

4 L Light Duty LPG Engine Evaluated for Heavy Duty Application

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ABSTRACT

Many applications of liquefied petroleum gas (LPG) to commercial vehicles have used their corresponding diesel engine counterparts for their basic architecture. Here a review is made of the application to commercial vehicle operation of a robust 4 L, light duty, 6-cylinder in-line engine produced by Ford Australia on a unique long term production line. Since 2000 it has had a dedicated LPG pick-up truck and cab-chassis variant.

A sequence of research programs has focused on optimizing this engine for low carbon dioxide (CO_2) emissions. Best results (from steady state engine maps) suggest reductions in CO_2 emissions of over 30% are possible in New European Drive Cycle (NEDC) light duty tests compared with the base gasoline engine counterpart. This has been achieved through increasing compression ratio to 12, running lean burn (to $\lambda=1.6$) and careful study (through CFD and bench tests) of the injected LPG-air mixing system.

The outcome of this work (and others) has stimulated a major Australian Government investment in gaseous fuels consistent with increased home consumption of LPG as over half of local LPG production is exported.

Thus the engine is evaluated in its application to trucks of 15 - 20 tonne GVM, needing 170 kW and high torque over the Euro truck steady state and transient test cycles; the European Stationary Cycle (ESC) and the European Transient Cycle (ETC) respectively. It is shown that this engine has the potential to deliver better than diesel CO_2 emissions with relatively low cost LPG components and likely meeting Euro 5 emission standards with only oxidation catalyst after treatment in contrast to the complex after treatment needed for diesel engine compliance with the diesel standard (e.g.

DPF and SCR using urea for particulates and oxides of nitrogen (NO_x) control).

The paper also comments on the relative merits of liquid-phase, gaseous phase and throttle body LPG injection.

Efficiency and emissions performance comparison are made with the same engine at a higher compression ratio optimised for compressed natural gas (CNG) and a 6 cylinder Euro 3 diesel engine of similar power. The data indicates that LPG can perform at a thermal efficiency equal to that of the diesel engine, but slightly worse at low NO_x levels. Although CNG has highest efficiency over both ESC and ETC, a significant NO_x control strategy is needed.

INTRODUCTION

Since the Second World War, diesel engines have been the power units used almost exclusively in heavy duty vehicles. Continued development of combustion systems have allowed efficiency increases of the order of 40% over the last forty years. Coupled with improvements in exhaust emissions, the diesel engine remains an attractive power unit except that its emissions of NO_x and particulates are higher than desirable and are the most difficult to control pollutants in diesel exhausts.

Table 1 is a brief classification of commonly used and potential engines for heavy vehicles. As can be seen, there are emerging opportunities for gaseous fueled heavy vehicles using LPG and CNG. Typically these gaseous fueled heavy vehicle engines are diesel engines converted to spark ignition. They currently use moderate compression ratios of around 10 to 12:1 compared with the diesel engine whose compression ratio is likely around 18:1. As compression ratio is an important dictate of efficiency, there is a tendency for gaseous fueled engines to be of lower energy efficiency than

diesel engines, although advanced technology such as high compression lean-burn can mitigate all of this disadvantage. LPG is a more readily stored fuel since it is liquid at pressures of about 5 atmospheres but CNG needs to be stored in high pressure vessels capable of withstanding pressures of up to 300 atmospheres. Also identified in the table is the possibility of using CNG or LPG in a dual-fuel mode whereby a proportion of the energy supplied to the engine is gas fumigated into the inlet manifold and a smaller proportion of the energy comes from diesel fuel injected in the normal way. The gaseous fueled engines have almost zero particulate level with little odor and are therefore also quite suited to operation in enclosed spaces such as foyers to railway stations, bus depots, down town city streets and the like.

(See [Table 1](#) after last section of paper)

The gas turbine has been a long time engine championed for bus and truck operation but poor part load efficiency and the need for waste heat recovery through the use of a regenerator mitigate its application.

Also seen in [Table 1](#) is the fuel cell. These have been used in experimental vehicles since 1954. The recent advent of the proton exchange membrane (PEM) fuel cell produced by Ballard in Canada has seen renewed enthusiasm for this power unit, on the basis that weight-to-power ratios have been reduced by 80%. Even so, the fuel cell has the undesirable need to run with hydrogen as fuel. Hydrogen is the most difficult to store of the gaseous fuels because of its low volumetric energy density, and liquid hydrogen or metal hydride storage is needed to obtain a reasonably compact, though difficult to manage energy source.

Although electric drives are receiving increased interest, necessary improvements in batteries and range limitations mean a viable solution is still distant.

Gaseous fueled engines offer a suitable medium term (about 2 to 4 years lead time) alternative to diesel engines with immediate environmental benefits until hybrids, fuel cells and electric vehicles can become practically viable. A brief review of LPG engines for heavy vehicles is now presented.

LPG ENGINES FOR HEAVY VEHICLES

Application of LPG to heavy vehicles to date has been limited. More recent development of LPG engines for heavy vehicles has focused primarily on liquid phase port injection of LPG. Liquid-phase is attractive as it mitigates some of the power loss which occurs with LPG due to the displacement of intake air by the gaseous fuel. Additionally, with liquid-phase injection, the evaporation of the liquid spray cools the intake air and increases its density, resulting in higher power output.

Studies with liquid-phase LPG for heavy vehicles have yielded particularly good results. An experimental and theoretical study by Sierens [1] into LPG liquid-phase injection showed that with an increase in compression ratio, significant gains in power were achievable over the baseline gasoline engine in the order of 10% to 15%. Similar results were achieved by Kang et al. [2] who conducted research into the performance of a liquid-phase LPG engine for heavy vehicles, showing that the liquid-phase system resulted in higher power and reduced emissions than the conventional gas mixer system. Additionally, an increase in volumetric efficiency and engine output by 10% was observed, as well as a decrease in manifold temperature. Consequently, exhaust gas temperatures were decreased and better control of the knocking phenomena, through optimisation of spark timing and compression ratio was possible. The engine was also successfully operated at $\lambda = 1.5$. A similar study with an 11 litre heavy vehicle engine also by Kang et al. [3] demonstrated higher engine performance and lower exhaust emissions than the baseline diesel engine. Other studies by these authors and others are available in references [4,5,6,7].

However, a significant drawback of liquid-phase injection is the high relative componentry cost, due to the low volume of LPG injector and fuel pump production when compared to equivalent gasoline components. Market research conducted by OEMs and others shows that as LPG is generally cheaper than gasoline (usually as the result of reduced government taxes applied), motorists additionally expect low LPG system cost [8,9,10]. However, the reality is that LPG injectors requiring bottom-feed to avoid vapour lock, are produced in low volume, and to have non-intrusive integration and performance, the cost increases [8]. Moreover, the high pressure LPG injection pumps manufactured presently in volumes of less than 100,000 per annum cost an order of magnitude greater than their gasoline equivalent. Hence, it is unlikely that liquid-phase systems will receive widespread popularity unless costs can be reduced. LPG Australia's survey of fleet managers, small business operators and private buyers of LPG vehicles showed that the choice to adopt LPG was primarily based on cost, and in some cases, the only reason for using the fuel [8]. Fleet managers and small business operators particularly stressed the need to minimise the cost of the LPG system. To mitigate these costs, the LPG system should be as simple and as low cost as possible, yet more efficient (to further reduce operating costs) and with lower greenhouse and particulate emissions.

Studies into fifth generation technologies of LPG direct injection (DI) systems are now also emerging [11, 12], however, research in this field is still limited and the cost of integration and control of DI systems may not be yet justified, as emission and performance requirements can currently be met with simpler systems (e.g. sequential port injection).

OBJECTIVE

In this study, a simple throttle-body injection (TBI) system is presented, being low cost and with low control complexity. The current study differs from previous work in that it is not based on a remodeled diesel engine, but on a light duty engine with capability for heavy vehicle application. Additionally, it offers a lower cost alternative to the trend towards liquid-phase injection of LPG.

Specifically, the objective of this paper is to demonstrate the suitability of an optimized LPG light duty engine's application to trucks or buses of 15 - 20 tonne GVM, needing 170 kW and high torque over the Euro truck stationary and transient cycles. Then to compare the engine with CNG and diesel fuelled power units. The starting point is a review of the engine's development for light duty application.

REVIEW OF LPG FOR LIGHT DUTY

Several research programs by the authors over a number of years have focused on improving the efficiency of the test engine under discussion for use with LPG fuel, and also CNG. The test engine is an Australian model 4-litre Ford Falcon AU II version, 6-cylinder in-line engine. To take advantage of the higher octane rating of LPG compared to gasoline (110 to 95 RON respectively), the engine was modified from its standard configuration with the compression ratio increased from 9.65:1 up to 11.7:1. Details are provided in [Appendix A](#). The engine was also tested at 12.9:1 with LPG, although not comprehensively [13]. The increase in compression was achieved by the use of modified pistons which had been optimised in prior research on a CNG version of the engine [14-15]. The engine specifications are shown in [Table 2](#).

Table 2. Test engine specifications

Bore × stroke	92.26 × 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 original research program focused on development of the engine for light duty application. In the study, various LPG systems were compared at different compression ratios against the baseline gasoline engine in OEM form [13, 16]. The details of the test apparatus and procedure are to be found in [Appendix B](#). 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. This increase in efficiency is attributable to the increase in compression ratio, liquid-phase injection and lean burn operation. Light duty results for the Euro NEDC cycle showed engine-out emissions also improved dramatically with best observed reductions in specific emissions of 88% for hydrocarbons (HC), 45% for NO_x , 40% for CO_2 and 92% for carbon monoxide (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 (PFI versus TBI). The differences in results between the various gaseous and liquid phase LPG systems were attributed 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 [17-18] 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 the surface area of the interface created, the greater the increase in diffusion rates and smaller droplet sizes quantities down to a SMD of 20 μm , which allows faster mixing.

It was also shown that variations in the air-fuel ratio between cylinders affects engine emissions and performance. With port fuel injection, despite having an injector located in each inlet port, significant variations in air fuel ratio between cylinders were shown to exist. A typical example is shown in

Figure 1 for the engine under discussion at $\lambda = 1.0$ for a world-wide mapping point (WWMP) equivalent to 1500 rev/min and 50Nm for this engine [13]. As can be seen, in all cases in Figure 1, the outer cylinders (cylinder numbers 1 and 6) have leaner mixtures than the other four cylinders, despite the location of a single injector in each port. This variation in air-fuel ratio has adverse consequences on engine performance, emissions, and combustion stability, which is particularly critical at lean burn operating conditions. Consequently, due to this cylinder maldistribution, all of the standard LPG fuelling systems tested could only be operated to as lean as $\lambda = 1.4$, before combustion instability became problematic.

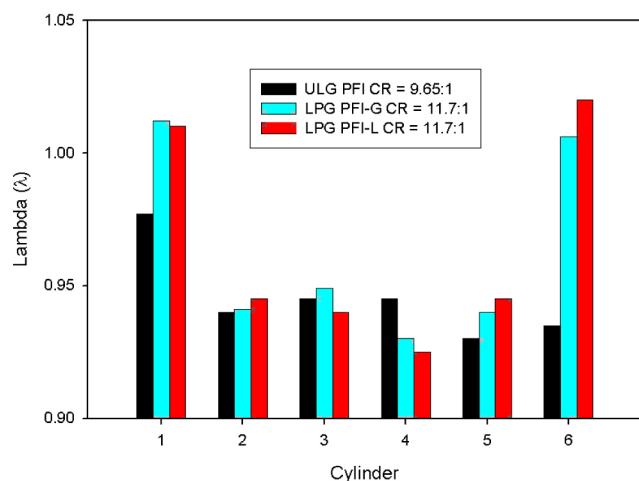


Figure 1. Cylinder-to-cylinder variations in lambda (λ) for three fuelling systems at 1500 rev/min, 50 Nm, MBT spark timing (ULG -unleaded gasoline, PFI-G - port fuel injection (gaseous phase), PFI-L - port fuel injection (liquid phase), CR - compression ratio).

To reduce these observed variations in air fuel ratio between cylinders and improve mixture quality, a throttle-body injection (TBI) system was designed, 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 that fired several times per engine revolution. The design was configured from CFD studies by Liew [19] 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 on the engine. For details, the reader is referred to these other works [19,20,21,22].

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 $\lambda = 1.6$ over most of the load range. At $\lambda = 1.6$ the NO_x levels could satisfy Euro 5 emission standards without the need for a reducing catalyst [20,21,22]. These benefits were achievable due to careful optimization of the location of injector nozzles and angles as well as optimization of injection parameters.

Figures 2 and 3 show the Euro weighted NEDC emissions and efficiency results for the test engine, comparing the TBI system against the existing gas mixer and gas and liquid phase port injection systems. The standard gasoline engine in OEM form is also compared. These results are now discussed.

For HC, all LPG results are much lower than the base engine gasoline data at all values of lambda. At stoichiometric, HC emissions are about 15% of those from gasoline for all LPG systems, and even up to $\lambda = 1.4$, 30% less than stoichiometric gasoline, with the increased compression ratio reducing the HC emissions in spite of the increased combustion chamber surface-area to volume ratio. All engine-out LPG HC emissions meet Euro5/6 standards out to $\lambda = 1.2$. It is noted that both PFI-gas systems, irrespective of compression ratio, follow the same curve from $\lambda = 1.0$ to 1.2 with slightly higher emission figures than TBI and PFI-liquid phase.

Additionally, both TBI and PFI-liquid phase systems at 11.7:1 compression ratio follow the same curve out as far as $\lambda = 1.3$ before the PFI-liquid phase system suffers deterioration in performance. At $\lambda = 1.4$, the reduction in HC of the TBI system over the standard gasoline case is 47%. At $\lambda = 1.6$, the TBI system has HC emissions above stoichiometric gasoline; however, these could be controlled with an oxidation catalyst.

The NO_x trends show reductions below those for gasoline, with the higher compression ratio results generally giving higher NO_x as might be expected. At the high compression ratio the liquid phase shows an advantage over the gas phase PFI because of lower mixture temperatures that result from the late fuel evaporation in the port and cylinder. Interestingly, the NO_x trend is the same for PFI-liquid and TBI CR 9.65 case in the region $\lambda = 1.2$ to 1.4. Leaning out to $\lambda = 1.6$ with TBI, the engine is able to achieve extremely low NO_x levels reaching 98% below stoichiometric gasoline and satisfying Euro 5/6 standards without the aid of a reducing catalyst. This is a significant advantage of very lean operation. The durability limitations of lean NO_x traps and lean NO_x catalysts, especially in the presence of sulfur fuels and additives, are well established as discussed in [23,24,25,26]. As the catalysts age, the regeneration cycles increase in frequency, with a fuel consumption penalty and corresponding deterioration in CO_2 emissions. The absence

of the need for a NO_x reduction catalyst can remove this drawback faced by current lean burn technology.

The CO trends are similar to those for HC. Emissions reductions from LPG use are in the range of 50 to 90% lower than those for gasoline. Euro 5/6 is satisfied with all the LPG cases with $\lambda = 1.1$ or higher. Inspection of Figure 2 shows that at stoichiometric, there are marginally higher CO emissions for the high compression trials compared to TBI at the standard compression ratio, except for TBI CR 11.7 where this increase is small. Additionally, there is a slight increase in CO emission level at stoichiometric for PFI-gas high compression compared to PFI-liquid phase high compression, also consistent with poorer mixing with the gas system.

As CO_2 is always the major contributor to greenhouse gas emissions from transport emissions, the benefits in this emission reduction is a significant argument for the application of this fuel. LPG is also advantageous over CNG, as methane itself is a greenhouse gas. As Figure 3 shows, the reduction in CO_2 emission for the TBI system is 33% at $\lambda = 1.2$ from gasoline stoichiometric, reducing further to a 36% reduction at $\lambda = 1.4$. This reduction is a combination of the 12% attributable to the change in H/C ratio of LPG to gasoline, increased compression ratio possible because of the higher octane number of the gaseous fuel and lean burn operation which reduces combustion temperatures and hence dissociation at peak combustion pressures.

The thermal efficiency results in Figure 3 exemplify the real increases in efficiency possible with gas technology over gasoline. Experience with bi-fuel cars in comprehensive vehicle tests across a range of LP gases [27] and the 9.65:1 CR results here demonstrate that ordinarily both fuels deliver fuel consumption on an energy basis within a few per cent. With the increase in compression ratio and choice of liquid fuelled LPG at $\lambda = 1.4$, the improvement in thermal efficiency is 26% relative to low compression ratio stoichiometric gasoline. Appendix C gives a breakdown of the contributions from compression ratio increase and the lean mixture shift in terms of concomitant CO_2 reduction. In Figure 3, the efficiency of the TBI system at CR 11.7 follows the same trend as the liquid phase case, only dropping slightly at $\lambda = 1.4$, with a maximum improvement of 23% over stoichiometric gasoline.

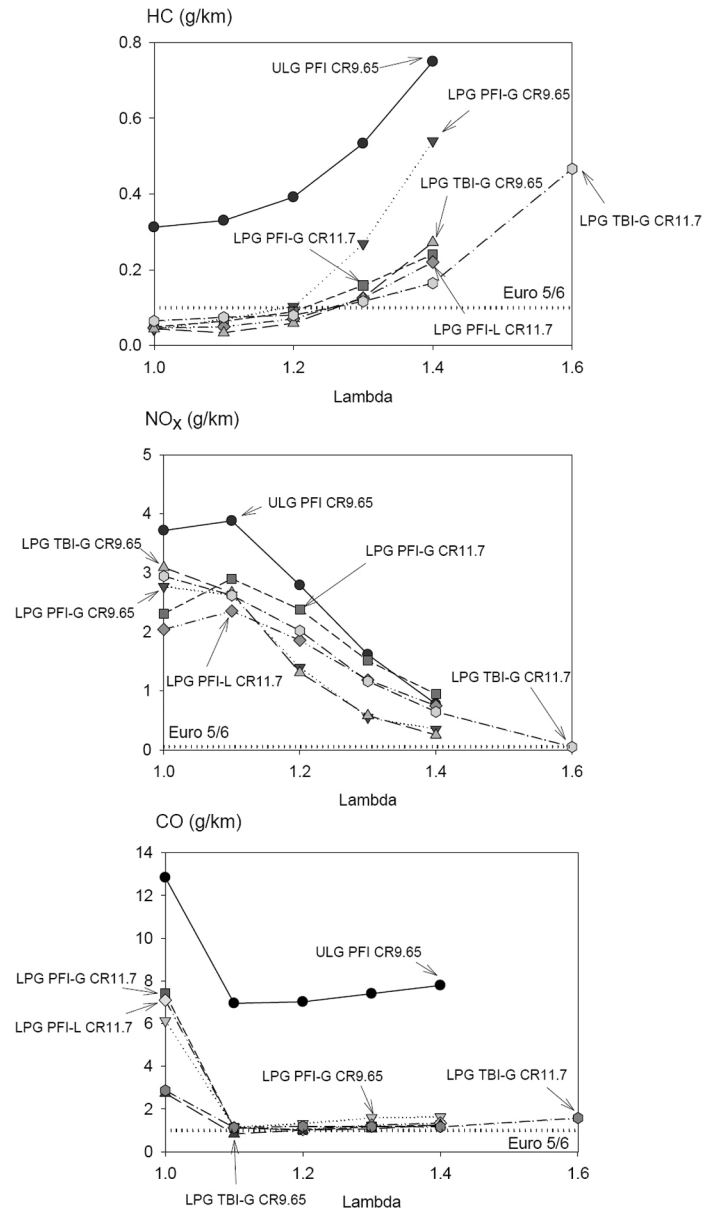


Figure 2. NEDC drive cycle weighted emissions of HC, CO and NO_x across the range of LP gas fuel supply technology compared with the reference gasoline (ULG).

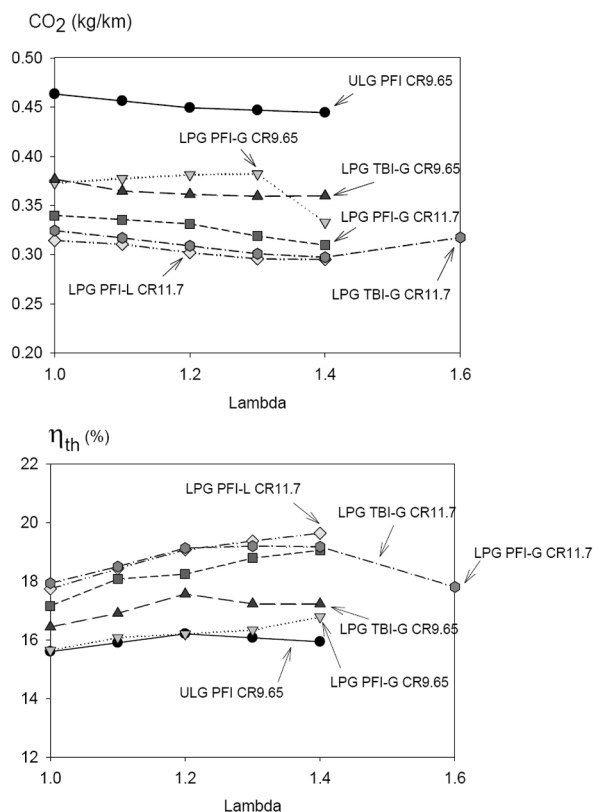


Figure 3. NEDC drive cycle weighted emissions of CO₂ and thermal efficiency across the range of LP gas fuel supply technology compared with the reference gasoline (ULG).

Based on the above results, the potential for LPG to reduce NO_x and CO₂ emissions in particular is very significant. Although highest efficiency was observed with the liquid phase port injection system, it is not able to operate near $\lambda = 1.6$ which is needed for the very low NO_x levels achieved with the TBI system. The presence of lean mixtures in the outer cylinders (as observed in Figure 1) results in misfiring of the leaner cylinders before a sufficiently overall engine lean mixture can be achieved.

Based on the preceding results and discussion, it is proposed that this engine has the potential to deliver better than diesel CO₂ emissions with relatively low cost LPG components and likely meeting Euro 5/6 emission standards with only oxidation catalyst after treatment in contrast to the complex after treatment needed for diesel engine compliance with the diesel standard. Moreover, the reduction in particulates by switching from diesel to LPG is also very significant.

The remainder of the paper considers the application of the LPG engine with the TBI system for heavy duty vehicles.

HEAVY-DUTY RESULTS

ENGINE MAP RESULTS - THERMAL EFFICIENCY AND EMISSIONS

Before presenting results for the Euro heavy vehicle stationary and transient cycles, the engine map results are presented for thermal efficiency and emissions. In this discussion, as a light duty engine is being considered for heavy vehicle application, the engine map is important for understanding the effects of the weightings of the various drive cycles to different regions of the engine map. These will differ considerably between the light duty NEDC cycle already presented and the heavy vehicle cycles to follow.

Thermal Efficiency

Figure 4 shows the thermal efficiency maps for four different values of lambda. These maps are presented for torques up to 300 Nm as the maximum torque varies with lambda as expected. The maximum torque at 3,000 rev/min is 365 Nm falling to 330 Nm at $\lambda = 1.6$. The lower load regions shaded out in the engine map for $\lambda = 1.6$ indicate where the engine cannot operate at such lean mixtures.

Inspection of Figure 4 shows that in the low torque region of the engine map, richer mixtures produce higher thermal efficiency values. This is true until the load is increased significantly, after which this trend reverses. At very light load, richer mixtures are needed to compensate for slower flame speeds caused by lower mixture density and increased residual gas fraction [28]. Consequently, higher efficiencies are observed at $\lambda = 1.0$ for a given operating point at low load conditions. Efficiency decreases in this low load region as lambda is increased. As load is increased, approaching full load, increases in efficiency may be achieved with leaner operation as observable in Figure 4. However, thermal efficiency only increases with lean operation until the cycle-by-cycle pressure fluctuations and the total duration of the burning process become too large and begin to adversely affect efficiency. The lengthening burn duration itself directly decreases efficiency, even in the absence of cyclic variations [28]. Figure 4 also shows that thermal efficiency deteriorates with increase in engine speed for all cases. This is not unexpected as the increase in engine speed increases the mechanical frictional losses.

From these results, it is important to understand the region that the engine operates in under normal driving patterns (or in specified drive cycles) in order to best devise a mixture control strategy for the engine map for high efficiency and low fuel consumption.

Hydrocarbons

Hydrocarbon emissions are shown in Figure 5. Although mixtures slightly lean of stoichiometric will reduce

hydrocarbon levels, increasing lambda to very lean mixtures has very adverse effects on hydrocarbon emissions. As the number of misfire cycles increase, the hydrocarbon levels will increase dramatically as combustion becomes increasingly unstable. Hydrocarbon levels, therefore, may be used as an indicator of the lean-limit.

Considering the lean region between $\lambda = 1.2$ and $\lambda = 1.6$ in [Figure 5](#), the HC levels are approximately five times higher at $\lambda = 1.6$, compared to those at $\lambda = 1.2$. Although these HC levels are substantial at $\lambda = 1.6$, modern oxidation catalysts are capable of cleaning such HC emissions in the exhaust with high efficiencies and good durability.

[Figure 5](#) also shows that hydrocarbon levels generally decrease with increasing engine speed. The primary effect of increases in engine speed on hydrocarbons is to reduce total heat transfer from the burned gases, therefore increasing crevice, wall quench, exhaust temperatures and the rate of oxidation of unburned hydrocarbons [29]. Hochgreb [29] has quoted that decreases in HC indices of the order of 20% to 50% are observed for increases from 1000 to 2000 rev/min, with decreasing sensitivity at higher speeds. [Figure 5](#) agrees quite well with her results.

With respect to the effects of load variation, the effect of load on HC emissions is more pronounced at low load conditions. At higher loads, the effect of load variation is less prominent. Hochgreb [29], quoting the works of other researchers has stated that small increases and decreases in HCs with load have been observed, depending on the particular operating conditions; typically with a sharper increase at very light loads, followed by a plateau.

Nitrogen Oxides

One of the major objectives if operating the engine at $\lambda = 1.6$ is to dramatically reduce NO_x levels. [Figure 6](#) shows NO_x emissions dropping dramatically with increase in lambda as the mixture is leaned. Operating at $\lambda = 1.6$, NO_x levels are approximately 50 to 75 times less than $\lambda = 1.2$ levels. The benefits of lean burn strategy are very apparent.

As already mentioned, a significant obstacle to the widespread adoption of lean burn strategies is the inadequacy of current NO_x exhaust aftertreatment technology. Current technology for lean NO_x catalysts is still not sufficient to provide high conversion efficiency for the life of a vehicle, despite some progress in catalyst technology [23,24,25,26, 30]. This presents a significant drawback to effective lean operation and hinders its widespread adoption. Several studies have shown that lean NO_x catalysts are highly sensitive to fuel sulfur levels and deteriorate in performance with age. Current fuel sulfur levels are generally too high to not affect catalyst performance. Mercaptans (sulfur containing compounds) are used as odorizers of gaseous fuels

for human detection of gas leaks into the atmosphere. The durability limitations of lean NO_x traps and lean NO_x catalysts, especially in the presence of sulfur fuels and additives, are well established as discussed in [23,24,25,26, 31-32]. The lean burn strategy avoids these limitations.

As NO_x formation is a function of the temperature history of the burned gas, and particularly, the peak temperature, increase in load causes increases in NO_x concentrations as observable in [Figure 6](#). The correlation with speed is less clear. It appears that speed has no significant effect on NO_x concentrations in the range tested. This would agree with the findings of other researchers, where the NO sensitivity to speed has been shown to be variable but usually small [29].

Carbon Monoxide

[Figure 7](#) shows the CO concentration contour plots. As can be seen, little change is observed with lambda in CO emission levels. This is not surprising. Although CO concentrations are predominantly dependent on the relative air-fuel ratio, in the lean region, CO concentrations vary little with equivalence ratio and are of order 10⁻³ mole fraction [28].

This accounts for the little change in results with increasing lambda. In any case, leaning of the mixture is the most effective way to decrease CO emissions without utilising exhaust aftertreatment.

It is noted also that CO concentration levels increase with engine speed. This may be attributed to the mixing process. With increasing engine speed, the residence time decreases giving the air and gaseous fuel shorter periods of time to mix adequately. The poor mixing gives rise to rich local regions that have high equilibrium CO, which then do not find sufficient oxygen in the mixing zone to burn out [33]. Additionally, poorer cylinder-to-cylinder air-fuel distribution may be present at increased speed. No cylinder-to-cylinder tests were conducted at speeds greater than 2000 rev/min, so it is possible that some deterioration in cylinder distribution occurs with the increased speed leading to rich mixtures in some cylinders giving rise to higher levels of CO.