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Effect of Hydrogen Enriched LPG Fuelled Engine with Converted from a Diesel Engine

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Abstract — The purpose of this study is to obtain low-emission and high-efficiency in LPG engine with hydrogen enrichment. The objective of this paper is to clarify the effects of hydrogen enrichment in LPG fuelled engine on exhaust emission, thermal efficiency and performance. The compression ratio of 8 was selected to avoid abnormal combustion. To maintain equal heating value of fuel blend, the amount of LPG was decreased as hydrogen was gradually added. The relative air-fuel ratio was increased from 0.8 to 1.3, and the ignition timing was controlled to be at MBT (minimum spark advance for best torque).

Key words : LPG (liquefied petroleum gas), LPi (liquefied petroleum injection), MBT (minimum spark advance for best torque), VCSCE (variable compression ratio single cylinder engine)

1. Introduction

Hydrogen, as an energy source, has some distinct benefits for its high efficiency, convenience in storage, transportation, and conversion[10]. Hydrogen has much wider flammable limit than methane, propane or gasoline and the minimum ignition energy is about an order of magnitude lower than for other combustibles[7][1].

Because of its excellent ignitability and high adiabatic flame temperature of hydrogen fuel, the ignition delay period, flame development duration, rapid burning duration and overall burning angle are remarkably shorter than those in gasoline and diesel engine[2][3][6].

Thomas *et al.*[11] described that hydrogen for fuel cell vehicles would eliminate local air pollution, substantially reduce greenhouse gas and oil imports, cost no more than current transportation fuels, and require little investment in new infrastructure.

From an environmental point of view there is an increasing interest among the supplier's to investigate LPG as a transportation fuel. It was found that the liquid petroleum gas, roughly a mixture of propane and butane, gives a benefit in terms of toxic hydrocarbons emissions and ozone formation due to its composition and CO_2 emission levels[4][5][9].

Comparing the properties of hydrogen and LPG, it is possible to obtain interesting fuel economy and emission reductions. However, today the concept of hydrogen enriched LPG fuel, as fuel for internal combustion engines, has a greater interest than pure hydrogen powered engines because it involves fewer modification to the engines and their fueling systems. In fact, hydrogen and gasoline can be burned together in a wide range of air-fuel ratio, providing such good performances, as high thermal efficiency and reduced pollutant emissions[8].

This research is an experimental investigation of the effects of the fuel supply method, namely Mixer and LPi methods, on engine performance and emissions characteristics. These experiments, in which an old compression ignition engine is converted into a spark ignition engine, should contribute to a definite reduction in automotive emissions.

For the past several years, the authors have been

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interested in retrofitting old engines, for example, a compression ignition engine that is over 5 years old or a diesel vehicle with over 100,000 km. For reducing the exhaust emissions of an old compression ignition engine, they are trying to use LPG instead of diesel fuel. In order to achieve this, the current diesel automobile's high compression ratio must be lowered and LPG must be supplied as the fuel. This leads to the question of how exhaust and other performance characteristics will vary depending on how the fuel is supplied.

2. Experimental Apparatus and Procedure

Figure 1 shows the schematic diagram of the experimental apparatus. The before and after image of the piston is shown in Fig. 2.

In order to determine the ideal compression ratio,

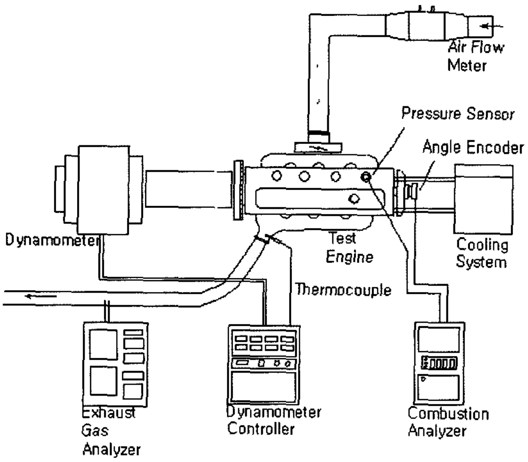

Fig. 1. Schematic diagram of experimental apparatus.

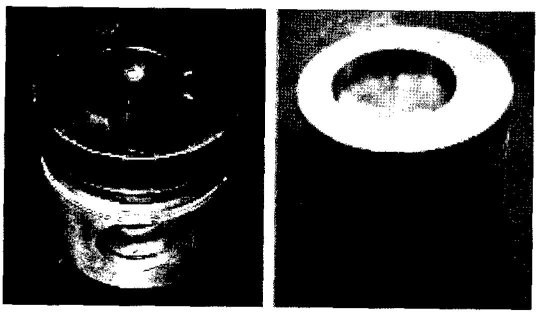

Fig. 2. Before and after images of the piston.

Table 1. Specification of test engine.

Engine type	OHV
Number of cylinder	1
Bore	130 mm
Stroke	140 mm
Displacement	1,858.2 cc
Range of compression ratio	7 to 14
Intake valve open	18 BTDC
Intake valve close	50 ABDC
Exhaust valve open	50 BBDC
Exhaust valve close	18 ATDC
Length of connecting rod	260 mm

the experimental engine was developed. The VCSCE (variable compression ratio single cylinder engine) used in this experiment was a single cylinder spark ignition that had been modified from a 6 cylinder 12 *l* diesel engine into engine. Major steps in the engine's fabrication are outlined below, and the specifications are listed in Table 1.

(1) The cylinder head was altered so that a spark plug could be inserted in the place of the injection nozzle, and the piston was modified into a bath tub type. In order to take advantage of the squish effect that occurs at the end of the compression process and subsequently optimize the mixture formation, a bath tub type piston was made.

(2) New crankshaft and crankcase were developed. A new flywheel was made so that the desired RPM could be obtained.

(3) A balance shaft and a flywheel were made to minimize vibration and acceleration, and the cooling and lubrication system were placed externally in order to more precisely determine engine performance.

All experiments were conducted at 1400RPM, MBT (Minimum spark advance for Best Torque), WOT (Wide Open Throttle), and a compression ratio of 8. The compression ratio of 8 was selected to minimize abnormal combustion. Upon determining the normal operation of the engine, LPG was supplied to achieve a relative air-fuel ratio of 0.8. To maintain equal heating value, the amount of LPG was decreased, and hydrogen was gradually added. In a similar manner, the relative air-fuel ratio was increased from 0.8 to 1.5 in increments of 0.1, and the ignition timing was controlled to be at MBT each time. The characteristics of LPG and hydrogen are listed in Table 2.

Table 2. Characteristics of LPG and hydrogen.

	C_4H_{10}	H_2
Low heating value, MJ/kg	45.84	120
Theoretical air-fuel ratio	15.5	34.3
Flammability limits	0.4 to 1.7	0.12 to 10.12
Density, kg/m^3	2.64	0.0899
Adiabatic flame temperature, K	2263	2657
Autoignition temperature, K	858	723
Turbulent burning velocity, m/s	0.4	1.7

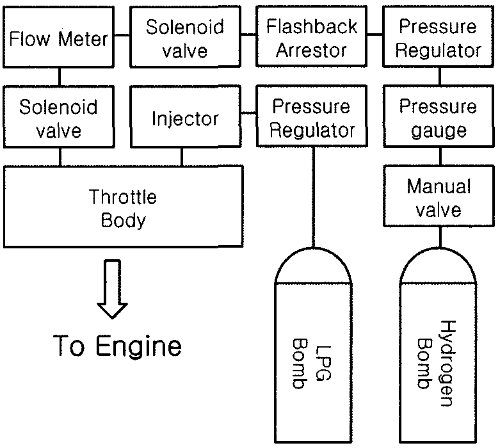

Fig. 3. Fuel supply system.

A desired mixture of LPG and hydrogen was used as the fuel system, and the fuel rate was controlled with a duty drive and a solenoid valve. LPG consumption was measured via a balance scale with a degree of precision of 1 g. High purity hydrogen at 200 bar was flown through the pressure controller, the mass flow meter, the solenoid valve, and the flame arrestor on its way to the intake. Figure 3 shows the fuel supply system. If the relative air-fuel ratio and the LPG consumption rate do not reach target values, the duty and main jet are controlled accordingly. Accordingly, the piston head was modified to reduce the compression ratio from 14 to 7. A surge tank was installed on the inlet side to minimize the experimental engine's intake pulsation, and an external pump was used for the coolant and the oil in order to minimize power loss.

Experimental parameters are dependent of the relative air-fuel ratio and hydrogen supplement rate. The hydrogen supplement rate is defined as follows:

Hydrogen supplement rate (%)

$$= \frac{m_{H_2} \times Q_{L(H_2)}}{m_{C_4H_{10}} \times Q_{L(C_4H_{10})} + m_{H_2} \times Q_{L(H_2)}},$$

where m_{H_2} is the mass flow rate of hydrogen, $m_{C_4H_{10}}$ the mass flow rate of LPG, $Q_{L(H_2)}$ the lower heating value of hydrogen, and $Q_{L(C_4H_{10})}$ the lower heating value of LPG (See Table 2). The 10% hydrogen supplement rate per heating value is equal to 55% hydrogen per total fuel volume and 20% hydrogen supplement rate per heating value is equal to 85% hydrogen per total fuel volume. The fuel supply system provides LPG/hydrogen mixtures based on same heating value. All experiments were conducted at 1400 rpm, MBT, and a compression ratio of 8. The compression ratio of 8 was selected to minimize abnormal combustion. To maintain equal heating value, the amount of LPG was decreased, and hydrogen was gradually added. In a similar manner, the relative air-fuel ratio was increased from 0.8 to 1.3 in increments of 0.1, and the ignition timing was controlled to be at MBT each time.

3. Results and Discussions

3-1. Motoring test

Figure 4 shows the theoretical cylinder pressure at different specific heat ratios and experimental cylinder pressure versus crank angle at the motoring condition. This figure shows similar trends overall between the pressure diagram taken from simulation and the experimental pressure. Also, it shows that there is no leakage in the manufactured engine. The cylinder pressure curve at BTDC (before top dead center) tends to

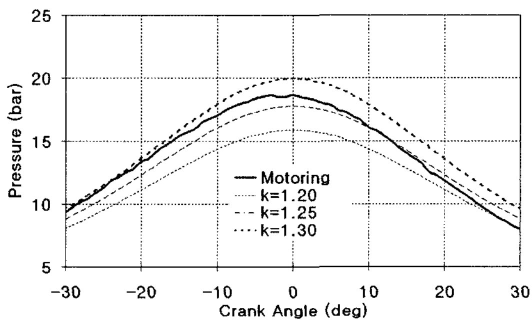

Fig. 4. Cylinder pressure versus crank angle at different specific heat ratio.

be a little low and the pressure curve at ATDC (after top dead center) tends to be a little high because it is thought that the effective compression ratio becomes low as the closing of the inlet valve is accomplished at ATDC.

3-2. Engine performance

Figures 5-8 show brake torque, power, thermal efficiency and BSFC (brake specific fuel consumption) as a function of relative air-fuel ratio (λ=(air/fuel)$_{actual}$/(air/fuel)$_{stoichiometric}$) with the addition of 0%, 10% and 20% H_2 at 1400 rpm, and a compression ratio 8. As shown in Figs. 5, 6, in general, torque, power and thermal efficiency decrease with the increase of hydrogen supplement rate.

The reason for the decreases in torque and power could be that the lack of oxygen increase with the increase of hydrogen supplement rate in the rich mixture zone and result the imperfect combustion. But there is no difference in thermal efficiency in the lean

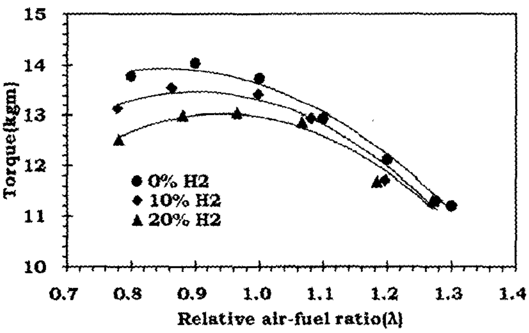

Fig. 5. Engine torque versus relative air-fuel ratio for hydrogen rates.

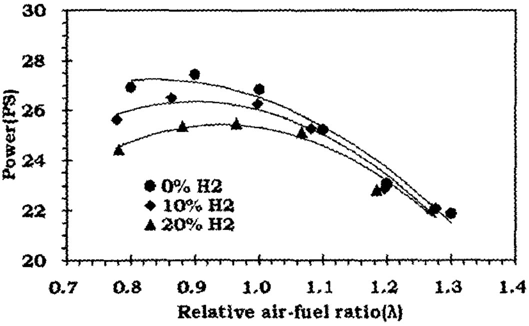

Fig. 6. Brake power versus relative air-fuel ratio for hydrogen rates.

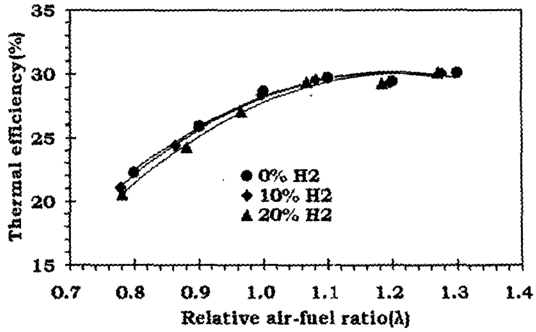

Fig. 7. Brake thermal efficiency versus relative air-fuel ratio for hydrogen rates.

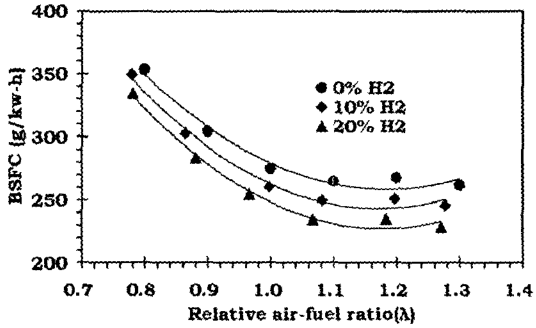

Fig. 8. Brake specific fuel consumption versus relative air-fuel ratio for hydrogen rates.

mixture zone. The reason for this result could be that the sufficient oxygen was supplied and hydrogen does much for the fast combustion because hydrogen has four times higher burning velocity.

Figure 7 shows thermal efficiency as a function of relative air-fuel ratio. As shown in this figure, thermal efficiency decreases with the addition of hydrogen. At λ=1, thermal efficiency shows a decrease of about 5% with the addition of 20% hydrogen. The reason for this increase in thermal efficiency could be that the hydrogen fuel burns all at once.

Figure 8 shows fuel consumption as a function of relative air-fuel ratio with the addition of 0%, 10% and 20% H_2 at 1400 rpm, MBT, and a compression ratio of 8. Fuel consumption that depends on thermal efficiency is defined as the mass flow rate per hour, and it may depend on the increase of brake power rather than the increase of fuel quantity. The reason for lower fuel consumption with increased hydrogen additions compared to LPG combustion would be the

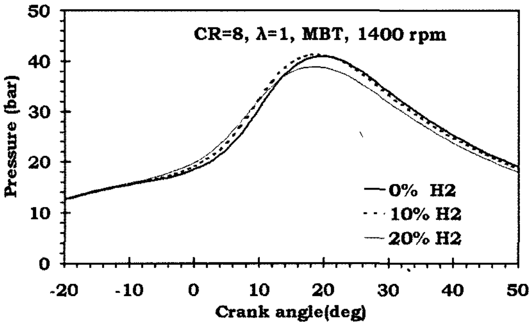

Fig. 9. Cylinder pressure versus crank angle with various hydrogen proportions.

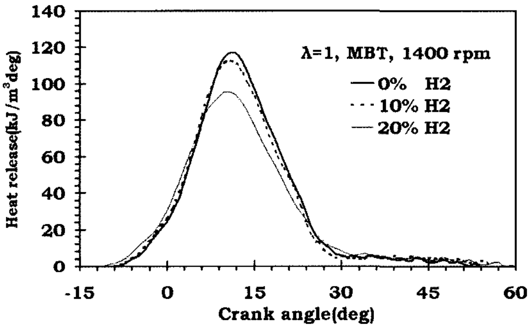

Fig. 10. Heat release versus crank angle with various hydrogen proportions.

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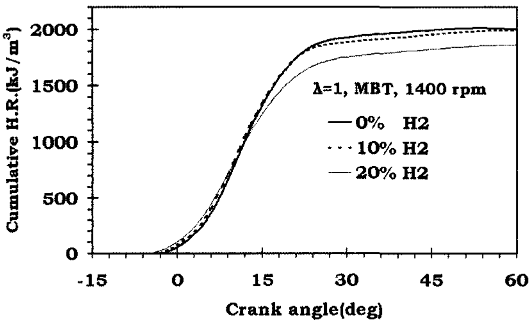

Fig. 11. Cumulative heat release versus crank angle with various hydrogen proportions.

ast flame propagation velocity of hydrogen.

Figures 9-11 show cylinder pressure, heat release and cumulative heat release as a function of crank angle for 250 cycles. Heat release rate was calculated by making a first law analysis of the average pressure versus crank angle variation. For the calculation the contents of the cylinder were assumed to behave as an ideal gas with the specific heat being dependent on temperature. The cumulative heat release was then calculated. The start of combustion was determined from the rate of pressure rise variation. In general, heat release, cylinder pressure and cumulative heat release decrease with the increase of hydrogen supplement rate, but the ignition timing decreases with the increase of it. From these figures, the fast burning characteristics of hydrogen permit much more satisfactory high-speed engine operation. This would allow an increase in power output with a reduced penalty for lean mixture operation.

3-3. Emissions

Figure 12 shows NOx emission as a function of relative air-fuel ratio with the addition of 0%, 10% and 20% H_2 at 1400 rpm, MBT, WOT and a compression ratio of 8. In general, NOx emission is the maximum at λ=1.2, and the addition of 20% hydrogen results in about 20% increase in the amount of NOx emission compared to that of pure LPG combustion. NOx emission ought to depend on the fast combustion of hydrogen fuel and the higher maximum temperature and pressure in the cylinder compared to LPG combustion. This fact results from the fast combustion of hydrogen fuel and the higher maximum temperature and pressure in the cylinder compared to LPG combustion (see Table 2).

Figures 13 and 14 show CO, CO_2 emissions as a function of relative air-fuel ratio with the addition of 0%, 10% and 20% H_2. The CO emission decreases if the relative air-fuel ratio is increased from 0.76 to 1.3, and the CO emission also decreases as hydrogen

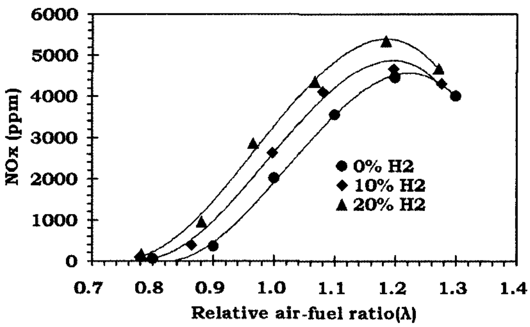

Fig. 12. NOx emission versus relative air-fuel ratio for hydrogen rates.

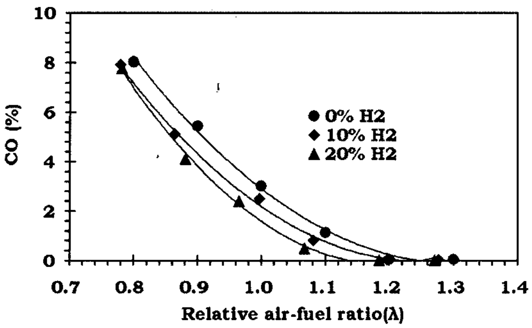

Fig. 13. CO emission versus relative air-fuel ratio for hydrogen rates.

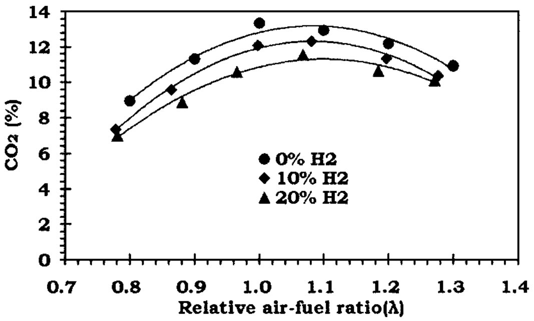

Fig. 14. CO_2 emission versus relative air-fuel ratio for hydrogen rates.

is added. For rich relative air-fuel ratios around λ= 0.8, CO emission exhibits a maximum value, and it appears that there is almost zero CO emission above λ=1.2. The figure shows that a great quantity of CO emission is produced with the rich air-fuel mixture, since an insufficient supply of air prevents all carbon of the fuel from becoming the perfect combustion gas CO_2. Therefore, CO emission can be represented as a function of the relative air-fuel ratio and might be considerably affected by it.

Unburned hydrocarbons from spark ignition engine exhaust are a primary component of smog. At present they are attributed to several sources, one of which is the gap between piston and cylinder wall. Unburned charge is pushed into this crevice during the compression stroke. The crevice is narrow enough to quench the flame front, leaving unburned crevice gases, so that during the power stroke, as the piston descends and the exhaust valve opens, these unburned gases re-emerge. Figure 15 shows THC emission as a function

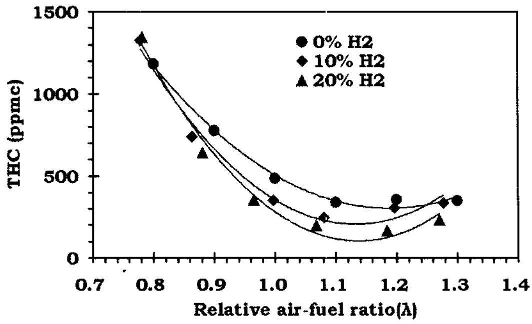

Fig. 15. Total hydrocarbon emission versus relative air-fuel ratio for hydrogen rates.

of relative air-fuel ratio with the addition of 0%, 10% and 20% H_2. THC emission tends to be similar to CO emission at rich relative air-fuel ratio as the ratio is increased from 0.76 to 1.3, but it is increased at lean side. Also, THC emission at the rich side decrease with the addition of hydrogen.

4. Conclusions

The results obtained are as follows.

(1) Thermal efficiency dencreases with the addition of hydrogen. At lambda=1, thermal efficiency was increased by about 5% with the addition of 10~20% hydrogen. The reason for this increase in thermal efficiency was that the hydrogen fuel burns all at once.

(2) NOx emission is the maximum at about the lambda=1.2, and the addition of 20% hydrogen results in about 20% increase in the amount of NOx emission compared to that of pure LPG combustion. This fact results from the fast combustion of hydrogen fuel and the higher maximum temperature and pressure in the cylinder compared to LPG combustion.

(3) CO emissions decreases if the relative air-fuel ratio (lambda) is increased from 0.8 to 1.3, and the CO emissions also decreases as hydrogen is added. CO emission can be represented as a function of the relative air-fuel ratio.

(4) THC emissions tends to be similar to CO emission at rich mixture conditions as the relative air-fuel ratio is increased from 0.8 to 1.3, but it is increased on the lean side. THC emission on the rich side also decrease with the addition of hydrogen.

Acknowledgments

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