high CO emissions that are produced at this point shown in Figure 6.5, which represents lost energy from the unburned fuel.

These findings show that the calibrations cannot be maintained across the full BMEP range without adjustments being made to the parameter settings. Poor calibrations, producing high exhaust emissions or fuel consumption, can be adjusted by changing the individual parameter settings as shown in Chapter 5.
6.3.2 Speed Sweeps

Figure 6.9 to Figure 6.14 show the responses for the speed sweeps at the two separate load levels. Figure 6.9 and Figure 6.12 show FSN and NO\textsubscript{x} emissions at 1.58 bar and 5.50 bar BMEP respectively. These figures clearly show that increasing NO\textsubscript{x} emissions limit the calibration as engine speed is decreased. Whereas, increased FSN values limit the range as speed is increased. An increase in either FSN or NO\textsubscript{x} results in the decrease in the other response, as is typically seen with the FSN-NO\textsubscript{x} trade-off.

An increase in the level of FSN is seen with increased engine speed as shown in Figure 6.9 and Figure 6.12. Longer combustion periods help to burn off soot and HC emissions [44]. However, in this case the increased engine speed means that there is less time available for oxidation of the soot formed later in the cycle. To combat this reduction in time available, the calibration needs to be adjusted by increasing the boost pressure to increase speed of mixing by increasing the available O\textsubscript{2} and the level of turbulence. EGR could be reduced to increase the level of O\textsubscript{2} available and hence increase the rate of mixing. Furthermore, main injection timing could be advanced to give more time for mixing and combustion before the end of MCC.

Increasing levels of NO\textsubscript{x} are seen with the reduction of engine speed as shown in Figure 6.9 and Figure 6.12. Shorter combustion duration is used to reduce the time that combustion gases are exposed to higher temperatures, reducing NO\textsubscript{x} formations [44]. However, in the case of reduced engine speed, the combustion gases spend more time at higher temperatures, allowing the Zeldovich kinetics more time to form NO, the primary constituent of NO\textsubscript{x} emissions. To combat the increase seen, boost pressure needs to be reduced, EGR levels need to be increased or main injection timing needs to be retarded. This will move the peak burn later in the cycle and reduce the amount of pre-mixed combustion, and the associated rapid burn rate, by allowing less time for mixing before start of combustion.

Emissions of HC show similar values and trends as those found for the BMEP sweeps discussed in the previous section. HC emissions at the higher load point
are low, as shown Figure 6.13, compared to the significantly higher HC values at the lower load point, as shown in Figure 6.10. Furthermore, HC emissions become increasingly bad with increased engine speed at this low load as increasingly incomplete combustion occurs.

The data for fuel consumption for these speed sweeps highlight that there is a BS FC penalty at low loads, as shown in Figure 6.10, with values in the region of 500 to 600 g/kWhr. At the higher load, as shown in Figure 6.13, values in the region of 300 g/kWhr are seen for BS FC. The higher BS FC seen at lower loads is due to the greater role that friction plays compared to the higher load points where it has a relatively small impact.

Emissions of CO and CO₂ for the speed sweeps show similar trends and values to those found for the load sweeps described previously. This highlights that there are higher levels of O₂ available to convert the carbon in the fuel to CO and CO₂ emissions at low loads, as shown in Figure 6.11, compared to that available at higher loads, as shown in Figure 6.14. Combustion quality can be seen to reduce at low loads by the increasing levels of CO and CO₂ emissions, shown in Figure 6.11, as engine speed reaches 3400 rpm. These increasing levels represent lost chemical energy from the fuel. Furthermore, for the increasing speed sweep from point A to D initially there is a steady and expected increase in FSN, as shown in Figure 6.9. However, as the speed reaches 3400 rpm, FSN rapidly reduces from 2.25 to 0.75 FSN. At the same point HC and BS FC levels significantly increase, as shown in Figure 6.10. This shows that the combustion quality has become very poor and inefficient use is being made of the injected fuel.

The findings outlined above show that the fixed calibrations cannot be maintained all the way between 1600 rpm and 3400 rpm and therefore adjustments need to be made to the parameter settings. As with the load sweeps, poor calibrations can be avoided by changing the individual parameter settings as shown in Chapter 5.
6.3.3 Operating Map Coverage

The area of the operating map that can be covered by a particular fixed calibration is of interest as it indicates, not only the region over which a particular calibration is useable, but also how many individual calibration settings are required across the whole operating map in order to result in a coherent calibration strategy. Fewer changes in the parameter settings make for a more robust calibration set up across the whole operating map. For the work undertaken here an upper limit of 50% from the starting point values was considered for the increase in the response values, taking FSN from 0.75 to 1.125 FSN and NO\textsubscript{x} from 4.0 to 6.0 g/kWhr.

Initially considering the load sweeps at the two different engine speed levels. The data at 1600 rpm, as shown in Figure 6.3, demonstrates that the calibration holds from the starting point of 1.58 bar, which is test point A, to 2.30 bar BMEP before FSN values reach the upper limit. While the second load sweep at this speed shows that the load was reduced from a staring point of 8.45 bar, test point C, to 6.00 bar BMEP before NO\textsubscript{x} emissions became problematic and could furthermore be increased to around 9.00 bar BMEP without reaching the FSN upper limit. These two starting points are wide spread in terms of BMEP values and resulted in a large mid-range region of 3.70 bar, between 2.30 bar and 6.00 bar, where the existing calibrations could not be successfully used. However, a similar sweep starting from test point B at 5.50 bar BMEP, shown in Figure 6.3, would cover much of this region.

The load sweeps at 3400 rpm have similar trends as those described above for the lower speed load sweeps and are shown in Figure 6.6. This figure shows that the calibration can be used from the starting point of 1.58 bar, test point D, to approximately 2.75 bar BMEP before FSN values become overly large. The second sweep at this speed shows that the load was reduced from the starting point of 5.50 bar, which is test point E, to 3.00 bar BMEP before running into problems with NO\textsubscript{x} emissions and could be increased up to around 6.50 bar BMEP before reaching the FSN upper limit. These two starting points were closer in terms of BMEP values than the load sweep at 1600 rpm and as such a
smaller mid-range region of 0.25 bar, between 2.75 bar and 3.00 bar BMEP, remained where the existing calibrations could not be successfully used.

Considering the speed sweeps at the two load levels. The data for 1.58 bar BMEP, as shown in Figure 6.9, shows that the calibration holds from the starting point of 1600 rpm, test point A, to around 2100 rpm before FSN values become unacceptably large; this same calibration also holds down to 800 rpm without NOx emissions reaching the upper limit. The second speed sweep at this load was reduced from the starting point of 3400 rpm, which is test point D, to around 2700 rpm before NOx became a problem and was increased to 3800 rpm without FSN values increasing too much. There is therefore a mid-range region of 600 rpm at this load, between 2100 rpm and 2700 rpm, where the existing calibrations reached unacceptable levels of either FSN or NOx emissions.

The speed sweeps at 5.50 bar BMEP have similar trends to those described above for the lower load speed sweeps, as shown in Figure 6.12. This figure shows that if the same upper FSN and NOx limits are considered, then the calibration holds from the starting point of 1600 rpm, test point B, to around 2200 rpm, where the FSN level increase limits further adjustment; this same calibration also holds down to 1000 rpm without NOx emissions reaching the upper limit. Engine speed was reduced from the starting point of 3400 rpm, which is test point E, to around 2800 rpm without problems with increasing NOx emissions and up to a maximum value of around 3800 rpm before FSN again became problematic. This shows that there is a region of 600 rpm at this load level, between 2200 rpm and 2800 rpm, where the existing calibrations reached unacceptable levels of FSN or NOx emissions. These are very similar ranges and cut off points to those found for the speed sweeps at lower load.

In summary FSN deteriorates with both increased speed and increased load. Whilst NOx deteriorates with both decreased speed and decreased load. There is similar and systematic coverage of the operating map as speed and load are adjusted for fixed calibrations. The coverage of the speed and load operating map, or the extent to which fixed calibrations can be taken across the operating
map, with speed and load sweeps with upper limits of 50% placed on FSN and BS NO\textsubscript{x} variations is summarised by the schematic in Figure 6.15.

6.4 Concluding Remarks

Moving between different speed and load points with very different fuel injection and engine operating parameter strategies can cause difficulties in terms of calibrating an engine, especially when running under transient conditions. Using the coincident point near the optimum trade-off on the FSN-NO\textsubscript{x} map as a starting point, it was shown that the parameter settings can not be held constant if values of FSN and NO\textsubscript{x} emissions are to be maintained as speed or load are changed. These changes of the parameter settings are driven by the changing AFR values, as different amounts of fuel are injected as BMEP is varied, and by the time available for mixing and combustion as engine speed is varied. Increasing the amount of fuel injected reduces the relative levels of locally available O\textsubscript{2}, resulting in more fuel-rich zones, which affects the levels of mixing and combustion. While increasing the engine speed reduces the amount of time available for the O\textsubscript{2} present to mix with the injected fuel and combust before the expansion stroke progresses too far and the combustion ceases.

In order to maintain the FSN and NO\textsubscript{x} values at the coincident point, it was shown that boost pressure needed to be increased with both increasing speed and increasing load. Conversely, EGR levels needed to be reduced with both increasing speed and increasing load. Furthermore, main injection timing needed to be advanced with increasing engine speed and increasing load. It was found that split main ratio needed to remain at 90:10, except at the low load and speed point, test point A, where a ratio of 60:40 was recorded. A ratio of 90:10 at this test point would result in a reduction of FSN and an increase in NO\textsubscript{x} emissions, as can be seen in Figure 5.17. The trends across the operating map for the setting of split main separation showed that with increasing engine speed, larger separations were required; while no clear trends were evident for split main separation in terms of load variations. The variations of the parameter settings between different operating conditions are systematic. This
indicates that it is possible to interpolate between separately calibrated points on the operating map to predict the parameter setting requirements at untested operating conditions. It is less clear, however, from the work undertaken here whether the relationships between these parameter settings and the operating condition are linear or otherwise.

Moving along lines of constant speed and constant load while holding the parameter settings constant highlights the influence that speed and load have on the responses. This also shows how far a particular fixed calibration can be moved across the operating map before the responses become too large to be acceptable. The work carried out here highlights the effects that speed and load have on combustion and shows how much the speed and load can be adjusted before fixed calibrations get into trouble in terms of high emissions. This therefore shows how much of the operating map can be covered with a number of fixed calibrations. Increases in NO\textsubscript{x} or FSN values are seen as combustion changes occur as the operating map is traversed with fixed calibrations. FSN deteriorates with increased speed and increased load, whilst NO\textsubscript{x} deteriorates with decreased speed and decreased load. These changes are due to either the fuelling as load is altered or the available time for mixing and combustion as speed is adjusted. With increased load more injected fuel results in an increase in locally fuel-rich zones, even though the overall equivalence ratio may remain lean [32], leading to increased levels of FSN. As load is decreased and less fuel is injected the relative local O\textsubscript{2} concentration is increased, leading to an increase in NO\textsubscript{x} emissions. Increasing engine speed means that there is less time available for mixing and combustion of the injected fuel, which results in increased FSN values. FSN levels are also elevated with increased engine speed as there is less time available later in the cycle for oxidation of the soot formed. Whereas decreasing engine speed increases the residence times of the combustion gases, which means that more time is spent at higher temperatures to facilitate NO\textsubscript{x} formations.

The initial slow rate of change of these trends in FSN and NO\textsubscript{x} show that both speed and load can be adjusted without running suddenly into trouble with a particular calibration. As such, this allows some flexibility when it comes to
the accurate matching of calibration settings with speed and load. At all the points considered similar trends and effects were seen with the speed and load adjustments, highlighting whether a particular calibration is going to get into difficulty with either NOx or FSN as the operating map is traversed. In order to ascertain the coverage of the operating map an upper limit increase of 50% from the starting values of FSN and BS NOx at the coincident point was considered as the operating map was traversed with fixed calibrations. This took smoke values from 0.75 to 1.125 FSN and NOx from 4.0 to 6.0 g/kWhr. This showed how much of the operating map could be covered by the five separate calibrations at the five operating conditions.

For the speed sweeps, a mid-range region of 600 rpm across speed range was seen where the existing calibrations could not be used at both of the load levels considered. These gaps would be covered by separate calibrations at around 2500 rpm at 1.58 bar and 5.50 bar BMEP. For the load sweeps at 1600 rpm, a mid-load region of 3.70 bar BMEP across the load range was seen where the existing calibration would not work. The degree of coverage for load sweeps was systematic, showing a total coverage of the operating map in the region of 3.00 bar BMEP from each test point. A separate calibration at around 5.00 bar BMEP at 1600 rpm would bridge this calibration gap. Furthermore, for the loads sweeps at 3400 rpm, a mid-load region of only 0.25 bar across the load range was seen where the existing calibration would not work. The starting points for the load sweeps at 3400 rpm were closer in terms of BMEP values than was the case at 1600 rpm, which resulted in this smaller mid-load region. A calibration at around 5.00 bar BMEP at 3400 rpm, instead of the 5.50 bar starting point used, would eliminate this gap in the calibration. This would also be consistent with the need for a calibration at 5.00 bar BMEP at 1600 rpm, as was indicated above.

The 50% increases in FSN and NOx values considered here were probably too large to be used in a practical application. However, by setting the FSN and NOx upper limits to lower levels more calibration changes would be necessary, and would need to be investigated, as the operating map is traversed. There is a balance to be made between the upper limits, and hence acceptable penalties,
for NO\textsubscript{x} and FSN emissions as speed and load are varied with fixed parameter settings and the number of points on the operating map where detailed investigations need to be undertaken. This balance depends on each individual engine calibration project and the requirements and goals of that project. It is evident, however, that any measure applied to reduce the number of points on the operating map that need to be calibrated will reduce the amount of testing and investigations required overall.
CHAPTER 7

DISCUSSION ON HOW TO DEVELOP A COMMON FUEL INJECTION STRATEGY ACROSS THE OPERATING MAP

7.1 Introduction

Calibrating a modern high speed turbocharged diesel engine with external EGR and a high pressure common rail fuel injection system, with the capability of delivering multiple injection strategies, is a challenging task. This defines the settings of the parameters in an engine ECU which are central to meeting targets for engine pollutant emissions, fuel economy, torque output and other features of performance. The additional degrees of freedom introduced by HPCR injection systems, such as split main injection capability, add to the increasing complexity of defining calibrations. This can defeat an intuitive approach to optimization and can require extensive experimental mapping to resolve [75]. Traditional calibration methods involve a combination of steady state engine dynamometer work and transient vehicle testing [115]. The selection and optimization of a set of steady-state speed and load points allows some confidence that the complete vehicle will perform within regulated emissions limits and the process of optimizing this set of operating points is generally an iterative one [15]. The issues surrounding transient calibration methods have not been covered in this thesis beyond the suggestions made in Chapter 6 that there is a requirement to have smooth parameter variations across the speed and load operating map.

The purpose of the work undertaken in this Chapter was to highlight the advantages of using split main injection strategies over single injection strategies on the engine responses at each of the five operating conditions considered and highlight how a fuel injection strategy could be selected as a first step in the emissions calibration process. The aim was to fix certain parameter settings that show repeatable gains earlier in the process and therefore reduce the amount and the complexity of the testing required. Leading on from this, methods or guidelines have been introduced to help in
the complex process of calibrating and optimizing the large numbers of parameters, and degrees of freedom, present in modern diesel engines. Understanding the effects of these parameters over the full range of operating conditions could be exploited to limit the amount of testing needed, and the aim here is to examine this possibility.

7.2 Comparison of Multiple and Single Injection Strategies

In Chapter 5 DoE techniques were used with a half factorial CCP test matrix to investigate the five parameters with split main injections under consideration. For the work undertaken here another half factorial CCP test matrix was executed to investigate the related single injection strategies. The same values of ranges were used for main timing, boost and EGR level as were used for the multiple injection DoE from Chapter 5, while considering only a single injection strategy, and these are shown in Table 7.1.

A comparison between a single injection and a range of multiple injection strategies at test point A, 1600 rpm 1.58 bar BMEP, is shown in Figure 7.1. There is always a decrease in NO$_x$ due to less fuel being injected early in the cycle. With ratio decreasing from 90:10 to 60:40 NO$_x$ can be reduced at all separations. However, there is always an FSN penalty, which becomes worse with decreasing ratios from 90:10 to 60:40 for all separations. The FSN-NO$_x$ trade off here is evident, but a reduction in NO$_x$ can be made with no significant FSN penalty. Figure 7.2 shows the FC and HC emissions data and confirms that a split separation of 0° CA with a ratio of 90:10 gives the best results at this test point.

Figure 7.3 shows the comparison between single and multiple injection strategies at test point B, 1600 rpm 5.50 bar BMEP. NO$_x$ emissions are improved for every split main profile used here. FSN reductions can also be seen at the higher ratios with smaller separations. NO$_x$ and FSN emissions can be simultaneously reduced and the greatest improvements are seen with a close-coupled ratio of 90:10 at a separation of 0° CA. Figure 7.4 highlights the
FC and HC emissions data and again shows that a split separation of 0° CA with a ratio of 90:10 gives the best results at this test point.

At test point C, 1600 rpm 8.45 bar BMEP, both FSN and NOx can be simultaneously reduced as shown in Figure 7.5. This is best illustrated with small separations of 0° CA and high ratios of 90:10. FC and HC emissions data are shown in Figure 7.6 and also show that a split separation of 0° CA with a ratio of 90:10 gives the best results at this test point.

Figure 7.7 shows test point D, 3400 rpm 1.58 bar BMEP, and also shows that a reduction in NOx can be achieved without deterioration in FSN, but the FSN reductions are small and only occur with the large separations of 15° CA and high ratios of 90:10. FC and HC emissions data are shown in Figure 7.8 and at this increased engine speed show that a split separation of 15° CA with a ratio of 90:10 gives the best results.

The final operating point at 3400 rpm 5.50 bar BMEP, test point E, shown in Figure 7.9, shows that significant reductions in both NOx and FSN can be made simultaneously when employing split main injections. Figure 7.10 shows FC and HC emissions data and again highlights that at this increased engine speed that a split separation of 15° CA with a ratio of 90:10 gives the best results.

It is clear from the graphs in Figure 7.7 and Figure 7.9 that at the higher engine speed of 3400 rpm that larger split main separation is required. This agrees with the work in Chapter 5, which showed that an increase in separation at higher speeds reduced FSN values. The observations above show that it was possible to achieve lower NOx or FSN with split main injection strategies compared to single injection strategies at the operating points considered here. However, it was not always possible to achieve simultaneous reduction of these two species as Corcione et al also found [2]. It is also clear that a ratio of 90:10 gave a better FSN-NOx trade-off at each point compared to moving towards a 60:40 ratio. The smallest separations were required at low engine speeds, while at higher engine speeds split main separation needed to be increased to maintain the best FSN-NOx trade-off.
There are a number of ways that split main injection strategies reduced the levels of FSN produced. Firstly, the reduction in soot with split main injections stemmed from the reduced quantity of fuel in the first injection event; it is this fuel that primarily gives rise to soot production even when the two injections are close-coupled [75]. Split main injections are effective at controlling the local equivalence ratio, \( \phi \), to lower values than are possible with single injections [12]. This meant that there was more \( O_2 \) locally available, which resulted in better mixing of the injected fuel and air. The improve mixing of the fuel and air results in the mixture becoming locally leaner, which reduces the high soot forming regions [19]. The timing, or separation, of the second injection is important for soot development because it dictates whether it would strike the soot producing fuel rich regions at the spray tip [19] from the first injection. Also the second injection entered a more turbulent fuel and air mixture in the combustion chamber, which resulted in the improved mixing of the second injection and more effective combustion. Furthermore, a sufficiently long dwell time between injection pulses prevented the fuel rich regions at the spray tip being replenished, which allows the soot previously formed by the first injection to be oxidised [19]. With the split main injection strategies, however, the second injection was entering the cylinder later in the cycle and could therefore result in increased levels of FSN and HC emissions if it was very late in the cycle and did not therefore fully combust.

\( NO_x \) emissions are strongly linked to in-cylinder temperature and pressure and the availability of \( O_2 \) in the combustion chamber. However, in most cases the temperature is the critical factor as \( NO_x \) production is an exponential function of combustion temperature [53]. Furthermore, when \( O_2 \) is not a constraint the rate of \( NO_x \) production is a function of the fraction of gas which is at a temperature above 2650 K [75]. The use of split main injections pushed some of the fuel later in the cycle, which reduces the peak in-cylinder temperatures resulting in reduced \( NO_x \) levels [19]. For \( NO_x \) production the first injection was again important; when the split main was close-coupled, the second injection extended the period over the cycle during which high temperatures were seen, which increased \( NO_x \) production. When the separations were large, the second
injection came too late in the expansion stroke to produce more of the high temperature gas associated with NO\textsubscript{x} production [75].

The levels of possible boost pressure and EGR that can be used change with the use of split main injection strategies; there is increased tolerance to both boost pressure and EGR level. The reduction in FSN values seen allows the addition of higher levels of EGR, which in turn helps to reduce NO\textsubscript{x} emissions. The reduction in NO\textsubscript{x} emissions seen allows higher boost pressures to be used, which then helps to further reduce the FSN levels. Furthermore, a split main injection strategy means that earlier main injection timings are possible as NO\textsubscript{x} emissions are reduced. An earlier injection would give more time for mixing and combustion and would reduce FSN levels further. The boost pressure used is limited by the peak in-cylinder pressures allowable and by the ability of the turbocharger to deliver pressures at each operating condition. The use of higher boost pressures is advantageous as it opens up the possibility of down-siting an engine for a given power output, which can reduce FC penalties and can reduce engine build costs.

7.3 Calibration of Parameter Settings

In Chapter 6 it was shown that the fuel injection and engine operating parameter settings could not remain unchanged as the speed and load operating map was traversed whilst maintaining NO\textsubscript{x} and FSN responses at fixed levels, and the reasons for this were given. Various approaches to define a steady state calibration strategy can be adopted [116, 117]. Each suggests a slightly different "pecking order" for the selection of the parameter adjustments, but the approach is to set a hierarchy of parameter adjustments that need to be made [118]. Each parameter needs to be fully optimized with iterations of each parameter in a loop to make certain that each is optimized with respect to the other parameters present and that none is fixed in isolation. There is not a "set in concrete" methodology for calibrating a diesel engine, but there are certain parameters that clearly need setting or optimizing before others as they are more unstable. These therefore need nailing down first so that following parameter adjustments are on a stable base. Pressure waves in the rail are a
restricting instability, as was shown in Chapter 4, and therefore the separation of pilot or split main may need to be determined early so that these waves do not cause problems for the setting of other parameters. Fixing some parameters early allows the number of variables under consideration at any one time to be minimised, but can reduce the possibility of locating the global optimum parameter settings [118]. The pecking order for the iterations outlined by Capon and Rottger [116, 117] can be brought together as: main injection profile and timing, pilot separation and quantity, rail pressure, boost pressure and EGR. Transferring this to the parameters considered in the experimental work undertaken in this thesis would suggest firstly determining the split main injection profile and then setting the main injection timing, followed by boost pressure and then EGR.

Set against the background outlined above, the following steps would reduce the amount of experimental work needed to cover the calibration requirements across the operating map and simplify the process. The first step would be to select sparsely populated test points on the speed and load operating map; an indication of how much of the operating map can be covered by individual calibrations at each operating condition was given in Chapter 6. It was shown in the comparison of multiple and single injection strategies in this chapter that at the operating conditions considered a split main ratio of 90:10 gave the best trade-off in terms of NOx and FSN emissions and also gave good results for both FC and HC emissions. The next step would therefore be to set the split main ratio to 90:10 for all operating conditions. Following this it would be necessary to set the split main separations; the requirement being for close coupled injections at low engine speeds; separations then need to be increased with increased engine speed. Then DoE testing of the remaining parameters would be recommended; in the case of the work undertaken in this thesis these remaining parameters would be main injection timing, boost pressure and EGR. The DoE testing would be undertaken instead of the iterative approach suggested by other calibration guidelines [15, 116, 117, 118], as an iterative approach may only detect local minima rather than the required global minimum or optimum trade-off for the responses as the parameters are adjusted. With just three parameters in this DoE, the amount of experimental
work would be greatly reduced from the 27 test points seen in Chapter 5 to 15 test points at each operating condition, if a similar design of CCP test matrix was considered.

The calibration steps above would be carried out at each of the test points on the operating map with the DoE techniques used in Chapter 5 and this would allow the parameter settings at a required point on the FSN-NO\textsubscript{x} map at each operating condition to be established. The information from Chapter 6, concerning the changes in parameter settings as speed and load are adjusted, would then allow a good prediction for the parameter settings to be made at operating conditions where experimental work had not been undertaken. If it is then found that the calibration at a certain untested operating condition is not as required then the information from Chapter 5, concerning the effects of individual parameters on the engine responses, would then allow for adjustments to be made in order to bring the calibration into line whilst also highlighting any penalties that would be associated with such changes in parameter settings. There are other responses beyond NO\textsubscript{x} and FSN that need to be taken into consideration when deciding whether a calibration is acceptable or not; namely fuel consumption, HC, CO and CO\textsubscript{2} emissions. It can be seen with the work undertaken in Chapter 6 that as the operating map is traversed with fixed calibrations the variations in FSN and NO\textsubscript{x} emissions are quite predictable and that if these two responses are well calibrated then the other responses should also be within acceptable values. A poor calibration that has high FSN values is also likely to have high CO, CO\textsubscript{2} and HC emissions as well as a fuel consumption penalty.

7.4 Concluding Remarks

The advantages of using split main injection over single injection strategies have been shown in this Chapter at the five speed and load operating conditions considered. For the main injection timing, boost pressure and EGR settings established at the coincident point, either FSN or NO\textsubscript{x} emissions could be reduced by using split main injection strategies. While at the higher load operating conditions it was possible to reduce both of these species with the
introduction of split main injection strategies. Consistently, split main injections with ratios of 90:10 gave the best trade-off on the FSN-NO$_x$ map both over single injections and split main injections with lower ratios (towards 60:40) at all operating conditions. At lower engine speeds, the smallest possible split main separations were required and as the speed was increased it was necessary to increase the separation. The FSN reductions seen with the use of split main injections allow for the possibility of using higher EGR levels, which in turn reduces the levels of NO$_x$ emissions produced, and which then allows for higher levels of boost to be used. Higher boost pressure is favourable as it reduces fuel consumption and increases the power produced by an engine. Increased power output then opens up the possibility of either downsizing a particular engine, which has potential savings for manufacture and production, or charging the customer more for the more powerful, higher performance engine in the vehicle they purchase.

A further intention of the work undertaken in this Chapter was to present some methods that could be used to simplify the process of establishing a steady state emissions and fuel consumption calibration. In Chapter 5 the effects that individual parameters had on the measured responses was highlighted. How parameter settings need to be adjusted over the operating map and the influence that engine speed and load had on the responses was considered in Chapter 6. Here the advantages of using split main rather than single injection strategies have been shown. And so by bringing this information together, this work has shown a way to reduce the amount of experimental work needed to calibrate an HPCR DI engine, which can be summarised as follows:

- Pick sparsely populated points on operating map; Chapter 6 gave an indication of how much of the operating map can be covered by individual calibrations at each operating condition.

- Set split main ratio to 90:10 at each speed and load test point and set split main separation as close coupled at low engine speeds and increase the separation with increased engine speed, as indicated by the work undertaken in this chapter.
• Conduct DoE testing at each of the speed and load test points for the remaining parameters, which would involve a reduced amount of testing with split main separation and ratio settings already fixed.

• From the DoE models the parameter settings at the required calibration point on the FSN-NO\textsubscript{x} map would then be defined.

• Make initial predictions for parameter settings at operating conditions that have not been tested, by considering the trends for the parameter settings across the operating map seen in Chapter 6 and interpolating between the parameter settings found from the DoE testing.

• Run these predicted points experimentally to establish the engine out responses. It would then be possible to make parameter adjustments as necessary to reach the required calibration point on the FSN-NO\textsubscript{x} map by considering the individual parameter effects, and the associated penalties, that were shown in Chapter 5.
DISCUSSION AND CONCLUSIONS

8.1 Discussion

The investigations reported in this thesis deal with the questions associated with calibration, that is the process of setting values for the fuel injection parameters, EGR level and boost pressure which best meet performance and emissions targets for the engine. The understanding of direct injection diesel engine combustion at different points on the operating map and how high pressure common rail fuel injection systems can be utilised to best effect support the development of the process. The investigations reported highlight the benefits that this technology, with multiple injection capabilities, can offer in terms of manipulating levels of exhaust emissions and fuel economy.

8.1.1 Damping of Pressure Oscillations in FIE System

The work outlined in this thesis was carried out on a single cylinder engine and so no experimental work was carried out to ascertain the level of impact of the fuel pressure variations on other injectors and cylinders. It is clear, however, that any techniques that reduce fuel pressure variations in the single cylinder engine would also have a beneficial impact on the fuel pressure and fuel delivery variations at the other injectors on a multi-cylinder common rail engine. Furthermore, the techniques outlined in Chapter 4 to establish the relationship between needle lift, injection pressure and the quantity of fuel injected could be applied to any fuel injection system.

The damping method developed to address the problems of pressure wave variations in the high pressure common rail fuel injection system is novel and a patent application for this has been filed [40]. The method used a second injection event close to the initial one in order to produce a second pressure oscillation to interfere with and cancel out the original oscillation produced by the operation of the injector. The second close sub-injection was positioned at 5.5° CA after the start of the initial injection at an engine speed of 1600 rpm.
This dampened the oscillations and gave independence to the positioning of the second split main injection in terms of the variation of fuel injected. Variations in injection pressures directly led to variations in the quantity of fuel injected due to the pressure differential variations across the injector. A further problem was the influence that the pressure variations had on the injector function and therefore the needle lift trace. Variations in the needle lift trace were seen at different injection pressures, which also had the effect of altering the quantity of fuel delivered by the injector.

The damping method outlined above potentially has a number of drawbacks. An increased number of injection events will lead to greater injector wear and hence shortened injector life on the engine. Also, an increased number of injection events can result in increased HC emissions due to fuel entering the cylinder from the injector nozzle sac volume at the end of each injection event [32]. This fuel enters the cylinder at low velocities and mixes poorly and may escape the primary combustion process.

### 8.1.2 Parameter Variations at Fixed Operating Points

A half factorial CCP test matrix was designed and DoE techniques were successfully used explore the response space in terms of the parameter settings at each of the points investigated on the speed and load operating map. The use of DoE made it possible to reduce the amount of testing required for the five parameters at each of the operating points considered by 89%, from 243 to 27 test points, whilst maintaining an accuracy of at least 10%. Non-linear second order polynomial equation models for the responses under consideration at each of the five operating points were produced. These models allow the user to explore the response spaces and to ascertain the parameter settings required to attain required response values.

The analysis of the sensitivities of NO\textsubscript{x} and FSN to the individual parameter variations were consistent at the speed and load operating conditions considered. This indicates how the parameters need to be adjusted in order to reduce FSN and NO\textsubscript{x} emissions, towards the optimum trade-off on the FSN-NO\textsubscript{x} map, at each of the operating conditions. It also highlighted any HC
emissions and fuel consumption penalties that were present. The sensitivity analysis highlighted which parameters had the largest effects on the responses at the different points on the operating map. EGR has the greatest impact on NO\textsubscript{x} emissions and generally has the second largest influence on FSN after boost pressure. Split main ratio and separation generally have less influence than the other parameters considered. The benefits achieved when using EGR to reduce NO\textsubscript{x} generally outweigh the FSN penalties seen. Increasing boost pressure generally reduces FSN values more than the associated NO\textsubscript{x} penalty away from low load conditions. The NO\textsubscript{x} penalties associated with increased boost pressure are outweighed by the possible reductions with the addition of EGR and the retarding of the main injection timing.

Combining the sensitivities of the FSN and NO\textsubscript{x} responses to adjustments of each parameter gives a measure of the linearity of parameter effects from a fixed point on the FSN-NO\textsubscript{x} map. This linearity shows how much a parameter can be adjusted before the behaviour of the parameter on the FSN-NO\textsubscript{x} map becomes unpredictable. This work showed that adjustments of certain parameters have to be treated with great care, as the effects on the responses across their full range of values cannot be accurately extrapolated from small adjustments. This shows which parameters are the most unpredictable, having the least linear behaviour in terms of FSN and NO\textsubscript{x} variations, and hence are the most constricting when it comes to adjustments of the parameters on the FSN-NO\textsubscript{x} map. EGR is the most linear at each of the operating points considered, followed by main injection timing; while boost pressure generally has the least linear behaviour.

The advantages of split main injection strategies over single injection strategies were highlighted at each of the speed and load operating points. It was shown that a split main ratio of 90:10 generally gave the best trade-off in terms of FSN and NO\textsubscript{x} emissions. A single injection strategy generally resulted in higher NO\textsubscript{x} emissions and lower FSN values, while reducing the ratio towards a 60:40 resulted in higher FSN values but lower NO\textsubscript{x} emissions. From this information it can be inferred that a split main ratio up to 95:5 could be the best trade-off in terms of both NO\textsubscript{x} and FSN reduction across the operating map.
Split main separation needed to be kept to the smallest setting at the low engine speed operating points of 1600 rpm. The separation was increased with increased speed in order to maintain the best FSN and NO<sub>x</sub> trade-off. The sensitivity analysis indicates that even larger split main separations, larger than the 15° CA used, may be required at the higher engine speed of 3400 rpm to give an improved trade-off on the FSN-NO<sub>x</sub> map.

8.1.3 Influence of Engine Speed and Load Variations

Increases in NO<sub>x</sub> or FSN are seen as combustion changes occur as the operating map is traversed with fixed parameter settings. FSN deteriorates with increased engine speed and load, whilst NO<sub>x</sub> deteriorates with decreased engine speed and load. These changes are due to either the fuelling, and hence AFR, as load is altered or the available time for mixing and combustion as speed is altered. At all the operating points considered similar systematic effects were seen, which allows a calibrator to have a clear idea of what will happen to the responses as the operating map is traversed with fixed parameter settings. This work led onto the idea of the coverage of the operating map with fixed calibrations with 50% upper limits on the NO<sub>x</sub> and FSN starting values. The interest was to see how far a fixed calibration could be moved across the operating map before these upper limits were reached. It was not anticipated that the five speed and load points would cover the whole of the operating map, but this work showed that there was systematic coverage of the operating map with speed or load adjustments.

Adjustments in the parameter settings needed to maintain FSN and NO<sub>x</sub> values at the coincident point, near the optimum trade-off on the FSN-NO<sub>x</sub> map, show systematic trends of these parameter settings with both speed and load variations. This allows the prediction of the parameter settings to be undertaken as the operating map is traversed. At low speed and light load operating conditions, the timing of fuel injection needs to be relatively retarded, boost pressure needs to be low and high levels of EGR can be used. The high AFR values found at low load points means that smoke production is not a problem and so there is plenty of scope for the addition of EGR for the reduction of NO<sub>x</sub> emissions. Boost is not a requirement at low loads as it is normally associated
with increasing the power output. Also at low load conditions increasing boost can lean out the air and fuel mixture too much resulting in poor combustion with high HC emissions and an associated fuel consumption penalty. High HC emissions tend to occur at lower load conditions while high FSN values tend to occur at higher loads. At lower loads the in-cylinder temperatures are lower and fuel entering the combustion chamber late in the cycle is therefore not burned resulting in HC emissions. While at higher loads this fuel is burned or partially burned, due to higher in-cylinder temperatures which continue later in the cycle, resulting in increased FSN levels. As engine speed and load are increased main injection timing needed to be advanced, boost pressure needed to be increased and EGR levels needed to be reduced. These calibrations are used in order to keep NO\textsubscript{x} emissions low without incurring large FSN penalties. It was found that split main ratio could remain unchanged at 90:10 across the operating map. Split main separation had to be increased with increased engine speed, but was not dependant on engine load.

8.1.4 Calibration Techniques Across the Operating Map

Traditionally the calibration of diesel engines was relatively straightforward as older generation engines had fewer controllable parameters. The level of complexity of modern diesel engines has been steadily driven up by increasingly strict emissions regulations and customer demands for a driving experience more akin to that of a gasoline engine. The increased level of complexity can defeat the traditional intuitive approach to calibration and can require extensive experimental mapping to resolve.

In Chapter 6 the influence that engine speed and load had on the responses was considered. It was also shown how parameter settings could be predicted across the operating map and that a good first estimate for these parameter settings could be made at points on the operating map that had not been experimentally investigated. These predictions would be a good starting point for investigating or calibrating the untested operating points and as such would reduce the amount of testing required to characterise a particular operating point. The sensitivity analysis in Chapter 5 and the multiple and single injection strategy comparisons in Chapter 7 allow the calibrator to adjust the individual
parameter settings at the fixed operating point in order to maintain response levels at or below required values. With this knowledge about effects of individual parameters, adjustments can be made to the first predicted calibration in order to adjust the responses as required, while having a clear idea of the associated penalties. A methodology to establish a steady state emissions calibration strategy was detailed in Chapter 7, which brought together the experimental work from the previous chapters. This showed ways of reducing the time consuming process of running full DoE explorations at each new operating condition, by predicting initial parameter settings and fixing the fuel injection parameters earlier to reduce the size of the experimental test matrix.

8.2 Suggestions for Further Work

Compared to a conventional rotary pump system for diesel fuel injection, like the York rotary distribution fuel pump initially used on the single cylinder test facility, the HPCR system provides a more finely atomised fuel spray. This finely atomised spray allows better mixing of the charge air and fuel to occur, which greatly reduces soot, but can also result in an increase in NOx emissions. From work undertaken by Needham and Bouthenet [119] it is clear that higher injection pressures in the region of 1000 to 1200 bar give FSN improvements, but there is an adverse affect on NOx emissions, as was shown in Figure 5.2, due to higher in-cylinder peak pressures and temperatures. By adjusting other parameters, such as EGR and main injection timing, it should be possible to maintain NOx values while reducing the levels of FSN produced with these increased injection pressures. The majority of the work undertaken in this thesis was carried out with an injection pressure of only 850 bar, which is low compared to the 1000 to 1800 bar typically achieved by common rail systems [28, 48, 119, 120]. It would be of great interest to re-run some of the split main injection work carried out in this thesis with higher injection pressures. The advantages of split main over single injection strategies should be further increased with increased injection pressure by emphasising the improved mixing of the injected fuel and charge air already seen with the use of split main injections. There would probably be a fuel consumption penalty present
with increased injection pressures, however, as the high pressure fuel pump must work harder, which is an increased load on the engine.

Good use could be made of the pressure waves present in the common rail caused by the action of the injector that were highlighted in Chapter 4. It may be possible to utilise the unexpectedly high pressures at the peaks of these oscillations to give higher than expected injection pressures. The experimental work needed to investigate this possibility was not carried out here, but it is clear that a compromise between hitting the peak injection pressures and position of the second split main injection, and therefore split main separation, would need to be investigated.

It would appear from the work undertaken in this thesis that one approach to the calibration issues raised could be to accept a NO\(_x\) emissions penalty at each point on the speed and load operating map and this would be an interesting area of research. Accepting such a NO\(_x\) hit would result in reduced FSN and HC emissions, depending on the operating condition, and improved FC. The focus would then be on finding an aftertreatment solution to resolve the increased NO\(_x\) emissions in the exhaust gases. This is already a challenging issue as NO\(_x\) formations occur at very high temperatures and therefore any reduction of this species must also occur at these very high temperatures, which makes the catalyst environment difficult to achieve. The author acknowledges that this is against the trend of using diesel particulate filters in the aftertreatment of diesel exhaust emissions in the automotive industry. There is some logic however in reducing the number of separate species that need to be dealt with by aftertreatment systems, which are already costly, while reducing FC in order to attain legislated emissions limits.
8.3 Conclusions

8.3.1 Damping of Pressure Oscillations in FIE System

- A relationship between needle lift, fuel injection pressure and quantity of injected fuel was established.
- A method to dampen the oscillations in the high pressure fuel in the common rail system was developed and applied.
- A significant reduction in the variation of the quantity of fuel delivered in the second part of the split main injection, as split main separation was varied, was demonstrated using this hydraulic damping method. This gave independence to the positioning of the second split main injection event.
- A patent was developed from this pressure variation damping work which has been filed with the Patent Office in association with the Ford Motor Company [40].

8.3.2 Parameter Variations at Fixed Operating Points

- The effects of the individual parameters on the measured responses at the different points on the operating map were highlighted with the data from the normal plots and confirmed with the sensitivity analysis. This shows the parameter adjustments needed to move to a required point on the FSN-NO\textsubscript{x} map and hence how to move closer to the optimum trade-off point. The associated penalties for the adjustment of each parameter in terms of HC emissions and FC were also highlighted. The parameter effects can be summarised as follow:
  - NO\textsubscript{x} reduction achieved with an increase in split main separation, a reduction in split main ratio (towards a 60:40), more retarded injection timing, decreased boost pressure and increased EGR.
  - FSN reduction seen with a decrease in split main separation, an increase in split main ratio (towards a 90:10), more advanced injection timing, increased boost pressure and decreased EGR.
  - HC emissions are most sensitive at low load conditions where incomplete combustion occurs and the low load means that the in-cylinder temperatures are not sufficiently high to oxidise any unburned fuel later in the cycle. Fuel consumption is also most
sensitive at low loads due to incomplete combustion and because with lower loads, engine friction becomes an increasingly large component of the power output of the engine. The methods outlined above for reducing FSN values can also be used to address both high HC emissions and FC.

- Split main injection strategies can always result in a reduction in NO\textsubscript{x} emissions compared to single injection strategies at different operating points with fixed main injection timing, boost pressure and EGR values. Furthermore, moving from a ratio of 90:10 to 60:40 generally reduces NO\textsubscript{x} emissions further.

- A split main injection strategy can reduce FSN values when compared to single injection strategies except at low loads where an FSN penalty is generally seen. Furthermore, reducing the ratio from 90:10 to 60:40 consistently increases FSN values.

- With the introduction of split main injection strategies it is always possible to achieve a reduction in FSN or NO\textsubscript{x} emissions, and at the higher load operating conditions it was possible to reduce these species simultaneously.

- The best compromise, in terms of FSN and NO\textsubscript{x} emissions, is therefore achieved by applying a split main ratio of 90:10 at all operating points for given main injection timing, boost pressure and EGR levels. At low engine speeds the smallest split main separation possible should be used and this then needs to be increased with increased speed.

8.3.3 Influence of Engine Speed and Load Variations

- Systematic trends were identified for the setting of the parameters at the different points on the operating map while keeping FSN and NO\textsubscript{x} values near the optimum trade-off on the FSN-NO\textsubscript{x} map. The following parameter adjustments were needed:
  - EGR needed to be decreased, boost increased and main injection timing advanced with increased speed and increased load.
  - Split main separation needed to be increased with increased speed, but separation did not need adjustment with changing load.
  - Split main ratio was generally left unchanged at 90:10 at the different operating points.
The effects of load and speed changes on NO\textsubscript{x} or FSN with fixed parameter settings were examined. FSN increased with increasing speed and with increasing load. NO\textsubscript{x} emissions increased with decreasing speed and with decreasing load. In general, NO\textsubscript{x} emissions increased as FSN decreased, and vice versa, when load or speed were changed.

8.3.4 Calibration Strategies

The individual parameter effects were investigated in Chapter 5, the influence of speed and load on parameter settings and engine out responses was looked at in Chapter 6 and in Chapter 7 the comparison of split main injection and single injection strategies was considered. Bringing this information together made it possible to outline a methodology to simplify the steady state emissions calibration process across the speed and load operating map. Ways to reduce the amount of experimental work needed to calibrate an HPCR DI engine with a split main injection strategy were also shown.


[16] Intergovernmental Panel on Climate Change, “Climate Change 2001”, UN Framework Convention on Climate Change, Published by UNEP and UNFCC, Edited by Michael Williams, July 2002.


<table>
<thead>
<tr>
<th>Level Year</th>
<th>EU Directive</th>
<th>CO (g/km)</th>
<th>NOx (g/km)</th>
<th>HC (g/km)</th>
<th>HC + NOx (g/km)</th>
<th>PM (g/km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Euro I 1992</td>
<td>91/44/EEC</td>
<td>2.72</td>
<td>-</td>
<td>-</td>
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<td>0.14</td>
</tr>
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<td>Euro II - IDI 1997</td>
<td>94/12/EC</td>
<td>1.00</td>
<td>-</td>
<td>-</td>
<td>0.70</td>
<td>0.08</td>
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<td>Euro II - DI 1997</td>
<td>94/12/EC</td>
<td>1.00</td>
<td>-</td>
<td>-</td>
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<td>0.04</td>
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<td>0.010</td>
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Table 1.1 – European Union emissions limits for diesel passenger cars [13, 14, 15, 18].
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<thead>
<tr>
<th>Solids</th>
<th>Liquids</th>
<th>Gases</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soot:</td>
<td>Hydrocarbons</td>
<td>Nitric oxide (NO)</td>
</tr>
<tr>
<td>- carbon nuclei</td>
<td>(soluble organic fraction - SOF):</td>
<td>Nitrogen dioxide (NO₂)</td>
</tr>
<tr>
<td>- agglomerated carbon particles</td>
<td>fuel derived</td>
<td>Hydrocarbons (HC)</td>
</tr>
<tr>
<td>Sulphates</td>
<td>- lube oil derived</td>
<td>Carbon monoxide (CO)</td>
</tr>
<tr>
<td>Ash:</td>
<td>Sulphuric acid</td>
<td>Carbon dioxide (CO₂)</td>
</tr>
<tr>
<td>- oil additives</td>
<td>Water</td>
<td>Water (H₂O)</td>
</tr>
<tr>
<td>- engine wear products</td>
<td></td>
<td>Oxygen (O₂)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Nitrogen (N₂)</td>
</tr>
</tbody>
</table>

Table 2.1 – The components of diesel engine exhaust [15].
Table 4.1 - Summary of fundamental frequencies and higher order harmonics caused by high pressure pump and rail pressure regulator for a motored test and values used in calculating the related crank angle values in the cycle.
### Table 4.2 - Summary of fundamental frequencies, harmonics, equation definitions and lengths of pipes used in calculations.

<table>
<thead>
<tr>
<th>Dimensions (mm)</th>
<th>Closed - Open Boundary</th>
<th>Closed - Closed Boundary</th>
</tr>
</thead>
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<tr>
<td>taken from fuel injection</td>
<td>135 M</td>
<td>135 M</td>
</tr>
<tr>
<td>system</td>
<td>60 D</td>
<td>60 D</td>
</tr>
<tr>
<td>schematic</td>
<td>90 E</td>
<td>90 E</td>
</tr>
<tr>
<td>Figure 4.11</td>
<td>65 F</td>
<td>65 F</td>
</tr>
<tr>
<td></td>
<td>62 G</td>
<td>10.5 J</td>
</tr>
<tr>
<td></td>
<td>10.5 J</td>
<td>9 K</td>
</tr>
<tr>
<td>Total length, l (m)</td>
<td>0.4225</td>
<td>0.4315</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fundamental c / (4 x l)</td>
<td>852</td>
</tr>
<tr>
<td>First harmonic (3 x c) / (4 x l)</td>
<td>2556</td>
</tr>
<tr>
<td>Second harmonic (5 x c) / (4 x l)</td>
<td>4260</td>
</tr>
<tr>
<td>Third harmonic (7 x c) / (4 x l)</td>
<td>5964</td>
</tr>
<tr>
<td>Fourth harmonic (9 x c) / (4 x l)</td>
<td>7669</td>
</tr>
<tr>
<td>Fundamental c / (2 x l)</td>
<td>1669</td>
</tr>
<tr>
<td>c / l</td>
<td>3337</td>
</tr>
<tr>
<td>(3 x c) / (2 x l)</td>
<td>5006</td>
</tr>
<tr>
<td>(4 x c) / (2 x l)</td>
<td>6674</td>
</tr>
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<td>Test Runs</td>
<td>Split Main Separation</td>
</tr>
<tr>
<td>-----------</td>
<td>-----------------------</td>
</tr>
<tr>
<td>1</td>
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</tr>
<tr>
<td>27</td>
<td>0</td>
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</table>

Table 5.1 – Half factorial CCP test matrix for 5 parameters showing the 27 required experimental test runs.
<table>
<thead>
<tr>
<th>Test Point A: 1600 rpm 1.58 bar BMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Split Separation - MS</strong>&lt;br&gt; (CA)</td>
</tr>
<tr>
<td><strong>Split Ratio - MR</strong>&lt;br&gt; (1st:2nd)</td>
</tr>
<tr>
<td><strong>Injection Timing - MT</strong>&lt;br&gt; (CA BTDC)</td>
</tr>
<tr>
<td><strong>Boost Pressure - BO</strong>&lt;br&gt; (bar)</td>
</tr>
<tr>
<td><strong>EGR - EG</strong>&lt;br&gt; (%)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point B: 1600 rpm 5.50 bar BMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Split Separation - MS</strong>&lt;br&gt; (CA)</td>
</tr>
<tr>
<td><strong>Split Ratio - MR</strong>&lt;br&gt; (1st:2nd)</td>
</tr>
<tr>
<td><strong>Injection Timing - MT</strong>&lt;br&gt; (CA BTDC)</td>
</tr>
<tr>
<td><strong>Boost Pressure - BO</strong>&lt;br&gt; (bar)</td>
</tr>
<tr>
<td><strong>EGR - EG</strong>&lt;br&gt; (%)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point C: 1600 rpm 8.45 bar BMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Split Separation - MS</strong>&lt;br&gt; (CA)</td>
</tr>
<tr>
<td><strong>Split Ratio - MR</strong>&lt;br&gt; (1st:2nd)</td>
</tr>
<tr>
<td><strong>Injection Timing - MT</strong>&lt;br&gt; (CA BTDC)</td>
</tr>
<tr>
<td><strong>Boost Pressure - BO</strong>&lt;br&gt; (bar)</td>
</tr>
<tr>
<td><strong>EGR - EG</strong>&lt;br&gt; (%)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point D: 3400 rpm 1.58 bar BMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Split Separation - MS</strong>&lt;br&gt; (CA)</td>
</tr>
<tr>
<td><strong>Split Ratio - MR</strong>&lt;br&gt; (1st:2nd)</td>
</tr>
<tr>
<td><strong>Injection Timing - MT</strong>&lt;br&gt; (CA BTDC)</td>
</tr>
<tr>
<td><strong>Boost Pressure - BO</strong>&lt;br&gt; (bar)</td>
</tr>
<tr>
<td><strong>EGR - EG</strong>&lt;br&gt; (%)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point E: 3400 rpm 5.50 bar BMEP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Split Separation - MS</strong>&lt;br&gt; (CA)</td>
</tr>
<tr>
<td><strong>Split Ratio - MR</strong>&lt;br&gt; (1st:2nd)</td>
</tr>
<tr>
<td><strong>Injection Timing - MT</strong>&lt;br&gt; (CA BTDC)</td>
</tr>
<tr>
<td><strong>Boost Pressure - BO</strong>&lt;br&gt; (bar)</td>
</tr>
<tr>
<td><strong>EGR - EG</strong>&lt;br&gt; (%)</td>
</tr>
</tbody>
</table>

Table 5.2 – DoE testing ranges for fuel injection and engine operating parameters for multiple injection investigation.
Table 5.3 – Summary of individual parameter effects on NO\textsubscript{x} and FSN across full range of parameter values, taken from normal plot data.
### Table 5.4 - Summary of individual parameter effects on HC and FC across full range of parameter values, taken from normal plot data.

<table>
<thead>
<tr>
<th>BMEP / N</th>
<th>1600</th>
<th>3400</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>HC</strong></td>
<td></td>
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</tr>
<tr>
<td>8.45</td>
<td>[↑↑↑↑↑]</td>
<td>[↑↑↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
<tr>
<td>5.50</td>
<td>[↑↑↑↑↑]</td>
<td>[↑↑↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
<tr>
<td>1.58</td>
<td>[↑↑↑↑↑]</td>
<td>[↑↑↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
<tr>
<td><strong>FC</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.45</td>
<td>[↑↓↑↑↑]</td>
<td>[↑↓↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
<tr>
<td>5.50</td>
<td>[↑↓↑↑↑]</td>
<td>[↑↓↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
<tr>
<td>1.58</td>
<td>[↑↓↑↑↑]</td>
<td>[↑↓↑↑↑]</td>
</tr>
<tr>
<td></td>
<td>MS MR MT BO EG</td>
<td>MS MR MT BO EG</td>
</tr>
</tbody>
</table>

MS overall from 0° to 15° CA  
MR overall from 60:40 to 90:10  
MT overall from 12° to -2° CA BTDC  
BO overall from 0 to 0.9 bar  
EG overall from 0 to 45%
<table>
<thead>
<tr>
<th>Test Point A: 1600 rpm 1.58 bar BMEP</th>
<th>Range of Linearity</th>
<th>Coverage of range (%)</th>
<th>Linearity Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS °CA</td>
<td>0.0 to 10.0</td>
<td>10.0</td>
<td>100.0</td>
</tr>
<tr>
<td>MR 1st</td>
<td>60.0 to 80.9</td>
<td>20.9</td>
<td>69.7</td>
</tr>
<tr>
<td>MT °CA BTDC</td>
<td>-1.5 to 6.0</td>
<td>7.5</td>
<td>93.8</td>
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<tr>
<td>BO bar</td>
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<td>0.4</td>
<td>78.0</td>
</tr>
<tr>
<td>EG %</td>
<td>10.0 to 45.0</td>
<td>35.0</td>
<td>100.0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point B: 1600 rpm 5.50 bar BMEP</th>
<th>Range of Linearity</th>
<th>Coverage of range (%)</th>
<th>Linearity Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS °CA</td>
<td>0.0 to 5.8</td>
<td>5.8</td>
<td>57.5</td>
</tr>
<tr>
<td>MR 1st</td>
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<td>17.1</td>
<td>57.0</td>
</tr>
<tr>
<td>MT °CA BTDC</td>
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<td>100.0</td>
</tr>
<tr>
<td>BO bar</td>
<td>0.3 to 0.51</td>
<td>0.2</td>
<td>42.0</td>
</tr>
<tr>
<td>EG %</td>
<td>8.6 to 25.6</td>
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<td>68.0</td>
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<table>
<thead>
<tr>
<th>Test Point C: 1600 rpm 8.45 bar BMEP</th>
<th>Range of Linearity</th>
<th>Coverage of range (%)</th>
<th>Linearity Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS °CA</td>
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<td>3.0</td>
<td>30.0</td>
</tr>
<tr>
<td>MR 1st</td>
<td>86.4 to 90.0</td>
<td>3.6</td>
<td>12.0</td>
</tr>
<tr>
<td>MT °CA BTDC</td>
<td>4.0 to 6.8</td>
<td>2.8</td>
<td>35.0</td>
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<tr>
<td>BO bar</td>
<td>0.9 to 0.90</td>
<td>0.0</td>
<td>10.0</td>
</tr>
<tr>
<td>EG %</td>
<td>0.0 to 15.0</td>
<td>15.0</td>
<td>100.0</td>
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</table>

<table>
<thead>
<tr>
<th>Test Point D: 3400 rpm 1.58 bar BMEP</th>
<th>Range of Linearity</th>
<th>Coverage of range (%)</th>
<th>Linearity Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS °CA</td>
<td>12.6 to 15.0</td>
<td>2.4</td>
<td>24.0</td>
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<td>MR 1st</td>
<td>82.5 to 90.0</td>
<td>7.5</td>
<td>25.0</td>
</tr>
<tr>
<td>MT °CA BTDC</td>
<td>0.0 to 8.0</td>
<td>8.0</td>
<td>100.0</td>
</tr>
<tr>
<td>BO bar</td>
<td>0.0 to 0.32</td>
<td>0.3</td>
<td>64.0</td>
</tr>
<tr>
<td>EG %</td>
<td>0.0 to 18.4</td>
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<td>92.0</td>
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</table>

<table>
<thead>
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<th>Range of Linearity</th>
<th>Coverage of range (%)</th>
<th>Linearity Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>MS °CA</td>
<td>10.9 to 15.0</td>
<td>4.1</td>
<td>41.0</td>
</tr>
<tr>
<td>MR 1st</td>
<td>60.0 to 90.0</td>
<td>30.0</td>
<td>100.0</td>
</tr>
<tr>
<td>MT °CA BTDC</td>
<td>5.0 to 12.0</td>
<td>7.0</td>
<td>100.0</td>
</tr>
<tr>
<td>BO bar</td>
<td>0.6 to 0.70</td>
<td>0.1</td>
<td>28.0</td>
</tr>
<tr>
<td>EG %</td>
<td>0.0 to 9.5</td>
<td>9.5</td>
<td>63.3</td>
</tr>
</tbody>
</table>

Table 5.5 – Zone of linearity for each parameter on FSN-NO\textsubscript{X} map at each operating point from the coincident point parameter settings. Also showing linearity ranking with most linear marked as 1st and least linear marked as 5th for each test point.
<table>
<thead>
<tr>
<th>Speed - BMEP (rpm - bar) Test Point</th>
<th>Split Main Separation MS (CA)</th>
<th>Split Main Ratio MR (1st : 2nd)</th>
<th>Main Injection Timing MT (CA BTDC)</th>
<th>Boost Pressure BO (bar)</th>
<th>EGR Level EG (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Sweep</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1600 - 1.58 A</td>
<td>10°</td>
<td>60:40</td>
<td>2°</td>
<td>0</td>
<td>27.5</td>
</tr>
<tr>
<td>1600 - 8.45 C</td>
<td>10°</td>
<td>90:10</td>
<td>4°</td>
<td>0.9</td>
<td>15</td>
</tr>
<tr>
<td>Load Sweep</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3400 - 1.58 D</td>
<td>15°</td>
<td>90:10</td>
<td>4°</td>
<td>0.25</td>
<td>10</td>
</tr>
<tr>
<td>3400 - 5.50 E</td>
<td>15°</td>
<td>90:10</td>
<td>9.2°</td>
<td>0.6</td>
<td>3</td>
</tr>
<tr>
<td>Speed Sweep</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1600 - 1.58 A</td>
<td>10°</td>
<td>60:40</td>
<td>2°</td>
<td>0</td>
<td>27.5</td>
</tr>
<tr>
<td>3400 - 1.58 D</td>
<td>15°</td>
<td>90:10</td>
<td>4°</td>
<td>0.25</td>
<td>10</td>
</tr>
<tr>
<td>Speed Sweep</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1600 - 5.50 B</td>
<td>0°</td>
<td>90:10</td>
<td>3°</td>
<td>0.45</td>
<td>20</td>
</tr>
<tr>
<td>3400 - 5.50 E</td>
<td>15°</td>
<td>90:10</td>
<td>9.2°</td>
<td>0.6</td>
<td>3</td>
</tr>
</tbody>
</table>

Table 6.1 – Fixed coincident point parameter settings for split main separation and ratio, main injection timing, boost pressure and EGR rates for speed and load sweep testing.
### Table 7.1 - DoE testing ranges for fuel injection and engine operating parameters for single injection investigation.

<table>
<thead>
<tr>
<th>Test Point A: 1600 rpm 1.58 bar BMEP</th>
<th>1</th>
<th>0</th>
<th>-1</th>
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</thead>
<tbody>
<tr>
<td>Injection Timing - MT (CA BTDC)</td>
<td>-2°</td>
<td>2°</td>
<td>6°</td>
</tr>
<tr>
<td>Boost Pressure - BO (bar)</td>
<td>0.5</td>
<td>0.25</td>
<td>0</td>
</tr>
<tr>
<td>EGR - EG (%)</td>
<td>45</td>
<td>27.5</td>
<td>10</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Test Point B: 1600 rpm 5.50 bar BMEP</th>
<th>1</th>
<th>0</th>
<th>-1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Timing - MT (CA BTDC)</td>
<td>-2°</td>
<td>3°</td>
<td>8°</td>
</tr>
<tr>
<td>Boost Pressure - BO (bar)</td>
<td>0.7</td>
<td>0.45</td>
<td>0.2</td>
</tr>
<tr>
<td>EGR - EG (%)</td>
<td>30</td>
<td>17.5</td>
<td>5</td>
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<table>
<thead>
<tr>
<th>Test Point C: 1600 rpm 8.45 bar BMEP</th>
<th>1</th>
<th>0</th>
<th>-1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Timing - MT (CA BTDC)</td>
<td>4°</td>
<td>8°</td>
<td>12°</td>
</tr>
<tr>
<td>Boost Pressure - BO (bar)</td>
<td>0.9</td>
<td>0.7</td>
<td>0.5</td>
</tr>
<tr>
<td>EGR - EG (%)</td>
<td>15</td>
<td>7.5</td>
<td>0</td>
</tr>
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<table>
<thead>
<tr>
<th>Test Point D: 3400 rpm 1.58 bar BMEP</th>
<th>1</th>
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<th>-1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Timing - MT (CA BTDC)</td>
<td>0°</td>
<td>4°</td>
<td>8°</td>
</tr>
<tr>
<td>Boost Pressure - BO (bar)</td>
<td>0.5</td>
<td>0.25</td>
<td>0</td>
</tr>
<tr>
<td>EGR - EG (%)</td>
<td>20</td>
<td>10</td>
<td>0</td>
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</table>

<table>
<thead>
<tr>
<th>Test Point E: 3400 rpm 5.50 bar BMEP</th>
<th>1</th>
<th>0</th>
<th>-1</th>
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</thead>
<tbody>
<tr>
<td>Injection Timing - MT (CA BTDC)</td>
<td>5°</td>
<td>8.5°</td>
<td>12°</td>
</tr>
<tr>
<td>Boost Pressure - BO (bar)</td>
<td>0.7</td>
<td>0.45</td>
<td>0.2</td>
</tr>
<tr>
<td>EGR - EG (%)</td>
<td>15</td>
<td>7.5</td>
<td>0</td>
</tr>
</tbody>
</table>
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Figure 5.13 – Hybrid half-normal plot showing regression coefficients or main parameter effects for BS NOx, FSN, BS HC and BS FC values at 1600 rpm 8.45 bar BMEP.
Figure 5.14 – Hybrid half-normal plot showing regression coefficients or main parameter effects for BS NOx, FSN, BS HC and BS FC values at 3400 rpm 1.58 bar BMEP.
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<table>
<thead>
<tr>
<th>Coincident point parameter settings</th>
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<tbody>
<tr>
<td><strong>Split main separation</strong></td>
</tr>
<tr>
<td><strong>Split main ratio</strong></td>
</tr>
<tr>
<td><strong>Main injection timing</strong></td>
</tr>
<tr>
<td><strong>Boost pressure</strong></td>
</tr>
<tr>
<td><strong>EGR level</strong></td>
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Coincident point parameter settings

<table>
<thead>
<tr>
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<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Split main separation</td>
<td>MS 0° CA</td>
</tr>
<tr>
<td>Split main ratio</td>
<td>MR 90:10 (1st : 2nd)</td>
</tr>
<tr>
<td>Main injection timing</td>
<td>MT 3° CA BTDC</td>
</tr>
<tr>
<td>Boost pressure</td>
<td>BO 0.45 bar</td>
</tr>
<tr>
<td>EGR level</td>
<td>EG 20 %</td>
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Figure 5.18 – FSN-NOx plot varying each parameter independently at test point B: 1600 rpm 5.50 bar BMEP.
Figure 5.19 – FSN-NOx plot varying each parameter independently at test point C: 1600 rpm 8.45 bar BMEP.
Coincident point parameter settings

<table>
<thead>
<tr>
<th>Parameter Setting</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Split main separation MS</td>
<td>15°</td>
<td>CA</td>
</tr>
<tr>
<td>Split main ratio MR</td>
<td>90:10</td>
<td>(1st : 2nd)</td>
</tr>
<tr>
<td>Main injection timing MT</td>
<td>4°</td>
<td>CA BTDC</td>
</tr>
<tr>
<td>Boost pressure BO</td>
<td>0.25</td>
<td>bar</td>
</tr>
<tr>
<td>EGR level EG</td>
<td>10</td>
<td>%</td>
</tr>
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</table>

Figure 5.20 – FSN-NOx plot varying each parameter independently at test point D: 3400 rpm 1.58 bar BMEP.
Coincident point parameter settings

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Setting 1</th>
<th>Setting 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Split main separation (MS)</td>
<td>15° CA</td>
<td></td>
</tr>
<tr>
<td>Split main ratio (MR)</td>
<td>90:10</td>
<td>(1st : 2nd)</td>
</tr>
<tr>
<td>Main injection timing (MT)</td>
<td>9.2° CA BTDC</td>
<td></td>
</tr>
<tr>
<td>Boost pressure (BO)</td>
<td>0.6 bar</td>
<td></td>
</tr>
<tr>
<td>EGR level (EG)</td>
<td>3 %</td>
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Figure 5.21 – FSN-NO\textsubscript{x} plot varying each parameter independently at test point E: 3400 rpm 5.50 bar BMEP.
Figure 5.22 – Percentage BS NOx change per unit parameter change from baseline for each parameter at each operating condition.
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Figure 6.5 – BS CO (upper) and BS CO$_2$ (lower) emissions for load sweep at 1600 rpm between 1.58 bar and 8.45 bar BMEP and single data point at 5.50 bar BMEP.
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Figure 6.7 – BS HC emissions (upper) and BS FC (lower) for load sweep at 3400 rpm between 1.58 bar and 5.50 bar BMEP.
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Figure 6.10 - BS HC emissions (upper) and BS FC (lower) for speed sweep at 1.58 bar BMEP between 1600 rpm and 3400 rpm.
Figure 6.11 – BS CO (upper) and BS CO$_2$ (lower) emissions for speed sweep at 1.58 bar BMEP between 1600 rpm and 3400 rpm.
Figure 6.12 – FSN (upper) and BS NO₅ (lower) emissions for speed sweep at 5.50 bar BMEP between 1600 rpm and 3400 rpm.
Figure 6.13 – BS HC emissions (upper) and BS FC (lower) for speed sweep at 5.50 bar BMEP between 1600 rpm and 3400 rpm.
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Figure 7.4 – Coincident point multiple and single injection modelled HC and FC data at test point B: 1600 rpm 5.50 bar BMEP. Parameter settings: MT 3° CA BTDC, BO 0.45 bar, EG 20%.
Figure 7.5 – Coincident point multiple and single injection modelled NOₓ and FSN data at test point C: 1600 rpm 8.45 bar BMEP. Parameter settings: MT 4° CA BTDC, BO 0.9 bar, EG 15%.
Figure 7.6 – Coincident point multiple and single injection modelled HC and FC data at test point C: 1600 rpm 8.45 bar BMEP. Parameter settings: MT 4° CA BTDC, BO 0.9 bar, EG 15%.
Figure 7.7 – Coincident point multiple and single injection modelled NOx and FSN data at test point D: 3400 rpm 1.58 bar BMEP. Parameter settings: MT 4° CA BTDC, BO 0.25 bar, EG 10%.
Figure 7.8 – Coincident point multiple and single injection modelled HC and FC data at test point D: 3400 rpm 1.58 bar BMEP. Parameter settings: MT 4° CA BTDC, BO 0.25 bar, EG 10%.
Figure 7.9 – Coincident point multiple and single injection modelled NOx and FSN data at test point E: 3400 rpm 5.50 bar BMEP. Parameter settings: MT 9.2° CA BTDC, BO 0.6 bar, EG 3%.
Figure 7.10 – Coincident point multiple and single injection modelled HC and FC data at test point E: 3400 rpm 5.50 bar BMEP. Parameter settings: MT 9.2° CA BTDC, BO 0.6 bar, EG 3%.
APPENDIX A – OPERATING PROCEDURE FOR TEST FACILITY

START UP PROCEDURE

1. Switch on all wall mounted mains sockets in test cell and control room
2. Turn on Signals emission equipment by completing steps 52 - 74
3. Check oil level (up to ‘C’ on dipstick if cold)
4. Check water level
5. Check fuel level
6. Switch the engine control panel on
7. Turn on the water pump and water heater
8. Turn on the cooling air supply to the injector, air pressure is 2 - 3 bar
9. Switch on compressor with green ‘on’ button in compressor housing
10. Switch on ±15V PS for temperature and pressure amplifiers
11. Switch on amplifier for the in-cylinder pressure transducer
12. Switch on the amplifiers for the rail and injector 2,000 bar pressure transducers at the mains socket in test cell and then on the ‘Excit Switch’ on the front panel, allow 30 mins to stabilize
13. Switch on ±15V PS for needle lift sensor in test cell
14. Switch on the rail pressure regulator RS232 (15V PS)
15. Switch on the fuel meter and smoke meter RS232 interfaces (15V PS)
16. Switch on dSPACE expansion box (ensure that whenever the dSPACE expansion box is switched on the RS232 interfaces are also on)
17. Turn on smoke meter with mains socket in test cell and powers up AVL fuel and smoke meter controller
18. Turn on fuel meter with toggle switch in control room
19. Switch on SCR PC
20. Load MATLAB on PC
21. In the MATLAB command window load “scrmmodel” by typing: ‘SCR’
22. Load dSPACE ControlDesk from desktop icon
23. In ControlDesk window load the experiment with: File, Open Experiment and Open “SCR_DO NOT ALTER”
24. Open exhaust back pressure valve fully while DRIVER is OFF
25. Ensure that the hardware connection is made between the PC and the CPU in the dSPACE expansion box - Refresh Hardware Connection: in ControlDesk menu system, Hardware, Initialisation, Refresh
26. If engine is running, switch off the input signal into the hardware trigger on the DS2001/B1 with toggle switch every time the model is compiled
27. Ensure that Boost Regulator Toggle Switch (24V PS) in control room is off, power supply must be off every time the model is compiled and built
28. In the ‘scrmodel’ window in Simulink build the model as follows: Tools: RTW Build (or ‘Control + B’), wait for ‘finished RTI build procedure for scrmodel’ to be display in MATLAB window
29. Turn on EGR Driver, EGR Motor, Boost Regulator and MAF Sensor toggle switches in control room
30. Turn on Exhaust Back Pressure Valve Driver in control room using red button (do not switch 2 other buttons) - once the driver is powered up DO NOT move the exhaust back pressure valve manually. (In dSPACE, fully closed or maximum = -7,800, take gradually up to about -3,400 to give a back pressure of approx 1.1 bar with boost pressure of 1.0 bar, need to keep back pressure differential between 0.11 bar and 0.15 bar. For no EGR flow, set EGR to zero, click CLOSE on EGR control in dSPACE.)
31. Plug in shaft encoder 6V PS and PS for the 50:50, 360 to 720 multiplier
32. Switch the hardware trigger on with toggle switch
33. In the ControlDesk window switch to ‘animation mode’
34. Turn on TEOM equipment by completing steps 75 - 82
35. Switch on the TTi TSX3510P Programmable PS for rail pressure regulator and set to output 0 - 10V for 0 – 2,000 bar
36. Check that black boxes in control room are as follows: Rail Regulator = ON, HP Pump disconnected (the Mprop solenoid on the HP pump)
37. Monitor the water and oil temperatures using ControlDesk, when the oil temperature has reached approx 45°C proceed to next step
38. Turn on the oil pump and the oil heater
39. Do not motor the engine until both oil and water are up to temperature and the oil and water temperature signal lights are green in dSPACE
40. Check that the cooling tower thermostat control is switched to auto
41. Switch on the cooling tower feed pump and check that the cooling tower water input into the rig is open in the test cell
42. Check that dSPACE ‘boost controller’ is set at 0 bar boost pressure (1 bar atmospheric)
43. Switch the air intake valve position from ambient to boost
44. Switch on the Bosch FIE by completing steps 83 - 89
45. Calibrate 2,000 bar pressure transducer amplifier after 30 mins of warm up as follows:
   a. switch filters to 1 Hz
   b. press RESET
   c. adjust BAL TRIM so LED is just RED to give 1 bar and check on dSPACE that approximately 1 bar is being read
   d. switch filters back to 10 kHz
46. Switch on the LP fuel pump and check that the indicator light is on
47. Turn on the fuel system heat exchanger cold water supply in test cell
48. Turn on the dyno control panel and press e-stop reset on the controller
49. Press the start button and set required motored engine speed
50. In dSPACE switch the ‘RP Regulator’ output from the PS to ‘on’ and set the desired rail pressure with the ‘RP regulator’ output; NEVER SET THE RAIL PRESSURE ABOVE 1,200 BAR
51. Check the ranges on Signals emission equipment are the same as those on dSPACE and check that the displayed values agree

START UP PROCEDURE FOR SIGNALS EQUIPMENT
52. Turn on mains power on back of EMISSIONS STACK, PRE-FILTER STACK and EGR
53. Turn on all DISPLAY PANELS on EGR STACK and EMISSIONS STACK
54. Turn on HEATED LINES on PRE-FILTER STACK
55. Turn on DISTRIBUTION OVEN on EMISSIONS STACK, need 60 mins to warm up before passing sample gases through
56. Power on to PRE-FILTER STACK
57. Turn on COOLER on EMISSIONS STACK, wait about 15 mins for red lights to go off - can only go as far as point 44 with red lights still on

58. Turn on GASES: \( \text{CO}_2 \) (10%), \( \text{N}_2 \) (99.999%), air (18 - 21% \( \text{O}_2 \)), \( \text{CO} \) (1%), propane (3000 ppm), hydrogen and helium mixture (40% and 60%), nitric oxide (5000 ppm)

59. Check GASES are at 2 - 3 bar

60. Check that COMPRESSED AIR LINE is at 3 bar

61. Check the AIR PRESSURE LOW light on PRE-FILTER STACK panel is off

62. Check H/L 1-3 on PRE-FILTER STACK set to 180° C

63. Check internal heated lines (\( \text{NO}_x \) and HC) digital readout on back of EMISSIONS STACK ~180° C

64. Check for green flashing light on DISTRIBUTION OVEN PANEL on EMISSIONS STACK and temperature dial set to 180° C

65. Check temperatures on PRE-FILTER STACK: (1) oven temperature ~195° C, (2) pump head temperature ~195° C, (3) sample gas temperature ~195° C and (4) electronics temperature ~30° C

66. Press ILLM then press SAMP on all display panels on EGR STACK and EMISSIONS STACK

67. MUST WAIT for all COOLER lights to go green on EMISSIONS STACK

68. Turn on PUMP on EGR STACK

69. Switch to MEASURE on PRE-FILTER STACK

70. Check for about 3 – 5 psi on pressure gauge on DISTRIBUTOR OVEN on EMISSIONS STACK

71. Press CAL on all display panels on EGR STACK and EMISSIONS STACK to calibrate system (does ZERO first and then SPAN)

72. Ready to take data when display panels show SAMPLE (4000VM \( \text{NO}_x \) warm up in 30 mins full accuracy in 60 mins, 3000HM THC warm up in 30 mins full accuracy in 60 mins, 7000FM \( \text{CO}_2 \) CO usable in 2 mins full accuracy in 60 mins, 8000M \( \text{O}_2 \) usable in 2 mins full accuracy in 60 mins)

73. Calibrations should be taken at the start of any experimental run and at the end to ensure that the calibration is good and has been maintained during the run
74. SPAN and ZERO calibrations should be no more than 10%. If they are then the pots on the inside of relevant analyser needs manual adjustment as the error is too large for the software, this adjustment must be made whilst the SPAN and ZERO buttons respectively have been pressed. SPAN and ZERO are found from FRONT SCREEN and press PAGE DOWN. When sampling check the analyser ranges

START UP PROCEDURE FOR HORIBA TEOM

75. Never power up the TEOM unit in test cell without the dedicated computer turned on and the software running

76. Turn on power switch in COMPUTER STACK, inside door "MDT-905 MFC", in control room

77. Turn on computer in COMPUTER STACK, inside door "MDT/DPAS CPU", in control room

78. When software is running, turn on mains power supply to TEOM unit in test cell

79. Turn on VACUUM PUMP, DILUTION AIR PUMP and BYPASS PUMP in COMPUTER STACK in control room

80. Leave for approximately 15 mins as unit goes through warm up process

81. When ready to sample put TEE VALVE to ON position to allow exhaust sample to be taken into TEOM unit

82. When not taking samples put TEE VALVE to OFF position to stop exhaust sample to be taken into TEOM unit unnecessarily to prolong life of filters

Filter Changing Procedure for Horiba TEOM

TEOM unit and pumps must be running and system fully warm before changing filter:

a. Make certain that the EXHAUST TEE VALVE is at OFF position to stop exhaust sample being taken into TEOM unit

b. Press <F2> to stop data collection

c. Turn off power supply to TEOM unit only in test cell

d. Open TEOM unit and remove old filter using filter exchange tool
e. Wipe filter exchange tool to remove any particulate matter
f. Insert new filter with filter exchange tool
g. Close TEOM unit and turn on power supply
h. Unit then goes through warm up process for approximately 5 minutes

**Bosch Injector Control**

83. Turn on mains voltage to Booster/amplifier and press "RESET" button on front panel (the Booster/amplifier can go into overload at other times too, shown by red "UBERLAST" LED on front panel, and the "RESET" button will need pressing)

84. Turn on mains voltage to TCU and switch on front panel "EIN/AUS" = "on/off" and check that in-cylinder pressure monitoring is on (= "EIN")

85. Switch on 15V PS to ASCET box and put switch on front panel to "Auto"

86. Switch on the oscilloscope

87. Ensure that the injector signal to the Booster/amplifier is disconnected from ASCET box until the desired injector signal is ready to be output

88. Switch on laptop and the ASCET software opens

(C:\ASCET32.RSF>ASCET)

89. For operation of Bosch Piezo FIE ASCET Software see relevant document ("Procedure For Bosch Piezo FIE ASCET Software")
SHUT DOWN PROCEDURE

1. Wind down the dyno speed and press the stop button
2. Switch off the dyno control panel
3. Switch off the water and oil heaters (leave the pumps on for a further 10 mins)
4. Turn rail pressure power supply control off and put zero in rail pressure value box in dSPACE
5. Switch the output from the TTi TSX3510P Programmable PS to off from dSPACE, this opens the RPR valve and releases all remaining rail pressure
6. Set boost pressure to zero in dSPACE
7. Switch off the cold water supply to the fuel system heat exchangers
8. Shut down Signals emissions equipment by completing steps 28 - 39
9. Switch off and close down Bosch FIE by completing steps 40 - 46
10. Shut down TEOM equipment by completing steps 47 - 54
11. Switch off the LP fuel pump
12. Turn off the cold water supply for the fuel system heat exchanger in test cell
13. Turn off fuel meter (toggle switch in control room)
14. Turn off smoke meter (mains socket in test cell) - also powers down AVL fuel and smoke meter controller
15. Switch off the dSPACE expansion box (ensure that whenever the dSPACE expansion box is switched on the RS232 interfaces are also on)
16. Switch off the rail pressure regulator RS232 interface (15V PS)
17. Switch off the fuel meter and smoke meter RS232 interfaces (15V PS)
18. Switch off the 2,000 bar pressure transducer amplifiers
19. Switch off power for needle lift sensor in control room
20. Disconnect the power supply to the shaft encoder output box and
21. Switch off the power supply to the temperature and Kulite pressure amplifiers
22. Switch off the in-cylinder pressure amplifier
23. Turn off Exhaust Back Pressure Valve Driver in control room using red button
24. Turn off EGR Driver, EGR Motor, Boost Regulator and MAF Sensor toggle switches in control room
25. Switch off the cooling tower feed pump
26. Switch off the 2 - 3 bar cooling air supply to the Bosch injector
27. Switch off the water and oil pumps and shut down the engine control panel

**SHUT DOWN PROCEDURE FOR SIGNALS EQUIPMENT**

28. Press STOP on all display panels on EGR STACK and EMISSIONS STACK
29. Turn off PUMP on EGR STACK
30. Switch to PURGE on PRE-FILTER STACK for at least 10 mins and wait for this 10 mins before going onto next point
31. Switch to STAND BY on PRE-FILTER STACK
32. Turn off COOLER on EMISSIONS STACK
33. Turn off DISTRIBUTION OVEN on EMISSIONS STACK
34. Power off to PRE-FILTER STACK
35. Press ILLM on all display panels on EGR STACK and EMISSIONS STACK
36. Turn off GASES
37. Turn off HEATED LINES on PRE-FILTER STACK
38. Turn off all display panels on EGR STACK and EMISSIONS STACK
39. Turn off mains power on back of EMISSIONS STACK, PRE-FILTER STACK and EGR STACK

**SHUT DOWN PROCEDURE FOR BOSCH FIE SYSTEM**

40. Switch output signal from the ASCET box off, remove the output BNC and use the signal switch on front panel
41. Save and close the ASCET software running on the laptop and switch off laptop
42. Switch off the power supply to the ETAS box
43. Switch off the TCU
44. Switch off the PS to the needle lift sensor
45. Switch off the oscilloscope
46. Switch off the Booster/amplifier

SHUT DOWN PROCEDURE FOR HORIBA TEOM

47. Stop sampling by pressing F3 on keyboard
48. Make certain that TEE VALVE is at OFF position to stop exhaust sample being taken into TEOM unit
49. PURGE for 10 mins by leaving all three pumps running
50. Turn off VACUUM PUMP, DILUTION AIR PUMP and BYPASS PUMPS in COMPUTER STACK in control room
51. Turn off mains power supply to TEOM unit in test cell
52. Press F10 on keyboard to return to DOS screen
53. Turn off computer in COMPUTER STACK, inside door "MDT/DPAS CPU", in control room
54. Switch off power switch inside door "MDT-905 MFC" in control room
APPENDIX B - EXHAUST GAS ANALYSIS

NO\textsubscript{x} AND HC EMISSIONS

Exhaust gas consists of water (H\textsubscript{2}O), nitrogen (N\textsubscript{2}), carbon monoxide (CO), carbon dioxide (CO\textsubscript{2}), oxygen (O\textsubscript{2}) and hydrogen (H\textsubscript{2}) along with oxides of nitrogen (NO\textsubscript{x}), hydrocarbon (HC) and soot emissions or solid carbon (C). It can be assumed that the exhaust gas is an ideal gas and hence the ideal gas relationship can be applied. Thus the analyser readings, which are on a volume basis, are linearly proportional to the mass or number of moles of the substance present in the exhaust gas sample. The mole fraction of each exhaust gas component can be determined from the following relationship:

\[
\bar{x}_i = \frac{n_i}{n_{\text{TOTAL}}} \quad (B.1)
\]

Where \(n_i\) is the number of moles of the component and \(n_{\text{TOTAL}}\) is the total number of moles of exhaust gas.

The exhaust gases were measured with a Signal Instrument Group MaxSys 900 Raw Test Bed Emissions Analysis System which determined the relative amount of each component as either a percentage, for CO, CO\textsubscript{2} and O\textsubscript{2}, or as parts per million (ppm), for NO\textsubscript{x} and HC emissions, on a volume basis [92]. Thus the mole fractions of each exhaust gas component can be given as follows:

\[
\bar{x}_{\text{CO}_2, \text{CO}, \text{O}_2} = \frac{n_{\text{CO}_2, \text{CO}, \text{O}_2}}{100} \quad (B.2)
\]

\[
\bar{x}_{\text{NO}_x, \text{HC}} = \frac{n_{\text{NO}_x, \text{HC}}}{1,000,000} \quad (B.3)
\]
Where $n_{\text{CO}_2, \text{CO}, \text{O}_2}$ are analyser readings for CO$_2$, CO and O$_2$ as a percentage and $n_{\text{NO}_x, \text{HC}}$ are the analyser readings for NO$_x$ and HC emissions as parts per million. The following is a definition of the mass of a substance:

$$m_i = n_i \times M_i \quad \text{(B.4)}$$

Where $M_i$ is the molecular weight and $n_i$ is the number of moles of the substance. And the following definition for mass fraction is taken from Heywood [32]:

$$x_i = \frac{m_i}{m_{\text{TOTAL}}} \quad \text{(B.5)}$$

Where $m_i$ is the mass of each component and $m_{\text{TOTAL}}$ is the total mass of the mixture. By considering equations (B.4) and (B.5) the following relationship can be shown for the mass fraction of a substance:

$$x_i = \frac{m_i}{m_{\text{TOTAL}}} = \frac{n_i M_i}{n_{\text{TOTAL}} M_{\text{TOTAL}}} = \frac{\dot{m}_i}{\dot{m}_{\text{TOTAL}}} \quad \text{(B.6)}$$

Where $\dot{m}_i$ is the mass flow rate of each component and $\dot{m}_{\text{TOTAL}}$ is the total mass flow rate of the mixture. Thus the mass flow rate of the individual components can be shown to be:

$$\dot{m}_i = \frac{n_i M_i}{n_{\text{TOTAL}} M_{\text{TOTAL}}} \times \dot{m}_{\text{TOTAL}} \quad \text{(B.7)}$$

Substituting in from equation (B.1) gives the mass flow rate in kg/hr of either NO$_x$ or HC emissions in the exhaust gas as follows:

$$\dot{m}_i = \frac{\bar{x}_i M_i}{M_{\text{TOTAL}}} \times \dot{m}_{\text{TOTAL}} \quad \text{(B.8)}$$
Where $\bar{x}_i$ is the mole fraction of either NO\textsubscript{x} or HC emissions in the exhaust gas, $M_i$ is the molecular weight of either NO\textsubscript{x} or HC emissions, $M_{TOTAL}$ is the total molecular weight of the exhaust gas and $\dot{m}_{TOTAL}$ is the total mass flow rate of the exhaust gases. The total mass flow rate of the exhaust gases can be found by considering the conservation of mass as follows:

$$\dot{m}_{TOTAL} = \dot{m}_{AIR} + \dot{m}_{FUEL} \quad \text{(B.9)}$$

Where $\dot{m}_{AIR}$ is the measured mass flow rate of the intake air and $\dot{m}_{FUEL}$ is the measured mass flow rate of the fuel.

**DRIED CO\textsubscript{x}, CO\textsubscript{2} AND O\textsubscript{2} EMISSIONS**

The mass flow rate of NO\textsubscript{x} and HC emissions can be found directly from the Signal Instruments analysers as shown above, however, when undertaking the analysis of CO\textsubscript{2}, CO and O\textsubscript{2} emissions the exhaust gas sample is passed though a cooler drier in the Signal Instruments equipment before entering the analysers. This cools the samples to 5° C and means that much of the water present in the exhaust gas sample is removed before analysis is undertaken, which has a small but significant effect on the CO, CO\textsubscript{2} and O\textsubscript{2} readings. This can be corrected for in the following way. The actual mole fraction of the exhaust components considering wet analysis can be found from equation (B.1) as follows:

$$\bar{x}_i\text{WET} = \frac{n_i}{n_{EXH}} \quad \text{(B.10)}$$

Whereas the mole fraction of the exhaust component in the cooled and dried exhaust gas, which is the dry analysis, is given by:

$$\bar{x}_i\text{DRY} = \frac{n_i}{n_{EXH} - n_{H2O}} \quad \text{(B.11)}$$
Where \( n_i \) is the number of moles of the exhaust component, \( n_{\text{EXH}} \) is the total number of moles in the exhaust gas and \( n_{\text{H2O}} \) is the number of moles of water removed from the exhaust gas and lost in the cooler dryer. To take account of this error it is necessary to calculate the amount of water removed from the exhaust sample when it is passed through the cooler dryer. The cooler drier reduces the exhaust sample temperature to 5° C and so the sample is not fully dried; the analysers therefore undertake a nearly-dry analysis of the exhaust gas sample. The relationship for the mass flow rate of \( \text{H2O} \) lost from the exhaust sample when passed through the cooler drier can be shown as:

\[
\dot{m}_{\text{H2O reduction}} = \dot{m}_{\text{H2O in exhaust sample}} - \dot{m}_{\text{H2O removed by drier}}
\]  

(B.12)

The amount of water left in the exhaust sample after it has passed through the cooler drier was found by referring to the psychrometric chart presented in Cengel and Boles [122] for air at atmospheric pressure, which is a good approximation for the exhaust gas under consideration. The chart indicates that reducing the sample temperature to 5° C results in a humidity ratio value of 0.55% as follows:

\[
w = \frac{\text{H2O in sample after drier (kg)}}{\text{dry air sample (kg)}} = 0.55%
\]  

(B.13)

Which can be rewritten as:

\[
\dot{m}_{\text{H2O in sample after drier}} = 0.0055 \times \dot{m}_{\text{dry air}}
\]  

(B.14)

As the humidity ratio value of 0.55% is very small, showing there is little \( \text{H2O} \) left in the exhaust gas sample after the drier, it is reasonable to say that the dried exhaust gas sample approximates to a dry air sample to give:

\[
\dot{m}_{\text{dry air}} \approx \dot{m}_{\text{exhaust sample after drier}}
\]  

(B.15)
Substituting this approximation into equation (B.14) gives:

\[ \dot{m}_{\text{H}_2\text{O in sample after drier}} \approx 0.0055 \times \dot{m}_{\text{exhaust sample after drier}} \]  

(B.16)

The relative amount of \( \text{H}_2\text{O} \) in the exhaust gas is calculated using the perfect combustion equation taken from Heywood [32] and modified to take into account how the combustion products change with varying relative air/fuel ratio, \( \lambda \), as shown below:

\[
\text{C}_a\text{H}_b + \lambda \left( a + \frac{b}{4} \right) (\text{O}_2 + 3.773 \text{ N}_2) = a\text{CO}_2 + \frac{b}{2} \text{H}_2\text{O} + \lambda \left( a + \frac{b}{4} \right) 3.773 \text{ N}_2 + c\text{O}_2
\]

(B.17)

from consideration of the fuel properties of light diesel fuel (\( \text{C}_a\text{H}_{1.8n} \)) given by Heywood [32] the value of \( c \) can be found by considering the oxygen balance as shown:

\[
c = \lambda \left( a + \frac{b}{4} \right) - \left( a + \frac{b}{4} \right)
\]

(B.18)

A spreadsheet was developed in order to calculate the number of moles of reactants and products, the percentage of \( \text{O}_2 \) present in both the wet and 0.55% wet exhaust samples and then the error between the wet and 0.55% wet exhaust samples for differing \( \lambda \) values. In the following equations, the 0.55% wet exhaust samples are referred to as dry exhaust samples. Mole fractions of \( \text{O}_2 \) present in wet and the 0.55% wet exhaust samples given by the analysers can be shown respectively as follows:

\[
\bar{x}_{\text{O}_2 \text{ WET}} = \frac{n_{\text{O}_2}}{n_{\text{EXH}}} = \frac{n_{\text{O}_2}}{n_{\text{CO}_2} + n_{\text{H}_2\text{O}} + n_{\text{N}_2} + n_{\text{O}_2}}
\]

(B.19)

\[
\bar{x}_{\text{O}_2 \text{ DRY}} = \frac{n_{\text{O}_2}}{n_{\text{CO}_2} + (0.0055 \times n_{\text{H}_2\text{O}}) + n_{\text{N}_2} + n_{\text{O}_2}}
\]

(B.20)
The error between these two values can be given as follows:

$$\text{Error} = \frac{\bar{x}_{\text{O}_2 \text{ DRY}} - \bar{x}_{\text{O}_2 \text{ WET}}}{\bar{x}_{\text{O}_2 \text{ WET}}}$$  \hspace{1cm} (B.21)

When the error data between the wet and 0.55% wet exhaust samples for each value of $\lambda$ were plotted against $\lambda$, a graph was produced as shown in Figure B.1. The best-fit curve from this graph of error against $\lambda$ gives the following relationship:

$$\text{Error} = 0.136 \lambda^{-1.026}$$  \hspace{1cm} (B.22)

As shown in Figure B.1 the error is greatest at higher relative air/fuel ratios, reaching a maximum value of around 12% when $\lambda$ has a value of 1. This occurs because there is relatively more H$_2$O in the exhaust gases around stoichiometric combustion and thus passing the sample through the cooler drier removes relatively more H$_2$O, which has a greater effect on the relative composition of the combustion products.

Bringing equations (B.21) and (B.22) together gives the following relationship, which was used during data processing to correct for the error introduced by the drying of the exhaust sample:

$$\bar{x}_{\text{O}_2 \text{ WET}} = \frac{\bar{x}_{\text{O}_2 \text{ DRY}}}{\text{Error} + 1} = \frac{\bar{x}_{\text{O}_2 \text{ DRY}}}{0.136 \lambda^{-1.026} + 1}$$  \hspace{1cm} (B.23)

Here $\bar{x}_{\text{O}_2 \text{ WET}}$ is the required wet exhaust analysis readings of either O$_2$, CO or CO$_2$ in percent and $\bar{x}_{\text{O}_2 \text{ DRY}}$ is the 0.55% wet value of either O$_2$, CO or CO$_2$ in percent given by the Signal analysers. Thus from equation (B.8) the mass flow rate in kg/hr of either O$_2$, CO or CO$_2$ emissions in the exhaust gas is given as follows:
\[ \dot{m}_i = \frac{\bar{x}_i M_i}{M_{\text{TOTAL}}} \times \dot{m}_{\text{TOTAL}} \]  \hspace{1cm} (B.24)

Where \( \bar{x}_i \) is the mole fraction of either O\(_2\), CO or CO\(_2\) emissions in the exhaust gas, \( M_i \) is the molecular weight of either O\(_2\), CO or CO\(_2\) emissions, \( M_{\text{TOTAL}} \) is the total molecular weight of the exhaust gas and \( \dot{m}_{\text{TOTAL}} \) is the total mass flow rate of the exhaust gases.

**PERCENTAGE EGR**

The EGR used here is defined as the ratio of cycle averaged mass flow rate of recycled exhaust gas and the mass flow rate of the total induced mixture, expressed as a percentage and shown as:

\[ \text{EGR} \% = \left( \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{MAN}}} \right) \times 100 = \left( \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{EGR}} + \dot{m}_{\text{AIR}}} \right) \times 100 \]  \hspace{1cm} (B.25)

Where \( \dot{m}_{\text{EGR}} \) is the mass flow rate of the EGR, \( \dot{m}_{\text{MAN}} \) is the mass flow rate of the gases in the air intake manifold and \( \dot{m}_{\text{AIR}} \) is the mass flow rate of the air intake. CO\(_2\) readings were taken directly as volumetric concentrations in percent using two CO\(_2\) Signal Analysers at the air intake manifold and at the exhaust. Equation (B.25) needed manipulation to enable percentage EGR to be found using data from the analysers. From equation (B.8) it can be shown that the mass flow rate of a CO\(_2\) in the EGR stream is given as follows:

\[ \dot{m}_{\text{CO}_2\text{EGR}} = \frac{\bar{x}_{\text{CO}_2\text{EGR}} M_{\text{CO}_2}}{M_{\text{EGR}}} \times \dot{m}_{\text{EGR}} \]  \hspace{1cm} (B.26)

Here \( M_{\text{CO}_2} \) is the is the molecular weight of CO\(_2\), \( M_{\text{EGR}} \) is the molecular weight of the EGR, which is the same as the exhaust gas, and \( \bar{x}_{\text{CO}_2\text{EGR}} \) is the mole fraction of CO\(_2\) in the exhaust gas and therefore also in the EGR stream.
Similar equations for the mass flow rates of CO\textsubscript{2} in the manifold and air intake can be defined as follows:

\[ \dot{m}_{\text{CO2 MAN}} = \frac{\bar{x}_{\text{CO2 MAN}} M_{\text{CO2}}}{M_{\text{MAN}}} \times (\dot{m}_{\text{EGR}} + \dot{m}_{\text{AIR}}) \]  \hspace{1cm} (B.27)

\[ \dot{m}_{\text{CO2 AIR}} = \frac{\bar{x}_{\text{CO2 AIR}} M_{\text{CO2}}}{M_{\text{AIR}}} \times \dot{m}_{\text{AIR}} \]  \hspace{1cm} (B.28)

Where \( M_{\text{MAN}} \) and \( M_{\text{AIR}} \) are the molecular weights of the manifold gas mixture and air intake respectively, while \( \bar{x}_{\text{CO2 MAN}} \) and \( \bar{x}_{\text{CO2 AIR}} \) are the mole fractions of CO\textsubscript{2} in the manifold and air intake respectively. The mass flow rates of CO\textsubscript{2} in equations (B.26), (B.27) and (B.28) are not known, but by considering the conservation of mass flow the following relationship can be shown:

\[ \dot{m}_{\text{CO2 EGR}} = \dot{m}_{\text{CO2 MAN}} - \dot{m}_{\text{CO2 AIR}} \]  \hspace{1cm} (B.29)

Therefore substituting into this gives the following relationship:

\[ \frac{\bar{x}_{\text{CO2 EGR}} M_{\text{CO2}}}{M_{\text{EGR}}} \times \dot{m}_{\text{EGR}} = \frac{\bar{x}_{\text{CO2 MAN}} M_{\text{CO2}}}{M_{\text{MAN}}} \times (\dot{m}_{\text{EGR}} + \dot{m}_{\text{AIR}}) - \frac{\bar{x}_{\text{CO2 AIR}} M_{\text{CO2}}}{M_{\text{AIR}}} \times \dot{m}_{\text{AIR}} \]  \hspace{1cm} (B.30)

The molecular weight of air is given by Rogers and Mayhew as 28.96 kg/kmol [93] and by assuming that the molecular weight of the exhaust gas, and hence also the EGR gas, is 29 kg/kmol [33], the mass flow rate of the EGR can be found as follows:

\[ \dot{m}_{\text{EGR}} = \dot{m}_{\text{AIR}} \left( \frac{\bar{x}_{\text{CO2 MAN}} - \bar{x}_{\text{CO2 AIR}}}{\bar{x}_{\text{CO2 EGR}} - \bar{x}_{\text{CO2 MAN}}} \right) \]  \hspace{1cm} (B.31)
If this relationship is substituted into equation (B.25) the following relationship for the percentage EGR is given by:

\[
EGR(\%) = \frac{x_{\text{CO2 MAN}} - x_{\text{CO2 AIR}}}{x_{\text{CO2 EGR}} - x_{\text{CO2 AIR}}} \times 100
\]  

(B.32)

Where \( x_{\text{CO2 MAN}} \) and \( x_{\text{CO2 EGR}} \) are the mole fractions of CO\(_2\) in the intake manifold and the EGR (actually measured in the exhaust) respectively and \( x_{\text{CO2 AIR}} \) is the mole fraction of CO\(_2\) in ambient air. Concentrations of gaseous emissions in the exhaust are measured in percent by volume by the CO\(_2\) analysers used here [92], which corresponds to the mole fraction multiplied by 100 [32]. The EGR rate reported here is the volumetric percentage of intake charge that is exhaust products and can be written as [12]:

\[
EGR(\%) = \frac{\%\text{CO2 MAN} - \%\text{CO2 AIR}}{\%\text{CO2 EGR} - \%\text{CO2 AIR}} \times 100
\]  

(B.33)

Where \( \%\text{CO2 MAN} \) and \( \%\text{CO2 EGR} \) are the percentage readings from the Signal analysers of CO\(_2\) in the intake manifold and the EGR (actually measured in the exhaust) respectively. The percentage of CO\(_2\) in ambient air, \( \%\text{CO2 AIR} \), is taken from Rogers and Mayhew tables [93] to be approximately 0.03%. As previously discussed the errors involved with passing the exhaust gas samples through the cooler drier when considering CO\(_2\) analysis need to be taken into account to remove any inaccuracies. However, both the exhaust and manifold CO\(_2\) samples are conditioned using cooler driers and hence any errors involved in not undertaking the full wet analysis are negligible. The calculation of the percentage EGR shown in equation (B.32) is therefore undertaken using CO\(_2\) values in percent as output by the Signal analysis equipment with no correction made for sample drying.
Figure B.1 - Error data between wet and 0.55% wet exhaust samples from perfect combustion equation.
APPENDIX C - DOE TECHNIQUES

DESIGN OF EXPERIMENTS

Design of Experiments (DoE) techniques were first developed in the 1930s by Fisher [123] and were mainly used in the agricultural industry. The benefits of using DoE for engineering applications have only been realised in recent years. The techniques can involve the need to get to grips with complicated statistical methods, however, for the majority of applications, including the application of DoE in this thesis, simple principles may be applied to produce satisfactory results. Much work has been published demonstrating the implementation of DoE in engineering studies and the benefits seen [124, 125]. Maintaining emissions at required targets and understanding their formation is a major objective in the engine development process; Piley [111] looks at the application of DoE in the optimisation of emissions targets. The main objective of DoE methodology is to streamline the experimentation process and the main benefits involved are the time and cost savings to the user. This is particularly valuable when the number of variables is large as is the case with this study into split main injections, which has five independently controlled parameters.

A statistically designed experiment, when used to model a response surface, attempts to fit some form of graduating function such as a polynomial to model the response [108]. Therefore in a two level experiment, the graduating function is a linear equation that relates the change in the inputs to the variations in the responses, which will result in a linear model to represent the effects of the parameters on the responses. If it is assumed that the responses will react in a non-linear way to variations of the input parameters, then at least a second order or three level investigation needs to be considered, which will result in a polynomial equation model. Using more than three levels allows more complex graduating functions to be used to produce the predictor equations, but this greatly increases the amount of testing required.
FRACTIONAL FACTORIAL AND CCP DESIGNS

Conducting a one-variable-at-a-time investigation by varying three parameters independently at two levels, to give linear models, would require $2^3 = 8$ experimental test runs. While varying three parameters independently at three levels, to generate non-linear models, would require $3^3 = 27$ test runs. Representations of these test matrices are shown in Figure C.1. Furthermore, a full factorial plan for five parameters at two levels will involve $2^5 = 32$ test runs; the test matrix for this is shown in Table C.1. The highest order interaction that occurs will be the 5-way interaction (A x B x C x D x E). This particular interaction is the 'generator' of the fractional design. Table C.2 shows the result for the interaction of the five parameters in the column headed 'ABCDE' and shows that half the runs produced are positive while the other half are negative. If just one of these half matrices were used for the experimental design, then it would no longer be possible to determine the individual effect of the 5-way interaction. These interactions can be assumed to be negligible [123], as they do not usually yield any useful information and have little effect upon the overall responses. Therefore, by considering the 5-way interaction to be negligible allows the number of test runs required to be halved. This design is known as a half fraction factorial of the original design and now requires just $2^{5-1} = 2^4 = 16$ experimental test runs. It is possible to reduce the size of the experimental matrix further, but this will be at the cost of further reducing the amount of information available about the higher order interactions.

To further reduce the amount of testing needed while not compromising the accuracy of the models produced, a central composite plan (CCP) test matrix can be used to produce models that are reliable and consistent over most of the design region [111]. The CCP is based on a two level linear model with additional axial and centre points included to help establish the curvature of the response surface [112] and give the required quadratic effects. The plan consists of three types of points: factorial, axial and centre points. The factorial corner points are used to determine the interactions, the axial points demonstrate the effect of varying each parameter while the others remain at
nominal conditions and the centre point gives the magnitude of the quadratic effect [62, 109]. A full factorial CCP for five parameters requires 43 test runs as shown:

\[(2)^5 + (5 \times 2) + 1 = 43\]

A half factorial CCP design reduces the number of the two level factorial corner points while leaving the axial and centre points unchanged. This allows the number of experimental test runs to be reduced to 27 as shown:

\[(2)^{(5-1)} + (5 \times 2) + 1 = 27\]

A representation of the full factorial and the half factorial CCP designs for just three parameters are shown in Figure C.2.

**MULTIPLE LINEAR REGRESSION AND ANALYSIS OF RESPONSES**

When undertaking a DoE experiment, information about the results is not available until all the tests have been completed. Once all the runs are completed and the test data collected, a regression analysis may be performed to model the responses to produce meaningful results and give information on the coefficients related to the main parameter effects and any interactions. If a given response depends on n variables then the relationship between the responses and the variables is characterised using a regression model. Multiple linear regression techniques are used to analyse the experimental responses as demonstrated by Montgomery and Gilchrist [62, 125] and yield a full set of regression coefficients and a value for the constant term. Thus an empirical model is produced, which represents the required response with respect to variations in the known variables. A particular modelled response, \( y \), is
therefore constructed from a combination of the regression coefficients, $\beta_n$, and the input parameter values, $x_n$, in the form:

$$y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \ldots + \beta_n x_n + \varepsilon$$

Least squares regression formulates predictions for the regression coefficients based on minimising the sum of the squares of the errors, $\varepsilon$. Any number of responses can be modelled in this way once the experimental data has been collected and the regression coefficients have been generated. The predictive formula may then be applied to produce response surface plots, normal plots and half-normal plots, and can determine which of the parameters or interactions have the most significant effect on the response in question and the magnitude of those effects. Surface plots for each combination of parameters highlighting their effects at each speed-load point were produced from the work undertaken in Chapter 5. Surface plots for BS NO$_x$ emissions at just one speed and load point, 1600 rpm 5.50 bar BMEP, are shown in Figure C.3 and Figure C.4. While surface plots for FSN values at the same operating condition are shown in Figure C.5 and Figure C.6.

The MATLAB Statistics Toolbox [126] statistical software package, along with a series of MATLAB m-files developed by Richardson [110] and modified by the Author for this investigation, was used to perform the linear regression analysis in this thesis. For an explanation of the theory behind linear regression, the reader is referred to Hicks and Turner [127].

**NORMAL PLOTS**

The central limit theorem [109] states that the sampling distribution is approximately normal if the sample size is sufficiently large. The coefficients generated by the linear regression model should be normally distributed and so they can be plotted against the normal distribution to distinguish between points of random variation and actual effects. Points that are within the normal distribution will plot on a straight line. Any points that do not plot on the normal distribution line have been affected by an external influence that has
caused the values to stray from this linear relationship. Normal plots are produced as follows: the regression coefficients are standardised to reduce the influence of the squared terms, which are not real physical terms, in the response equation. The standardised regression coefficients are then plotted in ascending order against the ascending normal scores, which are calculated based on the number of degrees of freedom in the analysis. For a 3-variable analysis there are nine degrees of freedom, which is taken from the number of coefficients in the response equation. In the normal distribution plot each degree of freedom is represented by an interval of \( \frac{100}{9} = 11.1\% \). Using normal distribution tables, as shown in Table C.3, the normal scores can be found by taking 9 equal steps of 11.1\%. The normal scores are measured from the middle of each interval. Thus the normal scores as shown in Figure C.7 are:

\[-1.5932, -0.9674, -0.5895, -0.2822, 0, 0.2822, 0.5895, 0.9674, 1.5932\]

Examples of normal plots of BS NO\(_x\) and FSN at 1600 rpm and 5.50 bar BMEP are shown in the top part of Figure C.8 and Figure C.9. These plots are used to differentiate between variables that have the largest effects on the responses and those that cause random variation or have no real effect on the response [62, 109].

Half-normal plots are produced in a similar way to full normal plots except the absolute values of the regression coefficients are considered. Only the positive half of the full normal scores is now considered and so the half normal scores, for a sample of size \( n \), is the positive half of a set of \( 2n \) full normal scores. The 9 equal steps are now taken over only the positive side of the normal distribution. The half-normal scores shown in Figure C.10 are:

\[0.0697, 0.2104, 0.3555, 0.5085, 0.6745, 0.8616, 1.0853, 1.3830, 1.9145\]

The advantage of using half-normal plots is that the relative influence of each parameter can be readily seen, however, it is not clear whether this influence is a positive or negative one. In the work carried out in this thesis the normal and
half-normal plots have been combined to give a 'hybrid half-normal plot', as shown in the lower half of Figure C.8 and Figure C.9. A hybrid half-normal plot is basically a half-normal plot with the data labels edited to show whether a particular variable has positive or negative effect on the response in question.

As shown in normal plot at the top of Figure C.8, the boost pressure (BO) has a positive effect on the BS NO\textsubscript{x} response, as does the interaction of main injection timing and EGR (MT\times EG). It can also be seen that MR has a small positive effect on the response. Furthermore, parameters MS, MT and EG have an increasing negative effect on the response. Normal plots give a clear indication whether a parameter has a positive or negative effect on a particular response and regression coefficients with the highest magnitude will account for the greatest effect on the response, while those with lower magnitudes will tend to be noise or random variation [128].

In the hybrid half-normal plot for BS NO\textsubscript{x} shown at the bottom of Figure C.8, increasing EGR levels have largest effect and reduce NO\textsubscript{x} as shown by the negative sign (-EG). The second influence on NO\textsubscript{x} here is the reduction seen with retarding main injection timing (-MT), while boost pressure (+BO) is shown as the next influence and increases NO\textsubscript{x} emissions. Split main separation is shown to have a small negative effect (-MS) and split main ratio is has a small positive effect (+MR). For FSN values, as shown in Figure C.9, EGR has the largest positive influence on FSN (+EG), while boost pressure is the second influence and decreases FSN values (-BO). Split main ratio is the next influence and reduces FSN (-MR) and separation increased FSN (+MR), while main injection timing has a small positive effect (+MT). It is clear that when comparing the two hybrid half-normal plots in Figure C.8 and Figure C.9 that the positive influence of EGR at reducing NO\textsubscript{x} outweighs the penalties associated with EGR and increased FSN values. Furthermore, it can be seen that the benefits of using boost in reducing FSN outweigh the penalties seen with increased NO\textsubscript{x} emissions.
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Table C.1 - Full factorial design for five parameters at two levels.
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Table C.2 - Fractional factorial design for five factors at two levels and showing the 5-way interactions in the column headed ‘ABCDEFG’.
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Table C.3 - Normal distribution tables.
Figure C.1 – Three parameter, two level test matrix design giving 8 test points (upper) and three parameter, three level test matrix design giving 27 test points (lower).
Figure C.2 – Three parameter, full factorial CCP design giving 15 test points (upper) and three parameter, half factorial CCP design giving 11 test points (lower).
Figure C.3 – Surface plots for BS NO\textsubscript{x} emissions for various combinations of parameters at 1600 rpm 5.50 bar BMEP.
Figure C.4 – Surface plots for BS NO\textsubscript{x} emissions for various combinations of parameters at 1600 rpm 5.50 bar BMEP.
Figure C.5 – Surface plots for FSN values for various combinations of parameters at 1600 rpm 5.50 bar BMEP.
Figure C.6 – Surface plots for FSN values for various combinations of parameters at 1600 rpm 5.50 bar BMEP.
Figure C.7 – Calculation of normal scores from the normal distribution.

\[ y = \frac{1}{\sigma \sqrt{2\pi}} e^{-\frac{1}{2} \left( \frac{x-\mu}{\sigma} \right)^2} \]
Figure C.8 – Full normal plot (upper) and hybrid half-normal plot (lower) for BS NOx at 1600 rpm 5.50 bar BMEP showing regression coefficients or main parameter effects and higher interactions.
Figure C.9 – Full normal plot (upper) and hybrid half-normal plot (lower) for FSN values at 1600 rpm 5.50 bar BMEP showing regression coefficients or main parameter effects and higher interactions.
Figure C.10 - Calculation of half normal scores from the normal distribution.