

Mixture Preparation Effects on Gaseous Fuel Combustion in SI Engines

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ABSTRACT

This paper presents a comparison of the influence of different mixture preparation strategies on gaseous fuel combustion in SI engines. Three mixture preparation strategies are presented for a dedicated LPG fuelled engine, showing varying results – gaseous phase port injection (PFI-G), liquid-phase port injection (PFI-L) and gaseous-phase throttle-body injection (TBI-G). Previous work by the authors has shown considerable differences in emissions and thermal efficiency between different fuelling strategies. This paper extends this work to the area of combustion characteristics and lean limit operation and closer analyses the differences between these systems.

A dedicated LPG in-line six cylinder engine with compression ratio increased to 11.7:1 (up from the standard 9.65:1) was tested over a range of speed/torque conditions representing most of the steady-state parts of the Euro drive-cycle for light duty-vehicles. The air-fuel ratio was varied from lambda 1.0 to the lean limit. Cylinder-to-cylinder measurements of air/fuel ratio and emissions were conducted. Additionally, sweeps of lambda at five important engine operating points were undertaken for each configuration and results of COV of IMEP, thermal efficiency, HC, NO_x and spark timing compared. Comparisons were also made at two world wide mapping points. Furthermore, in-cylinder pressure measurements were processed to compare combustion characteristics – 0-2.5% and 0-90% mass fraction burn durations – at important operating points.

It is shown that location of injectors as well as fuel state (liquid or gaseous) can have a considerable effect on combustion characteristics and engine stability. PFI-G is shown to have very slow burns, attributed to poor mixing as evidenced by emissions variations at a given lambda.

However, TBI-G using the same injectors as PFI-G is shown to approach PFI-L efficiency due to superior mixing, and to extend the lean limit to lambda 1.6 at which NO_x emissions are very low. The three injection systems are shown to deliver significantly different cylinder-to-cylinder air/fuel ratios and mixture homogeneity which along with temperature effects contribute to significantly different overall performance.

INTRODUCTION

In the current climate of increased fuel prices and global economic uncertainty, focus has intensified on the need to remove current dependency on oil reserves and look to alternative propulsion means to more economically and securely satisfy our mobility needs. As a consequence of recent record fuel prices, consumer preference has shifted towards smaller and more fuel efficient vehicles. Sales of larger vehicles, even in traditional large vehicle and truck markets have subdued considerably. Although always important, consumer demand for better fuel economy has increased in intensity.

In addition, global concern over the contribution of environmental pollution generated from human activity has intensified. Ever stricter emission targets and political solutions such as carbon taxes and emission trading schemes are undergoing widespread discussion. The transportation sector being the second largest emitter of greenhouse gas emissions (after power generation) is undergoing renewed scrutiny and social and legislative pressures. New emission targets are being legislated and must be met.

Consequently, the engineering community is faced with the demanding task of producing more efficient powertrains and vehicles, which will both satisfy

consumer demand for high fuel economy and comply with environmental mandates for low emitting vehicles. One such avenue to meet these demands which is seeing renewed interest is alternative fuels.

Currently, the dominant alternative fuels remain liquefied petroleum gas (LPG) and compressed natural gas (CNG). However, interest is increasing in transferable vegetable oils, alcohols (bioethanol) and hydrogen. Bioethanol as a food competitor has received very negative publicity in recent times, which may make it less attractive longer term.

For any alternative fuel to successfully compete with either diesel or gasoline, it must satisfy the following requirements, as mentioned by Zvirin [1]:

- Enable fast refuelling
- Fluently pass from the tank to the engine cylinders
- Effectively mix with the air
- Efficiently burn in the cylinders to produce adequate power and minimal amounts of pollutants in a wide range of ambient conditions.

Gaseous alternative fuels, such as LPG and CNG satisfy the above criteria. It is expected that gaseous fuels such as LPG, CNG and eventually hydrogen will play an increasingly greater role in road transportation. LPG and CNG are available abundantly in many parts of the world and are generally cheaper in cost compared to gasoline or diesel due to reduced government taxation. Moreover, they offer many environmental benefits which are well documented. In this study, experimental results for LPG will be presented, LPG vehicles being the most widely produced alternative fuel vehicles globally [2].

Previous work by the authors has shown considerable benefits and significant potential with LPG. At the Department of Mechanical & Manufacturing Engineering at The University of Melbourne, a comprehensive back-to-back study of various LPG systems was conducted by the last author [3-4]. The authors know of no other such comprehensive comparative study into LPG technology. The results have been published in earlier papers [4] and in addition to subsequent work by Liew [5] and Khan [6], has led to the subject of three theses [3, 5-6].

In Baker's study [3-4], various LPG systems were compared at different compression ratios against the baseline gasoline engine in OEM form. The LPG systems investigated covered second to fourth generation technologies including a gas mixer (GM) system, a throttle-body injection (TBI) system and port fuel injection (PFI) liquid and gaseous phase systems. Baseline results for the standard gasoline engine in OEM form were recorded for comparison. It was shown that selection of the fuelling system had a significant influence on emissions and performance of the LPG engine. With liquid phase port fuel injection and an increased compression ratio of 11.7:1, up from the

standard 9.65:1, a 26% improvement in thermal efficiency was recorded at $\lambda = 1.4$ over the baseline gasoline engine at stoichiometric conditions. Engine-out emissions also improved dramatically with best observed reductions in specific emissions of 88% for HC, 45% for NO_x , 40% for CO_2 and 92% for CO. In addition to the superior performance of the liquid phase system, differences between the results from the different gaseous-phase injection systems were also observed (port injection versus throttle body injection). Full results are available in earlier papers [4, 7-8].

These differences in results between the various gaseous and liquid phase LPG systems were attributed by the authors to differences in the mixing characteristics between the systems. For fluids of the same phase, the mixing process is primarily reliant on turbulence within the media to affect mixing. As Abraham's [9-10] model shows, turbulence at the periphery of the jet caused by the transfer of momentum of the gas jet to the adjacent air creates the shear forces necessary to generate vortices and promote mixing, but only at the interface of the gas jet with the surrounding air. This mixing process may be enhanced by additional bulk turbulence within the air stream. The mixing mechanism of fuel sprays is quite different. The very fine liquid droplets within the liquid spray have a much greater density than the surrounding air creating local mixing in their wake. Vaporisation of the droplets occurs as they travel through the air medium, leaving a vapour trail behind them. This mechanism greatly assists the bulk distribution of vapour throughout the air. The greater interface surface area created tends to increase diffusion rates and smaller droplet sizes down to a SMD of $20\mu\text{m}$ allow faster mixing [3-4, 9-10].

It has also been shown that variations in the air-fuel ratio between cylinders, as well as mixture homogeneity (i.e. homogeneity of the small fractions of the intake charge) affect engine emissions and performance. As Baker demonstrated, even with port fuel injection, despite having an injector located in each inlet port, significant variations in air fuel ratio between cylinders can exist. A typical example is shown in Figure 1, from Baker's work for the engine under discussion [3]. As can be seen, in all cases in Figure 1, the outer cylinders (cylinder 1 and cylinder 6) have leaner mixtures than the other 4 cylinders, despite the location of a single injector in each port. This variation in air-fuel ratio between cylinders has adverse consequences on engine performance, emissions, and combustion stability, more critical under lean burn operation. Consequently, due to this cylinder maldistribution, all of the LPG fuelling systems tested by Baker could only be operated to as lean as $\lambda = 1.4$, before combustion instability became problematic.

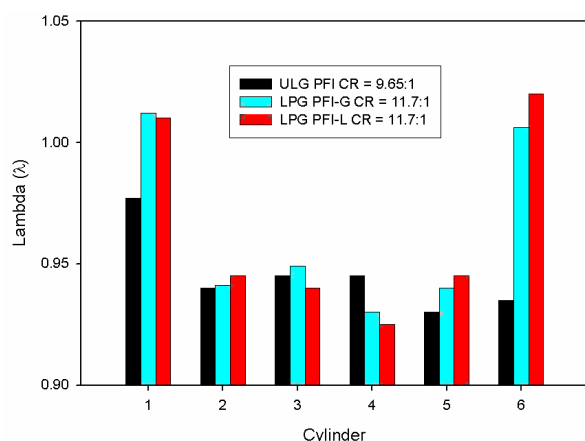


Figure 1. Cylinder-to-cylinder variations in lambda (λ) for three fuelling systems at 1500 rev/min, 50 Nm, MBT spark timing (ULG –unleaded gasoline, PFI-G - port fuel injection (gaseous phase), PFI-L – port fuel injection (liquid phase), CR – compression ratio).

To reduce variations in air fuel ratio between cylinders and improve mixture preparation, a throttle-body injection (TBI) system was proposed, which would provide adequate residence time and mixing length to allow the charge to be well mixed before it reached the combustion chamber, providing more even mixtures to each cylinder.

The TBI system consisted of two nozzles positioned at specific angles within the throttle-body with each nozzle fed by an individual injector. The design was configured from CFD studies by Liew [5] to minimise cylinder-to-cylinder air/fuel variations. The injector angles and location of the two injector outlet nozzles was studied in detail. Liew used Fluent® simulation, validated with a manifold test rig with injected propane gas to optimize the gas jet orientation and velocity at critical engine operating points. Further operating parameters were optimized experimentally. For details, the reader is referred to these other works [5-8].

Due to improved mixture preparation of the TBI system, significant improvements in combustion stability were achieved, such that the lean limit of the engine was extended from $\lambda = 1.4$ to $\lambda = 1.6$, where NO_x levels could satisfy Euro 5 emission standards without the need for a reducing catalyst [6-8]. These benefits were achievable due to careful optimization of the location of injector nozzles and angles as well as optimization of injection parameters.

Compared with the original gasoline sequential port injection system, best emissions observed with the TBI system were reductions of 88% for HC, 98% for NO_x , 92% for CO, 36% for CO_2 and an increase of 23% in thermal efficiency, making the system comparable to the liquid-phase port-injection system tested by Baker [3]. These results were achievable due to the superior mixing characteristics of the TBI system as evidenced from direct measurements of emissions and air/fuel ratio from

individual cylinders as well as analysis of combustion data [6].

Based on the work conducted into LPG engine research by the authors, it is proposed that mixing characteristics have a significant impact on the performance and emissions of LPG engines. The objective of this paper is to add to the body of evidence that the authors have gathered on this issue and compare three fuelling systems at a single compression ratio of 11.7:1 on a single engine. The systems compared are port fuel injection (PFI) systems in both gaseous and liquid phase (PFI-G and PFI-L, respectively), as well as a throttle body injection (TBI) system, injecting in gaseous phase (TBI-G). It is proposed that the mixture preparation of the various systems significantly affects the performance and emissions of the engine. Earlier studies investigated overall map or weighted drive cycle emissions and thermal efficiencies [5-6]. In this study however, for closer analysis, specific important steady-state operating points are investigated representing important parts of the engine's drive cycle and comparisons are made, providing further evidence on the effects of mixture preparation on engine emissions and performance.

TEST ENGINE AND PROCEDURE

The test engine in this study was an Australian model 4-litre Ford Falcon AU II version, 6-cylinder in-line engine. The engine is a suitable choice as it is quite representative of the type used in a large number of vehicles on Australian roads. Furthermore, it is envisaged that the size of the engine could see its possible application in heavy-duty vehicles. The engine, however, was modified from its standard configuration. To take advantage of the higher octane rating of LPG compared to gasoline (114 to 95 RON respectively), the compression ratio was increased from the standard 9.65:1 up to 11.7:1. This was achieved by the use of modified pistons which had been optimised in prior research on a CNG version of the engine [11]. The engine specifications are shown in Table 1.

Table 1: Test engine specifications

Bore x stroke	92.26 x 99.31 mm
Number of cylinders	6
Engine configuration	Vertical inline
Connecting rod length	153.85 mm
Compression ratio	11.7:1
Spark timing	MBT
Air/fuel ratio change (λ)	1.0 to lean limit
Coolant temperature	85 °C
Valve train	Single OHC, 2 valves per cylinder
Firing order	1, 5, 3, 6, 2, 4

The engine test-cell set-up is shown in Figure 2. Brake thermal efficiency and specific emissions were calculated from direct measurements for a wide-ranging torque-speed map at steady state conditions. Parameters systematically varied for each system include air/fuel

ratio, spark timing, injection timing, and compression ratio. The engine was not equipped with EGR. Details of the engine and experimental set-up can be found in [3, 4, 6-7].

The strategy employed was to standardise all settings, except the variable being tested; i.e. all tests were carried out at minimum spark advance for maximum brake torque (MBT), the broad band manifold setting was set to short runner and the VCT disabled. Only one parameter was varied at any one time while all other parameters were fixed. At each setting, the engine was tested for mapping points ranging from 1000 to 4000 rev/min in increments of 500 rev/min, at torques from no load to wide open throttle (WOT), in increments of 50 Nm.

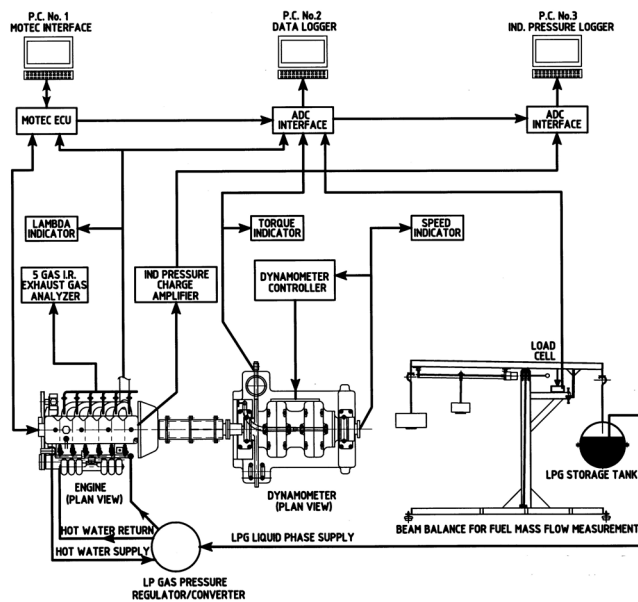


Figure 2. Engine test cell set-up.

The injection timing was optimised over the speed and load range, to minimize HC emissions and coefficient of variation of indicated mean effective pressure (COV of IMEP), while maximizing CO₂ (as an indicator of combustion efficiency). For assessing cylinder-to-cylinder air/fuel distribution, the exhaust gas composition was measured from each individual cylinder using a NDIR five-gas exhaust gas analyser. Additionally, a pressure transducer was installed in cylinder 6 for cylinder pressure analysis and post-processing for combustion analysis. Further details can be found in other works [3, 6].

RESULTS

Previous published work by the authors has considered overall engine map as well as NEDC drive-cycle weighted emissions and thermal efficiencies [4, 6-8]. The results were useful as they presented results representative of real-world driving conditions. In this study, however, in order to more closely examine the

underlying causes for the differences between the various fuelling systems, the results have been isolated to five important steady-state operating points. Results for COV of IMEP, HC, NO_x, MBT spark timing and thermal efficiency are presented, across a range of lambda values.

Six steady-state torque-speed operating points which constitute more than 72% of the engine's operating time in the NEDC drive-cycle were used in an earlier work to validate a NO_x model for the current engine under TBI mode [12]. The torque-speed points are shown in Table 2. These being important points of the engine's operation, they represent useful points at which to investigate the influence of varying lambda on COV of IMEP and the important emissions, namely NO_x and HC. Combustion analysis via processing of cylinder pressure data was also performed and results of 0-2.5% and 0-90% mass fraction burn (MFB) durations are presented. The results now presented are for five of the operating points with the idle point excluded, since the engine would normally operate near stoichiometric for satisfactory operation at idle. Following this, the three systems are compared at two world-wide mapping points (WWMP).

Table 2: Steady-state torque-speed points for NEDC cycle and For Falcon AU engine [12].

Point	Speed(rev/min) – Torque(Nm)	Frequency (%)	Thermal Efficiency (%)	NO _x (ppm)
1	500 – 20	18.0	13.3	85
2	1250 – 80	16.9	27.2	1275
3	1750 – 80	14.8	28.1	1500
4	1500 – 140	5.4	30.2	1970
5	2250 – 140	10.0	27.8	1860
6	2250 - 200	7.5	31.7	2100

COV of IMEP

Figure 3 shows the results for COV of IMEP (%) for the 5 drive cycle points across varying lambda. The solid lines represent PFI-G, the dashed lines represent PFI-L and the dotted lines TBI-G. The two colours - black and grey - are for different operating points, as labelled. It is important to note that the results are for cylinder 6 only, as the pressure transducer was located in only one cylinder of the 6-cylinder engine. Hence assumptions have to be made about the remaining cylinders from these results.

All graphs in Figure 3 show a consistent trend. COV of IMEP values are close for both PFI-G and PFI-L; however results are slightly lower for PFI-L, particularly closer to stoichiometric mixtures. As mixtures become leaner, the results between PFI-G and PFI-L appear to converge (generally at $\lambda = 1.4$). Values for TBI-G are significantly and consistently lower than both PFI-G and PFI-L systems, and the difference between these two port injection systems and the TBI system increases as lambda is increased.

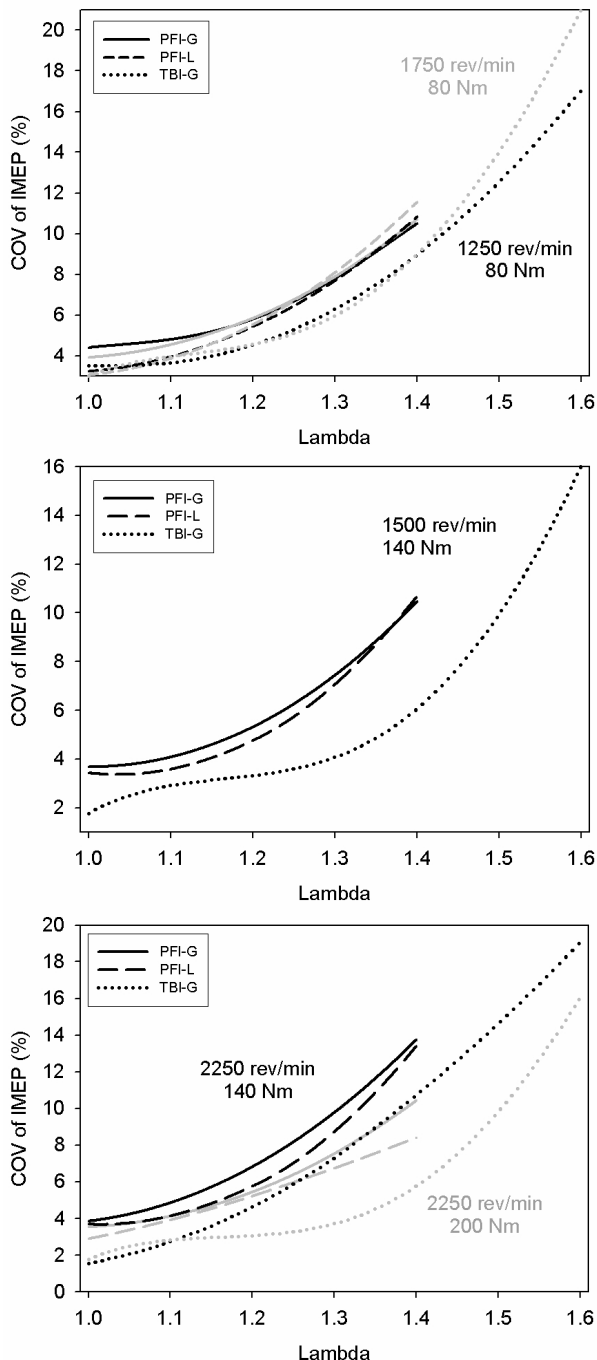


Figure 3 (a), (b) and (c). COV of IMEP (%) for 5 cycle points, comparing PFI-G, PFI-L and TBI-G.

Considering specific operating points, in Figure 3 (a), the increase in engine speed from 1250 rev/min to 1750 rev/min, has little effect on the combustion stability. Other load-speed points show similar values, in general; however, increasing load decreases COV of IMEP as one would expect (Figure 3 (c)). In any case, it is clear that air fuel ratio is the dominant factor affecting COV of IMEP.

Considering these results and comparing between the two port injection systems, it is proposed that the slightly improved results of liquid phase is due to poorer mixing of the PFI-G system. It is suggested that the short mixing length and short residence time of PFI-G do not allow the gaseous fuel to adequately mix with the air in the time available, leading to poor mixture homogeneity and ultimately, poorer combustion. In this respect, the study of Goulburn and Hughes [13] is particularly insightful. Via tests conducted with propane (fed by a vapouriser) and various devices to enhance mixing, it was demonstrated that mixing length plays a critical role in mixture homogeneity. From cylinder-to-cylinder mixture distribution measurements Goulburn and Hughes showed that thorough mixing does not take place between the vapouriser and the inlet manifold, however, this could be resolved by increasing the length of the inlet tract between the manifold and the vapouriser. Additionally, the length of the pipe required for complete mixing was shown to be heavily dependent on the initial segregation between air and fuel, although this was difficult to quantify. Goulburn and Hughes concluded that it was necessary to break the stream down into a series of smaller streams to speed up mixing and break down stratification patterns. The presence of multiple jets, it was concluded, tended to overcome two undesirable characteristics of all of the natural mixing means, namely: a relatively unmixed core inside a well mixed outer envelope (as is generally the case with gas mixers) and insufficient turbulence. This may account for the excellent distribution characteristics of the TBI-G system as reported in the authors' earlier works [6-8], which consisted of two individual jets intermittently injecting into the throttle body. Consequently, this may explain the improved combustion stability of TBI-G. Port injection of a gaseous spray in the test engine simply cannot provide similar mixing and distribution characteristics to TBI-G (ultimately leading to poorer combustion as shown in these results).

Closer consideration of Figure 3 will validate this. Both PFI-G and TBI-G use essentially the same injectors (the only difference is TBI-G has larger injectors as 2 injectors replace 6), yet the results are quite different. TBI-G, which has a long mixing length and by injecting upstream of the throttle plate, also uses the throttle plate to promote mixing, delivers a well mixed and more consistent charge to each cylinder. Cylinder-to-cylinder measurements across varying engine operating points by the authors for both systems has validated this [3-4, 6-8]. On the other hand, PFI-G does not have sufficient time to mix adequately in the short time interval between injection and spark discharge, resulting in local rich and lean regions within the combustion chamber at the time of combustion. Consequently, flame kernel establishment and development is highly dependent on the mixture concentrations in the vicinity of the spark plug, and as the flame propagates, the homogeneity of the charge throughout the rest of the combustion chamber.