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Performance Characterization of a Direct Injection LPG Fuelled Two-Stroke Motorcycle Engine

TEOH Yew Heng, Dr. Horizon GITANO-BRIGGS, Khairil Faizi MUSTAFA
University science Malaysia

ABSTRACT

To quantify the fuel consumption and emissions improvements of Direct fuel Injection (DI), measurements were taken from a two-stroke motorcycle engine while operating in premixed, and direct injection mode burning propane. The part-load lean combustion limit was investigated for the DI technique in both premixed and DI modes. Results were compared to a one-dimensional Computational Fluid Dynamics (CFD) model of the engine. Results indicate that a highly stratified mixture can not be achieved due to poor mixing. The DI technique yields significant reductions in HC emissions, and significantly improved fuel economy, though not as good at Gasoline Direct Injection (GDI). Finally an economic analysis indicates that a gaseous fuel DI retrofit system represents a significant cost savings over a gasoline DI retrofit system.

INTRODUCTION

The most serious problems facing carbureted two-stroke spark-ignition engines today are high exhaust emission and poor fuel economy. Despite these major weaknesses large numbers of carbureted two-stroke vehicles continue to ply the roads especially in developing countries. In Asia as a whole it is estimated that there are over 50 million two-stroke cycle engines powering mostly small motorcycles [1]. Emissions regulations may restrict the sales of new two-stroke vehicles, however large numbers of "legacy" vehicles will remain in operation for decades to come. Gasoline Direct Injection (GDI) retrofit systems have been developed which successfully address the fuel consumption and emissions issues. Commercially available GDI retrofit kits require a crankshaft mounted air compressor, fuel pump as well as separate fuel and air injectors. Direct Injection of gaseous fuels holds the possibility of achieving improvements similar to GDI. Because gaseous fuels are typically pressurized, however, gaseous fuel DI systems do not require fuel pumps. Also the fuel can be injected without pressurized air assistance, eliminating the need for an air compressor and separate air and fuel injectors. This should result in a significantly lower cost for the gaseous fuel DI system. Additionally with only minor adjustments the system may be used with many different gaseous fuels, such as Liquid Petroleum Gas (LPG), Compressed Natural Gas (CNG), biogas, hydrogen and propane. In the present study Direct Injection of propane was applied as a retrofit to an existing carbureted two-stroke motorcycle engine to determine the potential emissions and fuel economy improvements.

METHODOLOGY

To determine the effectiveness of direct injection of gaseous fuels a new two-stroke motorcycle engine was acquired. The engine is a carbureted Modenas Dynamik, with 120 cc swept volume typical of the small displacement motorcycles popular in South East Asia. Figure 1 gives the specifications of the engine. For testing the engine was mounted to a 20 kW eddy current dynamometer and instrumented with a combustion chamber pressure probe and crank shaft position encoder. Fuel consumption was measured gravimetrically and emissions were measured with a commercial 5-gas analyzer using non-dispersive IR measurements for HC, CO and CO_2, and additional sensors for O_2 and NOx.

Bore	54	mm
Stroke	51.8	mm
Displacement	118	cc
Dead Volume	12.5	cc
Compression Ratio	7.3	
Crankcase CR	1.2	
EPO	81	deg
XPO	110	deg
Ignition	-17	deg
Max. Power	12.5 kW @ 9,000 rpm	
Max. Torque	14.1 N-m @ 8,000 rpm	

Figure 1 Specifications of the test engine.

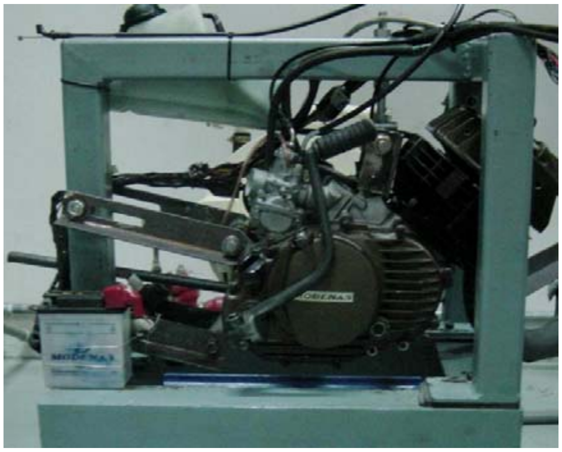

Figure 2 Engine coupled to dynamometer for testing

The test engine is shown in figure 2.The fuel injection controller is a Megasquirt II, operating in the “N-Alpha” mode using signals from the crankshaft position encoder, a Throttle Position Sensor (TPS), Manifold Air Pressure (MAP) sensor and a cylinder head temperature sensor. The fuel injection system, while being very flexible, does not offer the option of phasing the injection timing. To accomplish this, a circuit was designed which could read the 8 bit shaft encoder. The Most Significant Bit (MSB), which would normally be used to trigger the Electronic control Unit (ECU), was intercepted by the phasing circuit. Transmission of the trigger pulse to the ECU was delayed by a certain number of Least Significant Bit (LSB) transitions as shown in figure 3. The number of transitions was controlled by a knob on the front panel of the instrument. This could phase shift the crank shaft index signal by 0 to 360 degrees in 2.8 degree crank-angle increments.

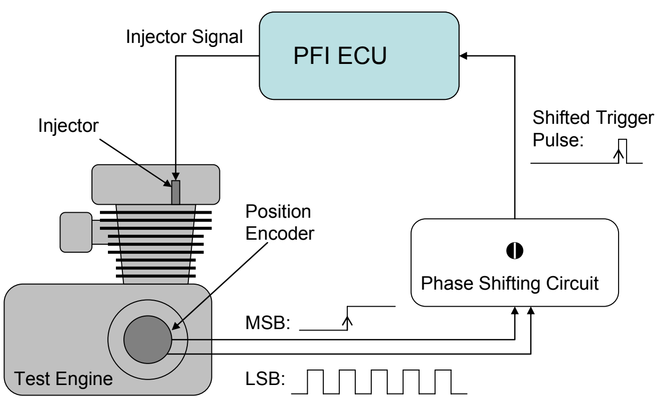

Figure 3 Schematic diagram of the injection shifting circuit

The fuel injector used in both premixed and direct injection modes is a Synerject Strata 1 air injector. The injector was mounted in the head near the spark plug. The head shape was is an “inverted bathtub” shape similar to that used in other two-stroke DI systems. As this injector is intended to receive lubricating oil from an up-stream air compressor, small amounts of two-stroke oil were added intermittently to insure proper functioning with minimal wear.

The engine operated satisfactorily on a variety of gaseous fuels, however for this study we used commercially available propane at a pressure of 9 bar. For our studies we regulated the pressure to 6.5 bar.

Initially the fuel injection system was tuned with a throttle body mounted injector. Once the basic fueling map was generated the injector was moved to the head, and the system tuning was refined. For a major portion of the work the injection timings were very early, with Start Of Injection (SOI) occurring before BDC. For relatively short injection timings this results in pre-mixed like behavior, as much of the fuel charge is short circuited out the open exhaust port with the scavenging air. As injection timing was retarded, less fuel was lost out the exhaust port, increasing the fuel retention in the cylinder. This, of course, results in less HC emissions and lower fuel consumption as the system shifts into DI mode. As SOI is retarded past Exhaust Port Close (EPC) combustion chamber pressure rises, reducing the fuel flow rate resulting in an increase in the Air Fuel Ratio (AFR). A key challenge with later injection timings is getting sufficient mixing between the air and fuel.

The most important operational area for fuel consumption is at intermediate speed and torques used while cruising. A model of the motorcycle was developed to determine the power required for a given cruising speed. The model includes rolling resistance and air drag. The vehicle and rider was modeled at a mass of 187kg, with a frontal area of 0.6m^2 and an aerodynamic coefficient of drag of 0.7. The tires and road were modeled with a coefficient of rolling resistance of 0.020 and air density was taken to be 1.18 kg/m^3. Calculating the power required as a function of steady-state rolling speed we get the curve of figure 4.

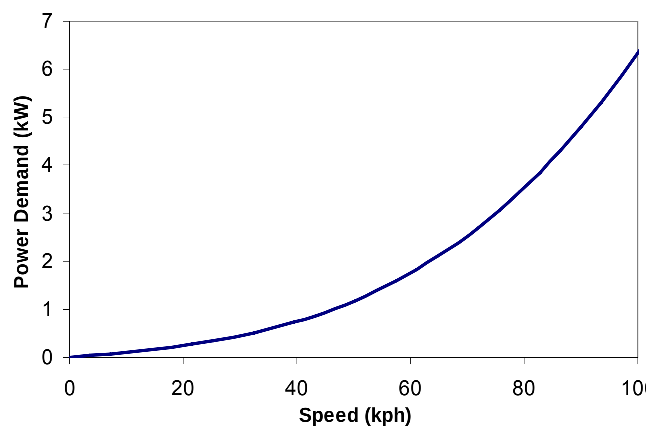

Figure 4 Engine power required as a function of steady state cruising speed

Motorcycle drive cycle measurements were measured using a variety of techniques including GPS and an innovative torque measuring rear sprocket shown in figure 5. As torque is applied to the sprocket, the outer ring with the chain drive teeth rotates with respect to the inner ring connected to the driven wheel. The rotation is resisted by 4 stiff springs, resulting in a relative displacement proportional to torque.

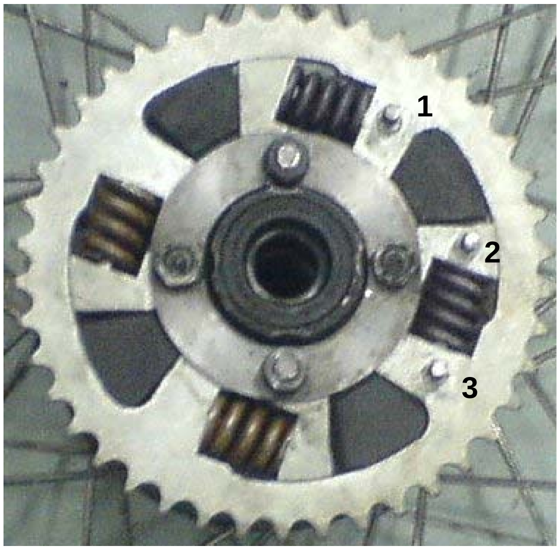

Figure 5 Special torque measuring sprocket used in the drive cycles studies

The outer ring is instrumented with two metal posts (labeled 1 and 3 in the figure). At the same radius the inner ring is instrumented with a similar metal post (labeled 2) half way between the outer ring posts. These posts are read by a variable reluctance speed pickup as the wheel rotates as giving a signal as shown in figure 6.

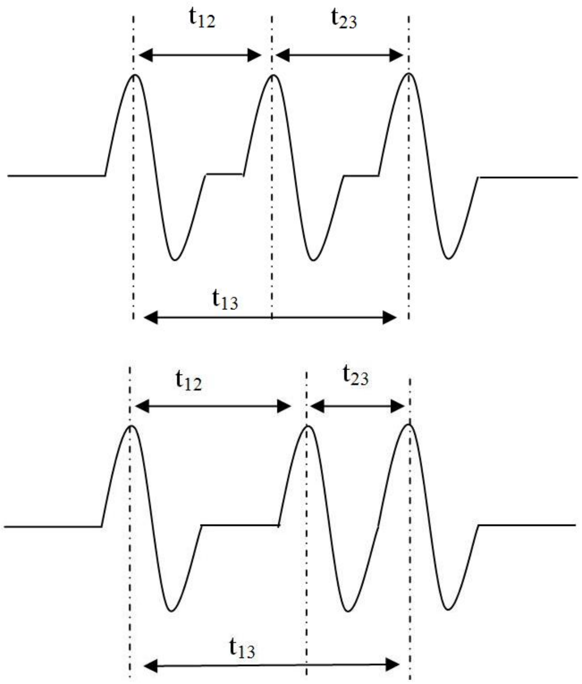

Figure 6 Signals from the torque measuring sprocket unloaded (top) and loaded (bottom)

The rotational velocity of the wheel of the wheel is inversely proportional to the time between the signal from the 1st and 3rd posts (which are both on the outer ring and have a constant angular spacing) as perceived by the speed pickup. The period between the 1st and 2nd post as a fraction of the period between the 1st and 3rd post is then a measure of the torque on the sprocket. At no load the period from the 1st to the 2nd signal is approximately 50% of the 1st to 3rd signal period.

Using a laptop based data logger, vehicle speed was acquired in this manner for a variety of typical driving cycles on the streets of Malaysia. One such run is shown in figure 7. The dominant cruising speed in mixed urban-rural motorcycle trips was determined to be approximately 55 kilometers per hour. According to the model, figure 4, this gives an engine power of around 1.3 kW. We therefore chose to perform our testing at a power of 1.3 kW and a mid-range engine speed of 3200 rpm and.

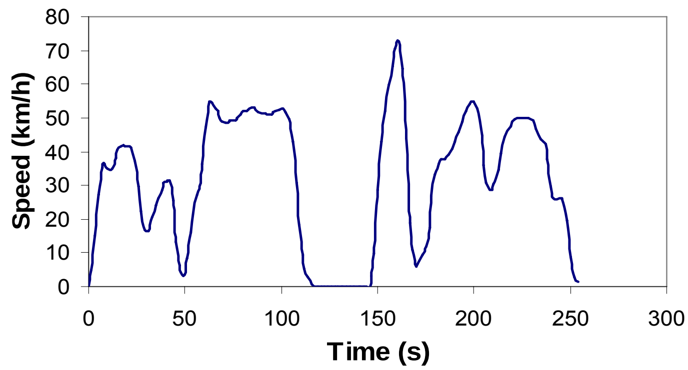

Figure 7 Speed history of a typical Malaysian motorcycle commute

RESULTS

The engine was started and warmed up with SOI at approximately BDC. Once the desired test speed and torque were achieved, SOI was then scanned both earlier and later while measuring torque and emissions. During the scan the injection duration was held fixed at 36° crank angle, which gave a slightly rich mixture, and the maximum break torque. For comparison purposes the injection duration was then shortened to 27°, increasing the AFR, and SOI was then scanned again. The resulting data for torque and HC emissions (hexane) are shown in figure 8 for injection durations of 36° (rich) and 27° (slightly lean).

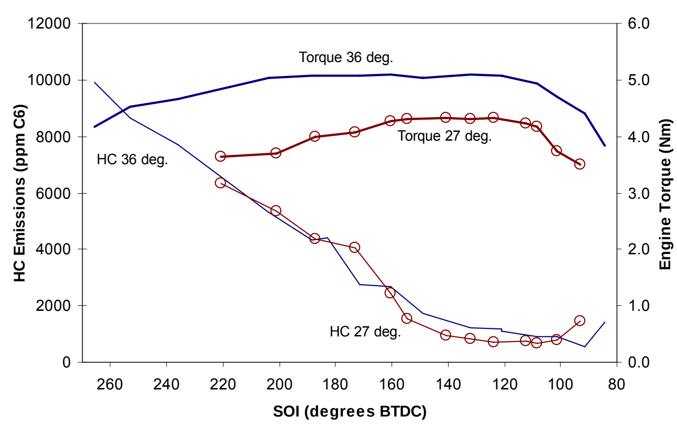

Figure 8 Torque (bold lines) and HC (fine lines) emissions as a function of Start of Injection timing for various injection durations.

At extremely early SOI (before BDC) much of the fuel is lost out the exhaust port, resulting in poor trapping, lower torque and high HC emissions. As SOI is moved later HC emissions decrease approximately linearly. This represents improving trapping of the fuel, which results in an increase in torque. When the fuel is injected near BDC (180 deg. before TDC) the engine is essentially operating in premixed mode.

As SOI is retarded further, HC emissions continue to drop, and torque is maximized at approximately 120 degrees before TDC. This maximum is more noticeable in the lean torque curve. At this point HC emissions have been reduced to 1100 ppm for the longer injection and 720 ppm for the leaner run. After this point torque drops off approximately linearly with SOI retard and HC emissions continue to decrease. Both injection durations attain a minimum HC emission of approximately 600 ppm at SOI between 90 and 110 degrees before TDC. At this point Torque has already decreased from its maximum by approximately 10%. Later SOI gives a sharp decrease in torque and increasing HC emissions as the engine begins to misfire intermittently.

At a given SOI the HC emissions are approximately the same for the two injection durations. This is because most of the fuel injected early on is short circuited out the exhaust port. Fuel injected later has a greater chance of being retained in the cylinder and contributing to torque. Shortening the injection duration moves EOI earlier resulting in less fuel being injected later and lower torque. However the amount of short circuited fuel remains approximately the same as it is injected, and lost to the exhaust port, well before EPC.

DISCUSSION

The basic shape of the torque curve and HC emissions compares well with that predicted by a Wave one-dimensional fluid dynamics model of the engine [2] shown in figure 9.

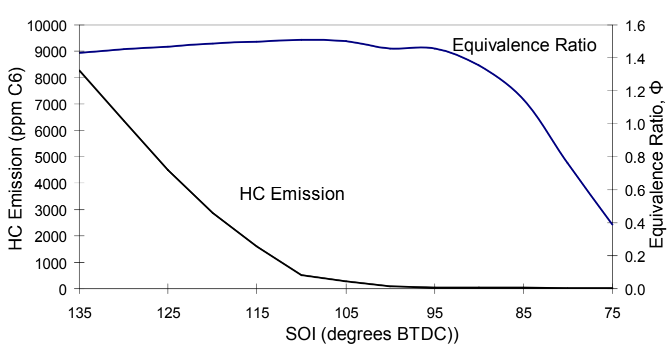

Figure 9 Trapped equivalence ratio and HC emissions versus SOI as predicted in a Wave computer simulation

Assuming reasonable mixing and combustion efficiencies, the torque produced by the engine should be proportional to the equivalence ratio in the region near Φ=1. While there were some differences in the specifics, the model (which assumed perfect mixing, ignition and combustion efficiency) does qualitatively predict the shape of the torque and HC emissions curves measured, confirming that the decrease in HC emissions and increase in torque are a direct result of improved fuel trapping in the cylinder.

From our data the HC emissions are minimized with SOI timings significantly earlier then EPC (EPC is at 81 degrees before TDC). Several factors come to play in this. First, the injector does not open instantaneously, it requires approximately 0.5 ms to fully open. During most of this time (the injector opening time) the flow area is relatively small. Similarly the injector does not close exactly when the solenoid voltage is removed; it is a spring biased system and requires a similar amount of time to close. These two factors move the effective injection timing later by approximately 10 degrees at 3200 rpm. Another factor is the gas transport delay. Even with injection timings before EPC fuel will not necessarily have time to traverse the distance from the injector to the exhaust port prior to closing of the exhaust port.

Given the 10 degree latency in the injector and injection durations of approximately 30 degrees, to simulate pre-mixed operation SOI should be set to 180 + 10 + 30/2 or about 205 degrees before TDC. At this point we recorded HC emissions of approximately 5000 ppm, similar to a previous study using a throttle body mounted injection system [3]. This again confirms that although very little visible smoke is present with premixed gaseous fuelled two-stroke engines, the hydrocarbon emissions are still quite high.

At SOI near 200 degrees BTDC the leaner injection timing gives 3.7Nm of torque, while at the optimum timing torque is 4.3Nm, or about 17% higher. This increase in power could be translated into a fuel savings of the same amount, i.e. instead of running at the higher power the throttle and fuel flow could be reduced to achieve the same power at the optimized injection timing. This would result in an approximately 17% reduction in fuel consumption.

With Gasoline Direct Injection systems, later injection timings are used for extra lean stratified charge combustion. Repeated experiments were performed using leaner fueling rates and relatively late injection, however we were unable to achieve fully stratified combustion with our current setup as later injection timings resulted in misfiring. Several studies have shown that gaseous fuels do not mix as readily as liquid fuels when injected into air due to their lower density [4]. This reduces their penetration and results in less homogeneous mixtures and combustion instability [5]. Operating at leaner air/fuel ratios will further aggravate this situation by increasing the combustion duration and reducing ignition stability [6].

In order to further reduce hydrocarbon emissions and operate leaner in-cylinder mixing of the propane and air will have to be enhanced. Three dimensional Computational Fluid Dynamics (CFD) modeling is currently underway to shed light on this situation with the hope of developing a combustion chamber and injector orientation more conducive to gaseous fuel-air mixing.

ECONOMIC ANALYSIS

The major components of a typical small motorcycle gasoline Electronic Fuel Injection system, a Gasoline Direct Injection retrofit and a gaseous fuel DI system are listed in figure 10.

	EFI	Gasoline DI	Gasious DI
Fuel Pump	X	X	
Regulator	X	X	X
Air Pump		X	
ECU	X	X	X
Injectors:			
Gasoline	X	X	
Air/Gas		X	X
Sensors:			
TPS	X	X	X
Crank Pickup	X	X	X
MAP	X	X	X
Temp	X	X	X
Pressure Tank			X
Est. Cost (US$)	125	250	150

Figure 10 Major components of various fuel injection configurations, and estimated costs

In large volumes (10,000 of units per month) typical EFI systems add a little more than 100 US$ to the price of a motorcycle. GDI retrofits are available for approximately 250 US$, however it is likely that this could come down to below 200 US$ in high volumes. Apart from the pressurized fuel tank the gaseous DI system should actually be less expensive than the gasoline EFI system. The gaseous fuel DI system requires a slightly more expensive direct injector, but does not require a fuel pump, separate fuel injector or an air pump. It is unlikely that the gaseous fuel DI system will be produced in greater numbers than the gasoline EFI system in the near future, so we estimate that the gaseous fuel DI system will cost slightly more than a gasoline EFI system, but significantly less than a GDI retrofit system.

According to our measurements Direct Injection of propane should reduce fuel consumption by approximately 17%. This is significantly less than GDI systems which may reduce fuel consumption by 30 to 40% over the whole operating range. Given similar fuel costs the gaseous fuel DI system should save slightly less than the GDI retrofit system, however in many countries gaseous fuels, such as CNG, cost much less than the equivalent amount of gasoline, making the gaseous fuel DI system potentially even more attractive than a GDI retrofit system.

Finally GDI retrofits have been reported to reduce HC emissions by approximately 88% [1]. Our best HC emissions numbers were 600ppm. Assuming premixed HC emissions of 5000ppm this represents an 88% reduction in HC emissions for the gaseous DI system as well.

CONCLUSIONS

- Direct Injection of propane can reduce HC emissions to approximately 600 ppm as compared with approximately 5000 ppm for premixed propane combustion
- Direct Injection can similarly reduce fuel consumption by 17% compared with premixed combustion
- Direct injection beginning before BDC effectively mimics pre-mixed behavior
- Optimum Start Of Injection is approximately 120 degrees BTDC for torque.
- Hydrocarbon emissions are minimized by SOI around 90 degrees BTDC. Later SOI timings result in degraded ignition stability. Earlier SOI timings results in greater amounts of short circuiting of fuel.
- Fully stratified combustion was not achieved due to misfiring.
- A gaseous fuel DI retrofit system should be economically competitive with existing GDI retrofits.

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CONTACT

For more information on this or related studies, please contact: Dr. Horizon Gitano-Briggs via email at: HorizonUSM@yahoo.com