

Improving the Efficiency of LPG Compression Ignition Engines for Passenger Cars through Waste Heat Recovery

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ABSTRACT

The turbocharged direct injection lean burn Diesel engine is the most efficient now in production for transport applications with full load brake efficiencies up to 40 to 45% and reduced penalties in brake efficiencies reducing the load by the quantity of fuel injected. The secrets of this engine's performances are the high compression ratio and the lean bulk combustion mostly diffusion controlled in addition to the partial recovery of the exhaust energy to boost the charging efficiency. The major downfalls of this engine are the carbon dioxide emissions and the depletion of fossil fuels using fossil diesel, the energy security issues of using foreign fossil fuels in general, and finally the difficulty to meet future emission standards for soot, smoke, nitrogen oxides, carbon oxide and unburned hydrocarbons for the combustion of the fuel injected in liquid state and the lack of maturity the lean after treatment system. LPG is an alternative fuel with a better carbon to hydrogen ratio therefore permitting reduced carbon dioxide emissions. It flashes immediately to gaseous form even if injected in liquid state for a much cleaner combustion almost cancelling some of the emissions (even if unfortunately not all of them) of the diesel and it permits a much better energy security within Australia. The paper presents a passenger car diesel engines converted to LPG. In this engine the efficiency is then improved recovering the waste heat. This recovery has impacts on both the in cylinder fuel conversion efficiency and the efficiency of the after treatment. Results of engine performance simulations are performed for a in-line four cylinder 1.6 litres LPG CI passenger car engine with a power turbine following the turbine of the turbocharger or an heat exchanger to recover the exhaust (and other) waste heat and compared with the

experimental results for the diesel without waste heat recovery.

CONVERSION OF DIESEL ENGINES TO LPG

As long as the price of crude oil increases and the threat of global warming and air pollution remain, there will be a need for alternative fuels. Liquefied Petroleum Gas (LPG) has emerged as a perfect solution to these challenges: it is inexpensive, clean burning and relatively low in carbon content [1, 2]. Diesel engines have been the backbone of the transportation industry. Valued for their power, fuel economy and durability, diesel engines are used worldwide. However, they are heavy emitters of particulate matter (PM) and oxides of nitrogen (NOx). By converting diesel engines to run on LPG we can significantly reduce the problem of diesel pollution whilst also improving emissions of Green House Gases (GHG). Such conversions are however not a simple matter of changing the fuel. Many technical problems have to be considered particularly with engine optimization and fuel injection control to ensure that the engine performances are maintained and the exhaust emissions are minimized. Research and development of LPG engines in general, and of conversion of diesel engines to LPG in particular, has always been underfunded for the limited interest towards LPG in the major markets.

Life-cycle emissions analysis of alternative fuels has shown the potential of gaseous fuels [1, 2]. There have been major advances in natural gas engines in recent years that mean that the present generation of natural gas vehicles has significantly lower emissions than the present generation of diesel vehicles. The emissions based on use in Original

Equipment Manufacturer (OEM) vehicles are lower in all categories, GHG, important criteria pollutants, and air toxics. The lower particulate emissions and noise levels compared with diesel make it particularly attractive for urban areas. The major uncertainty relates to upstream and in-service leakage, on the other hand already sufficiently reduced in the present generation of OEM compressed or liquid natural gas (CGN or LNG) vehicles, and the lack of sufficient refueling stations. The extra weight of CNG fuel tanks leads to slightly higher fuel consumption and loss of payload, but this is less of a problem with LNG vehicles due to the higher energy density. Similarly, a dedicated LPG vehicle produces significantly lower emissions of important criteria pollutants, and lower embodied emissions of GHG. Toxic air pollutants from tailpipe emissions of LPG vehicles are much lower than those of diesel vehicles, but the greater upstream emissions of air toxics results in the embodied emissions of air toxics from LPG being much the same as those from diesel [1, 2].

HD5 LPG has minimum propane content of 90% whereas the ratio of propane and butane varies widely in auto gas LPG. When compared with auto gas, HD5 LPG emits more NO_x but less particulate matter. Emissions of hydrocarbons are similar. The main benefit of HD5 LPG compared with auto gas is that the compression ratio can be altered to suit this higher-octane fuel. Therefore, the lower particulate emissions and lower noise levels compared with diesel make it attractive for use in urban areas. The major disadvantage of LPG is the lack of market penetration of LPG dedicated vehicles especially for heavy applications. There have been few serious attempts to convert diesel engines to LPG for the following reasons. Conversions have focused on CNG because its 130 octane number means only small or no change is needed from diesel engine combustion chamber shape. In many countries there is poor availability of both LPG and CNG outlets, and in many others where LPG is used for heating and cooking there is an excess of demand over supply. Furthermore, Engine manufacturers are reluctant to devote effort to developing engines for a niche or non-existent market.

In Australia LPG is used in automotive applications in cars (almost all taxis) and special applications in close proximity to workers in confined or enclosed spaces such as fork lift trucks, concrete mixers and similar plant because of the inherent low emissions of the fuel related to its gas phase and small molecule size relative to conventional fuels [3]. Australia is unique in having an established LPG infrastructure with over 3,500 retail outlets for LPG distributed across the continent well in excess of those in the whole of North America, where they are largely confined to West and East Coast regions. Furthermore Australia produces about 30% more LPG than it consumes and its availability will increase as more LNG plants come on stream as LPG is a liquid by-product from natural gas liquefaction. Inherently as a by-product LPG has only a minor energy production cost

compared with gasoline and diesel where significant increase in production energy have occurred to meet the recently introduced Australian fuel quality standards reducing sulphur levels and meeting composition and test requirements with strict quality control. Moreover the GHG emissions associated with LPG production according to both GREET and SIMA-PRO LCA analyses are only about two thirds of those of low sulphur diesel and about 60% of gasoline production. More important is that LPG has a 12% reduction in CO₂ emission (because of its lower C/H ratio), and whilst not as low as NG (Methane), it requires negligible compression energy compared with NG gas compression (~7-11% of gas energy) to CNG or liquefaction (~17%) to LNG. The challenge for further use of LPG is to produce an engine that has equal or near equal thermal efficiency to diesel to thus obtain the benefits of fuel composition (12%) and (8-20%) less production and distribution energy.

DUAL FUEL ENGINES DIESEL-LPG

The most part of the dual fuel gas-diesel internal combustion engines developed so far have been designed using the direct injection of the diesel fuel to ignite the homogeneous gas-air mixture produced injecting the gas in the throttle body or in the ports. This approach does not address the abnormal combustion phenomena operating with high octane number gaseous fuels high boost and high compression ratio engines, and does not permit to retain the high efficiencies of the diesel engine both full load and part load for the homogeneous combustion of the gas-air mixture. Despite the upstream premix of gas and air and the homogeneous combustion of the mixture within the cylinder is not optimal, the most part of the dual fuel gas-diesel internal combustion engines currently being considered in research papers is based on upstream throttle body or port fuel injection of gas [4,5,6,7,8,9,10,11,12,13,14,15,16,17,18,19]. However, a theoretically more sound solution has been proposed since more than two decades, and it is based on the direct injection of gas and diesel within the cylinder [20,21,22,23], with off-the-shelf products commercially available.

The characteristics of the diesel engine are retained operating with a small quantity of diesel fuel and a much larger quantity of natural gas [20,21,22,23]. The natural gas is injected at high pressure, thus the concept is called high pressure direct injection (HPDI). A unique injector concept makes possible the double injection to take place within a single injector that fit in the same location as the original diesel injector [20,21,22,23]. With this technology, diesel engines are able to run on natural gas without compromises in performance and fuel efficiency. Operation with LPG rather than CNG is considered here using similar hardware.

The major advantage of the Direct Injection diesel engine is the bulk, lean, diffusion combustion in a high compression ratio highly boosted engine with reduced heat losses to the

liner and head walls. The engines using the stoichiometric, premixed combustion controlled by a spark discharge suffer the disadvantage of the higher heat losses to the liner and head walls. The combustion is wall initiated, and the flame has to reach all the walls for burning all the homogeneously distributed fuel. Furthermore, the compression ratio is reduced to avoid the abnormal combustion phenomena.

Using LPG as the fuel of spark ignited internal combustion engines, the abnormal combustion phenomena, already suggests replacing the traditional throttle body or port fuel injection with the direct injection within the cylinder. Furthermore, the lean limit to the combustion stability running homogeneous is very well known to be much less extended than running lean stratified, with combustion rates much lower in locally much leaner conditions. In case of dual fuel operation of a diesel engine with LPG as the second fuel, it seems therefore logical to inject both the diesel and the LPG fuel within the cylinder to achieve the same as diesel efficiencies both full load and part load, with the load being controlled by the quantity of fuel injected.

The single dual fuel injector [20,21,22,23] has the advantage to be easily incorporated into an existing diesel engine with minimal or no modifications to the engine cylinder head, while the two injectors' option requires a new design of the cylinder head only possible on novel engines. Late-cycle, high-pressure direct injection ensures diffusion type combustion for the LPG and therefore it is expected to retain the high power, torque, and efficiency of the diesel engine with possibly the advantages of the higher power densities.

The LPG injector has to manage the high volumetric flow rates required for the LPG fuel much less dense than diesel and the carbonizing and thermal fluctuations that can result from the reduced cooling capacity of a gaseous fuel. The major disadvantage of the LPG-diesel fuel injector is the complex fuel routing for two fuels within one injector. This injector provides a small diesel pilot spray and a much larger LPG gas spray. The diesel averages only about 5% of the total energy input and is used to start the main combustion of high-pressure directly-injected LPG gas and the reduced pollutants of the LPG.

The Westport [20,21,22,23] High Pressure Direct Injection HPDI injector for diesel and LNG is a common-rail style injector that directly replaces the diesel injector. This injector may independently inject diesel and gas at up to 30 MPa of injection pressure. The diesel fuel is used as an ignition source, actuation fluid, lubricant and coolant. The injector body contains an outer gas needle and an inner diesel needle independently operated that produces an outer gas spray and an inner diesel spray. The injector uses gas as the primary fuel and a small amount of Diesel as the pilot injection fuel. The gas is injected at the end of the compression stroke, just like the diesel fuel is injected at the end of the compression

stroke in a diesel engine. Under the pressures found in the combustion chamber of a diesel engine, the gas requires higher ignition temperatures than diesel to maintain an acceptable ignition delay period of less than 1 ms. Therefore, to assist the ignition of the gas, a small amount of diesel fuel is injected into the engine cylinder followed by the main gas fuel injection. The pilot diesel combustion rapidly ignites the gas.

It is worth of note that while the Westport technology uses a pilot diesel injection preceding the main gas injection [20,21,22,23], there is no limitation to expand the dual fuel injection technology to consider different injection strategies, with both injections occurring for example almost simultaneously, or having for example the diesel injection following an earlier gas main injection or even no diesel injection at all, to explore alternative combustion modes of operation.

In the following section, the baseline diesel engine injection and combustion as well as the dual fuel injection and combustion will be represented through an equivalent injection and combustion process, where after an ignition delay chemically controlled, there is a premixed combustion chemically controlled and finally a diffusion combustion mixing controlled. The other injection strategies and the different combustion patterns that may be produced are not investigated here not because these are not interesting, but just because of the limited space to consider. Interaction with water direct injection and exhaust gas recycle are other areas worth of interest not considered here just for brevity.

The HPDI injector has never been tested with LPG and diesel, but only with LNG and diesel. Without any doubt, a similar injector could be developed with minimum resources to operate with LPG and diesel. The availability of such an injector is assumed here. The main injection of the LPG is supposed to occur after the pilot injection of the Diesel. The pilot injection of the Diesel amounts to roughly the 5% of the total fuel energy. The following paragraph deals with the modelling of the operation of the Diesel and the dual fuel Diesel-LPG engine. The subsequent section then deals with the brake efficiency improvements that may be achieved recovering the exhaust and coolant energy with organic Rankine cycles in the dual fuel engine.

The dual fuel engine operates compression ignition with the embedded advantages of the higher top brake efficiencies and the reduced penalties changing the load by quantity of fuel injected. The major advantage of the HPDI technology demonstrated with LNG [20,21,22,23] is the ability to burn diesel-like low Cetane high octane fuels. Following the diesel pilot injection, the LPG is expected to burn quickly as it has been demonstrated with the LNG.

MODELLING THE BASELINE DIESEL AND THE DUAL FUEL ENGINE

Simulations have been performed using the WAVE code [24]. WAVE is one of the market-leading 1D engine and gas dynamics simulation software package from Ricardo Software. This software is used worldwide since more than two decades in industry sectors including passenger car, motorcycle, truck, locomotive, motor sport, marine and power generation. WAVE enables performance simulations to be carried out based on virtually any intake, combustion and exhaust system configuration.

Modelling of traditional Spark Ignition and Compression Ignition engines with WAVE has been the subject of numerous studies, some of them published in peer-review Journals or peer-review papers. Only few details of the model, namely the modelling of combustion, need to be provided.

Despite every modern diesel engine now uses multi event strategies, and the one considered here in particular adopt a three event injection, with pre, main and post injection, an equivalent single injection is used to replace the actual injection event. Smoke and soot formation, as well as the combustion noise, other parameters that determine the fuel injection strategy in addition to brake specific fuel consumption, variance of brake mean effective pressure and formation of major pollutant (HC, NO, CO), are not considered in the model.

Combustion is modelled in this work using the WAVE diesel Wiebe Combustion object. Whilst the sub-model name including Wiebe implies a non-predictive curve-fit correlation, the Cetane number and the reference speed parameters used for calibration enable a semi-predictive diesel combustion model, predicting ignition and responding to changes in the trapped mixture conditions as well as engine operating speed. The reference speed for burn duration is typically the engine rated speed. This input is used to calibrate the burn duration. The Cetane rating of fuel used is used as a tuning parameter for ignition prediction and does not need to be precisely the chemical Cetane number of the fuel. The start of combustion timing and premixed burn fraction are automatically computed from cylinder conditions during fuel injection and the user-entered Cetane number.

The Wiebe combustion model uses functions similar to the correlations for pre-mixed and diffusion burn regimes. An additional third function has been added to represent the slow late burning (referred to as tail burning) at the end of the diffusion burning regime. In addition, the original functions were slightly modified to improve agreement with

experiment. The model used in WAVE for the mass fraction burned, W , is:

$$W = \overbrace{p_f \left\{ 1 - \left[1 - (0.75\tau)^2 \right]^{5000} \right\}}^{\text{premix}} + \overbrace{d_f \left\{ 1 - \left[1 - (cd_3\tau)^{1.75} \right]^{5000} \right\}}^{\text{diffusion}} + \overbrace{t_f \left\{ 1 - \left[1 - (ct_3\tau)^{2.5} \right]^{5000} \right\}}^{\text{tail}} \quad (1)$$

where p_f , d_f , t_f are the Mass fractions of the premix, diffusion, and tail burn curves, respectively, cd_3 , ct_3 the burn duration coefficients for the diffusion and tail burn curves, respectively and τ is the burn duration term.

The mass fraction of the premix burn is obtained from the ignition delay model. The mass fractions of the diffusion and tail burn curves are obtained using the following equations:

$$d_f = (1 - p_f)(1 - \alpha) \quad (2)$$

$$t_f = (1 - p_f)\alpha \quad (3)$$

$$\alpha = 0.6 * [\min(\Phi, 0.85)]^2 \quad (4)$$

where Φ is the trapped equivalence ratio. The burn duration coefficients for the diffusion and tail burn curves and the burn duration term are determined using the following equations:

$$cd_3 = \frac{0.055}{1 + 0.5 \min(\Phi, 0.85)} \quad (5)$$

$$ct_3 = \frac{3.7cd_3}{1 + 1.12 \min(\Phi, 0.85)} \quad (6)$$

$$\tau = \frac{\theta - \theta_b}{125 \left(\frac{RPM}{BRPM} \right)^{0.3}} \quad (7)$$

where θ is the crank angle, θ_b the crank angle at start of combustion, RPM the engine speed and $BRPM$ the user-entered reference speed. The burn duration varies inversely with $BRPM$. Higher values of $BRPM$ result in a shorter burn

duration and lower values of *BRPM* result in a longer burn duration.

Diesel combustion models in WAVE calculate the amount of air burned at each time step using an empirical relation. The entrainment of air is prescribed so that the burned-mass equivalence ratio varies as combustion progresses. WAVE uses a relationship dependent on the current value of the fraction of fuel mass burned beginning as a relatively rich burn and progressing to a lean burn such that, at the end of combustion, the air/fuel ratio of the burned mass is exactly the overall air/fuel ratio. Since the thermodynamic state of the burned mass is dependent on its air/fuel ratio, the time-dependent apparent heat release is affected by this schedule of air/fuel entrainment.

The ignition delay is calculated using the in-cylinder temperature and pressure (averaged over the delay period). In addition to the in-cylinder temperature and pressure histories, this extended correlation includes an additional dependency on the fuel Cetane number. This correlation is calculated as follows:

$$\Delta\theta_{del_{cyl}} = 323 \exp^{\min(2100 \cdot C / T_{cyl}^{avg}, 80.0) / P_{cyl}^{avg}} \quad (8)$$

$$C = \frac{67}{25 + Cetane} \quad (9)$$

$$T_{cyl}^{avg} = \sum_n \frac{T_c^n + T_c^0}{2} * \frac{\Delta\theta_n}{\theta_{n+1} - \theta_0} \quad (10)$$

$$P_{cyl}^{avg} = \sum_n \frac{P_c^n + P_c^0}{2} * \frac{\Delta\theta_n}{\theta_{n+1} - \theta_0} \quad (11)$$

Where *Cetane* is the user-entered Cetane number, T_c^n , P_c^n the current cylinder temperature and pressure, respectively, T_c^0 , P_c^0 the cylinder temperature and pressure at the start of injection, respectively, $\Delta\theta_n$ the time step size and θ_0 the crank angle at start of injection. Increasing the fuel Cetane number decreases the ignition delay, and this can be used as a tuning parameter. The premixed fuel mass fraction corresponds to the amount of cumulative fuel vaporized during the calculated ignition delay period.

A reference speed of 4,500 revolutions per minute and a Cetane number of 50 is used in the simulations. This Cetane number is the default set-up for the diesel combustion model. The reference speed is the maximum speed of the diesel engine converted.

Simulations have been performed for a 1.6 litres, four cylinder, TDI engine. The main parameters of the engine are proposed in [Table 1](#). Extensive engine dynamometer data is available to cover almost 200 points in the load and speed map. For what concerns the equivalent injection, the start as well as the duration of the injection process in crank angle degrees is simply taken proportional to the product of the mass of fuel injected by the speed of rotation. This simplified injection replaces the actual multi event injection in the timing and duration of the injection but not in the quantity of fuel to air introduced that is the one experimentally measured. Wall temperatures in the Woschni heat transfer correlation are kept constant over the full range of speeds and loads for sake of simplicity.

In terms of the gas exchange processes, this phenomena is complicated by the turbo charger operation, and difference in between the measured and the computed flow rates as well as the boost pressure may be larger in the low engine speed region, where the interpolation of the not very accurate maps of turbine and compressor may provide a much less precise result. Including combustion, the maximum cylinder pressure and the brake mean effective pressure are well computed, with remarkable exceptions only in the low engine speed region for the reason outlined before.

The brake mean effective pressure (BMEP) is a result of the gas exchange and combustion simulation prediction of the indicated mean effective pressure (IMEP) plus the use of an empirical simulation to predict the friction mean effective pressure

$$FMEP = IMEP - BMEP \quad (12)$$

Then Chen and Flynn correlation [25] is used to compute the FMEP. Nevertheless, the prediction of BMEP is accurate within a few percents except than in the low engine speed region for gas exchange reasons.

Comparison of computed and measured values of full load power output and fuel conversion efficiency and brake specific fuel consumptions (BSFC) at different engine speeds and loads provides an accuracy within a few percents. [Figure 1](#) presents the computed and measured fuel flow rate vs. BMEP and speed. The percentage error in the fuel flow rate for a given couple of BMEP and speed points is equal to the percentage error in brake specific fuel consumption (BSFC), being the fuel flow rate proportional to the product of BSFC and power, with this latter proportional to the product of BMEP and speed.

This result proves that despite the very simplified representation of the complex multi event injection, an equivalent single injection of all the fuel and a semi predictive model as the one outlined above may properly