18th Australasian Fluid Mechanics Conference
Launceston, Australia
3-7 December 2012

Comparison of Propane and Ethanol Fumigation of a Heavy Duty Common Rail Diesel Engine

Laurie Goldsworthy

National Centre for Maritime Engineering and Hydrodynamics University of Tasmania, Launceston 7250, Australia

Abstract

A heavy duty common rail marine diesel engine operating under load on a test bench is fumigated with either propane or vaporised ethanol mixed into the inlet air at various rates. Fumigation allows the addition of moderate quantities of alternative fuels to a diesel engine without significant modification of the engine. Carbon monoxide and hydrocarbon emissions and exhaust opacity tend to increase with increasing fumigation, with larger increases for propane compared with ethanol. Cylinder pressure and the electronically controlled two stage liquid fuel injection timing are recorded with a high speed data acquisition system. At high rates of fumigation very high pressure rise rates are measured. The apparent heat release rate and the fuel injection timing together allow analysis of the mechanism of the combustion process with fumigation. For both propane and ethanol, two distinct peaks in apparent heat release rate appear when the air/fumigant mixture strength approaches the flammability limit.

Introduction

Diesel engines can be readily configured to run in dual fuel mode, with gas mixed into the air intake while liquid diesel fuel is injected as normal, but at a reduced rate. This is sometimes called fumigation. Various gases and gas mixtures have been used for this purpose including methane, ethane, propane, butane, hydrogen, ethylene, liquefied petroleum gas, landfill gases and process gases.

The work described here was conducted as part of a project to assess alternative fuels for fishing vessels. Fishing vessel operators are looking to reduce fuel costs to make their operations more viable. Marine engines in such applications tend to be operated at steady loads with relatively high BMEP, for long periods of time. A common rail, electronically controlled marine engine with two stage injection is used in the present study. The fumigation of such an engine with two stage injection is of interest in its own right because of possible interactions of the added gas with the injection process. Two stage injection is used to reduce the severity of the second phase of combustion, by reducing the amount of fuel/air mixture formed during the delay period. Adding gaseous fuel to the intake air will increase the amount of fuel air mixture available during and after the delay period and thus impact on the ignition delay and combustion rate. If the overall combustion rate increases with fumigation, then thermal efficiency can be expected to increase, all other factors being equal.

Ethanol is a readily available biofuel which is liquid at atmospheric conditions. The most straightforward means of utilising ethanol is to add it to the air intake. Various techniques such as carburetion, continuous injection under pressure after the turbocharger and multipoint sequential port injection have been utilised. Injecting an atomised spray into the intake duct after the turbocharger utilises the elevated temperature of the air after compression to aid evaporation. Other methods of using ethanol include mixing anhydrous ethanol with normal diesel fuel or substitution of normal diesel fuel entirely with ethanol, plus an ignition enhancer. Adding ethanol to the inlet air stream allows the use of aqueous ethanol. Flowers et al **[1]** demonstrate that the energy costs of removing water from ethanol produced by fermentation become very significant as the azeotropic concentration is approached.

Liquefied Petroleum Gas (LPG) is a readily available fuel which can be stored as a liquid at moderate pressures. Systems for adding LPG to the intake air of diesel engines are available commercially. Such systems increase maximum engine power and can lead to fuel cost savings where the LPG is appropriately priced. LPG as Autogas normally contains around 60% propane and 40% butane. In remote areas, LPG is often supplied as propane so that the same infrastructure can be used for heating purposes and for automotive use. In the present tests, the fuel used is 97% to 99% propane, the remainder being primarily ethane, then butane. Only trace amounts of other gases (<0.1%) are present. Both propane and ethanol can be produced from renewable sources.

Numerous studies have been published on fumigation of diesel engines with ethanol and LPG/propane (see for example **[2-7]**). Where thermal efficiency has been found to increase it is generally at high engine load and of the order of 5% gain. Some studies report reduced thermal efficiency at light loads. With fumigation, NOx emissions tend to decrease and CO and HC emissions tend to increase. Surawski et al **[8]** found that ethanol fumigation increased the volatility of particles and increased the concentrations of particle related reactive oxygen species. Thus, the effects of ethanol usage on the nature of particle emissions may be undesirable. Natural gas can be used to substitute for up 80% to 90% of the diesel fuel energy supply**[9]**, while maximum substitution rates for LPG/propane and ethanol are limited by knock to around 30%.

The aim of the present work is to compare the effect of propane and ethanol fumigation on changes in thermal efficiency and exhaust emissions in a heavy duty turbocharged common rail diesel fuel engine with two stage injection, and to understand the combustion processes involved. Apart from previous work of the present author**[10]** there are no published studies on fumigation with two stage injection.

Test Procedures

The engine is a Cummins QSB5.9-305MCD 6 cylinder in-line, common rail, electronically controlled marine diesel engine rated at 224 kW (305 bhp) at 2600 rpm, turbocharged with water cooled aftercooler, compression ratio 17.2:1, bore 102 mm, stroke 120 mm, no EGR.

A gas analyser is used which consists of an NDIR bench for CO and HC and electrochemical cells for NO and O_2. A NOx convertor allows measurement of total NOx. A heated ceramic filter and refrigerated drier are used to condition the sampled exhaust gas. A Bosch opacity meter draws directly from the unfiltered exhaust stream.

Labview is used as the data acquisition system. A high speed card (NI6143) is run at 200,000 samples per second for each channel, representing a sampling interval of 0.054 crank angle degrees at 1800 rpm. Cylinder pressure, the engine's crank marker pulses and the injector activation voltage are sampled with the high speed card. The high sampling rate allows capture of fine details of the pressure development which leads to good detail in the apparent heat release rate. A slow speed card (NI6036) records exhaust temperature, flow rates of normal diesel fuel and ethanol, inlet manifold pressure, inlet manifold air temperature and fuel rail pressure. The net flowrate of the normal diesel fuel is measured with a MacNaught M1SSP-1 elliptical gear flowmeter, which gives 1000 pulses per litre. Ethanol flow rate is measured with a MacNaught M05SSPI-1H elliptical gear flowmeter with carbon bearings, which gives 1552 pulses per litre. Propane flowrate is measured by weight. The tank weight is counterbalanced and a PT4000 load cell with maximum range 50 kg is used to measure mass of fuel consumed. The manufacturer's quoted precision of the cell is 0.023% of full scale, or 0.012 kg. The load cell output was calibrated by adding and removing known weights at the tank attachment point.

A water cooled AVL QC34C piezoelectric pressure transducer was fitted to cylinder 5 by Cummins USA. It is used in conjunction with a Kistler type 5002 charge amplifier with a 180 kHz low pass filter. The transducer has a measuring range of 0 to 250 bar and linearity of ±0.2% of full scale output.

For timing of injection, the engine control unit (ECU) uses a toothed wheel and sensor on the crankshaft with two missing teeth at top dead centre (TDC) on cylinder 1. The crank angle position at TDC on cylinder 5 was found from hot motored tests to be at 35% of the width of the pulse from the eighth tooth after the missing teeth, measured from the trailing edge.

The injection activation voltage pulses from the ECU for cylinder 5 are recorded with the high speed card. The voltage activates a solenoid inside the injector, which in turn initiates fuel injection. The actual fuel injection will begin a short time after the injector activation voltage rises. Studies **[11,12]** have shown that with a common rail injection system the delay is typically about 0.3 ms. At 1800 rpm this delay time represents about 3 CAD.

A typical data set recorded with the high speed card is shown in Figure 1. The two stage fuel injection can be seen. A short pulse of fuel (pre-injection) is injected from about 25 degrees before Top Dead Centre (TDC).

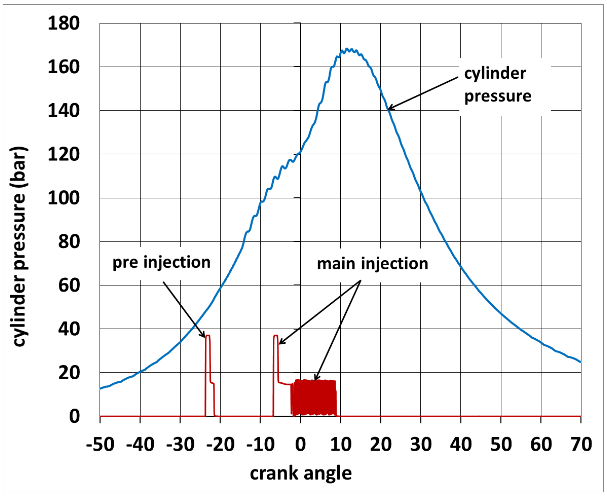

Figure 1 Cylinder pressure (unsmoothed) and fuel injection pulses, for a single cycle. 1800 rpm, 17 bar BMEP, normal diesel fuel only. Top Dead Centre is at 0 degrees crank angle. The mass of fuel injected in each pulse is proportional to the pulse duration.

To produce more precise data on the start of combustion and overall combustion rates, the apparent heat release rate is calculated from the measured cylinder pressure.**[13]** This is the difference between the energy released due to combustion of the fuel and the energy lost by heat transfer from the combustion space. A combination of exponential smoothing and Savitsky-Golay filtering **[14]** is used.

For comparing values of brake thermal efficiency at a given load and speed as ethanol rate is changed, uncertainties in the absolute values of dynamometer load and engine speed can be neglected. The main sources of error are in the flowrates of the fuels and the relative heating values. It is estimated that ethanol, propane and diesel fuel flowrates are measured with a 95% confidence interval of ±2%. The 95% confidence interval for thermal efficiency changes at a given engine load and speed setting is estimated at ±3%.

Speed (rpm)	1800
Torque (Nm)	794
BMEP (bar)	17
Brake Power (kW)	149.7
baseline conditions (no added fumigant)	
Inlet manifold gauge pressure (bar)	0.83
Start pre injection activation pulse (CAD before TDC)	23.7
pre injection pulse duration (CAD)	0.2
Start main injection activation pulse (CAD before TDC)	6.7
main injection pulse duration (CAD)	15.4
Fuel flowrate (litre/hour)	35.6
Exhaust temperature (°C)	518
Brake thermal efficiency (%)	37.6
Overall air to fuel ratio by mass	18.5

Table 1 Engine operating conditions. For each test, speed and torque are held constant while propane or ethanol are added.

The torque demand of the dynamometer is held constant and speed is maintained at a constant value by the ECU. As ethanol or propane substitution rate is increased, the ECU automatically decreases the flowrate of normal diesel fuel to compensate for the added fumigant and thus maintain constant engine speed and load. This is different from the often used dual fuel approach of fixing the rate of liquid fuel injection to a small constant value just high enough to achieve ignition, then increasing the load by adding more gas.

2.2 Fuel Properties

Industrial Methylated Spirits (IMS) is used as the ethanol source. The nominal composition is 95% ethanol and 5% water by mass, with minute amounts of denaturant. Density of the IMS is measured at with a hydrometer at 807±1kg/m^3. For density 807 kg/m^3, the composition is 93% ethanol by mass **[15]** at 20ºC. For reference, a mixture of ethanol and water is azeotropic at about 95.5% ethanol by mass where the density is 803 kg/m^3 at 20ºC. The azeotropic mixture represents the upper limit of ethanol concentration achievable by simple distillation.

Fuel properties are summarised in Table 2.

Fuel	**Density (kg/m^3)**	**Lower Heating Value (LHV) (MJ/kg)**
standard fuel	840	43.2
93% ethanol	807	24.9
propane	not applicable	46.3

Table 2 Fuel properties

Results

The test point has a BMEP less than the maximum rated BMEP at the given speed, to avoid excessively high peak pressure at high fumigant substitution rates. At this test point, the ECU retards injection timing as more fumigant is added, so this needs to be taken into account when interpreting the results. Table 3 compares energy substitution percentage with percentage reduction in normal diesel fuel. Energy substitution percentage is the percentage of total fuel energy from the fumigant and is calculated from the mass flow rates and heating values. Also shown is brake thermal efficiency, calculated from the measured fuel flow rates and heating values. Engine power output is nominally constant. It is apparent that for ethanol, the energy based percentages are lower than the fuel replacement values, and vice-versa for propane. The inconsistencies are presumably due to experimental error in measurement of flow rates and uncertainties in the fuel heating values. The same normal diesel fuel was used for both series of tests and all tests were conducted at the same ambient conditions. If the heating value or flow rates of the ethanol were higher than the values used, then the energy percentage would increase and the thermal efficiency would decrease, and vice-versa. Changes in thermal efficiency are not significant compared with the estimated error in brake thermal efficiency of ±3%.

ethanol						
fuel replacement %	0.0	8.5	19.2	26.4	33.8	35.5
energy %	0.0	8.0	17.8	24.3	31.3	34.3
thermal efficiency %	38.1	38.3	38.7	39.2	39.6	38.8
propane						
fuel replacement %	0.0	9.1	19.0	25.3	34.2	
energy %	0.0	9.7	21.3	25.9	35.1	
thermal efficiency %	38.1	37.9	37.1	37.8	37.6	

Table 3 Fuel replacement percentage by mass, energy supply percentage and brake thermal efficiency for both fumigants.

At substitution rates greater than around 20%, the engine becomes noticeably noisier. For more than 30% fumigant, the engine is knocking severely. The measured cylinder pressure for the higher substitution rates is shown in Figure 3. Shown are representative individual cycles with knock intensity similar to the mean of 100 cycles. Quantification of knock intensity is detailed in **[10]**. The 9% and 19% ethanol by energy plots are excluded from his figure for clarity. The 9% ethanol pressure development essentially overlaps the baseline. As substitution rate is increased, the peak of pressure development shifts towards TDC, even though injection timing is retarded. Similar results are found for propane. At 36% ethanol the pressure rise rate at the start of combustion is very high and the cylinder pressure oscillates significantly.

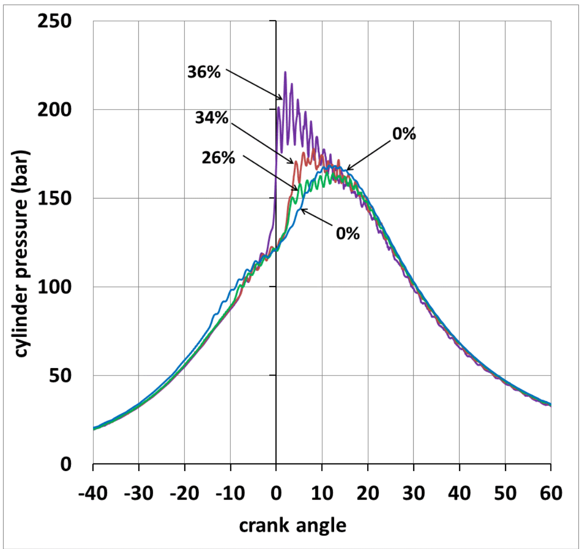

Figure 3 Cylinder pressure (unsmoothed) at various ethanol substitution rates. The curves are labelled with fuel replacement percentage.

The combustion development is better illustrated in the apparent heat release rate plots in Figure 4. The behaviour of ethanol and propane are essentially the same within experimental error. In normal diesel engine operation, liquid fuel evaporates and mixes with air prior to ignition. The flammable fuel air mixture ignites after a chemical delay, during which the full combustion chain mechanism is established via numerous preliminary reactions. Normal diesel fuel ignites readily under these conditions. Ethanol and propane are more dependent on a high temperature ignition source. It is postulated that with fumigant added to the cylinder air, the region of flammable mixture formed after injection of liquid fuel will be extended. Regions which would otherwise be too fuel lean will be flammable due to the presence of the fumigant. Thus the early combustion rate is enhanced. Calculations show that above about 25% ethanol or propane by energy, the fumigant/air mixture strength is above the flammability limit at compression temperatures, as estimated with the Burgess-Wheeler Law.**[10]**

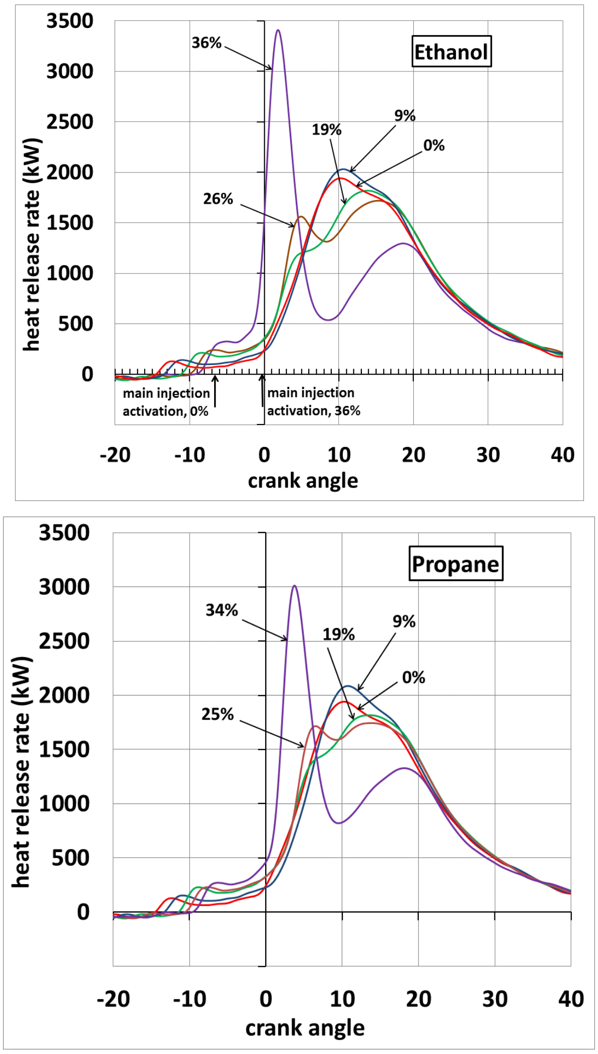

Figure 4 Apparent heat release rate at high fumigant substitution rates. The curves are labelled with fuel replacement percentage. For 0% fumigant, the pre injection activation pulse commences at - 23.7 CAD and the main injection activation pulse begins at -6.7 CAD. Both injection pulses shift together towards TDC as fumigant rate increases, with a shift of about 6 CAD for the highest fumigant rates.

With 9% fumigant, the form of the heat release rate plot does not change significantly compared with the baseline, but the onset of high heat release rate occurs slightly later and the maximum heat release rate is higher. With 19% fumigant, the maximum heat release rate is lower than the baseline and occurs later. For 25% propane and 26% ethanol the heat release rate shows a double peak, with the first peak occurring earlier than the baseline maximum, and the second peak occurring later. This pattern continues with the first peak becoming larger and the second peak diminishing as fumigant rate is increased. The first peak at the higher fumigant rates occurs earlier with increased fumigant rate, even though injection timing occurs later than at low fumigant rates. For the highest fumigant supply rates, the propane/air or ethanol/air mixture is ignited by the small pulse of normal diesel fuel injected in the pre-injection and begins to burn independently at a high rate before the main charge of normal diesel fuel is injected. This combustion could involve flame propagation and/or autoignitive propagation. The rapid pressure rise rates result in severe knock. A distinct minimum in the heat release rate occurs at around the end of injection of the main liquid fuel pulse. It is postulated that the pre-mixed combustion of the fumigant depletes the oxygen in the region of the main injected fuel pulse, leading to a minimum in the heat release rate until the injected liquid fuel finds sufficient oxygen to burn.

After the pre injection pulse, there is a distinct period of positive heat release rate before the main injection pulse. The length of the delay between the start of the pre-injection pulse and the start of positive apparent heat release rate does not increase significantly with increased fumigant. As the fumigant rate increases, the ECU retards the injection timing. The pre-injection pulse and main injection pulse move together. As a result of the retarded injection timing, the point at which positive heat release rate starts shifts towards TDC with increasing fumigant. However, the onset of the main combustion process, as indicated by the rapid increase in heat release rate at or just after TDC, does advance at higher fumigant rates.

For both ethanol and propane, emissions of NOx decrease with increasing fumigant rate, while exhaust opacity and emissions of CO increase. NOx emissions decrease more for ethanol. Emissions of unburnt hydrocarbons are insignificant throughout for ethanol, while for propane they increase significantly at high fumigant rates. Exhaust opacity increases at a greater rate for propane compared with ethanol. At low fumigant rates around 5% to 10% by energy, CO and HC emissions are lower than for standard fuelling and higher fumigant rates.

Conclusions

Addition of ethanol or propane by fumigation is a relatively simple way of introducing a biofuel to a diesel engine, while at the same time reducing NOx emissions and potentially increasing thermal efficiency. Addition of fumigant increases early combustion rates, but no significant changes in thermal efficiency are measured at the moderate BMEP operating condition tested. Generally emissions of CO, HC and smoke increase with fumigant addition, except at low fumigant rates around 5% to 10% by energy, where these emissions are lower than for standard fuelling and higher fumigant rates.

At fumigant rates where the strength of the induced fumigant/air mixture exceeds the lower flammability limit at the compression temperature, the combustion involves premixed combustion, high pressure rise rates, rapid oscillation of cylinder pressure around the peak values and two distinct heat release rate peaks. This regime occurs for ethanol or ethanol rates greater than 25% by energy, when the BMEP is 17 bar. These tests give useful insight into the mechanism of combustion with fumigation and highlight the dangers of excessive fumigant addition in a heavy duty engine. Further work could involve testing of different injection timing strategies.

Nomenclature

BMEP	Brake Mean Effective Pressure
CAD	Crank Angle Degrees
CO	Carbon Monoxide
ECU	Engine control Unit
HC	Hydrocarbons
LHV	Lower Heating Value
NOx	Oxides of Nitrogen
rpm	revolutions per minute
TDC	Top Dead Centre

Acknowledgements

This project was financed by the Australian Fisheries Research and Development Corporation, Project number 2007/200 SESSF Industry Development Subprogram: alternative fuels for fishing vessels.

The author acknowledges the contribution of a number of undergraduate students who assisted with data collection and processing. These include Stephen Stothart, John MacKenna, Mohammed Nishan, Ahmed Iruhas, Tangi Brusq and Yann Morimont. The author also acknowledges the assistance of AMC technical staff. Global Fuel Systems donated and installed the LPG (propane) system. APA donated the LPG tank.

References

[1] Flowers, D., S. Aceves and J. Frias, Improving Ethanol Life Cycle Efficiency by Direct Utilisation of Wet Ethanol in HCCI Engines, *Journal of Society of Automotive Engineers,* 2007, **20077037**: pp 1070-1078.

[2] Dong, J., X. Gao, G. Li, and Z. Xintang, Study on diesel-LPG Dual Fuel Engines, *SAE 2001-01-3679,* 2001.

[3] Gibson, C.M., A.C. Polk, N.T. Shoemaker, K.K. Srinivasan, and S.R. Krishnan, Comparison of Propane and Methane Performance and Emissions in a Turbocharged Direct Injection Dual Fuel Engine, *Journal of Engineering for Gas Turbines and Power,* 2011, **133**.

[4] Patterson, J., A. Clarke and R. Chen, Experimental Study of the Performance and Emissions Characteristics of a Small diesel Genset Operating in Dual-Fuel Mode with Three Different Primary Fuels, *SAE Technical Paper Series,* 2006, **2006-01-0050**.

[5] Hayes, T., L. Savage, R. White, and S. Sorenson, The Effect of Fumigation of Different Ethanol Proofs on a Turbocharged diesel Engine, *SAE 880497,* 1988.

[6] Chen, J., D. Gussert, X. Gao, C. Gupta, and D. Foster, Ethanol Fumigation of a Turbocharged diesel Engine, *SAE 810680,* 1981.

[7] Situ, R., R. Brown, Z. Ristovski, U. Kruger, and D. Hargreaves, Analysis of dual fuel compression ignition (diesel) engine., in The Seventh International Conference on Modeling and Diagnostics for Advanced Engine Systems (COMODIA), 2008: Sapporo, Japan

[8] Surawski, N., B. Miljevic, B. Roberts, R. Modini, R. Situ, R. Brown, S. Bottle, and Z. Ristovski, Particle emissions, volatility, and toxicity from an ethanol fumigated compression ignition engine, *Environmental Science and Technology,* 2010, **44**(1): pp 229-235.

[9] Sterling, D. and L. Goldsworthy, Energy Efficient Fishing: a 2006 review - Part A Alternative fuels and efficient engines - Project No. 2005/239 www.frdc.com.au/commercial/energy.php, 2006. Fisheries Research and Development Corporation, Australian Government

[10] Goldsworthy, L., Combustion Behaviour of a Heavy Duty Common Rail Marine diesel Engine Fumigated with Propane, *Experimental Thermal and Fluid Science,* 2012, **42**: pp 93-106.

[11] Kim, M.Y., S.H. Bang and C.S. Lee, Experimental Investigation of Spray and Combustion Characteristics of Dimethyl Ether in a Common-Rail diesel Engine, *Energy & Fuels,* 2007, **21**: pp 793-800.

[12] Mancaruso, E., S. Merola and B. Vaglieco, Study of the multi-injection combustion process in a transparent direct injection common rail diesel engine by means of optical techniques, *International Journal of Engine Research,* 2008, **9**: pp 483-498.

[13] Heywood, J.B., Internal Combustion Engine Fundamentals 1988: McGraw-Hill (USA).

[14] Orfanidis, S.J., Introduction to Signal Processing 1996, Englewood Cliffs, NJ: Prentice-Hall.

[15] Green , R.H. and D.W. Perry, Perry's Chemical Engineers Handbook 2007: McGraw Hill Professional, USA.