

A Simulation Study on Performance Characteristics of a Gasoline Fuel Injection Motorcycle Engine with Separated Addition of H₂, CH₄, and LPG

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Abstract: Hydrogen, methane, and liquefied petroleum gas (LPG) have the fast flame velocities that could affect to the combustion process of gasoline-gas fuelled engines. This leads to reduce in time loss (due to finite combustion process) and to gain higher combustion efficiency and brake thermal efficiency. Thus, the primary objectives of this paper are to study the effects of addition of H₂, CH₄, and LPG on performance of a gasoline fuel injection motorcycle engine. The combustion process characteristics is predicted by Vibe 2-Zone model and the Woschni 1987 model is used to measure heat transfer process in AVL Boost software. The simulation process bases on characterizing of the motorcycle engine specification and simulating with using gasoline to demonstrate durability of simulation model. Then, the motorcycle model is evaluated when it use gasoline-gas fuel with different gas concentrations (from about 5%, 10%, 15%, 20% and 25%). The obtained results are used to compare each other's engine characteristics for each type of added gases.

Keywords: Motorcycle Engine, Gasoline Fuel Injection, Gasoline-Gas Fuel, Vibe 2-Zone Model, Simulation

Introduction:

One of the contemporary issues is carbon dioxide emission and air pollution. The study present a solution for this global problem is application gas fuel in the engine as dual fuel. In addition, motorcycle productions and applications in Vietnam are continually climbed and cause lots of effects on environment. Many researches and statistics is showed as alert for this.

In recent years, decreasing CO₂ emissions is the main goal in new vehicle development and technology research. Following 2015 SMMT's report [1], average new car CO₂ emissions, which is 181g/km, fall off 2.9% from 2013 to 2014. Moreover, registrations of alternatively-fuelled vehicles (AFVs) rose by 58.1% in 2014 to 51,739 units to account for a record 2.1% share of the market. A handful of hydrogen vehicles were also registered ahead of full commercial sales beginning in 2015. However, it has some disadvantages with using alternative fuel on motorcycle in ASEAN countries, especially in Vietnam, which is the second big motorcycle market in the world.

For the report of Vietnam government in 2012 [2], vehicle emission is one of air pollution resources which is Urban Issue in Vietnam. It led to the target of them that is raising use efficiency and gaining more effect energy economization and conversion. On the other hand, in Ho Chi Minh City, amount of motorcycles has dramatically increased to over 7.4 million by 2015, which accounts for 1/6 total number of motorcycles in the whole country. This is a problem which pressurize into transportation infrastructure and environment protection. And addition of gas fuel which is one of alternative and renewable energies on hydrocarbon fuel can be seen the solution for this contemporary problem.

To make good profit on gaining fuel economy, decreasing CO₂ and toxic exhaust gas emissions is application gasoline-gas, fuelled to motorcycle

engine. With fast flame propagation, hydrogen, methane and LPG are potential solution for motorcycle engines that don't need to change a lot about engine component. Experimental studies of Changwei Ji and et al. [3,4] about applying hydrogen on motorcycle by addition hydrogen injector in intake manifold. Engine brake thermal efficiency on the average increases from 25.12% to 28.35% when hydrogen addition level is 3% by compared with that of the original engine. The peak in-cylinder temperature raises and advances, while the temperature for post combustion duration goes down with the addition of hydrogen. HC and CO₂ decrease and CO and NO_x increase when the hydrogen volume fraction is higher. In independent work, Rusu and et al. [5] is measured combustion characteristics of SI engine using gasoline and addition hydrogen in case of lambda is 1 and 1.25 and at 3000 rpm. In the contrary, the result of simulation research of Kamil and Rahman [6], brake torque, brake power and maximum cylinder pressure is continuously decreased when hydrogen mass fraction is increased to 20%. In addition, the studies of Ceviz and et al. [7] and Aritra Chatterjee and et al. [8] indicate that the hydrogen-gasoline engine have higher brake thermal efficiency when it operate at lean condition mixture, especially lambda is 1.

In short, a simulation study on effect of hydrogen, methane and LPG volume fraction in mixture with gasoline is necessary for application to motorcycle. This study will create a lot of data for experimental study in the future or will use to compare in same conditions.

A port fuel injection, four-stroke, single cylinder motorcycle engine is used in this simulation. The engine specifications are showed in Table 1. Besides manufacturer parameters, the others are measured in actual engine.

Engine specifications and model parameters:

Table 1: Engine Specifications

Engine model	4-stroke
Bore (mm)	53.5
Stroke (mm)	55.5
Compression ratio	9.6
Piston pin offset (mm)	0
Displacement (cc)	124.76
Piston surface area (mm ²)	2248
Cylinder surface area (mm ²)	2472
Number of strokes	4
Max torque (N.m)	9.2 N.m/6500 rpm
Max power (kW)	6.9 kW/8000 rpm
Number of valve	2
Coolant type	Water
Fuel system type	Port Fuel Injection

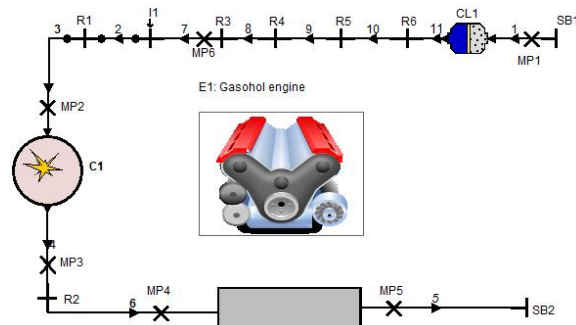
Simulation model:


Figure 1: Manifold Injection Motorcycle Engine Model

Figure 1 shows the model of manifold injection motorcycle engine in AVL Boost software. C1 represent the cylinder, MP1 to MP5 represent the measuring point and CL1 and PL1 represent the air cleaner and the plenum. R1 to R6 symbolize of restriction and P1 to P11 symbolize of Pipe. Finally, SB1 and SB2 are boundary conditions of the model.

Materials:

Table 2 describes properties of hydrogen, gasoline and LPG.

Table 2: Gas Properties

	Gasoline	H ₂	CH ₄	LPG
Molecular Mass (kg/kmol)	111.2	2.015	16.04	51
Lower Heating Value (kJ/kg)	43500	12000	50044	45996.5
Liquid Density	0.7 – 0.75	0.0708	0.421	0.600
A/F stoichiometric ratio	14.5	34.3	17.2	15.56
Research Octane number	91-99	>120	140	100.4 [8]
Flame velocity (m/s)	0.37-0.43	1.85	0.38	0.432

Combustion model:

Vibe 2-Zone Model is used to evaluate combustion process of this port manifold injection SI engine based in AVL BOOST. This model divided the combustion chamber into unburned and burned gas regions [9]. On the other hand, the assumption that burned and unburned charges have the same temperature is dropped. Instead the first law of thermodynamics is applied to burned charge and unburned charge respectively [10].

$$\frac{dm_b u_b}{d\alpha} = -p_c \frac{dV_b}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_{wb}}{d\alpha} + h_u \frac{dm_b}{d\alpha} - h_{BB,b} \frac{dm_{BB,b}}{d\alpha} \quad (1)$$

$$\frac{dm_u u_u}{d\alpha} = -p_c \frac{dV_u}{d\alpha} - \sum \frac{dQ_{wu}}{d\alpha} + h_u \frac{dm_B}{d\alpha} - h_{BB,u} \frac{dm_{BB,u}}{d\alpha} \quad (2)$$

Where,

Index b burned zone,

Index u unburned zone,

$dm_{b,u} u$ Represent for the change of the internal energy in cylinder,

$p_c \frac{dV}{d\alpha}$ Represent for the piston work,

$\frac{dQ_F}{d\alpha}$ Represent for the heat input,

$\frac{dQ_w}{d\alpha}$ Represent for the wall losses,

$\frac{dm_B}{d\alpha}$ Represent for the enthalpy flow from the unburned to the burned zone due to the conversion of a fresh charge to combustion products. Heat flux between is neglected.

$h_{BB} \frac{dm_{BB}}{d\alpha}$ Represent for the enthalpy due to blow-by,

In addition, the sum of the volume changes must be equal to the cylinder volume change and the sum of the zone volumes must be equal to the cylinder volume as follow:

$$\frac{dV_b}{d\alpha} + \frac{dV_u}{d\alpha} = \frac{dV}{d\alpha} \quad (3)$$

$$V_b + V_u = V \quad (4)$$

The amount of mixture burned at each time step is obtained from the Vibe function specified by the user. For all other terms, like wall heat losses etc., models similar to the single zone models with an appropriate distribution on the two zones are used.

Emission model:

The amount of CO is estimated following 2 reactions based on Onorati et al. [10, 12] in Table 3.

Table 3: Two Reactions to predict CO production

Stoichiometry	Rate
CO+OH = CO ₂ +H	$r_1 = 6.76 \cdot 10^{10} \cdot e^{\left(\frac{T}{1102.0}\right)} \cdot c_{CO} \cdot c_{OH}$
CO ₂ +O = CO+O ₂	$r_2 = 2.51 \cdot 10^{12} \cdot e^{\left(\frac{-24055.0}{T}\right)} \cdot c_{CO} \cdot c_{O_2}$

The rate of CO production is calculated as:

$$r_{CO} = C_{const} \cdot (1 - \alpha) \cdot (r_1 + r_2) \quad (4)$$

$$\alpha = \frac{r_{CO,act}}{r_{CO,equi}} \quad (5)$$

With

c is Molar Concentration in Equilibrium

r_i is reactions rates based on the model

The NO_x formation model in AVL Boost based on Pattas and Häfner [11] which incorporates the well-known Zeldovich mechanism [10]. The rate of NO_x production is following 6 reactions in Table 4.

Table 4: Reactions in NO_x formation mechanism

	Stoichiometry	Rate	k ₀ [cm ³ ,mol,s]	a [-]	T _a [K]
R1	N ₂ + O = NO + N	$r_1 = k_1 \cdot c_{N_2} \cdot c_O$	4.93E13	0.0472	38048.01
R2	O ₂ + N = NO + O	$r_2 = k_2 \cdot c_{O_2} \cdot c_N$	1.48E08	1.5	2859.01
R3	N + OH = NO + H	$r_3 = k_3 \cdot c_{OH} \cdot c_N$	4.22E13	0.0	0.0
R4	N ₂ O + O = NO + NO	$r_4 = k_4 \cdot c_{N_2O} \cdot c_O$	4.58E13	0.0	12130.6
R5	O ₂ + N ₂ = N ₂ O + O	$r_5 = k_5 \cdot c_{O_2} \cdot c_{N_2}$	2.25E10	0.825	50569.7
R6	OH + N ₂ = N ₂ O + H	$r_6 = k_2 \cdot c_{OH} \cdot c_{N_2}$	9.14E07	1.148	36190.66

The NO's concentration is estimated following equation:

$$r_{NO} = C_{PPM} \cdot C_{KM} \cdot (2.0) \cdot (1 - \alpha^2) \left(\frac{r_1}{1 + \alpha AK_2} + \frac{r_4}{1 + AK_4} \right) \quad (4)$$

$$\alpha = \frac{r_{NO,act}}{r_{NO,equi}} \cdot \frac{1}{C_{KM}} \quad (5)$$

$$AK_2 = \frac{r_1}{r_2 + r_3} \quad (6)$$

$$AK_4 = \frac{r_4}{r_5 + r_6} \quad (7)$$

Where:

C_{PPM} : Post Processing Multiplier

C_{KM} : Kinetic Multiplier

c : Molar Mass

r_i : reactions rates of Zeldovich mechanism

The following major sources of unburned hydrocarbons can be estimated in SI engines [10, 13]:

1. A fraction of the charge enters the crevice volumes and is not burned since the flame quenches at the entrance.
2. Fuel vapour is absorbed into the oil layer and deposits on the cylinder wall during the intake and compression. The following desorption takes place when the cylinder pressure decreases during expansion stroke and complete combustion cannot take place any more.

3. Quench layers on the combustion chamber wall which are left as the flame extinguishes prior to reaching the wall.
4. Occasional partial burning or complete misfire occurring when combustion quality is poor.
5. Direct flow of fuel vapour into the exhaust system during valve overlap in PFI engines.

Hydrocarbon emission level described by equation below [10] depends strongly on mass in the crevices through two first mechanisms:

$$m_{crevice} = \frac{pV_{crevice} \cdot M}{RT_{piston}} \quad (8)$$

Where:

Crevice denotes mass of unburned charge in the crevice (kg)

p denotes cylinder pressure (Pa)

C_{crevice} denotes total crevice volume (m³)

M denotes unburned molecular weight (kg/kmol)

R denotes gas constant (J/(kmol.K))

T_{piston} denotes piston temperature (K)

Finally, total hydrocarbons released into the exhaust gases undergo a complex mechanism of oxidation due to the existing high temperature in the chamber. For predict the oxidation speed of the amount of HC, Arrhenius equation is used [10, 14]:

$$\frac{dC_{HC}}{dt} = -F_{Ox} \cdot f \cdot A_{Ox} \cdot \exp\left(\frac{-T_{Ox}}{T}\right) C_{O_2} C_{HC} \quad (9)$$

Where:

C stand for concentration of HC and O₂ (kmole/m³)

F_{Ox} stand for HC post-oxidation multiplier

f stands for HC post-oxidation scaling factor

T_{Ox} stand for activation temperature, default=18790.0 (K)

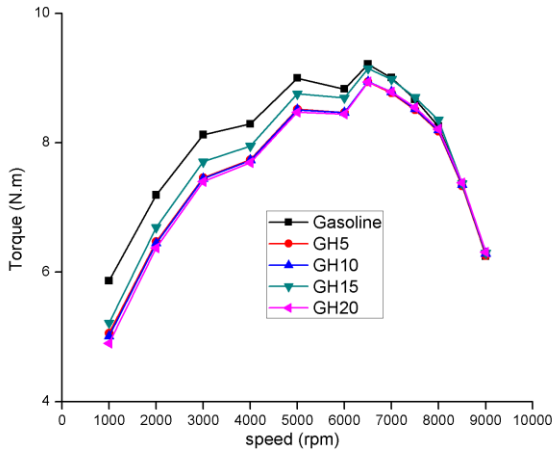
A_{Ox} stand for frequency factor, default=7.7E12 (m³/kmole/s)

Simulation conditions:

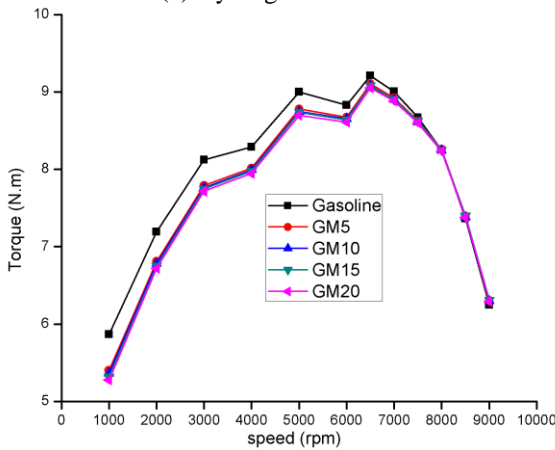
The simulation process is operating with gasoline, gasoline-hydrogen, gasoline-methane and gasoline-LPG as fuel. In each case, the concentration of addition gas is changed from 5% to 20% (step is 5%) and lambda value is set as 1 for all. Engine angle speed is changed from 1000 rpm to 9000 rpm. Especially, 6500 rpm, 7500 rpm and 8500 rpm is added to clearly measure parameters from the speed at maximum torque to the speed at maximum power. The start of combustion angle is set at 10 BTDC.

Results and discussion:

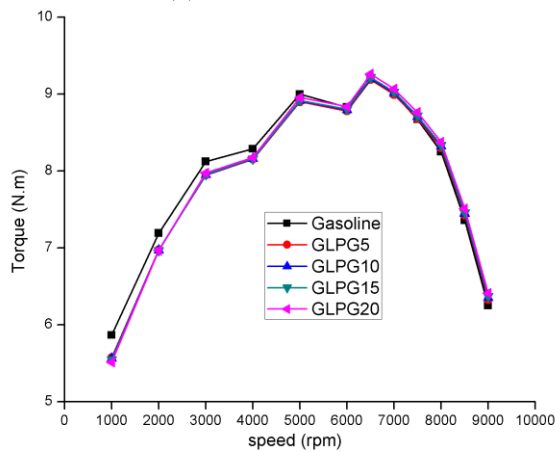
Performance characteristics



(a) Hydrogen-Gasoline



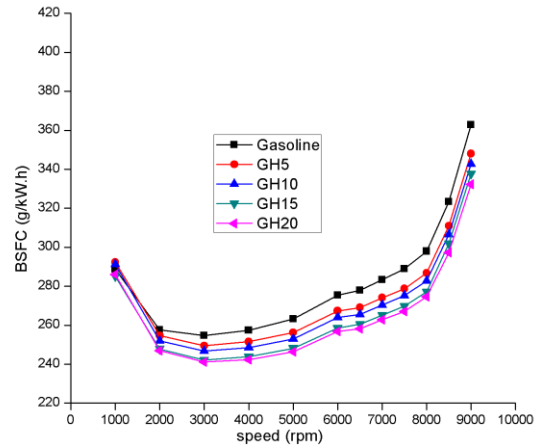
(b) Methane-Gasoline



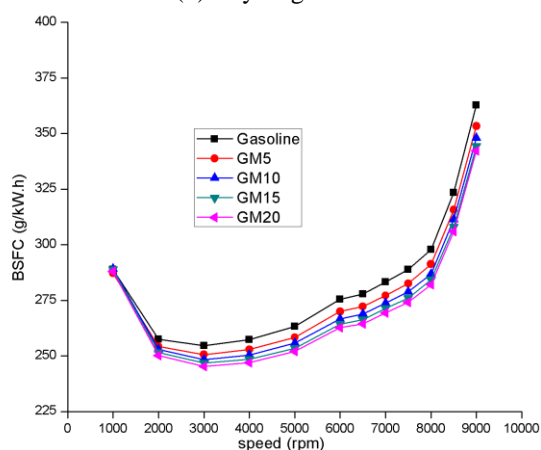
(c) LPG-Gasoline

Figure 2: Brake Torque variations with engine speed

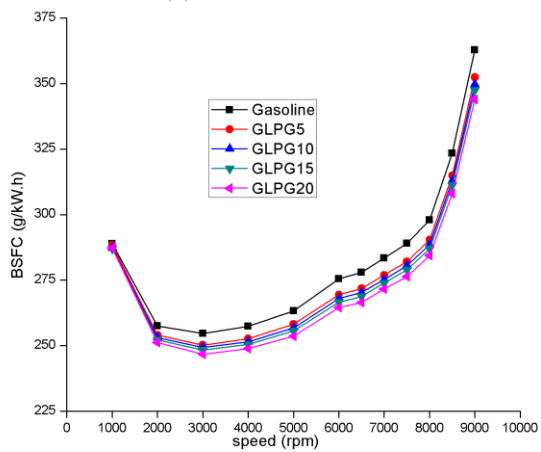
According to Figure 2(a), it shows that the change of brake torque when hydrogen concentration is form 5 to 20%. Brake torque of the engine goes down while increasing hydrogen. Despite lower heating value increasing, the energy density of fuel mixture decrease and this led to decrease brake torque. At 6500 rpm, it slightly drops 3% from 9.21 N.m to 8.92 N.m. In addition, with shortly combustion duration, the engine using higher addition hydrogen have more time to heat transfer to coolant and the brake torque go down.



(b) Hydrogen-Gasoline



(b) Methane-Gasoline



(c) LPG-Gasoline

Figure 3: Brake specific fuel consumption variations with engine speed

In the comparison with hydrogen, gasoline-methane fuel and gasoline-LPG fuel have brake torque nearly that of original in Figure 2(b) and 2(c). In 20% methane blended, at 6500 rpm, there is a reduction of 0.16 kW in the brake torque and also 20% LPG blended which have approximately equalled to brake torque of the original. This is due to the decrease of energy density of the mixture when enriches methane and LPG is slower than that of hydrogen. For all the blends, the maximum torque is at 6500 rpm, because of the volumetric efficiency rising when the engine

speed goes up from 1000 rpm to 6500 rpm. However, when it reach over 6500 rpm, the time for cycle go down and led to low quality for gas exchange and the torque is dropped.

The line chart in Figure 3(a) indicates effects of each hydrogen volume fraction ratio on brake specific fuel consumption. With the fast flame velocity and highest lower heating value, hydrogen addition which can burn fuel completely, bring us the high economical of fuel. In case that hydrogen concentration is increased, the BSFC is continually reduced and touch the minimum at 20% hydrogen in volume in which BSFC is significantly fall down 5.2%. This could be due to the lower heating value and the flame velocity of hydrogen addition fuel is higher that of gasoline. Hence, if mix more hydrogen in fuel, BSFC will decrease.

Moreover, not only hydrogen but also methane and LPG bring up high benefit with using this mixed with gasoline in Figure 3(b) and 3(c). As can be seen, at 3000 rpm, BSFC is marked fallen 3.6% for 20% methane-gasoline blends and 3.11% for 20% LPG-gasoline blends. As the flame speed and lower heating value of mixture grew, the fuel is burned more than gasoline and made good conversion of fuel. In the overview, BSFC is lowest at 3000 rpm. This is cause by cycle time at 3000 rpm is appropriate for fuel burning process, although the volumetric efficiency of this is lower than 6500 rpm.

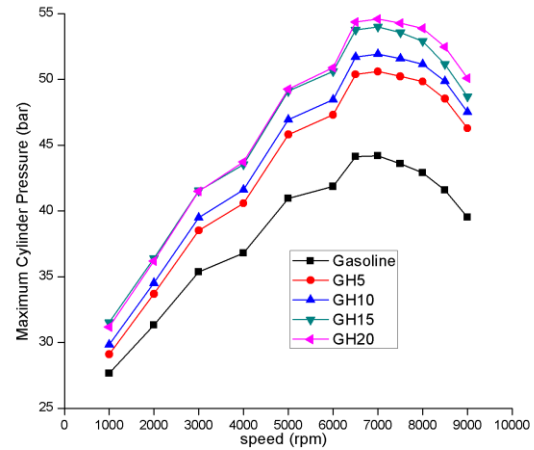
Combustion characteristics:

According to the Figure 4, there is one of important characteristics of combustion process which is the profiles of peak cylinder pressure under the stoichiometric condition for various hydrogen concentrations. In all case, the pressure rise continuously from begin to 6500 rpm, because volumetric efficiency is maximum at 6500 rpm and time of cycle at this take advantage of combustion process. As more percentage of hydrogen is added, the peak cylinder pressure significantly raises. At 6500 rpm, with 20% hydrogen blended, the peak pressure has approximately 1.22 times as gasoline to 54.35 bars. This is because of combustible identify and high flame velocity of hydrogen, combustion process is faster and gain high pressure.

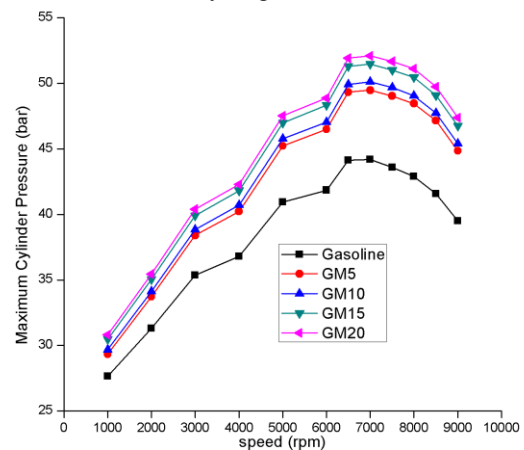
On the other hand, methane and LPG is also gas fuel, so it gains higher pressure than gasoline. While having 20% concentration addicted methane-gasoline fuel maximum pressure peaks at 17.62% to 51.91 bars and that of LPG-gasoline fuel is at 16.75% to 51.52 bars. LPG has more carbon in fuel than methane; therefore, flame speed of them is slower case of methane.

Figure 5 displays the profile of peak cylinder temperature against engine speed under different hydrogen volume fractions. The trend of maximum temperature of cylinder with engine speed is described in Figure 5(a) for the gasoline-hydrogen blends under stoichiometric condition. In case of

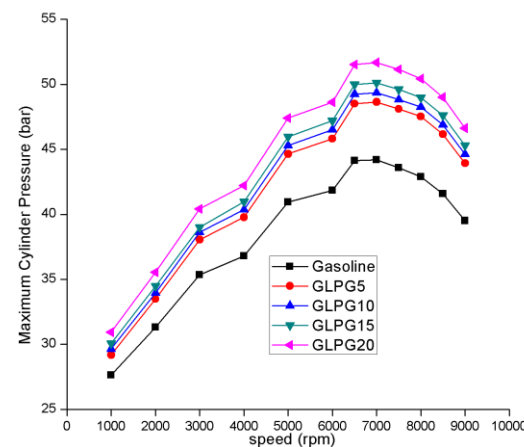
20% hydrogen enriched, at 5000 rpm, the cylinder temperature rises to 4% which is 2606.36 K. When amount of hydrogen addition increases, flame propagation speed of mixture fuel is improved and combustion process takes place quickly before engine will change expansion stroke; therefore, not only peak pressure but also the cylinder temperature steadily rise.



(a) Hydrogen-Gasoline



(b) Methane-Gasoline

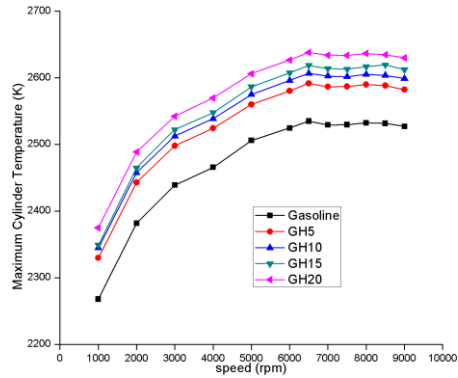


(c) LPG-Gasoline

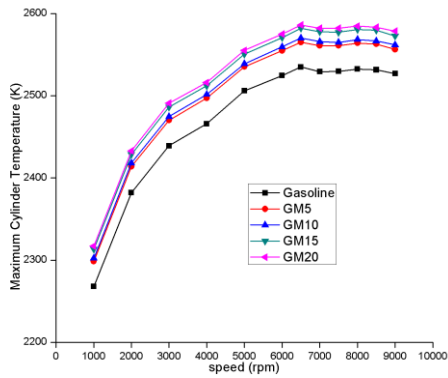
Figure 4: Peak cylinder pressure vs. engine speed and hydrogen volume fractions

Besides adding hydrogen, the quality of combustion process is raised by enhancing methane and LPG in mixture fuel in figure 5(b) and 5(c). With 20 methane

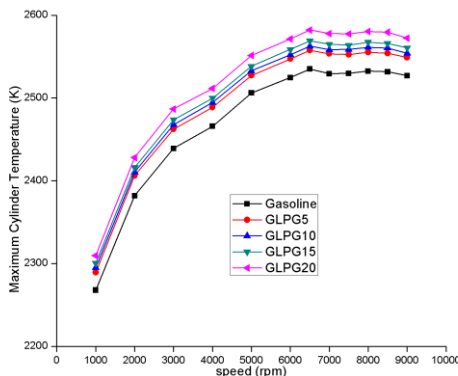
percentage of mixture, the cylinder temperature rises to 1.9% and to 1.8% with LPG at the same fraction. Because the flame velocity of LPG is faster than methane, but methane has shorter combustion duration, combustion process of them is too quickly and gain high cylinder temperature.



(a) Hydrogen-Gasoline



(b) Methane-Gasoline



(c) LPG-Gasoline

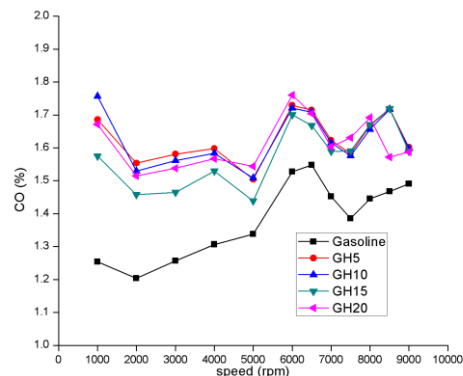
Figure 5: Peak cylinder pressure vs. engine speed and hydrogen volume fractions

At the same fraction of mixture fuel, the temperature of cylinder is continually gone up from 1000 rpm to 5000 rpm and after 5000 rpm, it is approximately stable. For the reason that volumetric efficiency, which is a characteristic to measure gas exchange process, is grown up when increasing engine speed. In addition, the time for heat transfer for coolant is decreased if speed increase. Thus this leads to improve that temperature. On the other hand, when the engine speed reaches over 6500 rpm, the quality of gas exchange process is low due to shorter time for

intake stroke and also reduction of the time for heat transfer.

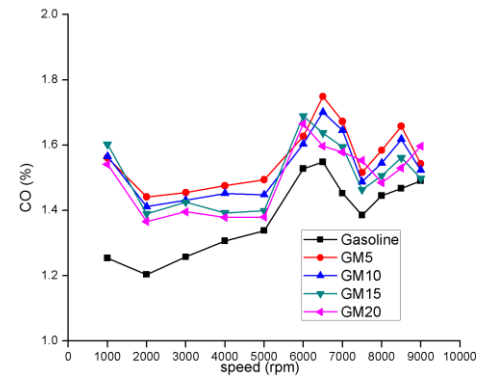
Emission characteristics

Figure 6 illustrates effect hydrogen addition volume ratios to CO emission concentration. Normally, when part of gasoline in fuel is down, amount of CO creation is decreased. This is cause by the concentration of carbon in fuel declined, but the temperature of engine go up when fuel is addicted to hydrogen and this made the reaction that reaction of CO formation happen quickly. It has the same result of some studies [4, 8]. Hence, the maximum concentration of CO is founded at the 6500 rpm which has maximum temperature and create highest brake torque. Although the concentration for each addition ratio is different, the amount of CO in 4 cases is fluctuated from 1.3% to 1.7%.

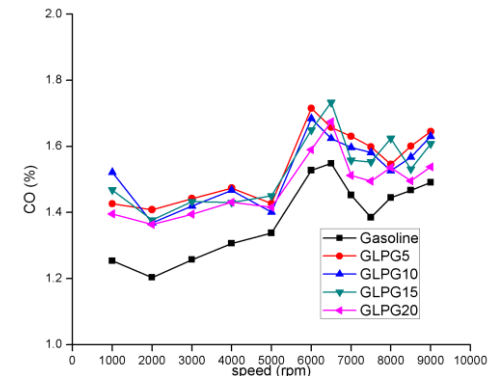


Hydrogen-Gasoline

(a)



(b) Methane-Gasoline



(c) LPG-Gasoline

Figure 6: Variation of CO with speeds for various hydrogen volume fractions

A simulation study on performance characteristics of a gasoline fuel injection motorcycle engine with separated addition of H₂, CH₄, and LPG

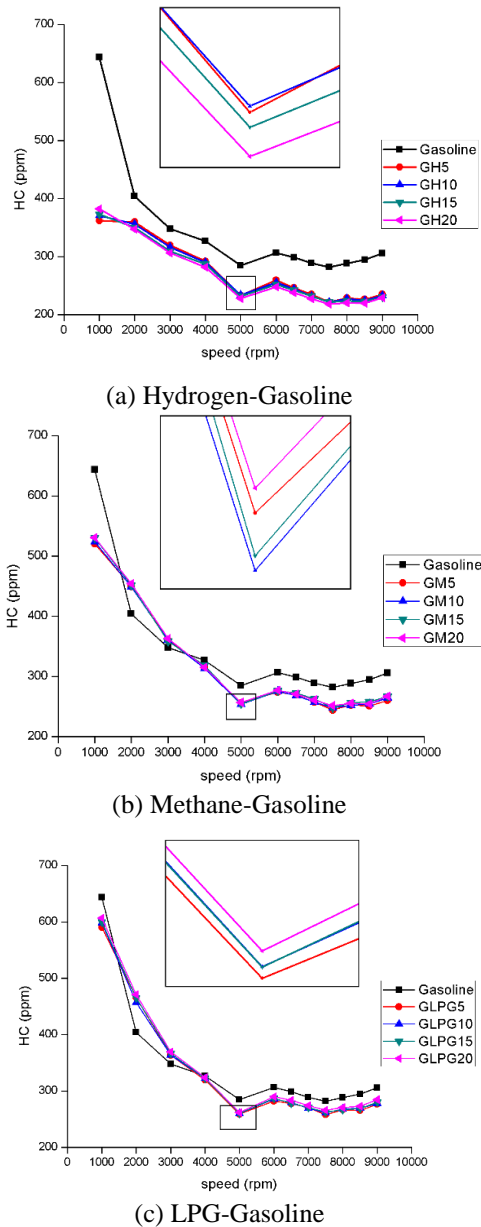


Figure 7: Hydrocarbon emission variations vs. engine speed and Hydrogen fraction

Also methane and LPG blends, amount of CO emission is lower than that of hydrogen. This is due to the mixture with methane and LPG has the lower temperature and the CO reaction is slower. Therefore, with 5% methane volume ratio, CO emissions slightly rise to 0.12% and to 6.7% with LPG in the same fraction. This is described in experimental research [7, 8] in same stoichiometric ratio condition.

The diagram gives us the hydrocarbon emission variations with engine speed at each hydrogen fraction in figure 7(a). The chart demonstrates for high efficiency and low pollution of hydrogen addition into gasoline. At 5000 rpm, only 5% hydrogen fraction, the hydrocarbon emission sharply drops into 18.14% and touch minimum with 20% hydrogen which fall 19.72% than gasoline. These

main reasons are high combustion quality and cylinder temperature due to fast flame velocity of hydrogen. In addition, hydrogen is no carbon fuel and this cause carbon part in fuel and in exhaust decrease significantly.

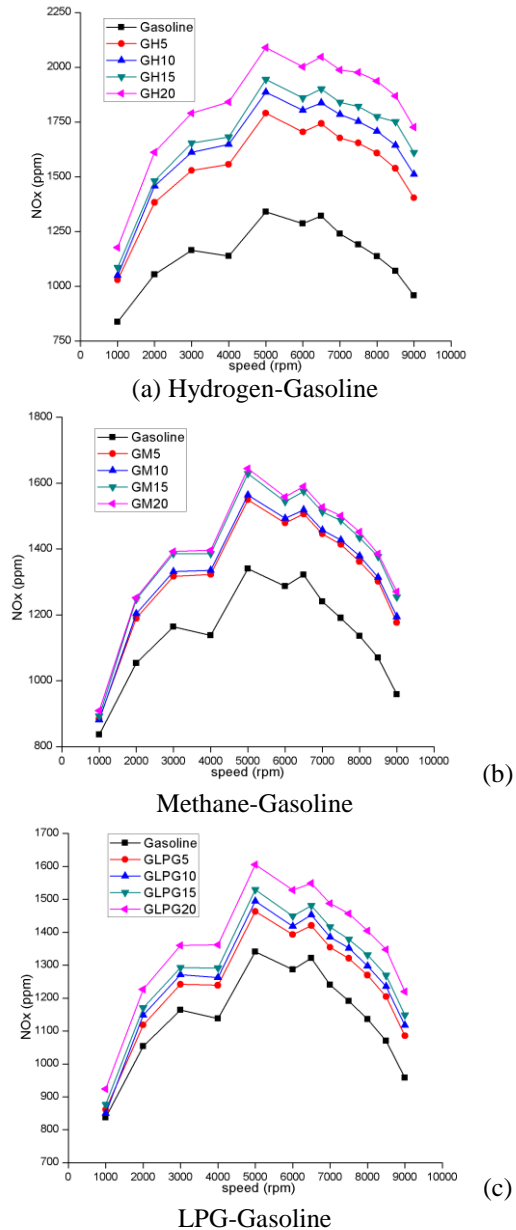


Figure 8: NOx emission variations vs. engine speed and Methane fraction

Not only hydrogen addition but also methane and LPG addition are gaining profits with engine using gas-gasoline fuel in figure 7(b) and 7(c). With having lower carbon than LPG, a reduction of hydrocarbon have 20% methane fraction is steep drop into 9.5% and more than LPG which is 7.7%. This is caused by the cylinder temperature for methane addition is higher than LPG addition and it is good condition for hydrocarbon oxidation. For all fractions, the hydrocarbon emission is dramatically dropped from 1000 rpm to 5000 rpm. Increase of cylinder

temperature could be caused by the improvement of gas exchange as engine speed is increased. This result is corresponding with that of combustion characteristics. In the contrary, the rapid cylinder temperature gives only disadvantage which is grown NO_x emission.

Three diagrams in Figure 8 present the effects of speed and hydrogen, methane and LPG volume fraction in fuel on NO_x emission. In three type of gas, hydrogen-gasoline blends are highest NO_x emission. This is due to hydrogen-gasoline have the fast flame propagation and shortest combustion duration, temperature in cylinder rapid rise which is good condition for creation of nitrous oxide. At 5000 rpm, the NO_x emission peak at over 56% with 20% hydrogen fraction in fuel. This drawback of hydrogen gives a requirement which must decrease it when application in vehicle hydrogen-gasoline fuel.

With having lower cylinder temperature, methane-gasoline and LPG-gasoline fuel are comparable to NO_x emission. With 20% methane concentration, the NO_x release in exhaust gas is increased to 22.68% and with LPG at the same ratio, there is 19.77% jumped. In all cases, the amount of NO_x is highest at 5000 rpm because the cycle time at this speed is advantage for reaction happened completely.

Conclusion:

This study describes the simulation of intake manifold fuel injection motorcycle engine using hydrogen, methane, LPG blended in fuel. Hydrogen enriches gives the best fuel economical profit and HC emission but it is low comparable with NO_x emission.

Methane and LPG which come from hydrocarbon fuel bring us more selections as the solution to NO_x emission of hydrogen case but it has high BSFC than gasoline. In general, this research gives simulation data for experimental study in the future. And with catalyst development, it is need to solve the NO_x emission problem which is described.

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