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A Study on the Performance of an LPG (Liquefied Petroleum Gas) Engine Converted from a Compression Ignition Engine

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Abstract — The purpose of this study was to investigate the reduction of exhaust gas temperature in a LPG engine that had been converted from a diesel engine. A conventional diesel engine was modified to a LPG (Liquefied Petroleum Gas) engine by replacing the diesel fuel injection pump with a LPG fuel system. The research was performed by measuring the exhaust gas temperature upon varying spark ignition timing, air-fuel ratio, compression ratio, and different compositions of butane and propane. Engine power and exhaust temperature were not influenced by various butane/propane fuel compositions. Finally, among the parameters studied in this investigation, spark ignition timing is one of the most important in reducing exhaust gas temperature.

Key words : Liquefied petroleum gas (LPG), LPG composition, Exhaust gas temperature, Power, Heat release, Mass fraction burned

1. Introduction

Diesel vehicles are most widely used for commercial purposes because they have excellent thermal efficiency, low fuel consumption and low carbon dioxide emission. However, because the exhaust emissions of diesel engines contain a large amount of PM (particulate matter), diesel vehicles pose a serious threat to the environment and human health. Both NOx and particulate emissions are affected by combustion temperatures, albeit in opposite ways. Combustion temperature, in turn, is related to injection timing. Thus, NOx emissions generally increase and particulate emissions decrease when injection timing is advanced. In addition, for a realistic comparison of the effects of diesel fuel quality on emissions, injection timing and exhaust gas temperature must be considered[1][2].

LPG (Liquefied petroleum gas) has long been used as an automotive fuel in light duty vehicles such as passenger cars and taxies. LPG is suitable as automotive fuel because it is free of sulfur and aromatic compounds[3][4]. It has already been widely used as fuel for gasoline engines. If LPG could be used in a diesel engine, the PM emissions could be reduced since LPG is a gaseous material and will vaporize as soon as it leaves the injector. LPG was assumed to be used in spark ignition engines primarily for its environmental benefits; however, LPG also offers some reduction of petroleum usage, since approximately 50~60% of the LPG appropriated for motor fuel use currently comes from natural gas[5][6].

As petroleum use increases worldwide as a result of industrial expansion and an increase in the number of automobiles, the dependency on petroleum is becoming greater each day. With the increase in the number of automobiles, automobiles are replacing commercial industries as the main culprit of environmental pollution and are becoming a serious problem.

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Of these automobiles, diesel vehicles emit an especially large quantity of smoke and NOx. In Korea, diesel vehicles only comprise 34% of the total vehicle count, but they produce 65% of the total pollutants, including 90% of NOx and 99% of PM. Because of these reasons, regulations concerning NOx and PM are incrementally becoming stricter. Clean fuel vehicles must be developed in order to satisfy such regulations, and research is taking place concerning alternative fuels. LPG is being considered as a prospective alternative fuel source since its use is advantageous from an environmental as well as an economic standpoint as much infrastructure already exists for its use. Research regarding the conversion of diesel vehicles to LPG vehicles is being conducted since diesel vehicles account for a large portion of the environmental pollution[7]-[9].

Therefore, this research will study the effects of spark ignition timing, compression ratio, air-fuel ratio and change in fuel mixtures on power output and exhaust gas temperatures of a LPG engine that had been converted from a diesel engine.

2. Experimental Setup and Method

This experiment uses a water-cooled, 4 cylinder, 3568 cc, direct-injection diesel engine from a 2.5 ton truck that had been converted to run on LPG. The major specifications of the engine are shown in Table 1.

Unlike the conversion from a gasoline engine, the conversion of a LPG engine from a diesel engine is technically complex. In other words, the conversion from a compression ignition engine to a spark ignition engine requires major engine modifications. Accordingly, the piston head was modified to reduce the compression ratio from 21 to 9. The before and after shape of the piston is shown in Fig. 1.

Table 1. Engine specifications.

Items	Specification
Number of cylinder	4
Combustion type	Pre-mixture spark ignition
Displacement volume	3,568 cc
Bore× stroke	104×105 mm
Maximum power	88/3200 PS/RPM
Compression ration	9

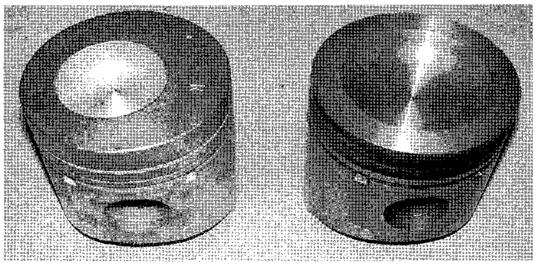

Fig. 1. Before and after images of the piston.

In order to supply combustion energy to the combustion chamber, the injection nozzle was removed, and the coolant path was utilized to install a spark plug. Also, the ignition coil and the distributor were installed. The diesel fuel spray pump was removed, and the distributor was installed by connecting it to the driving gear of the fuel pump. To control more precise spark timing, an electronic, optical type distributor with 360 slits was used. Spark timing was controlled by using an ECU.

The mixture is controlled through an ECM (electronic control module) that receives signals from an oxygen sensor mounted on the exhaust and calculates the appropriate stoichiometric ratio. The schematic diagram of the experimental apparatus is shown in Fig. 2. In this experiment, the LPG composition was changed by varying the ratio of butane and propane.

The engine used in the tests, which had been con-

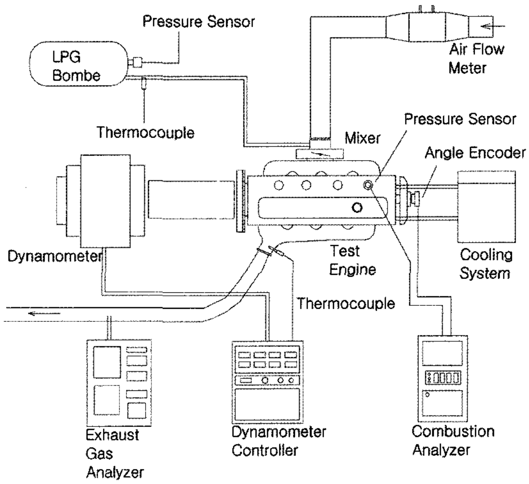

Fig. 2. Schematic diagram of experimental apparatus.

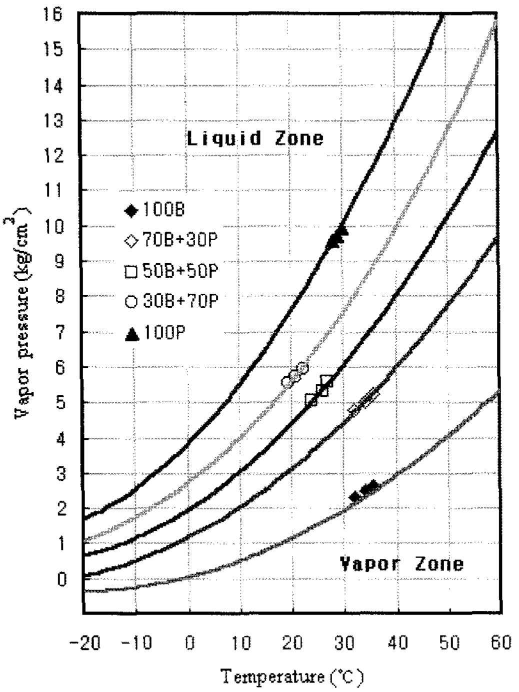

Fig. 3. Vapor pressure of butane/propane fuel composition.

verted from a diesel to a LPG engine, was connected to a dynamometer, and the RPM (revolution per minute) and load were controlled. Fuel composition was varied through the mass ratio by, for example, filling the tank with the desired amount of butane and then adding the propane to achieve the desired composition. In Fig. 3, the fuel composition was confirmed by measuring the tanks pressure and temperature and comparing them to the appropriate vapor pressure plot.

The tests were done by first finding the MBT (Minimum spark advance for Best Torque) spark ignition timing for 100% butane and then, using this point as a reference, finding the MBT spark ignition timing for each of the fuel compositions. The fuel compositions used for the experiment were 100% butane, 100% propane, 50% butane 50% propane, 70% butane 30% propane, and 30% butane 70% propane. The change in fuel rate resulting from the propane increase was controlled through the duty ratio of the mixer. In this experiment, exhaust gas temperature was measured by inserting a thermocouple 8cm away from the end of the exhaust manifold where exhaust gas is collected.

Spark ignition timing was controlled with an ECU (electronic control unit), and the air-fuel ratio was varied by either changing the duty value of the mixer or changing the diameter of the main jet, which is located in the main pathway of the fuel system.

3. Experimental Results and Discussion

Figure 4 shows power output and exhaust gas temperature as a function of spark ignition timing at full load. Exhaust gas temperature decreases about 30℃ for every 4° of spark advance. This decreases in temperature is thought to happen because as spark timing is retarded, ignition starts late and the extended combustion process during the expansion stroke leads to high temperature exhaust gas, but as spark timing is advanced, the combustion period during the expansion stroke is shortened and the exhaust gas temperature becomes relatively lower. In addition, power output increases due to spark advance and the corresponding increase in maximum cylinder pressure, and the increased expansion ratio that results from the high pressure increases the expansion-cooling effect and also decreases exhaust gas temperature.

Figure 5 shows cylinder pressure for varying spark ignition timing at 3000 RPM and full load. The figure shows that as spark is advanced, maximum pressure increases, and the crank angle at maximum pressure

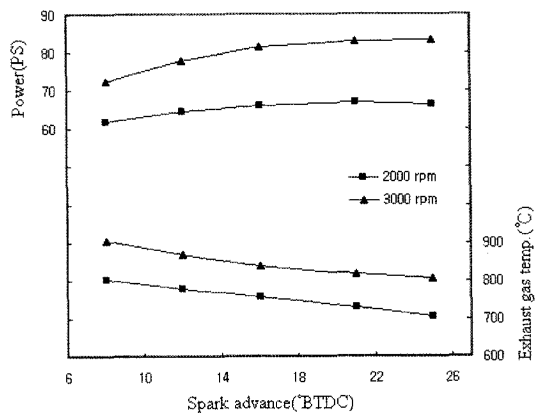

Fig. 4. Power and exhaust gas temperature for various spark ignition timing.

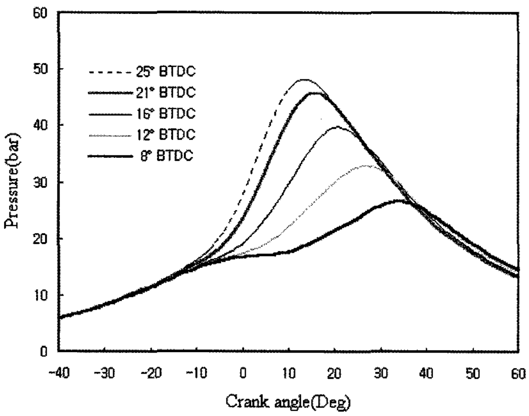

Fig. 5. Cylinder pressure curve for various spark ignition timing at 3000 RPM and full load.

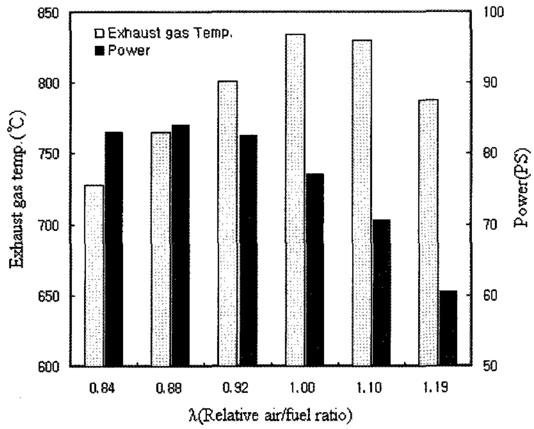

Fig. 7. Power and exhaust gas temperature for various air-fuel ratios at 3200 RPM and full load.

also increases. When the spark timing is at BTDC (Before Top Dead Center) 8°, the severe retardation causes a significant decrease in cylinder pressure, and when compared to the spark timing of BTDC21°, the maximum cylinder pressure is 42% lower. Also, maximum pressure occurs 41° after ignition at BTDC8° compared to 37° at MBT spark timing. This is thought to cause the exhaust gas temperature to increase as spark timing is retarded.

Figure 6 shows the heat release rate curve. The starting point of heat release is directly proportional to spark timing, and since the period of heat release also increases as spark is retarded, the end point of heat release is delayed even more in such cases. As the figure shows, the point of maximum heat release

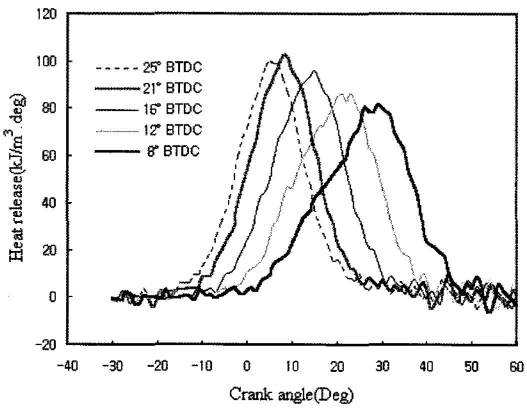

Fig. 6. Heat release curve for various spark ignition timing at 3000 RPM and full load.

is delayed about 23° for every 4° of spark retardation. When the spark timing is significantly retarded to 8° and BTDC12°, maximum heat release rate is low, but the heat release period is long, causing the heat release end point to be greater than ATDC (After Top Dead Center) 40° and the exhaust gas temperature to be higher as a result.

Figure 7 shows exhaust gas temperature and power output for various air-fuel ratios at 3200 RPM and full load. In order to investigate the dependence on only the air-fuel ratio, spark ignition timing was set at BTDC21°, which is the MBT spark timing for λ=1. The experimental results show that while exhaust gas temperatures decrease for both rich and lean mixtures, the decrease was most significant for rich mixtures. Also, power output increases for rich mixtures but decreases for lean mixtures. Compared to the air-fuel ratio of λ=1, the exhaust gas temperature decreased by around 30 and the power output increased by around 5 HP (Horse Power) for λ=0.92, but the exhaust gas temperature decreased by only about 4 and the power output decreased by around 6 PS for λ=1.1.

When the mixture is rich, combustion speed becomes faster and the cylinder pressure increases. As a result, increased cooling effect due to the larger expansion ratio and short combustion period is thought to decrease the exhaust gas temperature and increase the power output. When the mixture is lean, fuel consumption in the combustion chamber is reduced, and

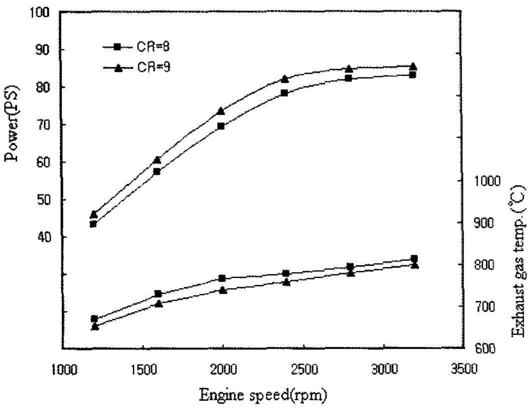

Fig. 8. Power and exhaust gas temperature for various compression ratios at full load condition.

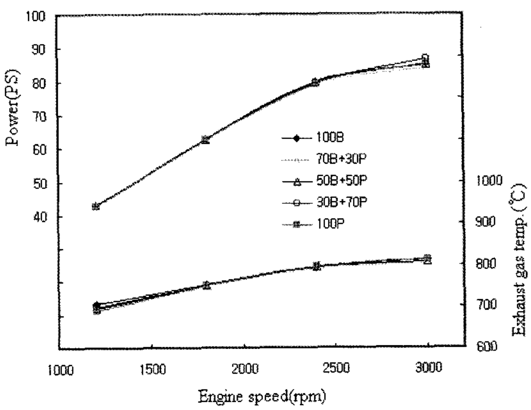

Fig. 9. Power and exhaust gas temperature for various butane/propane fuel compositions at full load condition.

the combustion speed becomes slower. Combustion chamber is decreased due to the resulting decrease in cylinder pressure, but the cooling effect is low because of the decreased expansion ratio. Also, the longer combustion period leads to an increase in exhaust gas temperature that is greater than the reduction in combustion temperature.

Figure 8 shows power output and exhaust gas temperature for varying compression ratios at full load. In order to investigate the dependence on only the compression ratio, spark ignition timing was set throughout the experiment to the MBT spark timing for the compression ratio of 9.

Experimental results show that as compression ratio increased from 8 to 9, power output increased by an average of 3PS at all engine speeds while exhaust gas temperature decreased by an average of 18. It is thought that the increase in maximum pressure causes the power increase while the increase in expansion ratio causes the decrease in exhaust gas temperature. Therefore, it is thought that, in order to decrease exhaust gas temperature and increase power, the compression ratio should be raised so long as knocking does not occur. However, the resulting change in MBT spark timing and exhaust gas temperature must be taken into consideration.

Figure 9 shows power output and exhaust gas temperature as a function of engine speed at different butane/propane compositions. Experimental results show that power output and exhaust gas temperature do not change significantly for various fuel compositions.

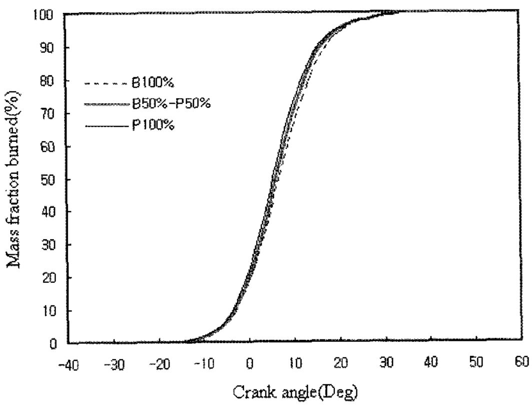

Fig. 10. Mass fraction burned versus crank angle curves for various butane/propane fuel compositions at 2400 rpm and full load.

Figure 10 shows the mass fraction burned for varying fuel compositions. Mass fraction burned was also shown to have little dependence on varying the fuel composition. Crank angle for 10% and 90% combustion was almost identical at 3°BTDC and 17°ATDC, respectively, and the crank angles for 10~90% combustion was also almost identical at 20°. Accordingly, it is thought that varying the fuel composition at the same compression ratio of 9 will not have a significant effect on power output and exhaust gas temperature, and considering the fact that the spark timing at knocking for 100% propane is about 3°more advanced than that of 100% butane, higher compression ratios should yield better results.

4. Conclusions

Upon modifying a diesel engine into a LPG engine and investigating the effects of various experimental parameters on power output and exhaust gas temperature, the following conclusions could be made.

1) Power increases by an average of 4% for every 4° CA of spark advance before MBT spark timing, and exhaust gas temperature is decreased by about 30°C for every 4° CA of spark advance.

2) Compared to the air-fuel ratio of λ=1, the exhaust gas temperature decreased around 30 and the power output increased by around 7% for λ=0.92, but the exhaust gas temperature decreased by only about 4 and the power output decreased by around 8% for λ=1.1.

3) As the compression ratio increases from 8 to 9, power output at all engine speeds increase by around 4%, and the exhaust gas temperature decreased by an average of 18.

4) With all other parameters held constant, MBT spark timing did not change significantly for varying fuel compositions, and the resulting power and exhaust temperature trends were similar.

5) In conclusion, if the optimal compression ratio for the given fuel composition is chosen and the corresponding MBT spark timing found, optimal power output and exhaust gas temperature values should be obtained.

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References

1. Kajiwara, M.; Sugiyama, K.; Sagara, M.; Mori, M.; Goto, S.; Alam, M. "Performance and Emissions Characteristics of an LPG Direct Injection Diesel Engines", SAE paper 2002-01-0869, 2002.
2. Min, B.H.; Chung, J.T.; Kim, H.Y.; Park, S. "Effects of Gas Composition on the Performance and Emissions of Compressed Natural Gas Engines", KSME International Journal, 2003, 16(2), 219-226.
3. Heywood, J.B. Internal Combustion Engine Fundamentals, McGraw - Hill International Editions, 1988.
4. Vincent, M.W.; Richards, P.; Cook, S.L. "Particulates reduction in diesel engines through the combination of a particulate filter and fuel additive", SAE paper 982654, 1998.
5. Yokota, H.; Nakajima, H.; Kakaegawa, T. "A new concept for low-emission diesel combustion (2nd report: reduction of HC and CO emission, and improvement of fuel consumption by EGR and MTBE blended fuel)", SAE paper 981933, 1998.
6. Zelenka, P.; Egert, M.; Cartellieri, W. "Ways to meet future emission standards for heavy sports utility vehicles (SUV)", SAE paper 2000-05-0288, 2000.
7. Lee, K.H.; Lee, C.S.; Ryu, J.D.; Choi, G.M. "Analysis of Combustion and Flame Propagation Characteristics of LPG and Gasoline Fuels by Laser Deflection Method", *KSME International Journal*, 2002, 16(7), 935-941.
8. Arcoumanis, C.; Nagwaney, A.; Hentschel, W.; Rake, S. "Effect of EGR on spray development, combustion and emissions in a 1.9L direct-injection diesel engine", SAE paper 952356, 1995.
9. Lejeune, M.; Lortet, D.; Michon, S. "Combustion development for Euro 4", SAE paper 2002-30-0021, 2002.