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LPG Direct Injection: An Alternative Fuel Solution to the Two-Stroke Emissions Problem

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The high exhaust emissions and poor fuel economy of carbureted, crank-case scavenged two stroke engines are well documented. Despite their high fuel consumption and poor emissions they remain a popular power source for small transports especially in developing countries due to their low cost and high power to weight ratio. To meet more ridged emissions requirements some two stroke manufacturers have switched from carburetors to Direct fuel Injection (DI). Direct Injection of the fuel greatly reduces the emissions and simultaneously improves the fuel economy of the vehicle. This technique is even being applied at a retrofit to existing vehicles. DI systems for small two-stroke engines are typically the air-assist Orbital system, which requires an air pump, separate fuel pump and both an air and fuel injector. In some countries another attractive alternative exists, namely the direct injection of a gaseous fuel such as Compressed Natural Gas (CNG) or Liquid Petroleum Gas (LPG). Many developing countries have an existing LPG infrastructure as it is often used for cooking. Also many developing countries have natural gas reserves, and it is a potentially renewable resource in the form of biogas. In some countries both LPG and CNG may have lower costs than the equivalent amount of gasoline.

A gaseous fuel DI system may represent significant cost savings over an air-assisted gasoline DI system as the gaseous system does not require a fuel pump, air pump or liquid fuel injector. Additionally the elimination of the air pump greatly improves the flexibility of the gaseous DI system for retrofit applications.

In this paper we compare gasoline and premixed LPG as an alternative fuel for retrofit application to a 110cc two-stroke motorcycle. Engine performance, emissions and fuel consumption are measured on a dynamometer at various speeds and throttle settings. Results are compared for the original gasoline engine, the same engine running on pre-mixed LPG and air. Initial results are also presented with the engine operating on Direct Injected LPG at idle.

Results indicate that HC emissions (on a mass basis) and fuel consumption similar for premixed LPG and gasoline. At idle HC and CO emissions are greatly reduced with Direct Injection of the LPG fuel.

Keywords: Liquid Petroleum Gas, Direct Injection, Two-Stroke Engines, Alternate Fuels, Vehicle Emissions

1. INTRODUCTION

Crankcase scavenged, carbureted engines are a popular choice for small transports due largely to their low purchase price. Because of fuel short-circuiting during the scavenging process however, they are plagued by very high hydrocarbon emissions and high fuel consumption. These problems are most noticeable in the urban centers of developing countries, where the high concentration of carbureted two-stroke vehicles may account for up to 70% of the total measured atmospheric hydrocarbons (Pundir 1994). While many countries are now banning the sales of new two-stroke vehicles, or implementing emissions requirements which effectively prohibit new carbureted two-stroke vehicles from entering the market, the issue of the pre-existing or legacy vehicles remains unsolved by these measures.

Generally the only thing an end-user can do to improve emissions of a two-stroke vehicle is to insure that the machine is properly maintained and tuned. However even a well tuned carbureted two-stroke engine will have hydrocarbon emissions approximately 10 times worse than an equivalent carbureted four-stroke engine, and the two stroke will consume approximately 35% more fuel over the same drive cycle as the four-stroke (Archer, 2001).

Relatively few options exist for the improvement of emissions and fuel consumption of an existing carbureted two-stroke vehicle (EPA 2000). We will consider three possible options here: addition of a catalyst, switching to an alternate fuel, and direct fuel injection. First, while oxidation catalysts can effectively reduce HC emissions via their conversion to CO_2 with excess exhaust oxygen, they do nothing to address the fuel consumption issue. Additionally it is rather difficult to effectively implement a catalyst on a two-stroke engine. Large amounts of exhaust gasses are re-circulated at idle resulting in frequent "missing" or failure of the spark to ignite the air-fuel mixture. At idle a carbureted two-stroke engine may only fire every other cycle (four-stroking), or every third or fourth cycle (eight-stroking as seen in figure 1). This results in relatively cool emissions gasses containing a lot of unburned hydrocarbons.

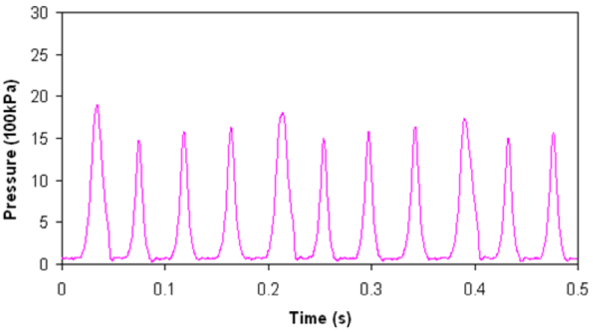


Fig. 1 Combustion pressure trace of a crank-case scavenged, carbureted two-stroke engine at idle. The broad pressure peak every fourth cycle represents cycles where the mixture was effectively ignited. The lower pressure peaks are "misfiring" cycles (from Gitano-Briggs 2001).

Under these conditions the catalyst tends to operate below its light-off temperature resulting in exceptionally poor conversion efficiencies. If the engine is subsequently operated at high power levels the exhaust temperatures will rise, lighting off the catalyst, and potentially destroying it with the heat created by the oxidation of the large amounts of hydrocarbons stored up on the surface of the catalyst. Finally as approximately 35% of the fuel which nominally escapes the combustion process unburned at high power levels the catalyst will have to oxidize very large amounts of hydrocarbons, resulting in damagingly high temperatures. For example, if we assume a 10kW engine operating near peak power is losing 35% of its fuel, and converting the rest with a 30% efficiency, then the total heat load in the exhaust catalyst unit will be (10kW / 0.3) * (0.35 / 0.65) or approximately 18kW from the short circuited fuel.

Retrofitting the engine to use an alternate fuel is an appealing possibility. Many countries have programs for converting popular two-stroke vehicles to either Liquid Petroleum Gas (LPG) or Compressed Natural Gas (CNG). Typically these conversions consist of simple gaseous carburetors, premixing the fuel with the incoming air. Engines thus converted to premixed LPG or CNG are popularly believed to be much cleaner than the original gasoline engine due to a significant reduction in visible smoke produced. They do nothing, however, to address the fuel short-circuiting, and subsequently suffer from both poor fuel efficiency and high hydrocarbon emissions, albeit in the form of a gaseous fuel which is less likely to condense and form visible "smoke". Both CNG and LPG do generally reduce the emissions of CO and many other of the higher molecular weight hazardous air pollutants as they are fully vaporized before combustion and consist exclusively of lower molecular weight hydrocarbons. Finally premixed gaseous fuel systems generally have lower power than premixed gasoline systems due the fuel displacement effect. Assuming stoichiometric combustion, and taking C_8H_{18} for gasoline 1.65% of the incoming mixture is fuel. Assuming it is completely vaporized this reduces the possible intake air charge by 1.65%. The process of vaporization cools the air however, typically resulting in an overall increase of volumetric efficiency of approximately 2%. Assuming stoichiometric combustion of LPG, and taking it to be pure propane, C_3H_8, approximately 4% of the incoming mixture is fuel, resulting in a 4% decrease of volumetric efficiency, or about 6% less than gasoline (Sinor 1992). The heating value of LPG is typically slightly higher than gasoline (46 Mj/kg for LPG compared to 44 MJ/kg for gasoline), potentially giving it a slight advantage in power. LPG also has a higher octane rating, allowing the usage of higher compression ratios. In a premixed retrofit application the compression ratio is typically left unchanged, resulting in a slightly lower power for the LPG version of the engines. For example, if the gasoline engine has a 2% increase in volumetric efficiency, and the LPG engine suffers a 4% decrease, with heating values of 44,000 MJ/kg and 46 MJ/kg for gasoline and LPG respectively, the LPG engine has 0.96 / 1.02 * 46,000 / 44,000 or 98% of the maximum power output of the gasoline version of the engine.

A solution that addresses both the emissions and fuel consumption problems of two-stroke engines is direct fuel injection (DI). With direct injection the fuel is injected directly into the combustion chamber instead of being premixed with the intake air. Air is still lost during the scavenging process, and while it may contain some two-stroke lubricating oil, it is no longer loaded with large amounts of fuel. A special injector is required, but fortunately the head of two-stroke engines affords much more flexibility for adding such equipment than does a four-stroke head. Ideally fuel is injected after the exhaust port is closed (EPC), eliminating fuel short-circuiting. A special injection system is required as the fuel must be completely atomized and mixed with the air in the relatively short period of time between and the spark. Assuming EPC at 110 degrees before top dead center (BTDC), and a spark timing of 20 degrees BTDC, an engine running at 6000 rpm with constant angular velocity will only have ((110 - 20) / 360) / (6000 / 60) or 2.5 ms for fuel vaporization and mixing. One way of accomplishing this demanding task is to use a high-pressure injection system similar to diesel injection. This requires a high pressure pump, and diesel type injector. There are variations on this technique using alternate methods for producing the high pressure necessary to pump the fuel, such as the so-called "water hammer" pump, and a solenoid boosted pressure injector, but we will not address these here as they are not commonly used. Another semi-direct injection technique which shows good promise is compression wave injection. However by far the most common direct injection retrofit technique for two stroke engines is the air assisted injection system. In air assisted DI systems a relatively low pressure port-type fuel injection is used, but the fuel is injected into a small cavity directly above the head. Pressurized air is supplied to the cavity from an air pump. A solenoid operated poppet valve connects the cavity to the combustion chamber. When the poppet valve, or blast valve as it is called, is opened, the compressed air in the mixing cavity carries the fuel into combustion chamber, finely vaporizing the fuel. This type of system has been successfully commercialized as a retrofit for gasoline two-stroke engines in India, and the Philippines (Lorenz 2005). One of the major disadvantages of this type of retrofit system is the necessity of adding an air pump which is typically linked directly to the crankshaft. In a pre-existing engine design this modification can be rather invasive, and therefore expensive.

Liquid fuel injectors are unsuitable for the injection of gaseous fuels as the orifice sizes are too small, significantly reducing the gaseous fuel's flow rate due to flow choking. However, the blast valve of the air assisted DI system is obviously suitable for the injection of gaseous fuels as it injects a significant amount of air in nominal operation. Furthermore many of the fuel injection components of the air assisted DI system can be used in a gaseous fuel DI system. In fact the use of a gaseous fuel eliminates the need for several components from the air assisted DI system, representing a major cost reduction. First as the gaseous fuel is already pressurized a fuel pump is no longer required. Additionally the liquid fuel injector may be eliminated, keeping only the gaseous injector. Finally, and perhaps most significantly for a retrofit application, the air pump is completely eliminated. Considering the maturity of the various components, we have therefore decided to focus on the development of a gaseous fuel direct injection system for retrofit applications on two-stroke engines using components from the air assisted fuel injection DI system. This paper presents the initial work involved in converting a carbureted, crankcase scavenged two-stroke engine to direct fuel injection, including base line measurements of the gasoline engine, and comparison to premixed LPG operation.

2. TEST VEHICLE

The vehicle chosen for this study is the Suzuki RG 110, a typical example of the two-stroke motorcycles popular in Malaysia. A list of the RG 110's specifications can be found in figure 2.

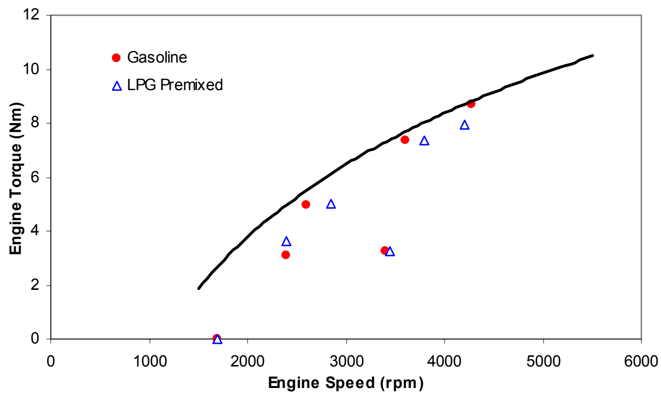
Number of cylinders	1
Swept volume	109.9cc
Bore	54.0mm
Stroke	48.0mm
Trapped Compression Ratio	6.7:1
Max engine speed	7500 rpm
Max power (at 7000 rpm)	10.2 kW

Fig. 2 Specifications of the Suzuki RG 110

The Suzuki RG 110 is a crankcase scavenged, carbureted two-stroke engine with separate two-stroke oil injection system. This has the added benefit of allowing us to use the same oil injection system throughout the conversion to direct fuel injection. Our vehicle was purchased second hand with approximately 20,000 km. The only modifications made to the gasoline engine prior to baseline testing were tuning of the carburetor and adjusting and cleaning the oil injection system. The test vehicle can be seen in figure 3.

Fig 3 The Suzuki RG 110 used in this study

For testing the output sprocket was connected to the shaft of an eddy current dynamometer. All tests were performed in 5th gear, having an overall gear reduction of 2.94 from the engine speed. All speed and torque numbers presented here have been converted by this ratio back to engine torque and speed numbers without accounting for gear loss which may comprise approximately 4%. In each engine configuration the engine was tested at a number of speed-torque combinations representative of normal Malaysian riding patterns. The test points may be seen in figure 4 along with the wide-open throttle (WOT) torque curve of the carbureted gasoline engine.



Fi.g 4 Test points and gasoline WOT torque curve (solid line)

At each test point the dynamometer was set to the appropriate speed and the engine's throttle was then opened until the corresponding torque was achieved. The throttle position was maintained for the duration of the test which was typically 2 minutes. At the beginning of the test the fuel tank (petrol or LPG) was weighed. Exhaust emissions were continually sampled during the test by an Autocheck (SPTC) Gas and Smoke emissions analyzer. At the end of each test the fuel tank was again weighed. This final fuel weight was subtracted from the initial fuel weight and divided by the time between readings to calculate the fuel consumption rate. The emissions were taken as an average over the duration of the test. The exhaust analyzer has both a gasoline and LPG setting for hydrocarbon calculations. When calculating hydrocarbon emissions in gasoline mode the part-per-million (ppm) concentrations of hydrocarbons are referenced to hexane, C_6H_{14}, and in LPG mode they are referenced to propane, C_3H_8.

For premixed and direct injection LPG the engine was modified for electronic fuel injection by the addition of a throttle position sensor (TPS), manifold air pressure sensor (MAP) and an engine speed pickup. A Megasquirt II electronic control unit (ECU) was connected to the sensors and the gaseous fuel injector, and tuned for maximum power operation at each of the test points (Reference #3). When testing LPG the fuel injection system was tuned for maximum power at each of the test points, resulting in rich operation for LPG. For premixed LPG the gas was introduced into the intake manifold up-stream of the throttle body. For DI operation the fuel was injected directly into the head very near the spark plug.

3. EXPERIMENTAL RESULTS

The idle fuel consumption can be seen in figure 5. Both of the pre-mixed cases have very similar fuel consumptions.

Technique	Fuel Consumption gm/sec
Premixed Gasoline	0.068
Premixed LPG	0.07

Fig. 5 Idle Fuel Consumption comparison

As the test points were all at moderate speeds compared to the engine redline rating of 7500 rpm, we would expect better efficiencies at higher power levels as a result of the reduced pumping losses with a more open throttle. Indeed this can be seen in the fuel consumption data of figure 6, where the higher load points tend to have lower break specific fuel consumption (BSFC) numbers. The BSFC appears slightly worse for premixed LPG, as a result of rich tuning.

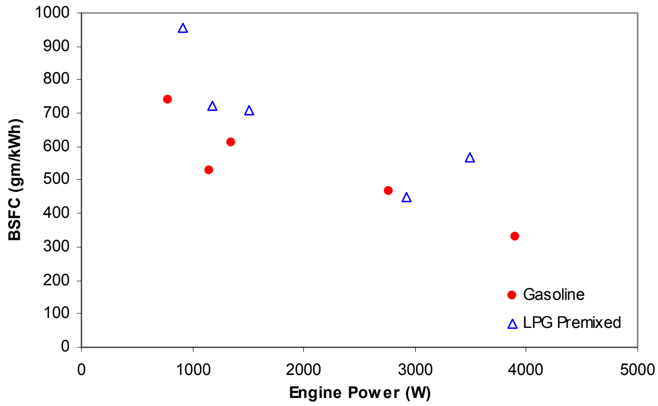


Fig. 6 Break Specific Fuel Consumption as a function of power for the premixed techniques

The hydrocarbons emissions are measured using a non-dispersive infrared sensor. The HC emissions are converted to parts-per-million assuming propane in LPG mode and hexane in gasoline mode. A comparison of raw HC emissions in ppm can therefore be deceptive as molecular weight of the emissions from the LPG fueled engine are much lighter than the gasoline fueled engine. The part-per-million HC emissions of the two-stroke engine burning LPG and gasoline as fuel can be seen in figure 7. The LPG fueled version has significantly higher HC emissions in terms of ppm.

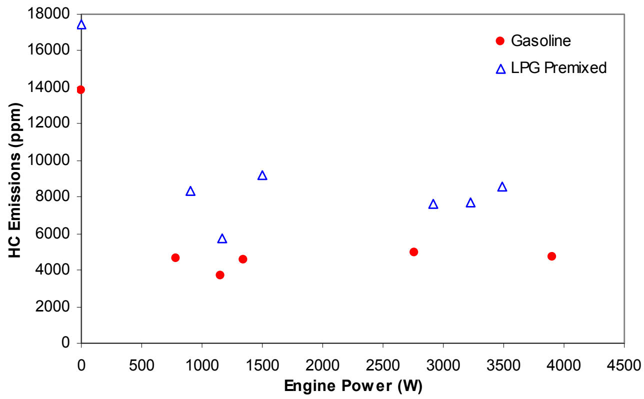


Fig. 7 Hydrocarbon exhaust emissions versus power of the premixed techniques

For an equivalent HC mass comparison we may convert the LPG HC emissions to hexane by multiplying by the molecular weight ratio (hexane/propane) of approximately 86/44 or 1.95. Converting the LPG emissions, figure 8, we see that burning LPG we generally have slightly better HC emissions than burning gasoline using a consistent molecular weight basis. The engine was operated on direct fuel injection using LPG at idle. While the HC emissions were significantly better with the DI technique, they are not as good as reported by others (Lorenz 2005). The major reason for this is that the local blend of LPG is approximately 70% butane, 25% propane and 5% other components. This blend gives relatively low vapor pressure, 3.5 bar at room temperature. With such low pressure the fuel must be injected very early, allowing some fuel short circuiting, and correspondingly higher HC emissions.

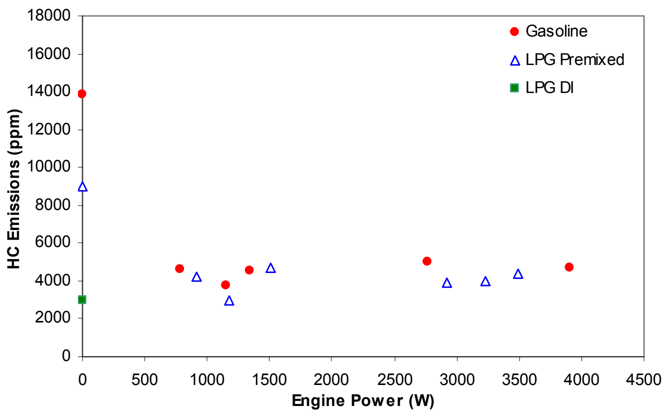


Fig. 8 Data of fig. 7 converting LPG emissions to hexane equivalent emissions

Any incompletely vaporized liquid fuel, either in the form of suspended droplets or wall films, may not completely burn, any wind up contributing to the HC emissions of liquid fueled engines. The bulk of the HC emission from both premixed versions, however, is due to short-circuiting of the fuel during scavenging. Gasoline HC emissions may cool and condense out resulting in visible smoke, while LPG HC emissions, predominantly propane or butane, remain as a gas. Figure 9 shows the difference in visible smoke emissions when running on gasoline (top) and LPG premixed (bottom).

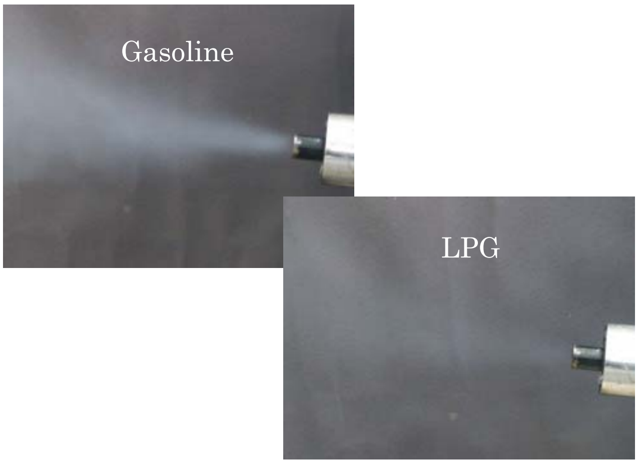


Fig 9. Exhaust smoke from premixed gasoline (top) and premixed LPG (bottom). The gasoline powered version has noticeable more exhaust smoke than the LPG fueled version.

LPG is generally thought to reduce the CO emissions. From our data shown in figure 10 the emissions of CO are higher when using premixed LPG. This is likely a result of rich tuning for maximum power which was performed at each test point when using LPG. When operating on directly injected LPG at idle, however, the CO emissions are greatly reduced.

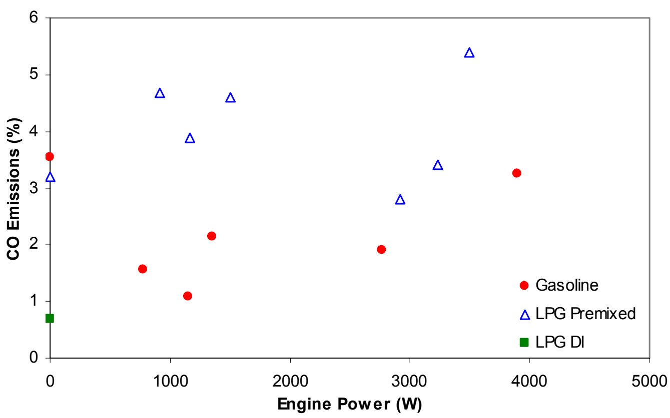


Fig. 10 CO emissions versus power for the various techniques versus engine power.

Higher propane content LPG will have higher vapor pressure, allowing later injection timings, and reduced fuel short-circuiting. Future work will focus on improving further emissions via the use of high propane content LPG.

4. CONCLUSION

(1) Premixed LPG gives higher HC emissions than gasoline on a ppm basis due the lighter molecular weight of the fuel, but similar HC emissions in terms of grams of hydrocarbons.

(2) Exhaust smoke emissions are significantly less for premixed LPG compared to gasoline

(3) Direct Injection of LPG significantly reduces both HC and CO emissions at idle.

(4) High propane content LPG is required for high vapor pressure and effective direct injection.

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