

Development of a Direct Injection High Efficiency Liquid Phase LPG Spark Ignition Engine

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ABSTRACT

Direct Injection (DI) is believed to be one of the key strategies for maximizing the thermal efficiency of Spark Ignition (SI) engines and meet the ever-tightening emissions regulations. This paper explores the use of Liquefied Petroleum Gas (LPG) liquid phase fuel in a 1.5 liter SI four cylinder gasoline engine with double over head camshafts, four valves per cylinder, and centrally located DI injector. The DI injector is a high pressure, fast actuating injector enabling precise multiple injections of the finely atomized fuel sprays. With DI technology, the injection timing can be set to avoid fuel bypassing the engine during valve overlap into the exhaust system prior to combustion. The fuel vaporization associated with DI reduces combustion chamber and charge temperatures, thereby reducing the tendency for knocking. Fuel atomization quality supports an efficient combustion process. Furthermore, the spray-guided combustion process has significant thermodynamic benefits over wall or air guided combustion resulting in near optimal combustion. Injection timings and compression ratio are optimized for best performances over Wide Open Throttle (WOT) conditions running homogeneous at stoichiometry or spray-guided stratified lean of stoichiometry. The paper also explores advantages of Jet Ignition (JI) versus standard SI in the lean spray-guided stratified mode of operation. Effects of fuel properties on gas exchange, mixture formation, in-cylinder charge cooling, wall-spray interactions, turbulent combustion evolution and knock occurrence are taken into account. Results are presented in terms of brake mean effective pressure, specific fuel consumption, efficiency and specific CO₂ production showing significant improvements of LPG

over gasoline for the reduced green house gas formation potential. This paper is a contribution to the development of a high efficiency gaseous fuel SI engine for the Australian market.

INTRODUCTION

LPG has been and continues to be a widely used alternative fuel to gasoline and diesel on the worldwide basis. The popularity of LPG as an alternative fuel over the years is evidenced by its place in the clean air programs. LPG is a colorless odorless gas, typically comprised of mainly propane, butane, or a combination of these two constituents. The composition of LPG as an automotive fuel varies in a broad range from one country to another, depending on the cost and availability of the fuel in relation to alternative fuels, especially conventional fuels, gasoline and diesel. LPG fuel for vehicles is actually a mixture of various hydrocarbons which are gases at atmospheric pressure and temperature but which liquefy at higher pressures.

LPG is a natural derivative of both natural gas and crude oil. The composition of commercial LPG includes propane, propylene, butane and heavier HC, moisture content, residual matter, pentane and heavier HCs, and some sulfur, either fuel sourced or introduced as a mercaptan deodorant. The remainder is normally n-butane, with isobutene and other higher isomers also present. The limitation on propylene and other unsaturated hydrocarbons (olefins) is in their low octane numbers which means low knock resistance. Table 1 presents octane numbers for LPG components and gasoline, while Table 2 summarizes some of the basic

other properties of gasoline and LPG. A concern with propylene is its photochemical reactivity, which is higher than that of propane. This could be an important factor in formation of smog. Propylene does not occur in LPG obtained from natural gas processing plants but may be found in the LPG resulting from petroleum refinery operations.

The high octane rating of propane suggests that optimized propane engines would have higher thermal efficiencies than those of equivalent gasoline engines. This is because the higher octane rating of propane compared to gasoline allows higher compression ratio for the engine and higher compression ratio improves thermal efficiency and allows more power to be produced by the engine. Conversely, the lower density of propane would lower the volumetric efficiency compared with gasoline powered engine. However, propane (when injected in liquid phase) has higher heat of vaporization, and this may help decrease the temperature of mixture thus increasing the density with positive effects on the volumetric efficiency. Moreover, liquid phase injection increases the resistance to pre-ignition and knock. The lower air-to-fuel ratio for gasoline means that the chemical energy released per kg of stoichiometric mixture burnt during combustion is slightly greater than for propane despite the lower heating value of gasoline.

Table 1 - Octane numbers of LPG components and gasoline.

Component	Formula	Research Oct. Num.	Motor Oct. Num.
Propane	C_3H_8	111.5	100
n-butane	C_4H_{10}	95	92
Isobutene	C_4H_{10}	100.4	99
Propylene	C_3H_6	100.2	85
n-butane-1	C_4H_{10}	100	80
n-butane-2	C_4H_{10}	101	83
Gasoline	C_8H_{18}	92-95	83-86

Table 2 - Properties of LPG, gasoline and Diesel fuels.

Property	LPG (HD5)	Gasoline	Diese I
Res. Octane No.	106-111	92-95	Low
LHV (MJ/kg)	46.33	44.2	43.25
LHV (MJ/l)	23.63	31.82	35.9
Relative CO ₂ per kJ	0.885	1	1.028
Rel. Density	0.51	0.74	0.83
Evap. Enthalpy (kJ/kmol stoic. mix)	820	662	330
Stoic. A/F Ratio (vol. basis)	24.1	60.1	100
Stoic. A/F Ratio (mass basis)	15.7:1	14.7:1	14.5:1
Flammability limits (Φ)	0.55-2.35	0.85-3.55	-

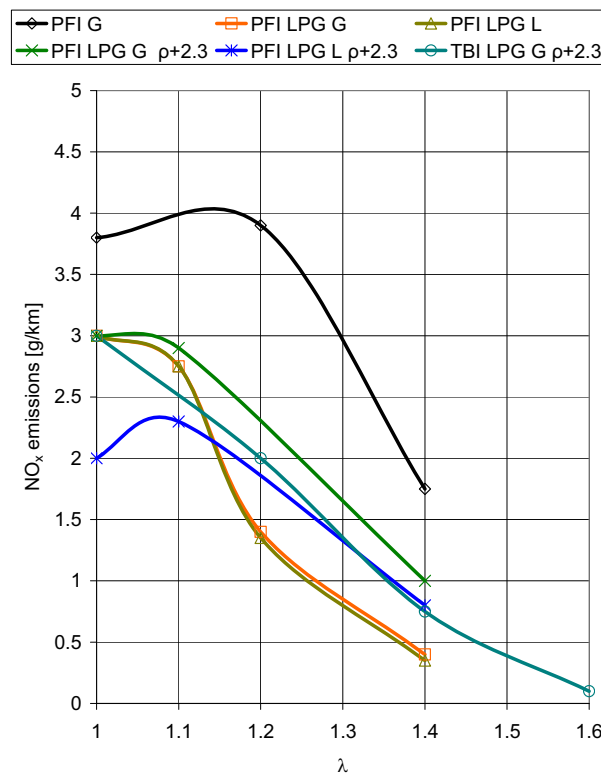
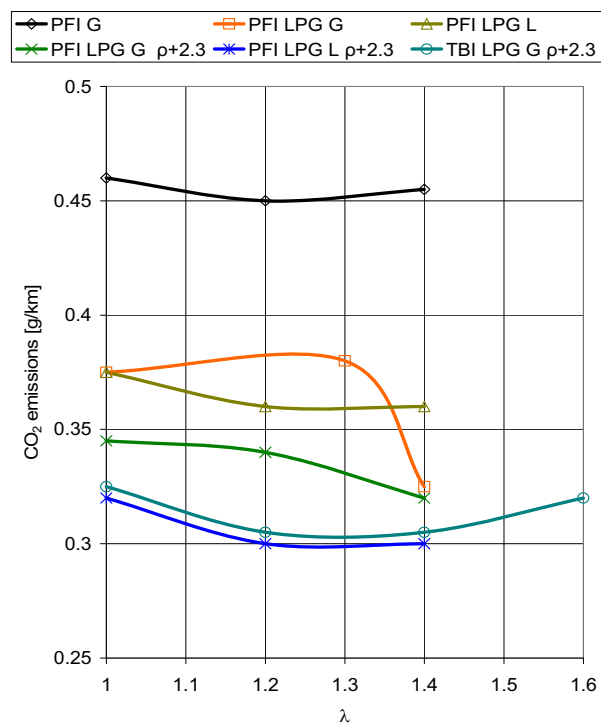


Figure 1 - CO₂ and NO_x results on the NEDC drive cycle for port (PI) and throttle body (TBI) injection (six mapping point measurements weighted).

In addition, the inherent properties of LPG already mentioned, in times of greenhouse gas concern, as a viable alternative to gasoline and diesel its H/C ratio of 2.63:1 compared with H/C ratios of around 1.8 to 1.85:1, respectively, offer potentially a much lower CO₂ specific emissions. Also, being a fuel with simple molecular structure, primarily being a mixture of propane and butanes, the fuel has a low tendency to form aromatics or particulates on combustion.

A deterrent to the adoption of LPG, even though it is always sold at a price very competitive with gasoline (usually as the result of reduced government taxes applied) is the high cost of conversion or manufacture of the engine and related LPG components because of the high component costs due to the small volume of sales compared to gasoline or diesel vehicles. To mitigate these costs the LPG system should be as low cost as possible, yet more efficient (to further reduce operating costs) and with lower greenhouse gas and particulate emissions.

In Australia, LPG is the dominant alternative fuel with some 550,000 vehicles on the road according to 2002 figures. This accounts for an LPG consumption of approximately 8% of all fuel used by cars in the country. In addition to about 50% excess production capacity of LPG, nearly 40% of refueling outlets supply LPG such that motorists can travel the whole continent with confidence entirely on LPG. Indeed sufficient that Ford Australia markets a range of dedicated fuel vehicles (about 10% of gasoline vehicle sales).

Figure 1 presents a comparison of existing LPG technologies for operation during the NEDC drive cycle (six mapping point measurements weighted). CO₂ and NO_x results are shown across a range of LPG gas fuel supply technologies, PFI LPG liquid injection (PFI LPG L), PFI LPG gas injection (PFI LPG G), throttle body LPG gas injection (TBI LPG G), compared at increased compression ratio ρ with the reference PFI gasoline (PFI G). Extending the opportunity presented by externally injected LPG is direct injection.

The advantages of DI injection have been explored in previous projects. These experimental studies of DI LPG amongst several other liquid fuels have been carried out in an optical accessible high compression engine with a 4-valve cylinder head [6]. Both conventional SI and Hydrogen Assisted Jet Ignition (HAJI) were explored and compared. Figure 2 shows thermal efficiency at the WWMP (World Wide Mapping Point, 250 kPa BMEP and 1500 rpm engine speed). Best efficiency is about $\lambda=2.15$ and spark timing 345 °CA, lowest NO_x is about $\lambda=1.7$ and spark timing 365 °CA and the 10% CoV of IMEP boundary is shown. These results are for the HAJI system. At this load and speed the best standard SI performance was 26% brake efficiency with $\lambda=1.5$.

The images of the liquid fuel spray assist in calibrating the injector model spray penetration, while the enflamed

combustion images seen in figure 3 give an indication of the extent of vapor penetration. It follows that it would be useful to determine whether the inherent advantages of LPG fuel with higher octane rating and wider flammability limits than gasoline still offer advantages in a multi-cylinder engine as expected from the research engine studies.

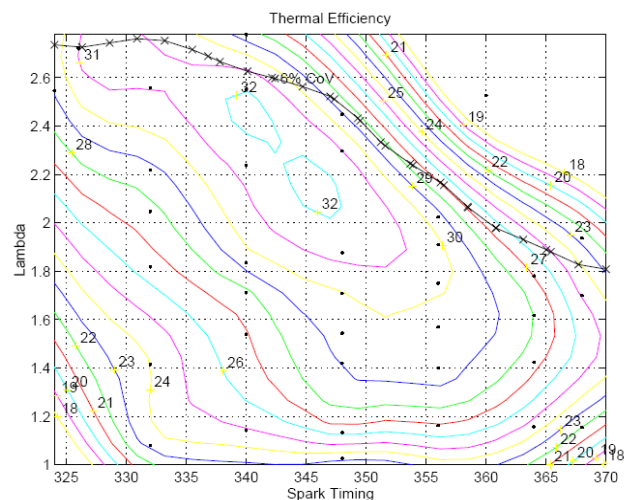


Figure 2 - Thermal efficiency contours for an optical access engine with DI butane and HAJI at 250 kPa BMEP and 1500 r/min with differential inlet valve opening assisted swirl (base engine efficiency is 26% at $\lambda=1.5$) [6].

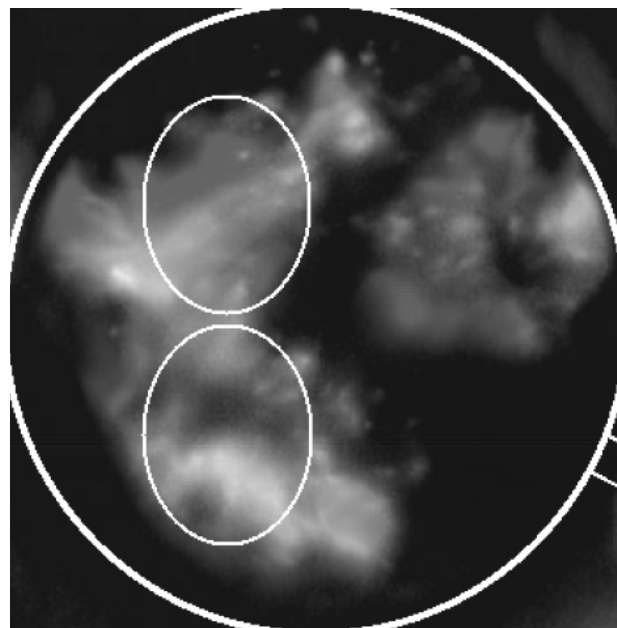


Figure 3 - Overlaid images of 10 sequential cycles of enflamed regions of lean burn DI butane 21 °CA after ignition (9 °CA ATDC) at $\lambda=1.7$ and 1500 rpm [6].

LPG is an important and widely available fuel in Australia because of reduced running costs. With increasing government pressure to reduce CO₂

emissions and reduce dependence on imported oil there is motivation to further improve engine efficiency. Gasoline technology has benefited from the adoption of DI and aim of the paper is to establish if DI of liquid phase LPG may produce similar gains. The paper evaluates potentials relative to gasoline of liquid phase LPG for engine operation homogeneous stoichiometric and spray-guided stratified lean with DI [20-28]. The paper also explores additional advantages of JI [6 and 8-15] with reference to standard SI.

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

ENGINE MODEL

Modeling is done by using GT-POWER [1]. GT-POWER is the industry-standard engine simulation tool, used by most leading engine and vehicle makers and their suppliers. In view of the fact that the computational approach used in GT-POWER is very well known, only parts of specific interest here are discussed in this section. Fuel is considered to be indolene (gasoline) or propane (LPG). Properties of indolene and propane are presented in Table 3.

Table 3 - Properties of propane and indolene fuels.

Property	Propane	Indolene
Stoichiometric fuel-to-air	0.064	0.069
Stoichiometric air-to-fuel	15.6	14.5
H_{vap} [J/kg]	370000	350000
density [kg/m ³]	545	750
C	3	7.93
H	8	14.8
O	0	0
N	0	0
LHV [J/Kg]	4.64E+0	4.40E+0
T_{crit} [K]	369.8	568.8
p_{crit} [bar]	42.5	24.9
LHV* [J/Kg _{mixture}]	2.97E+0	3.03E+0
	6	6

The DI injector is modeled as a sequential injector with prescribed fuel-to-air ratio, start of injection (SOI), injector delivery rate, injected fluid temperature and vaporized fluid fraction. The user imposes the fuel-to-air ratio, and the resultant injection pulse width (TI) is calculated for each injection event. The vaporized fluid fraction is an important parameter of the DI model with liquid fuels. Assuming the same injection pressures and timings, the evaporation in the air is most likely limited by the energy available in the air with which it mixes and may limit the evaporation, thus propane will vaporize pretty much the same than indolene. Critical pressure and temperature would also have an influence. An increase in injection pressure would generally result in smaller droplets and more evaporation for the same

fuel. This is assuming that the fuel evaporates in the air, and does not impinge on the piston. This assumption is mostly true with DI indolene and propane. However, a certain fraction may evaporate due to contact with the piston, but for sake of simplicity, the fuel impingement on the piston is neglected.

Homogeneous stoichiometric and spray-guided lean stratified modes of operations are considered. Homogeneous operation is supposed to be possible with both early and late injection, because injector opening times are small even at higher engine speeds. Conversely, spray-guided stratified operation is only considered with late injection. For homogeneous stoichiometric operation, early injection refers to Start of Injection (SOI) about exhaust valve closure (EVC), while late injection refers to SOI about intake valve closure (IVC). Conversely, for spray-guided lean stratified operation, late injection refers to SOI following IVC and producing an End of Injection (EOI) approaching the ignition timing.

Ignition in the main chamber may occur by standard spark plug discharge or by a jet of reacting gases. The engine operating stoichiometric homogeneous has a standard SI. Conversely, the engine operating in jet controlled lean stratified mode has the standard spark plug ignition replaced by JI. With jet ignition, additional propane gas fuel is injected in a pre chamber which is connected to the main chamber via calibrated orifices, creating fuel rich conditions in the pre chamber. Following ignition by a conventional spark plug in the pre chamber, the main chamber globally lean mixture is then ignited by the jets of hot products and radicals from the pre chamber, enabling a faster combustion of the globally lean main chamber mixture. Flame speed enhancement of up to six times has been measured [13-15] when JI has been applied to ignite homogeneous lean mean chamber mixtures of several different liquid and gaseous fuels. The use of JI to ignite a nearly stoichiometric mixture close to the JI nozzle surrounded by air is a novelty considered here.

In-cylinder flow characteristics are used to calculate in-cylinder heat transfer. The engine cylinder flow model calculates a mean radial velocity, axial velocity, swirl, tumble, turbulence intensity, and turbulence length [1]. An effective velocity is calculated and used in the calculation of a heat transfer coefficient using the Colburn analogy [1].

The in cylinder volume is represented with a two zone model [1]. At the start of combustion the cylinder is divided into two zones, an unburned zone and a burned zone. All of the contents of the cylinder at that time start in the unburned zone, including residual gases from the previous cycle. At each time step, a mixture of fuel and air is transferred from the unburned zone to the burned zone. The amount of fuel-air mixture that is transferred to the burned zone is defined by the burn rate. This burn rate is prescribed or calculated by the combustion