

# 070 THE LEAN BURN DIRECT-INJECTION JET-IGNITION TURBOCHARGED LIQUID PHASE LPG ENGINE

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## Abstract

This paper explores the use of Liquefied Petroleum Gas (LPG) fuel in a 1.5 liter Spark Ignition (SI) four cylinder gasoline engine with double over head camshafts, four valves per cylinder equipped with a novel mixture preparation and ignition system comprising centrally located Direct Injection (DI) injector and Jet Ignition (JI) nozzles. The engine is turbocharged for high power density and high efficiency running lean. The LPG fuel is injected in liquid phase in the main chamber. The new generation fast actuating, high pressure, high flow rate injector is capable of injection shaping and multiple events and produces bulk lean stratified mixtures. The jet ignition system uses a second direct injector to inject a small amount of fuel in a small pre chamber, where a spark plug ignites a slightly rich to stoichiometric mixture then igniting the main chamber mixture through multiple jets of hot reacting gases. Bulk ignition and combustion of the lean, jet controlled, stratified main chamber mixture resulting from coupling DI with JI enables high efficiency burning of main chamber mixtures with fuel-to-air equivalence ratios reducing almost to zero for a throttle less control of load diesel-like.

**Keywords:** Gas engines, direct injection, bulk jet ignition, lean burn stratified bulk combustion.

## 1. INTRODUCTION

Liquefied Petroleum Gas (LPG) is one of the gaseous fuels of major interest in Australia because of its cost advantage over gasoline and its nationwide distribution network. This paper evaluates LPG fuel's potential for operation in an engine targeting high efficiency.

LPG is a mixture of hydrocarbon gases used as a fuel in vehicles. Varieties of LPG include mixes that are primarily propane, primarily butane or including propane and butane (usually 60% and 40%). Propylene and butylenes are frequently also present in small concentrations. LPG is typically derived from fossil fuel sources, being manufactured during the refining of crude oil, or extracted from oil or gas streams as they emerge from the ground. At normal temperatures and pressures, LPG is a vapor and to remain as a liquid is supplied pressurized. The pressure at which LPG becomes liquid depends on composition and temperature. To illustrate the range it is approximately 2.2 bar for pure butane at 20 °C, and approximately 22 bar for pure propane at 55 °C. LPG is also a low carbon emitting hydrocarbon fuel, emitting less CO<sub>2</sub> per kWh than gasoline but more than CNG. LPG burns cleanly with no soot and very few sulfur emissions unless mercaptans are added as an odorant. LPG has a typical specific calorific value of 46.1 MJ/kg compared to 42.5 MJ/kg for diesel and 43.5 MJ/kg for gasoline. Its energy density per unit volume is lower than either gasoline or diesel. LPG has a RON of 109 and a MON of 96. Therefore, LPG allows higher compression ratios than gasoline.

Main chamber direct injection of LPG in liquid phase with fast actuating, high flow rate, high pressure injectors capable of injection shaping and multiple events and main chamber jet ignition, with ignition by spark in a small volume pre chamber with pre chamber mixture preparation by pre chamber direct injection, has never been considered before for both stationary and

transport applications. In large volume pre chamber ignition systems for large gas engines, the pre chamber fuel is not negligible, the cylinder head design is strongly affected, and pre chamber combustion is important also in itself and not just in initiating main chamber combustion. In standard direct injectors, the direct injector is a traditional low cost, slow actuating, solenoid, low pressure, low flow rate injector. Finally, with standard main chamber spark ignition, combustion is wall initiated with a relatively small energy supply in just one location [7-17].

This paper considers a new generation fast actuating, high pressure, and high flow rate injector capable of injection shaping and multiple events producing a bulk lean jet controlled stratified mixture. High energy bulk ignition is then achieved by using pre chamber jet ignition. The proposed ignition pre chamber is very small in size, just a few percents of the combustion chamber volume at top dead centre and about 1cm<sup>3</sup>, and designed to be fitted within the traditional spark plug thread  $\square=14$  mm. The jets of reacting gases from the ignition pre chamber enhance the rate of combustion of the main chamber mixture and allow bulk ignition and combustion for reduced heat losses and faster heat release.

Fuel properties play a key role in development of the direct injection mixture preparation system. Figure 1 presents fluid properties of methane, propane and hydrogen along isothermal lines T=300 K [1]. This picture clearly states problems and opportunities of gas injection with variable pressure levels. Late direct injection overcomes the air displacement effects of port fuel injection of gaseous fuels. However, development of a direct injector providing adequate flow rates is difficult. Propane (C<sub>3</sub>H<sub>8</sub>) has critical temperature, pressure and density respectively T<sub>c</sub>=369.8 K, p<sub>c</sub>=42.5 bar and  $\rho_c=220.0$  kg/m<sup>3</sup>, while the normal boiling point is 231.1 K [1]. Methane (CH<sub>4</sub>) has critical temperature, pressure

and density respectively  $T_c=190.6$  K,  $p_c=46.0$  bar and  $\rho_c=162.7$  kg/m<sup>3</sup> while the normal boiling point is 111.7 K. Hydrogen (H<sub>2</sub>) has critical temperature, pressure and density respectively  $T_c=33.1$  K,  $p_c=13.0$  bar and  $\rho_c=31.3$  kg/m<sup>3</sup> while the normal boiling point is 20.4 K. At a temperature  $T=300$  K, propane is vapor for pressures below 10.0 bar, and liquid above. Conversely, methane is vapor for pressures below 48.4 bar, and supercritical above. Hydrogen is vapor for pressures below 13.5 bar, and supercritical above. Therefore, a dedicated LPG engine may inject fuel in liquid phase, features quite difficult if not impossible with CNG and hydrogen.

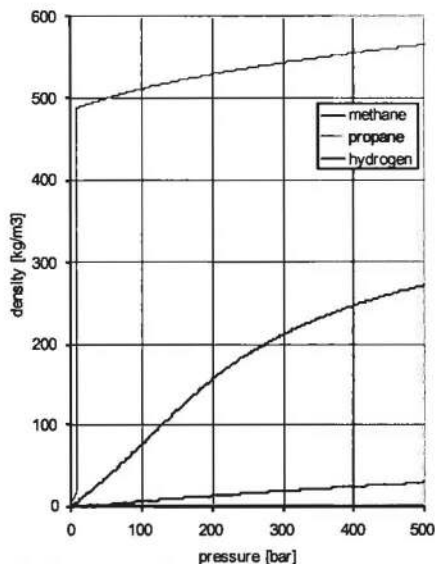


Figure 1 – Propane, methane and hydrogen isothermal density data for  $T = 300$  K [1].

Injections in liquid phase also offer the advantages of the temperature drop by the heat of vaporization of the liquid fuel, with embedded advantages in terms of knock resistance and charge density. Injection in liquid phase also improves the opportunities to achieve a bulk lean stratified mixture.

The coupling of JI and DI technologies allows development of an engine permitting operation with overall fuel-to-air equivalence ratios reducing almost to zero, because combustion is always started in the JI pre chamber providing a very small amount of fuel is injected there to produce an ignitable mixture, and the jets of hot reacting gases from the JI pre chamber may extend combustion to globally very lean main chamber mixtures providing only a minimum amount of fuel is behind the JI nozzle thus replicating Diesel-like light load operation.

The lean, jet controlled, stratified engine operating the new generation fast actuating, high pressure, high flow rate injector capable of injection shaping and multiple events and jet ignition is an innovative concept

by the Authors described in better details in the following section.

## 2. THE LEAN BURN DIRECT-INJECTION JET-IGNITION ENGINE

The lean burn direct injection jet ignition [2] is an internal combustion engine developed to more efficiently and completely burn within the cylinder of a four stroke engine with or without supercharging or turbo charging for non stationary or stationary applications lean mixtures of a variety of fuels, including fuels liquid and vapor under normal conditions. This engine uses a fuel injection and mixture ignition system comprising:

- One direct main chamber fuel injector per engine cylinder;
- One jet ignition device per engine cylinder, the latter being made of:
  - One pre chamber connected to the main chamber through one or more calibrated orifices;
  - One direct pre chamber fuel injector;
  - One pre chamber spark plug;
- All the ancillaries required to supply the fuel at the desired pressures by the direct injectors and operate the direct injectors and the spark plug.

The fuel injection and mixture ignition system operation is given as follows.

One fuel is injected directly within the cylinder by a main chamber direct injector operating one single or multiple injections to produce a lean stratified mixture. This non homogeneous mixture is mildly lean in an inner region surrounded by air and some residuals from the previous cycle. The extension of the inner region may be reduced in size to achieve mean chamber average mixtures ranging from slightly to extremely lean.

This mixture is ignited by one or more jets of reacting gases that issue from a pre-chamber connected to the main chamber via calibrated orifices, sourced from the same or an alternative fuel that was injected into it by a direct injector and then ignited by a spark plug discharge.

The pre-chamber has a very small volume and necessitates a small quantity of fuel to produce there a mixture richer than stoichiometry and necessitating a small energy to be ignited.

Combustion started in the pre chamber moves to the main chamber through one or more jet ignition nozzles, with one or more jets of reacting gases bulk igniting the main chamber mixture. The jets of reacting gases enhance combustion of lean stratified mixtures within the main chamber through a combination thermal energy and the presence of active radical species.

With reference to homogeneous direct injection or port injection and main chamber spark ignition, non homogeneous direct injection and jet ignition offer the advantage of:

- Much faster, more complete, much leaner combustion;
- Less sensitivity to mixture state and composition;
- Reduced heat losses to the main chamber walls.

This is because of better fuel for same chamber averaged lean conditions, combustion in the bulk of the

in cylinder gases, heat transfer cushion of air between hot reacting gases and walls, very high ignition energy at multiple simultaneous ignition sites igniting the bulk of the in cylinder gases, aided by large concentrations of partially oxidized combustion products initiated in the pre chamber that accelerate the oxidation of fresh reactants.

Advantages of the system are as follows:

- Larger brake efficiency (ratio of engine brake power to total fuel energy) and therefore reduced brake specific fuel consumption (ratio of engine fuel flow rate to brake power) for improved full load operation of stationary and transport engines.
- Efficient combustion of variable quality fuel mixtures from globally near stoichiometry to globally extremely lean for load control mostly throttle less by quantity of fuel injected for improved part load operation of non stationary engines.
- Opportunity in the ultra lean mode to produce near zero  $\text{NO}_x$ .

In four valve pent roof combustion chamber applications, charge motion controlled by jet and shaped piston surface/wall or fully jet controlled configurations are possible, while jet ignition may be single nozzle or multiple nozzles. Figure 2 presents a sketch of the cylinder head layout of the 4 valve Direct Injection Jet Ignition engine. Figure 3 presents a picture of the jet ignition assembly.

Prototypes Jet Ignition devices have been built using low cost gasoline direct injection injectors providing the requested low flow rates with gaseous fuel and small diameter racing spark plugs placed at the top of a jet ignition pre chamber having volume about  $1 \text{ cm}^3$ .

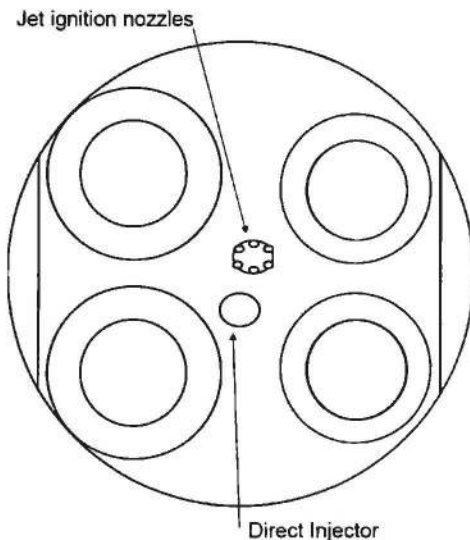


Figure 2 – Cylinder head layout of the 4 valve Direct Injection Jet Ignition engine.

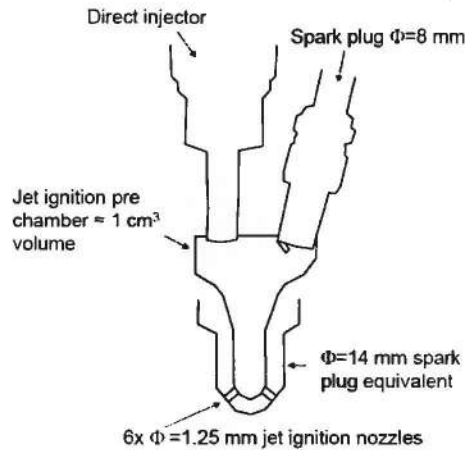


Figure 3 – Jet ignition assembly for the 4 valve Direct Injection Jet Ignition engine.

### 3. TURBOCHARGED LIQUID PHASE LPG ENGINE RESULTS

The new technology described above has been applied to develop a high power density turbo charged multi cylinder engine.

This engine is a V-Four 1.5 liters engine, with bore 78 mm, stroke 78 mm, 4 valves per cylinder, intake valve seat insert inside diameter 32 mm, exhaust valve seat insert inside diameter 26 mm, connecting rod length 109 mm, and pent roof combustion chamber. Valve timings for the intake and exhaust with valve lash 0.15 and 0.20 mm respectively are IVC  $80.2^\circ \text{ CA ABDC}$ , EVO  $80.2^\circ \text{ CA BBDC}$ , IVO  $41.3^\circ \text{ CA BTDC}$  and EVC  $32.0^\circ \text{ CA ATDC}$ . Maximum intake and exhaust valve lifts are 12.3 and 10.3 mm respectively. Intake and exhaust forward discharge coefficients for a valve lift to diameter ratio 0.5 are respectively 0.70 and 0.64. The compression ratio is  $\text{CR}=14.2$ .

The engine has been modeled with GT-POWER [3, 4 and 5]. GT-POWER is the industry-standard engine simulation tool, used by most leading engine and vehicle makers and their suppliers. Better details of the model are presented in [4, 5]. The model has been derived from a validated high performance engine model with PFI of gasoline [6]. Main differences are stroke and lengths of primary intake and exhaust pipes to accommodate reduced maximum torque and power engine speeds, compression ratios, the Jet Ignition (JI) replacing the standard spark plug, the DI injector and the fuel.

The compression ratio has been selected on the basis of knock index computations. The compression ratio  $\text{CR}=14.2$  produces knock index results with propane injected in liquid phase close to those obtained for the validated high performance engine model with PFI of gasoline. Knock index results with methane are smaller than those with propane, because the compression ratio of the flexi fuel gas engine is limited by the RON of propane.

These values are close to those proposed for the AVL CNG DI system in the NICE project [7], where a

CR=14 is used for naturally aspiration and a CR=13 is used for charged operation.

Results for the engine running WOT stratified with late injection lean  $\lambda=1.65$ , have been obtained running the GT-POWER model.

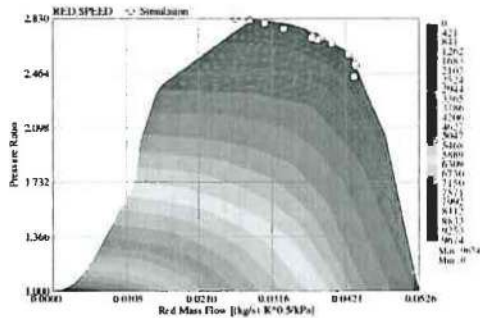


Figure 4 – Compressor reduced speed map and operating points with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

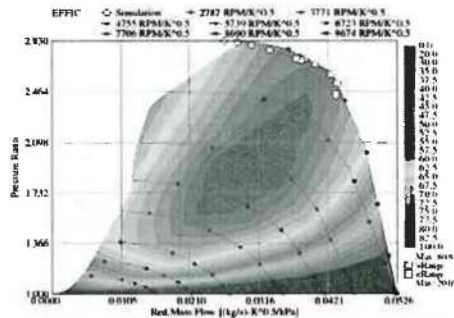


Figure 5 – Compressor efficiency map and operating points with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

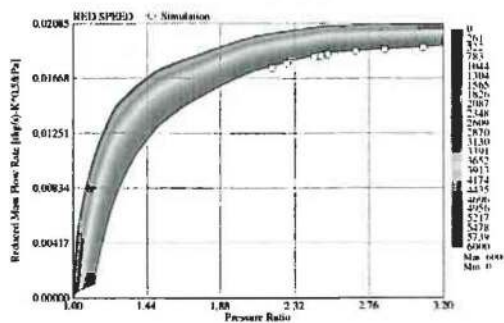


Figure 6 – Turbine reduced speed map and operating points with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

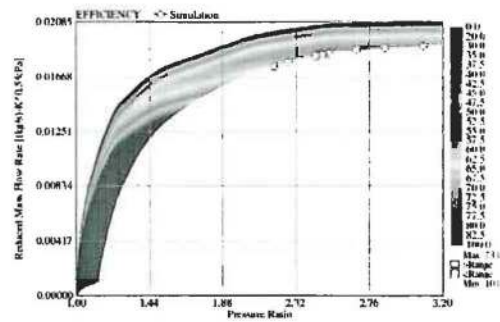


Figure 7 – Turbine efficiency map and operating points with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

Figures 4 to 7 present turbine and compressor reduced speed and efficiency maps vs. pressure ratio and reduced mass flow. The operating points are also superimposed.

Figures 8 to 12 present friction, indicated and brake mean effective pressure FMEP, IMEP and BMEP, brake efficiency  $\eta_b$  and brake specific fuel consumption BSFC results with  $\lambda=1.65$ .

BSFC are as low as 170 g/kWh and brake efficiencies  $\eta_b$  are up to 46% running lean  $\lambda=1.65$ . Maximum BMEP values reach almost 24 bar for an incredibly high low end torque while providing power densities larger than the naturally aspirated gasoline.

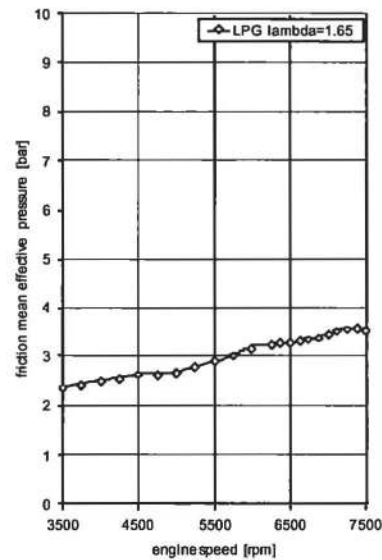


Figure 8 – FMEP results with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

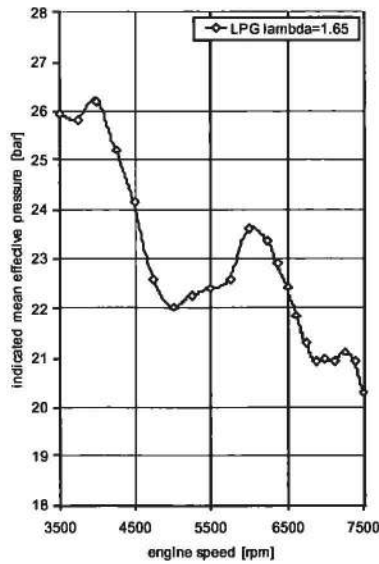


Figure 9 – IMEP results with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

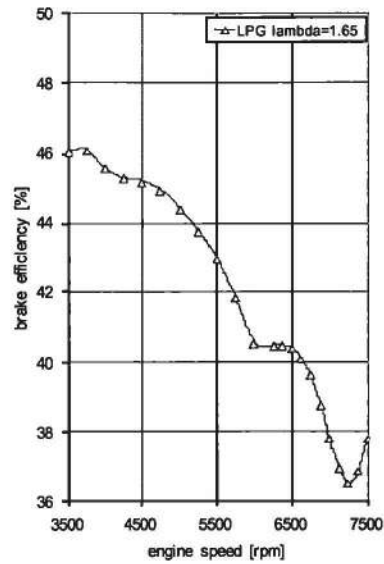


Figure 11 -  $\eta_b$  results with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

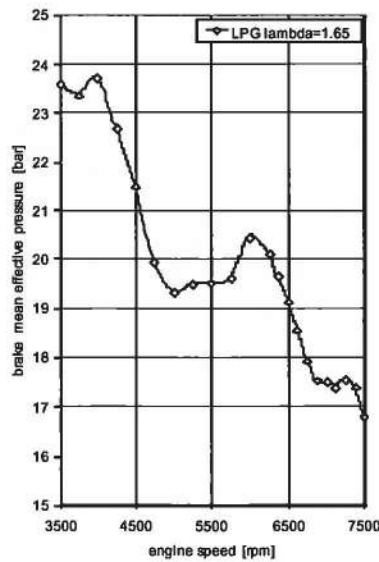


Figure 10 - BMEP results with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

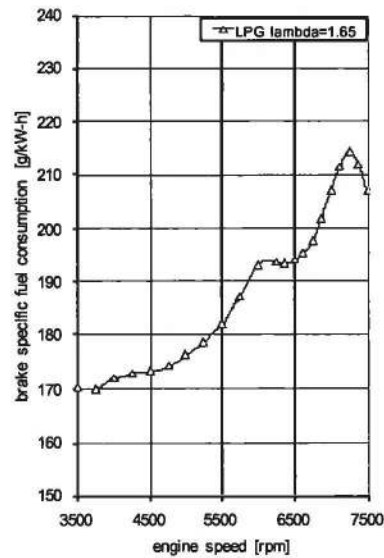


Figure 12 - BSFC results with  $\lambda=1.65$  at WOT and MBT or knock limited spark timing.

Further improvements of brake efficiencies  $\eta_b$  about 1-2% are possible running leaner  $\lambda$  about 1.7-2.2 over the speed range 3500-7500 rpm [18].

Considering the fuel a hydrocarbon  $C_nH_m$ , indolene has  $n=7.93$  and  $m=14.8$ , while propane has  $n=3$   $m=8$ .

Therefore, combustion of 1 kg of indolene fuel produces 3.17 kg of  $CO_2$ , while combustion of 1 kg of propane fuel just 3 kg. For that reason, the LPG engine has huge potentials for reducing BSFC and  $CO_2$  production over the load range with reference to gasoline.

#### 4. CONCLUSIONS

Coupling of Jet Ignition (JI) and Direct Injection (DI) allows development of an engine permitting operation, with overall fuel-to-air equivalence ratios reducing almost to zero from nearly stoichiometric for Diesel-like throttle less control of load.

Combustion always starts in the jet ignition pre chamber providing a very small amount of fuel is injected there to produce an ignitable mixture. The jets of hot reacting gases from the JI pre chamber then extend combustion to a main chamber mixture that may be globally very lean having only a minimum amount of fuel available behind the JI nozzle thus replicating Diesel-like light load operation.

Within the limits of accuracy of the proposed engine model and when reference is made to the naturally aspirated gasoline, a dedicated liquid phase LPG engine turbo charged may have better than naturally aspirated gasoline power densities, huge improvements in maximum torque, huge improvements in efficiencies and huge reductions of CO<sub>2</sub> production.

The turbocharged engine has BSFC as low as 170 g/kWh or brake efficiencies  $\eta_b$  as high as 46% while maximum BMEP is 23 bar running lean  $\lambda=1.65$ .

Maximum power densities running  $\lambda=1.65$  are more than 140 hp/liter (more than 100 kW/liter) with LPG.

Further improvements of brake efficiencies  $\eta_b$  about 1-2% are possible running leaner  $\lambda$  about 1.7-2.2 over the speed range 3500-7500 rpm [18].

This paper is a contribution to the development of a high efficiency gas fuel SI engine for the Australian market.

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