Numerical Modeling of a Jet Ignition Direct Injection (JI DI) LPG Engine

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Abstract

The paper presents simulations of the operation of a liquefied petroleum gas (LPG) engine fitted with Direct Injection (DI) and Jet Ignition (JI). The liquid propane rapidly evaporates and mixes with the air after injection in the main chamber (MC). The mixture in the MC may be much closer to stoichiometry than a diesel. Combustion within the MC is limited by the turbulent mixing rather than the vaporization and diffusion processes of the injected fuel of the diesel. The engine may have diesel like efficiencies and load control by quantity of fuel injected. The engine may also have a better specific power at the low engine speeds typical of the diesel. This design also works at the high engine speeds impossible for the diesel.

Keywords: internal combustion engines, direct injection, jet ignition, liquefied petroleum gas

1. Introduction

The Jet Ignition (JI) device is a small pre-chamber (PC). This PC is united to the main chamber (MC) through calibrated orifices. It contains one direct injector and one spark plug, as it is shown in Fig. 1. This figure presents a sketch detailing the JI device (a). The JI device is a small volume with fitted one direct injector and one spark plug. It is attached to the MC volume through adjusted orifices. The sketch refers to a 14 mm thread design to fit the space available in the cylinder head of a spark ignition engine for a traditional spark plug. This figure then presents earliest (b) and latest (c) experimental flame fronts generated by a 6 nozzle central JI in a pent roof combustion chamber. Image (b) is from [6], image (c) is from [23]. The figure finally presents side (d) and (e), bottom (f) and top (g) views of the DI JI combustion system comprising the JI PC and the MC. The images (d) to (g) provide an overview of the volume where the fuel injected by the two injectors and air mixes and then ignites starting combustion from the spark plug location in the PC for a configuration previously studied by CFD. The PC mixture is locally homogeneous and close to stoichiometric, preferably slightly rich of stoichiometry. The spark discharge ignites the PC mixture. The hot reacting gases issued from the PC travel at high speed across the MC bulk-igniting the MC mixture. The MC mixture may be homogeneous or stratified, stoichiometric or lean.

Compared with classic Ricardo PCs used in indirect injection engines, the JI PC is a very small volume. It accounts for only a very few percent of the top dead centre (TDC) combustion chamber volume. It only receives a few percent of the total fuel rather than the totality of the fuel. Therefore, the JI PC is not expected to suffer the same downfalls of the classic Ricardo PCs that have been discontinued for Diesel engines in favor of direct injection (DI).

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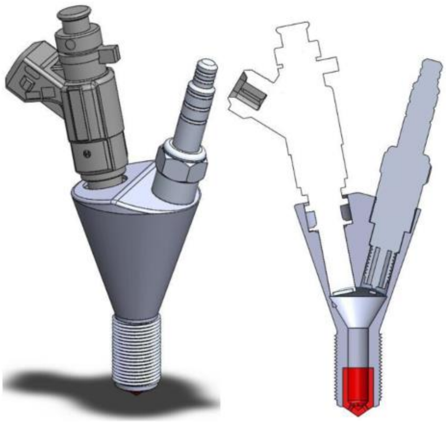
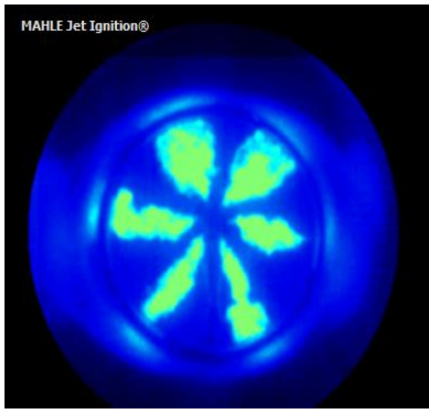
The specific design used here was first proposed by Professor Harry Watson in the early 1990s. Professor Watson started to work on this concept back in the 1980s. Initially, it was intended to be used with hydrogen as the JI PC fuel and almost any fuel for the MC. Then, it has been then extended to a variety of different fuels for both the JI PC and the MC, including LPG.

The JI device here considered has been studied in The University of Melbourne, Parkville, Australia over more than 25 years. The JI device has been operated with hydrogen or with other fuels. The United States patent "Internal combustion engine ignition device" [1] was filed in 1994, and published in 1997. Papers have been published even earlier, as for example [2, 3 and 4]. Many Master theses, PhD theses, journal articles and conference papers have been published presenting the studies performed under the guidance of Professor Harry Watson, with references [2-8] just a few examples of the many.



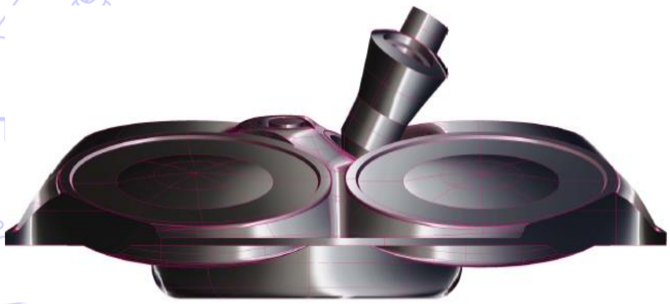
(a) JI device

(b) JI combustion visualization

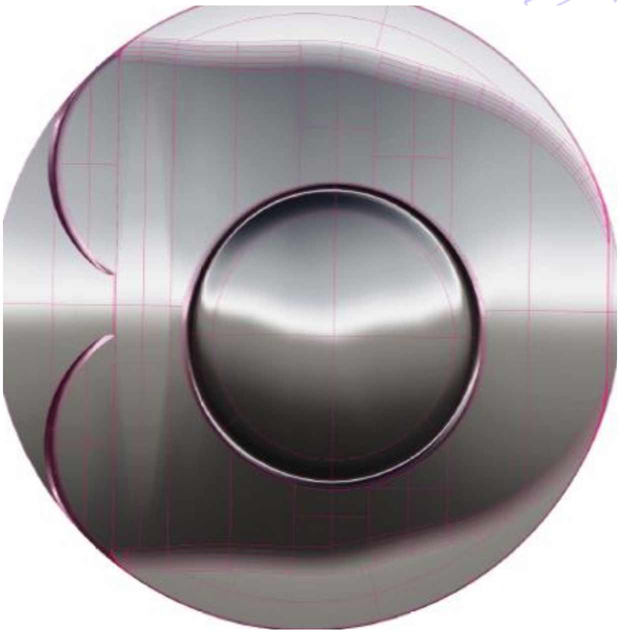


(c) JI combustion visualization

(d) side view of JI PC and MC



(e) side view of JI PC and MC



(f) bottom view of JI PC and MC

(g) top view of JI PC and MC

Fig. 1 DI JI combustion system

The coupling of JI to DI with traditional and alternative fuels, liquid or gas was more recently covered in the Australian patent [9] filed in 2009. The author is working on this subject since 2006, with references [10-18] reporting on the subject of jet igniting lean to stoichiometric, homogeneous to stratified, mixtures of a variety of conventional and alternative fuels, liquid or gaseous.

Recently, other groups as for example Mahle [19-23], have adopted JI for passenger car as well as racing engines applications. Mahle has been working so far mostly with homogeneous mixtures and without fully exploring the opportunities given by the coupling with DI. As JI of lean, homogeneous mixtures has permitted much faster combustions and diesel like thermal efficiencies, the technique is finally receiving the attention it deserves, with reference [24] only one example of the many headlines the innovation has recently received.

It is not the purpose of this paper to re-propose experimental and computational results that may be found in the above references, even if the younger readers may possibly ignore the existence of the oldest papers. This paper discusses the performances of a LPG directly injected JI engine based on simulations.

2. DI JI State-of-the-Art

The JI makes the MC ignition process much faster and widespread. This leads to a much faster and complete combustion within the MC. JI has been initially coupled to port fuel injection and homogeneous mixtures and then extended by the author [9-18] to DI and homogeneous or stratified mixtures. Lean homogeneous or stratified mixtures as well as homogeneous stoichiometric mixtures have all been investigated. With late direct MC injection, JI produces better lean combustion results than with early direct MC injection or port injection. This is because the MC fuel may be concentrated in a smaller area at the center of the combustion chamber with local conditions close to stoichiometry. A patch of a stoichiometric mixture is much easier to ignite and burn than a patch of a lean mixture. The direct MC injection may also occur at least partially after the JI occurs, thus modulating the premixed and diffusion combustion in ultra-lean operation. The operation of a conventional gasoline engine is characterized by the combustion of a premixed, stoichiometric, homogeneous mixture started in a tiny area between the spark plug electrodes close to the wall. The combustion evolution is then described by the travel of the flame front that develops across the chamber and finally extinguishes at the walls. Combustion of the fuel occurs at the flame front where patches of burned and unburned mixture are pushed together by turbulence. With JI, combustion is controlled by the turbulent mixing time same of traditional spark ignition (SI). The turbulent mixing time is the ratio of the turbulent length scale, about constant for a given engine, and a turbulent velocity that is proportional to the mean piston speed. Opposite to the compression ignition diesel combustion, the homogeneous spark ignition gasoline combustion has a time duration that reduces with the engine speed. This translates in roughly constant combustion duration in terms of crank angle degrees. This is the reason why spark ignition gasoline engines for racing applications may run even well above 20,000 rpm, while compression ignition diesel engines for racing applications do not exceed the 4,500 rpm speed.

The introduction of a JI device has already proven to be effective in producing faster and more efficient bulk ignition of the MC mixture [1-23]. The hot jets of partially burned gases rich in radicals permit indeed to start combustion all over the paths of the jets thus translating in combustion by multiple flame fronts than quickly burn the available fuel. After ignition, the rate of combustion is still limited by the turbulent mixing of burned and unburned patches, thus permitting roughly constant MC combustion duration in terms of crank angle degrees.

The method has been applied mostly to the ignition of lean, homogeneous mixtures produced by port or early DI, and more recently also to the ignition of lean, stratified mixtures produced by late DI. In this case, the mixture is globally lean, but locally close to stoichiometric. The bulk-ignition of the lean stratified mixture produces diesel-like fuel conversion efficiencies and ability to reduce the load throttle-less by mixture quality. As the MC DI may also partially occur after the ignition event is started, the system may permit premixed only, diffusion and premixed, or diffusion only MC fuel combustion.

JI may also help with homogeneous change compression ignition (HCCI) operation. If the JI is phased at top dead centre, near knocking mixtures that haven't auto-ignited yet may be ignited and burned quickly. This makes stable and repeatable a process that otherwise is often erratic. This opportunity is called controlled HCCI.

As previously mentioned, the JI fuel combustion has been covered by the group of the author in [10-18], embracing either homogeneous or stratified premixed combustion as well as stratified premixed and diffusion combustion. The JI of only homogeneous mixtures has also been covered by other groups, as for example Mahle [19-23]. These references do not constitute a survey of the literature on jet ignition, but only the prior background information on the specific JI application proposed in the paper, where the interested reader may certainly find further information.

3. Design of a LPG DI JI Engine

Results presented here refer to a directly injected engine, turbocharged with intercooler, fueled with LPG. Liquefied petroleum gas (LPG) is a more environmentally friendly alternative to diesel and gasoline, also enhancing sustainability of transport, energy economy and energy security. The advantages of LPG are discussed in [36]. Liquefied Petroleum Gas (LPG) is a fossil fuel clean burning alternative to gasoline and diesel. While gasoline and diesel are complex mixtures of heavier hydrocarbons liquid at ambient pressure and temperature, LPG is in the most part of the markets mostly propane C_3H_8. With better hydrogen to carbon ratio, and the availability in gaseous form at ambient temperature and pressure, LPG-fueled vehicles produce significantly lower amounts of carbon dioxide (CO_2) and harmful emissions, as the for example the particulate typical of the diesel or the non-methane hydrocarbon (NMHC) emissions typical of the gasoline. LPG is also usually less expensive than gasoline or diesel.

The paper presents simulations performed with a previously validated model of the operation of a LPG engine fitted with DI and jet ignition. The LPG fuel is injected into the MC by a direct injector and ignited by jet ignition. The parameters of th e injection and ignition are adjusted to command the amount of premixed or diffusion combustion. Aim is to produce the best fuel conversion efficiency for every requested load and speed within the prescribed constraints (pressure and temperature, combustion rate, combustion stability, pollutants formation). The emissions issue is not addressed in this work. While computer codes may work reasonably well to assess the fundamental aspects of mixture formation and combustion evolution, their predictions of pollutants are still far from being accurate. Starting from a turbocharged directly injected engine architecture, the cylinder head had a direct LPG injector and the JI device accommodating another LPG injector and the spark plug fitted.

The volume of the JI PC is larger than the small volume of the first hydrogen assisted JI (HAJI) concept, and operated with LPG rather than hydrogen, where the JI assembly is designed to replace a normal spark plug, but obviously also significantly smaller than the typical indirect injection Diesel engine PCs of the past.

The role of the JI PC is to provide bulk ignition of the MC mixtures homogeneous or stratified as created by the direct injector, as well as to produce suitable conditions for the MC to have the gaseous fuel injected later to burn diffusion controlled. The fuel injected before the JI event burns premixed. The fuel injected after the JI event burns diffusion.

The DI JI engine permits 3 modes of combustion.

- Gasoline-like, premixed if all the main LPG fuel is injected before the JI event.
- Diesel-like, diffusion if all the main LPG fuel is injected after the JI event.
- Mixed gasoline/Diesel like injecting the LPG fuel before and after the JI event.

The gasoline-like, premixed mode may include a homogeneous charge compression ignition (HCCI)-like mode. In this case, an amount of main LPG fuel smaller than the threshold value producing top dead center auto ignition is ignited at top dead center by the jet ignition. This translates in a more robust, stable and repeatable HCCI operation unaffected by small changes in properties and composition of the fuel air mixture.

Thanks to the opportunity to run the different combustion modes described above with mixtures from extremely lean to stoichiometric, the novel design improves the power and torque density of the engine as well as the fuel efficiency and the pollutants' emission over the load and speed map.

Turbulent JI systems for homogeneous premixed spark ignition engines using small PC systems are now receiving significant attention by Original Equipment Manufacturers (OEM) as well as their suppliers [19-24].

The major advantage of JI systems for racing applications is that they permit very fast burn rates due to the ignition system producing multiple, distributed ignition sites. The homogeneous main charge mixture about stoichiometric is then consumed rapidly and with minimal combustion variability.

Coupling direct injection to JI in the proposed combustion concept permits finely tuned premixed and diffusion modes of combustion. This permits faster and more complete combustion events. In addition to the single fuel operation, the dual fuel operation permits to burn almost any MC fuel with the help of JI "quality" fuel as methane, propane, hydrogen or gasoline.

4. Results and Discussion

The contribution proposes computational results over the full range of speeds and loads for a 1.6 litre high speed DI engine modified to accept the DI of Liquefied Petroleum Gas (LPG) fuel and jet ignition.

The validation of the baseline diesel engine model vs. experiments is shown in the Appendix. The baseline diesel engine model has been extensively validated versus detailed experiments. The figures below present the computed vs. experimental brake specific fuel consumption (BSFC) vs. the brake mean effective pressure (BMEP) at different speeds. Full load results for BMEP, BSFC, brake efficiency, brake power, fuel and air flow rate, maximum cylinder pressure (Cylinder 1) and boost are also presented vs. the engine speed. Combustion is modelled through Wiebe function parameters experimentally derived that are tabulated vs. speed and load. The modifications of this model to represent the proposed DI JI LPG engine follow best practices and detailed CFD simulations of combustion for diesel and DI JI gas engines. As a model, even a good and verified one, is still a model in spite of its sophistication and legacy applications, dynamometer testing is certainly needed to support the model findings.

The simulation results are shown here as maps of performance parameters vs. speed and load (in revolution per minute RPM and BMEP) that are sufficient to assess the potentials of the technology in terms of fuel efficiency. The major area of concern is the modelling of combustion. The rates of heat release were previously investigated for a variety of fuels experimentally and by using detailed chemistry models as STAR-CCM+ [25] and SRM suite [26], as well as with the turbulent combustion models of GT-SUITE [27] and Ricardo Wave [28]. All the above computer tools have been extensively validated by the developers and by the users. These experimental and computational activities are reported in [2-8] and [10-18, 19-23].

The net effect of JI is to make the turbulent combustion much quicker. If we consider for example the simple spark ignition turbulent combustion model of [28] (but similar argument applies to the eddy break up model used in CFD simulations), the turbulent combustion multiplier is doubled by JI vs. standard spark ignition.

The flame speed model of [28] is based on, and extends, the work of [29-33]. The development of the flame is modelled as a turbulent entrainment process. This entrainment is followed by the burning behind the flame front. The rate of entrainment for unburned gas is a function of the flame front area and the entrainment velocity:

$$\frac{dM_e}{dt} = \rho_u A_e \left(S_t + S_l \right) \tag{1}$$

where M_e is the mass of entrained unburned gas, r_u the unburned gas density, A_e the flame front area, S_t the turbulent flame speed, S_l the laminar flame speed.

The equation for the burned mass, M_b, is then:

$$\frac{dM_b}{dt} = \frac{M_e - M_b}{\tau} \tag{2}$$

where τ is a time constant.

The burn is assumed to occur at the laminar flame speed, over a length scale typical of the micro scale of turbulence. The time constant is the ratio of the Taylor micro scale λ and the laminar flame speed:

$$\tau = \frac{\lambda}{S_l} \tag{3}$$

The model uses three empirical constants, a proportionality factor between turbulence intensity and turbulent flame speed, C_s, a constant relating to initial flame development representing an ignition delay, C_k and a constant relating the Taylor micro-scale to the turbulent length scale, C_l.

In the Eddy Break Up (EBU) Model, quite popular in the CFD simulations of few decades ago [34, 35], premixed combustion under hypothesis of very fast chemistry is controlled by the break-up of eddies from the unburnt mixture, with duration of this break up controlling the rate. The rate of combustion S_{fu} is thus proportional to the concentration of reactants, fuel or oxidizer, and products, divided by a turbulent mixing time ratio of the kinetic energy k to its dissipation rate ε.

$$S_{fu} = -\rho \frac{\varepsilon}{k} min\left[C_R m_{fu},\ C_R \frac{m_{ox}}{s},\ C_R^{'} \frac{m_{pr}}{1+s} \right] \tag{4}$$

where m_{fu}, m_{ox} and m_{pr} are the mass fractions of fuel, oxidizer and products, s is the stoichiometric ratio, ρ is the density of the mixture. As a net effect of jet ignition, the proportionality factors C_R and $C_R^{'}$ are doubled vs. spark ignition.

Obviously, a better description of the combustion evolution is possible by coupling a detailed chemical kinetics with direct simulation of turbulence. As the time and space scales may become prohibitive, time and space average equations are used to represent turbulence [13, 14]. It is however not the subject of the paper to discuss these modelling and computational details.

The engine operation is modelled here by using Ricardo Wave [28], with combustion represented through the Wiebe correlation with empirically and computationally adjusted tabled constants. Figs. 2 to 8 are results of computations made by using Ricardo Wave [28], where however the combustion rates are set up by using prior measurements and computations of turbulent combustion with detailed chemical kinetics and simplified approaches.

The engine is a 1.6-liter turbocharged engine with intercooler of bore x stroke 80 x 80 mm. The compression ratio is 12.0:1. The engine has four valves per cylinder of diameter 21.6 and 19.2 mm respectively the intake and the exhaust. Maximum valve lifts are 7.44 mm for both the intake and the exhaust. Valve timings are 350/550 (-10/10) the intake, and 160/360 (-20/0) the exhaust (360 degrees' crank angle is the intake top dead center and 0-720 degrees' crank angle is the combustion top dead center). The LPG fuel supposed to be injected high pressure – 500 bar common rail injection system – through small orifices vaporizes and mixes very quickly.

Figs. 2 to 8 presents the map of maximum cylinder pressure, lambda, efficiency, power, pressure out of compressor, pressure and temperature inlet of turbine vs. speed and load. The maximum pressure increased with the load and it is within the 150 bar limit deemed acceptable in novel Gasoline Direct Injection (GDI) applications. The values of lambda are maximum at low speeds and low load conditions, up to values of 2.7. The engine is working near stoichiometry at top loads. The efficiency (ratio of brake power to fuel energy flow rate) is maximum for slightly lean of stoichiometry mixtures, reducing at higher rpm mostly because of the increasing frictions. The brake power reaches top values about 5,000 rpm. This is the result of the design for the low speeds typical of diesel engines of the intake system. The turbocharger boosting is strong especially at low speeds, with up to almost 2 bar of pressure at the exit of the compressor. The pressure intake of turbine increases with speed and load up to almost 3 bar. The temperature intake to turbine reaches about 900 °C about the peak power operating point. These results appear qualitatively correct, with however some changes expected in the eventual testing of the proposed engine design, with opportunity to do even better thanks to the fine turning of the injection events to the spark discharge starting ignition in the MC.

The use of JI permits (1) to work much closer to stoichiometry than the Diesel and (2) the opportunity to run much higher speeds, as the MC combustion duration is controlled mostly by the turbulent mixing of burned and unburned patches rather than the time needed to vaporize, mix with air and auto-ignite. The rate of heat release may also be larger than in the Diesel, providing much closer to isochoric conditions. The higher loads are premixed, the lower loads mixed premixed/diffusion. The mixed premixed/diffusion operations work very close to the Diesel operation.

The baseline Diesel engine CAE model is validated versus detailed experimental measurements. CFD, CAE and detailed chemical kinetics simulations have been previously performed for Diesel and gasoline engines, as well as dual fuel diesel ignition, or alternative fuel and gasoline JI and validated vs. the experimental evidence. Therefore, even if there are no experimental result obtained on the proposed LPG engine featuring jet ignition, all the building blocks of the model have been previously validated.

The 1D software and the tuning of the Wiebe function has been previously fully validated against dynamometer data for the baseline Diesel engine. The Wiebe function parameters with JI of LPG have then be redefined follow prior 3D CFD simulations following best practices developed through validation works for Diesel, spark ignition and JI combustion. The specific changes done to the 1D model in order to represent the behavior of turbulent JI systems are the tabled Wiebe function parameters obtained from the CFD studies.

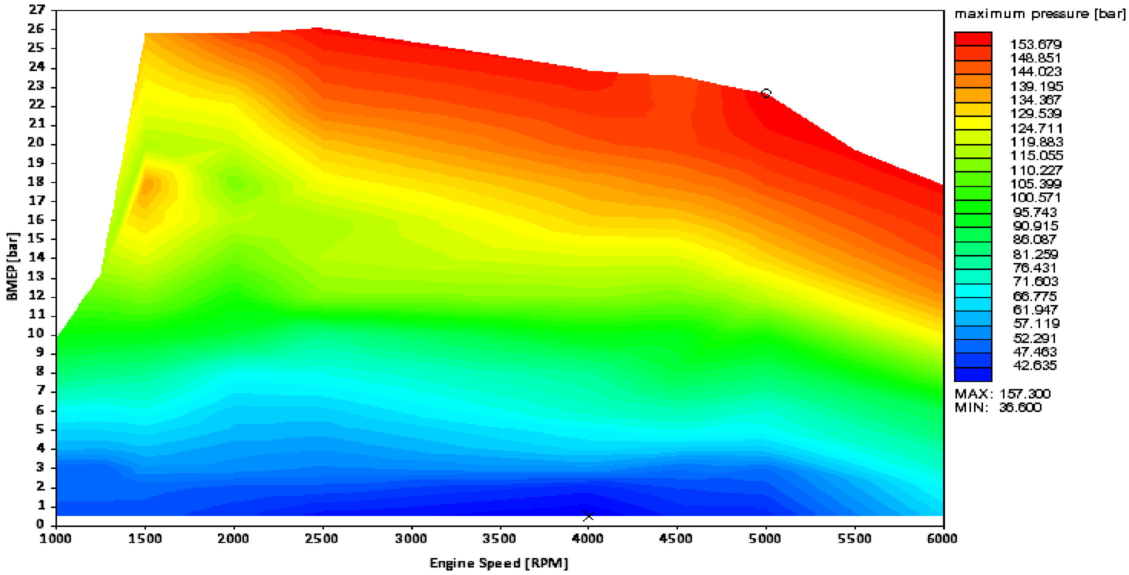


Fig. 2 Map of maximum pressure vs. speed and load (BMEP)

The benefits offered by JI vs. spark ignition are Diesel-like load control throttle-less thanks to the ability to run with ultra-lean mixtures, combustion rates generally faster and more complete, and reduced heat losses working lean stratified. The advantages versus commonly used direct and or port fuel injection spark ignition engine systems may be obtained by comparing the efficiency map of Fig. 4 with the efficiency map of reference traditional turbocharged gasoline engines. The major differences are higher efficiency over the most part of the load range.

The benefits offered by JI vs. the Diesel are the opportunity to run much closer to stoichiometry and much higher engine speeds. As the Diesel combustion is limited not by the turbulence mixing but the time needed by the fuel to vaporize and mix with the air and then ignite, the combustion duration in terms of crank angle degrees increased dramatically with the engine speed. This is the reason why even Diesel engines for racing applications do not exceed the 4,500 rpm. With JI, the rpm range of engine speeds of Diesel-like operation may be increased. An assessment for constant operating conditions against commonly used DI Diesel engine systems may be obtained by comparing the efficiency map of Fig. 6 with the efficiency map of reference turbocharged Diesel engines. While, at low speed, the Diesel engine may be superior, the JI engine permits much higher loads, and has an efficiency curve stretched towards higher rpm not suffering the drastic drop of performances approaching the 4,500 rpm.

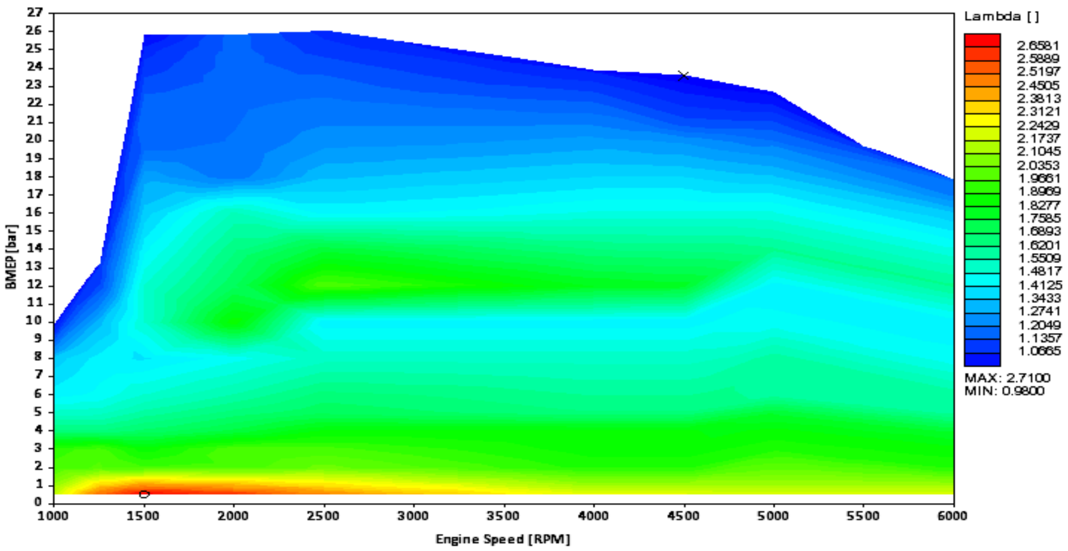


Fig. 3 Map of lambda vs. speed and load (BMEP)

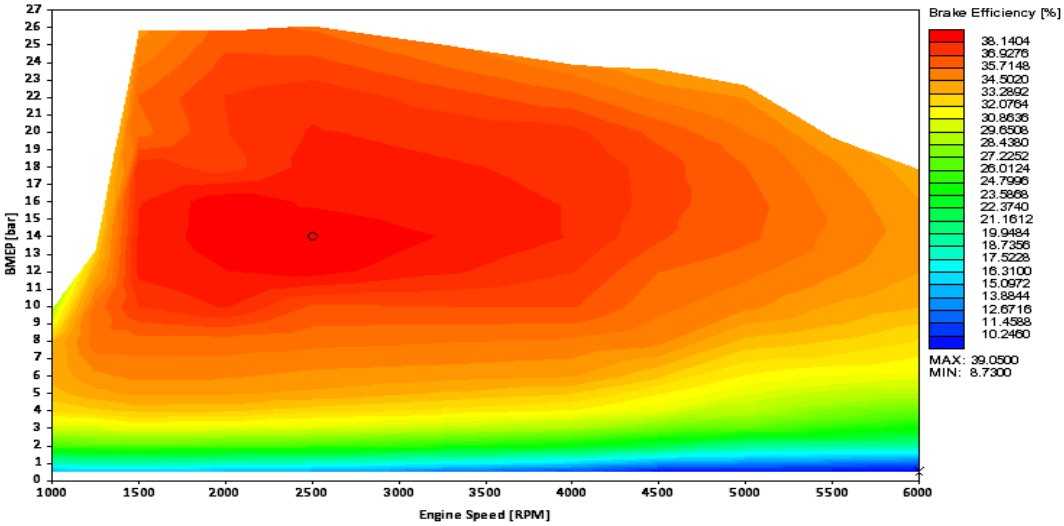


Fig. 4 Map of efficiency vs. speed and load (BMEP)

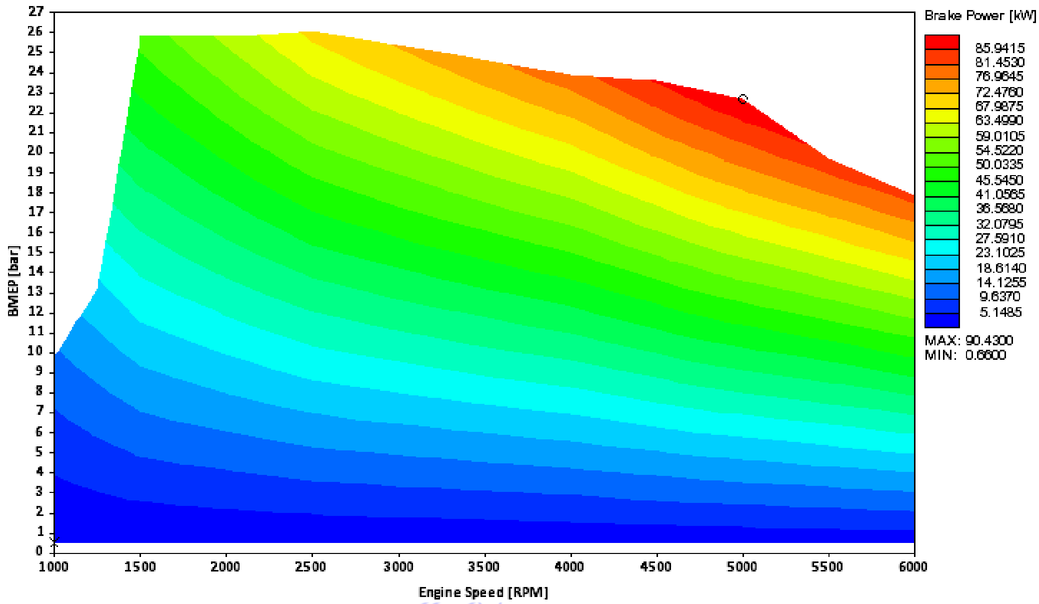


Fig. 5 Map of brake power vs. speed and load (BMEP)

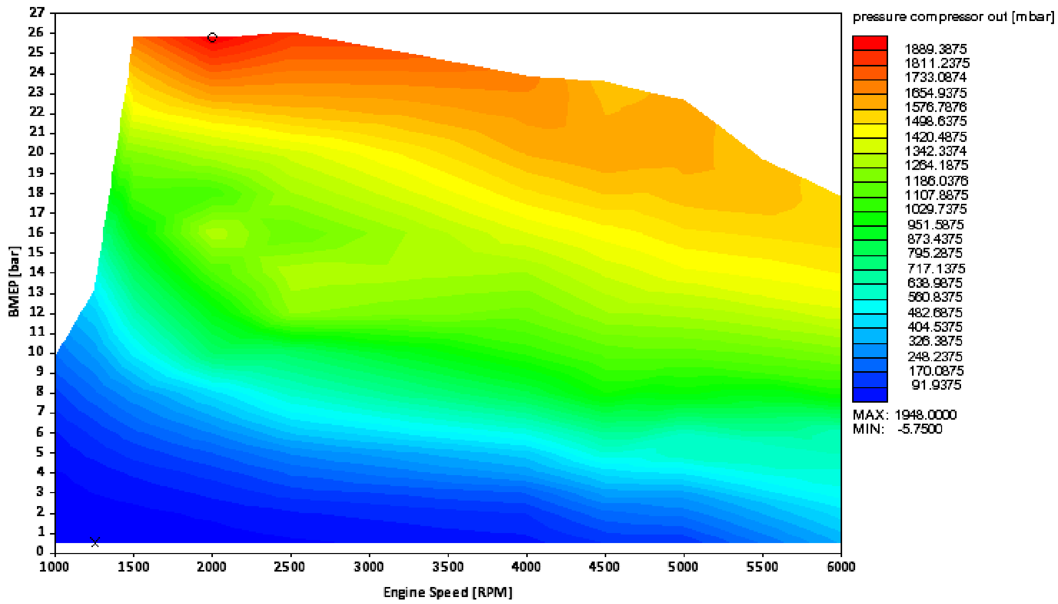


Fig. 6 Map of pressure out of compressor vs. speed and load (BMEP)

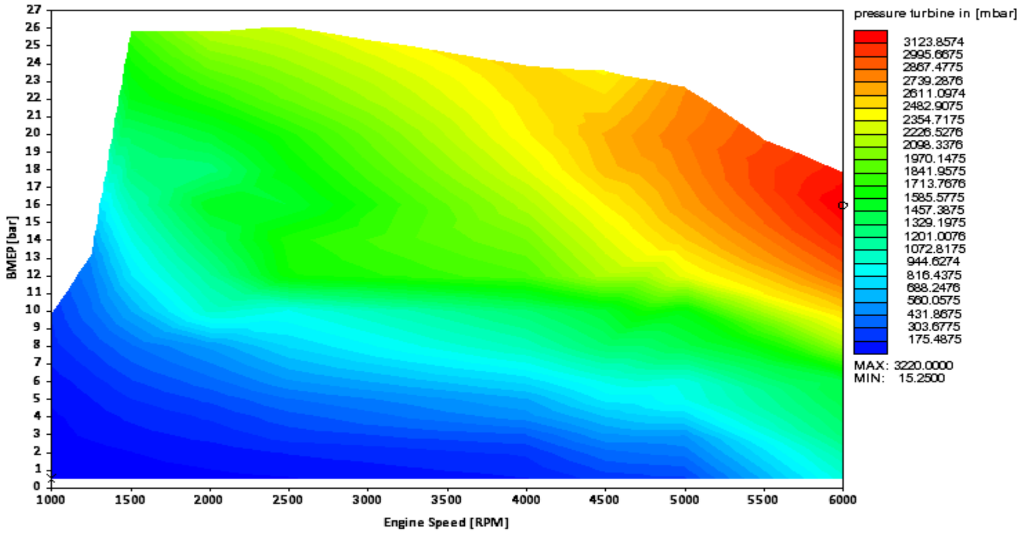


Fig. 7 Map of pressure inlet of turbine vs. speed and load (BMEP)

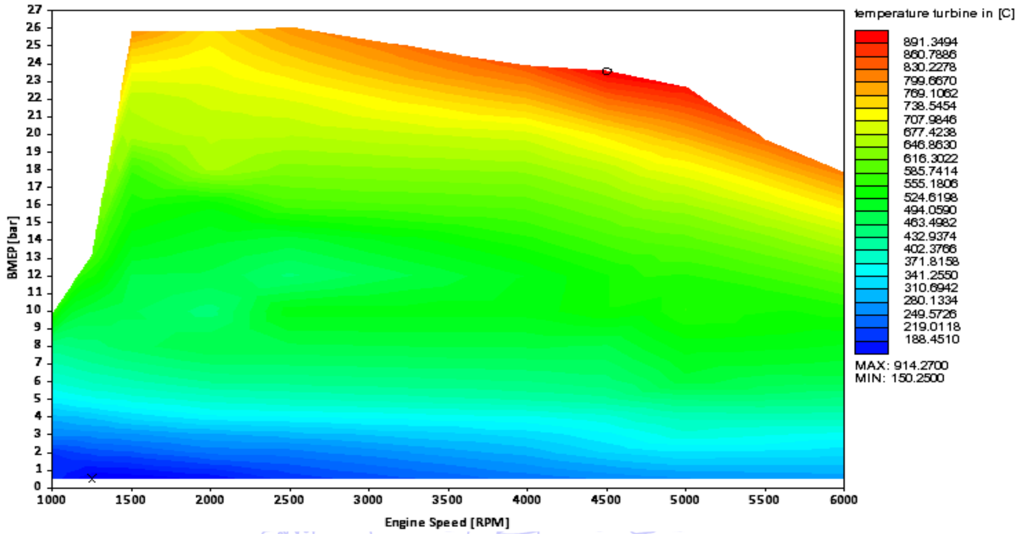


Fig. 8 Map of temperature inlet of turbine vs. speed and load (BMEP)

5. Conclusions

Jet ignition (JI) permits much faster and more complete combustion thanks to the opportunity to bulk ignite lean mixtures with multiple jets of hot reacting gases. Simulations have been presented for the operation of a 1.6 litre engine fuelled with LPG. The engine equipped with JI has about same of Diesel engine efficiency at low loads and low speeds, but may further increase both loads and speeds thanks to the premixed combustion controlled by the turbulent mixing. A model, even a good and verified one, is still a model in spite of its sophistication and legacy applications. Real-world dynamometer testing and comparison of major performance and efficiency parameters are certainly needed.

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List of Symbols and Acronyms

A	area
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
C	constant
CFD	computational fluid dynamic
DI	direct injection
EBU	Eddy Break Up
JI	jet ignition
k	turbulence kinetic energy
LPG	liquefied petroleum gas
m, M	mass
MC	main chamber
NMHC	non-methane hydrocarbon
OEM	Original Equipment Manufacturers
PC	pre-chamber
s	stoichiometric ratio
S	speed
T	time
TDC	top dead centre
ρ	density
τ	time scale
λ	Taylor micro scale
ε	turbulence dissipation rate

Appendix – Validation of the baseline diesel engine model

The baseline diesel engine model has been extensively validated versus detailed experiments. Combustion is modelled through Wiebe function parameters experimentally derived that are tabulated vs. speed and load. The figures below present the computed vs. experimental BSFC vs. the BMEP at different speeds. Full load results for BMEP, BSFC, brake efficiency, brake power, fuel and air flow rate, maximum cylinder pressure (Cylinder 1) and boost are also presented vs. the engine speed. The modifications of this model to represent the proposed DI JI LPG engine is supported by detailed computational fluid dynamic (CFD) simulations of combustion for diesel and DI jet ignition gas engines. As a model, even a good and verified one, is still a model in spite of its sophistication and legacy applications, dynamometer testing is certainly needed.

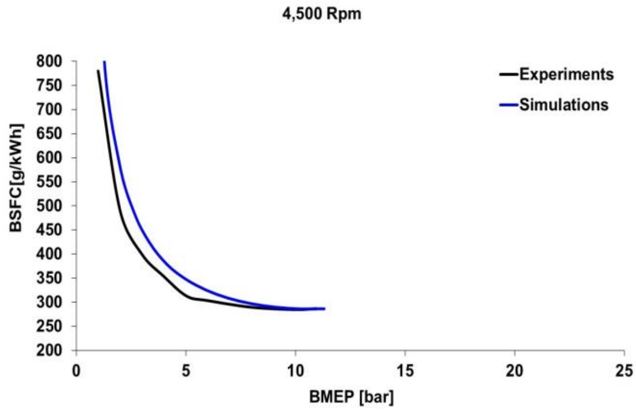

Fig. A1 BSFC vs. BMEP, 4,500 rpm

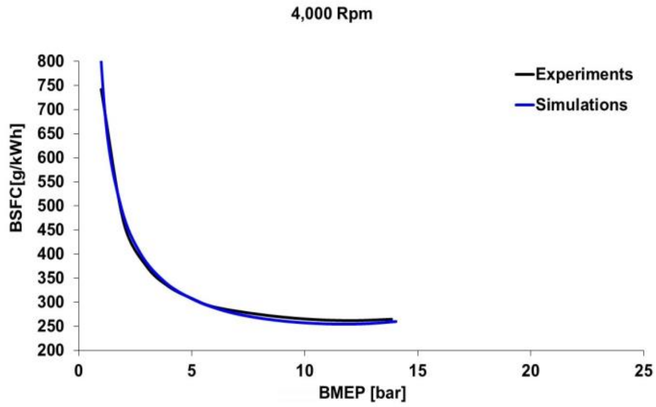

Fig. A2 BSFC vs. BMEP, 4,000 rpm

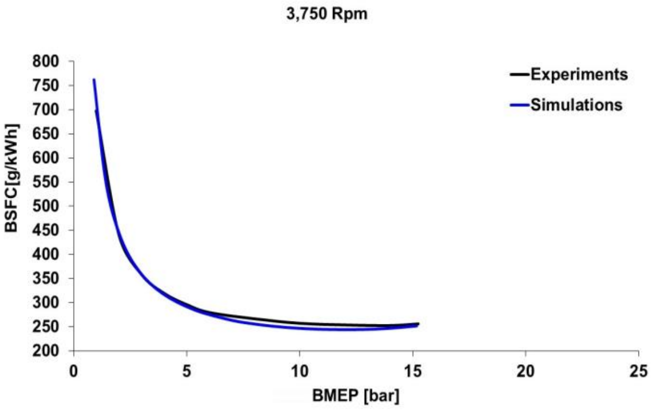

Fig. A3 BSFC vs. BMEP, 3,750 rpm

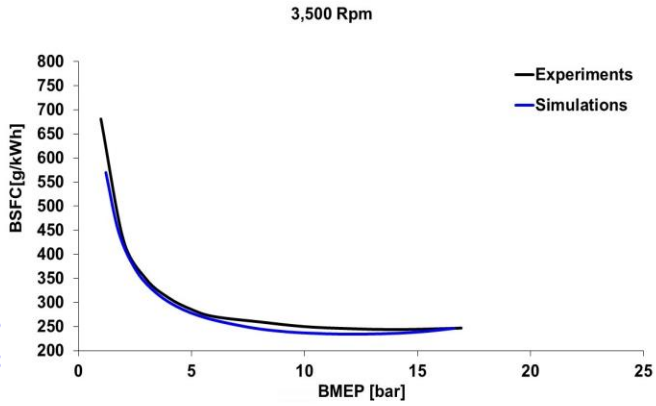

Fig. A4 BSFC vs. BMEP, 3,500 rpm

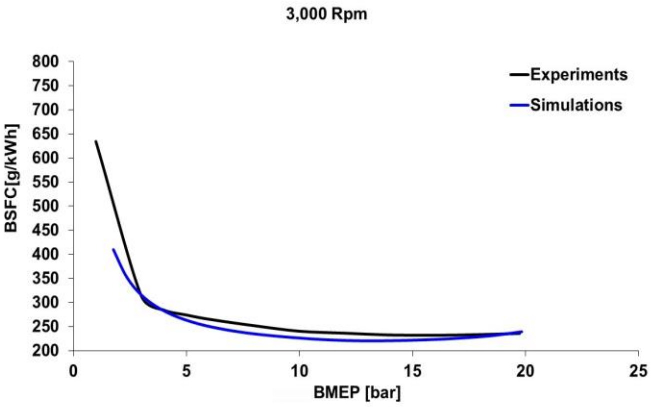

Fig. A5 BSFC vs. BMEP, 3,000 rpm

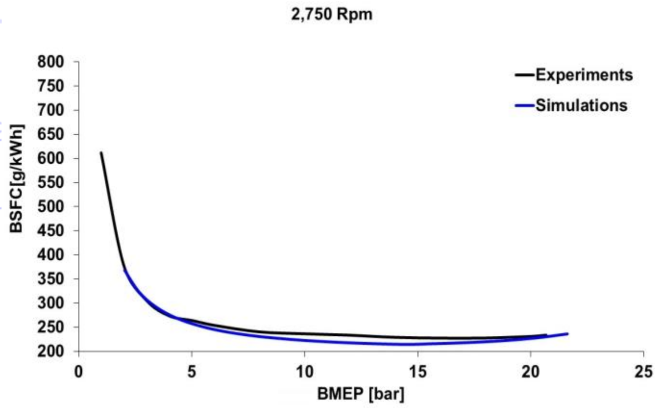

Fig. A6 BSFC vs. BMEP, 2,750 rpm

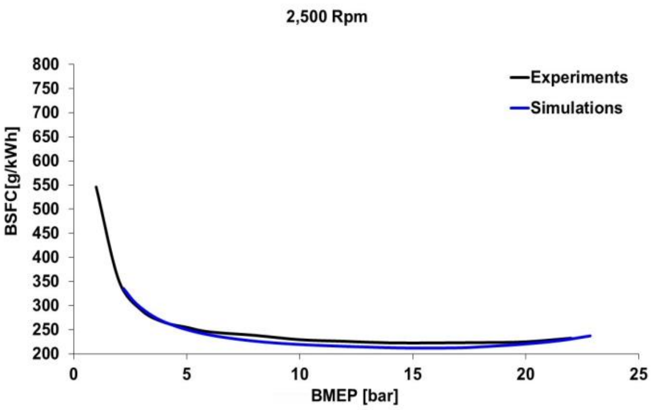

Fig. A7 BSFC vs. BMEP, 2,500 rpm

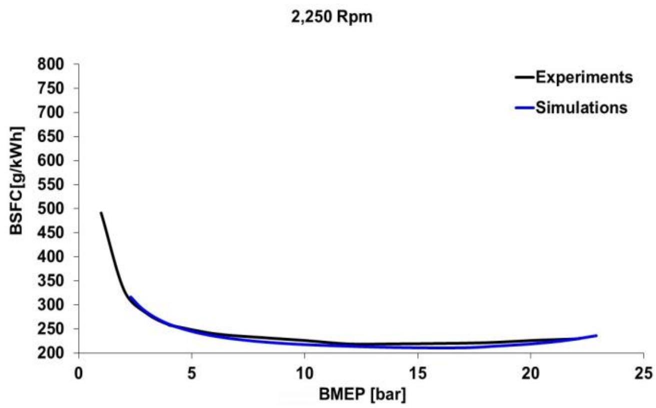

Fig. A8 BSFC vs. BMEP, 2,250 rpm

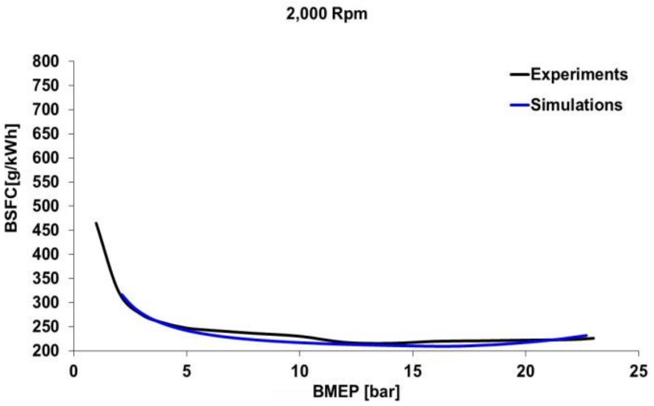

Fig. A9 BSFC vs. BMEP, 2,000 rpm

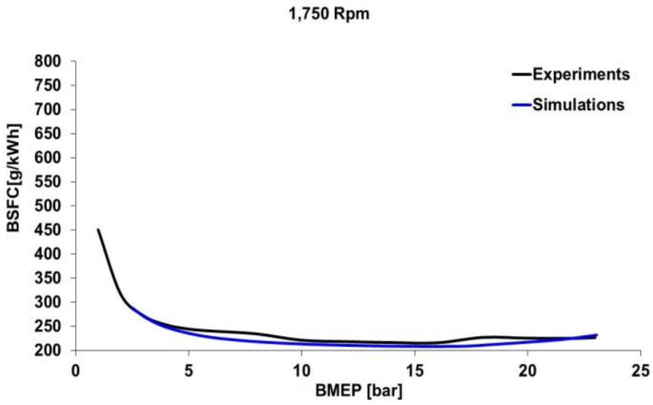

Fig. A10 BSFC vs. BMEP, 1,750 rpm

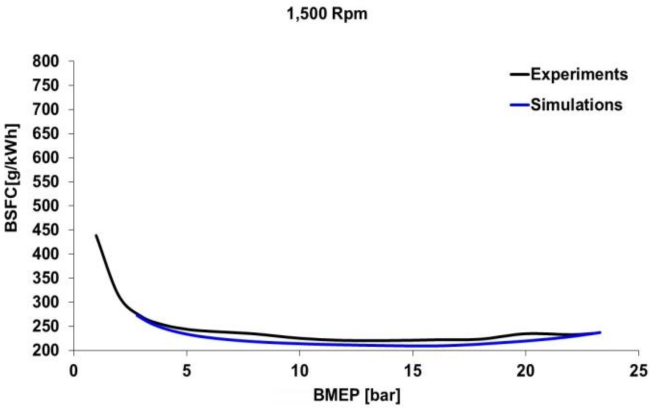

Fig. A11 BSFC vs. BMEP, 1,500 rpm

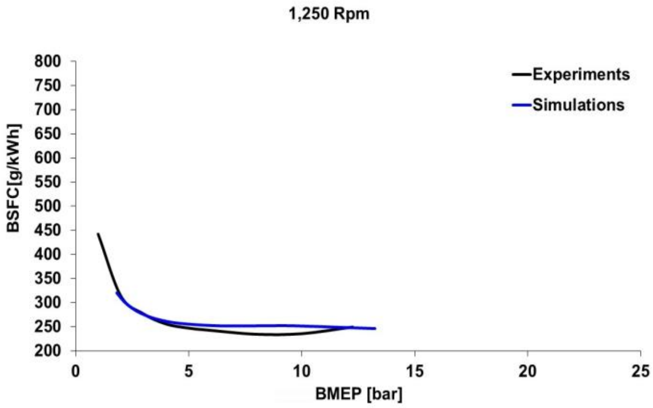

Fig. A12 BSFC vs. BMEP, 1,250 rpm

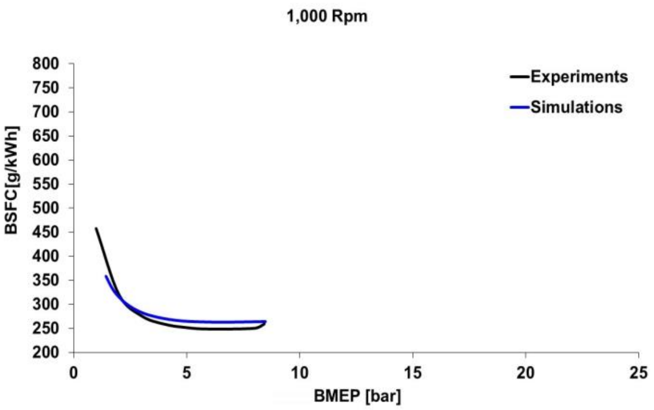

Fig. A13 BSFC vs. BMEP, 1,000 rpm

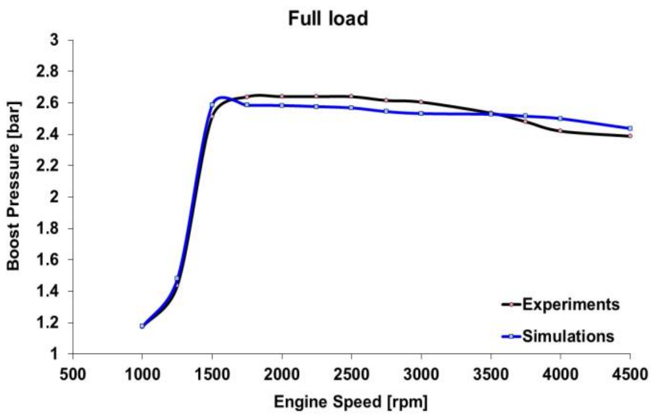

Fig. A14 Boost pressure vs. speed, full load

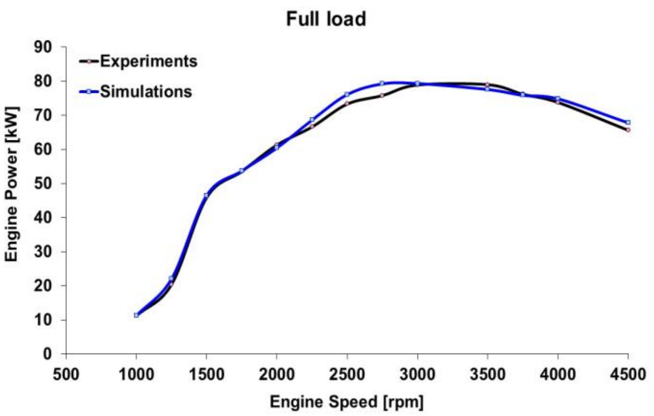

Fig. A15 Power vs. speed, full load

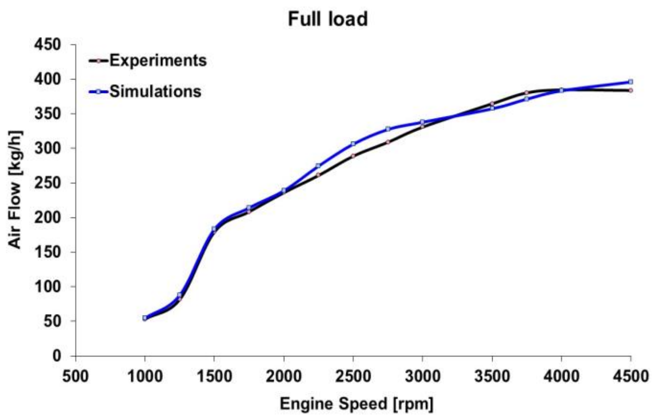

Fig. A16 Air flow vs. speed, full load

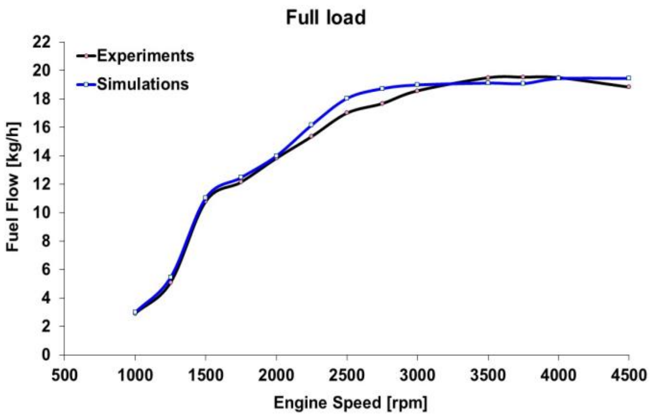

Fig. A17 Fuel flow vs. speed, full load

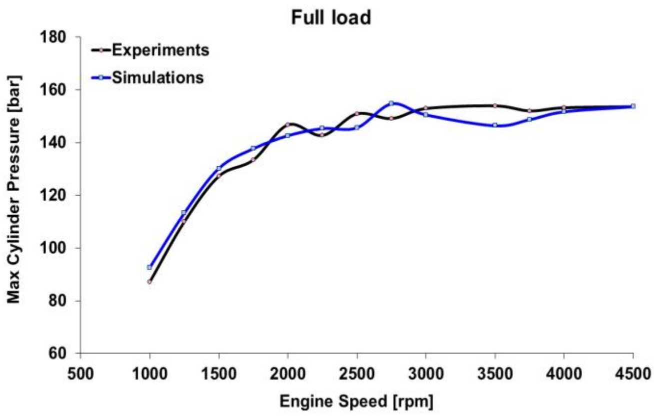

Fig. A18 Max. cylinder pressure vs. speed, full load

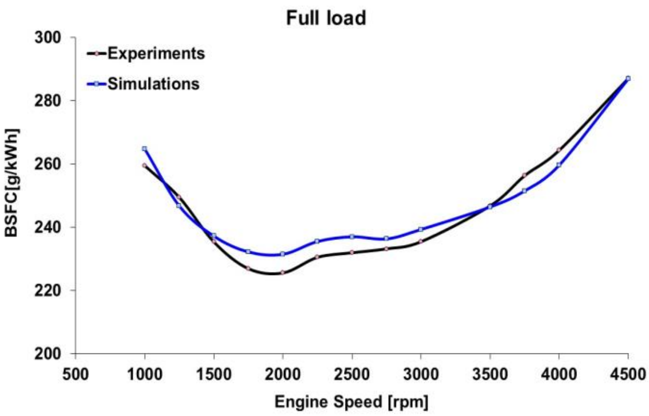

Fig. A19 BSFC vs. speed, full load

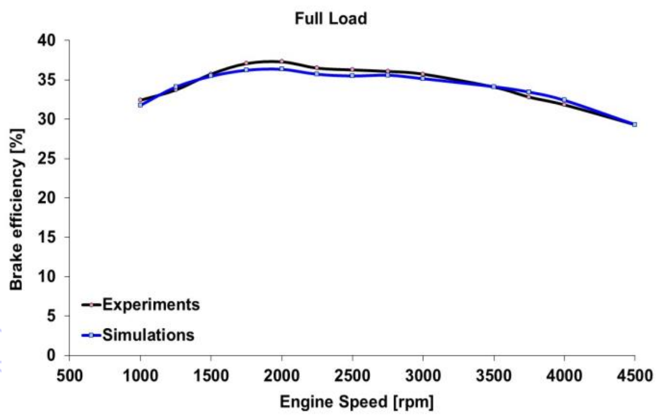

Fig. A20 Brake efficiency vs. speed, full load

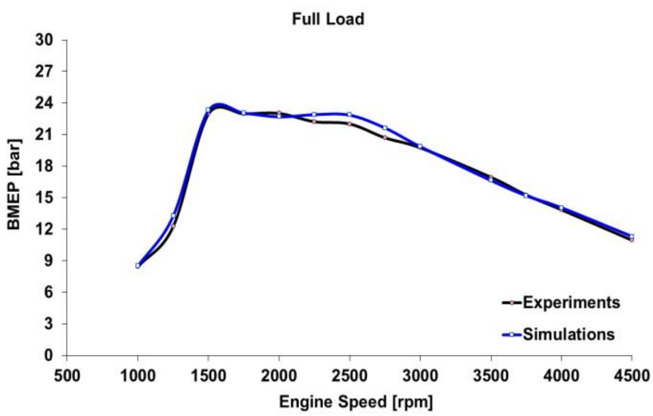

Fig. A21 BMEP vs. speed, full load