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Development and performance analysis of a Miller cycle in a modified using diesel engine

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Abstract — The objective of the research was to study the effects of Miller cycle in a modified using diesel engine. The engine was dedicated to natural gas usage by modifying pistons, fuel system and ignition systems. The engine was installed on a dynamometer and attached with various sensors and controllers. Intake valve timing, engine speed, load, injection timing and ignition timing are main parameters. The results of engine performances and emissions are present in form of graphs. Miller Cycle without supercharging can increase brake thermal efficiency and reduce brake specific fuel consumption. The injection timing must be synchronous with valve timing, speed and load to control the performances, emissions and knock margin. Throughout these tested speeds, original camshaft is recommended to obtain high volumetric efficiency. Retard ignition timing can reduce NO_x emissions while maintaining high efficiency.

Key words : Miller Cycle, Intake Valve Timing, Injection Timing, Ignition Timing, Emissions, Natural Gas Engine

1. Introduction

In engineering, the Miller cycle is a combustion process used in a type of four-stroke internal combustion engine. The Miller cycle was patented by Ralph Miller, an American engineer, in the 1940s. This type of engine was first used in ships and stationary power-generating plant, but has recently been adapted by Mazda for use in the Mazda Millennia which is also known as a Eunos 800 in some countries(1,2).

A traditional Otto cycle engine uses four "strokes", of which two can be considered "high power" - the compression stroke and power stroke. Much of the internal power loss of an engine is due to the energy needed to compress the charge during the compression stroke, so systems that reduce this power consumption can lead to greater efficiency(3,4).

In the Miller cycle the intake valve is left open longer than it normally would be. This is the "fifth" cycle that the Miller cycle introduces. As the piston moves back up in what is normally the compression stroke, the charge is being pushed back out the normally closed valve. Typically this would lead to losing some of the needed charge, but in the Miller cycle the piston in fact is over-fed with charge from a supercharger, so blowing a bit back out is entirely planned. The supercharger typically will need to be of the positive displacement kind (due its ability to produce boost at relatively low RPM) otherwise low-rpm torque will suffer(5).

The key is that the valve only closes, and compres-

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sion stroke actually starts, only when the piston has pushed out this "extra" charge, say 20 to 30% of the overall motion of the piston. In other words the compression stroke is only 70 to 80% as long as the physical motion of the piston. The piston gets all the compression for 70% of the work.

The Miller cycle "works" as long as the supercharger can compress the charge for less energy than the piston. In general this is not the case, at higher compressions the piston is much better at it. The key, however, is that at low compressions the supercharger is better than the piston. Thus the Miller cycle uses the supercharger for the portion of the compression where it is best, and the piston for the portion where it is best. All in all this leads to a reduction in the power needed to run the engine by 10 to 15%. To this end, successful production engines using this cycle have typically used variable valve timing to effectively switch off the Miller cycle in regions of operation where it does not offer an advantage[6,7].

In a typical spark ignition engine, the Miller cycle yields an additional benefit. The intake air is first compressed by the supercharger and then cooled by an intercooler. This lower intake charge temperature, combined with the lower compression of the intake stroke, yields a lower final charge temperature than would be obtained by simply increasing the compression of the piston. This allows ignition timing to be advanced beyond what is normally allowed before the onset of detonation, thus increasing the overall efficiency still further[8].

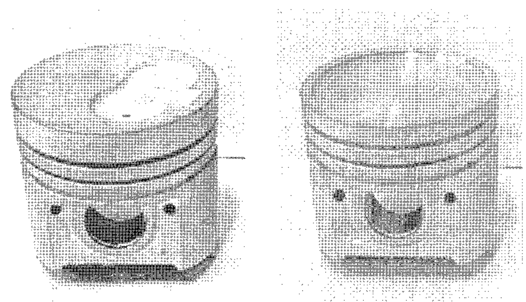
Ju Hee Lee[9] researched on the thermal efficiency on an industrial engine with Miller cycle. A diesel engine was retrofitted to natural gas engine for better duration. He changed the closing time of intake valve for adapting Miller cycle. Intake cam lift compensation test was added on the EIVC test and also effective compression pressure compensation test was added on the LIVC test. LIVC test at 51 degree-ABDC (After Bottom Dead Center) bettered the fuel consumption ratio around 5-8% and brake thermal efficiency around 2-3%. LIVC test at 77 degree-ABDC bettered the fuel consumption ratio and brake thermal efficiency around 3-7% and 1-2% respectively. The quantity of NO_x was reduced 5-10%.

The objectives of the work were to study the ignition timing in a natural gas engine. And also to find the tendency of engine efficiency in different intake valve closures and injection timings. In this research, the effects of injection timing on the efficiencies and emissions will be studied under the compression ratio of 9, speed of 1500 rpm, 2000 rpm and 2500 rpm with the equivalence ratio of 1.0.

2. Experimental apparatus and method

A diesel engine was dedicated for using with natural gas by modifying the pistons. Compression ratio has been reduced to 9 : 1. Fuel pump and fuel injectors are replaced by spark plugs. The engine was installed to an eddy current dynamometer. This experiment was mainly to compare the differences among three intake valve closures. Notify that the intake valve opening time and exhaust valve timing were not changed. Changing camshaft profiles was the way to this experiment. Each camshaft was also tested in various loads. Every load, four different injection timings were tested to achieve the objective. In each injection time, many ignition timings were tested to find the MBT. A diesel engine (Daedong 4A220A-S1) was totally dedicated to natural gas engine. The pistons were redesigned from the diesel compression ratio of twenty-two to the compression of nine as shown in figure 1. Diesel pump and injectors were replaced by spark plugs. Table 1 shows the dedicated engine specification. Figure 2 demonstrates the Daedong 4A220A-S1 natural gas diesel engine located on the dynamometer and attached with several sensors in the engine test laboratory.

This research was focusing on the intake valve timing and injection timing. Therefore, three camshafts are used



(a) Compression ratio of 22 (b) Compression ratio of 9

Fig. 1. The original and modified pistons.

Table 1. Engine specifications.

Item	Natural Gas Diesel Engine (dedicated engine)
Type	4-cylinder, 4-stroke engine
Displacement	2,197 cc.
Bore (mm.)	87
Stroke (mm.)	92.4
Compression Ratio	9.0
Fuel Supply System	Gas Injectors

Table 2. MBT timing at 25% load.

25% load	1500 rpm.	2000 rpm.	2500 rpm.
Camshaft No.1	24° - 27°	27° - 30°	30° - 33°
Camshaft No.2	24° - 30°	30° - 33°	33° - 42°
Camshaft No.3	33° - 39°	33° - 36°	51°

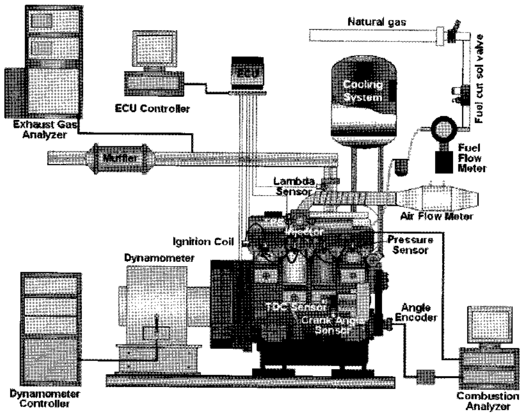


Fig. 2. Schematic diagram of the test engine.

for giving three different valve timings. Each valve timing was tested in 25%, 50% and 100% loads. The speeds of 1500, 2000 and 2500 rpm are experimented in each load. Three different injection timings are tested in every speed. MBT was found by changing the ignition timing. The compression ratio of nine and equivalent air-fuel ratio of 1 are the test condition. The ignition timings were varied between 15 and 54 degree BTDC with the interval of 3 degrees. The first camshaft was installed in the natural gas diesel engine. All the sensors were connected. The engine then started warming up until the cooling water temperature reached 80°C. In the main area of MOTEC ECU control screen, injection timing and ignition timing can be inserted.

Before collecting the data in each ignition timing, the engine must be running under the condition of equivalent air-fuel ratio of 1 (air-fuel ratio of 16.83). So adjusting the amount of injected natural gas was needed. The dynamometer data acquisition system collected the output data such as power, torque, engine speed, temperature, etc. While the exhaust gas analyzer collected the data of CO, CO_2, NO_x, O_2, and THC.

Camshaft no.1, intake valves start opening at 8 degree BTDC during the exhaust stroke. The maximum lift is at 103.5 degree ATDC in the intake stroke. The intake valves close 35 degree ABDC. Camshaft no.2, intake valves start opening at 8 degree BTDC during the exhaust stroke. The intake valves close at 51 degree ABDC (16 degree later than camshaft no.1). Therefore, the maximum valve lift period is between 103.5 and 119.5 degree ATDC in the intake stroke. Camshaft no.3, intake valves start opening at 8 degree BTDC during the exhaust stroke. The intake valves close at 77 degree ABDC (42 degree later than camshaft no.1). Therefore, the maximum valve lift period is between 103.5 and 145.5 degree ATDC in the intake stroke.

3. Results and discussions

The ignition timing with gaseous fuel operation is perhaps the most important adjustment that can be made to accomplish best engine performance. Ignition timing affects nearly all the major operating parameters that include specific fuel consumption, power output, efficiency and tendency to knock. Knocking occurred in this experiment when the ignition timing was too early. This occurred in camshaft no.3 at ignition timing 45° BTDC, the speed of 1500 rpm and 25% load. This situation is presented in table 3 to 5. MBT timing obviously depends on valve timing and speed, referring to the results. MBT timing also slightly depends on load. Nevertheless, there was not any evidence that injection timing affected the change in MBT timing. The following tables show the MBT timing in degree BTDC. As speed increases, the spark must be advanced to maintain optimum timing because the period of the combustion process in crank angle degree increases. Comparing among three camshafts, later intake valve closure needs more advanced ignition timing. Lower load, especially at low speed, also desires more advanced ignition timing. CO and O_2 emissions are less than one percent in the exhaust gas. The variation of CO concentration in the exhaust is minimal because CO emission levels are hardly affected by spark advance variation. The highest level of hydrocarbon emission (about 900 ppm), much lower than THC level (1000 ppm – 3000 ppm) for gasoline engines under normal operating conditions, was observed. The MBT ignition timing brings high power output, high brake thermal efficiency and low SFC. This ignition timing is desirable. There is another point of view if the exhaust emissions are considered. Look in figures 6 to 11, these figure are presenting the results of every camshaft at WOT in the speed of 2000 rpm. Camshaft no.1 has MBT of 14.879 kg-m at ignition timing of 24° BTDC. Camshaft no.2 has 14.2165 kg-m at 30° BTDC. In addition, camshaft no.3 has 11.825 kg-m at 33° and 36°

Table 3. MBT timing at 50% load.

50% load	1500 rpm.	2000 rpm.	2500 rpm.
Camshaft No.1	24°	27°	30°
Camshaft No.2	24° - 27°	27° - 30°	33° - 36°
Camshaft No.3	30° - 33°	36° - 39°	45° - 48°

Table 4. MBT timing at WOT.

WOT load	1500 rpm.	2000 rpm.	2500 rpm.
Camshaft No.1	21° - 24°	24° - 27°	30° - 33°
Camshaft No.2	24°	27°- 33°	33°
Camshaft No.3	33°	33°- 36°	42°- 45°

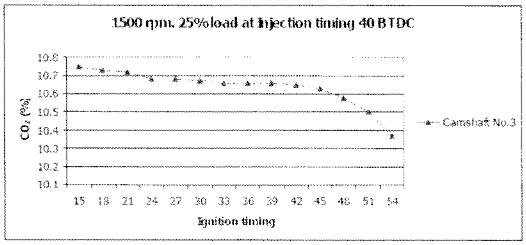


Fig. 3. CO_2 Concentration according to the knocking.

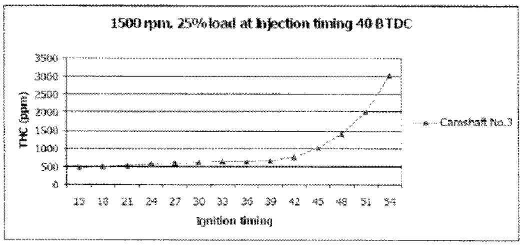


Fig. 4. THC concentration according to the knocking.

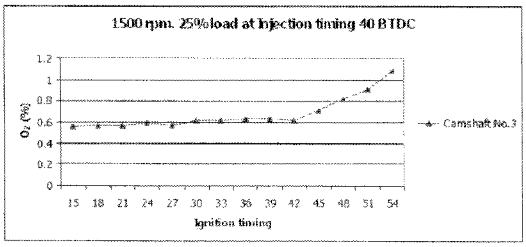


Fig. 5. THC concentration according to the knocking.

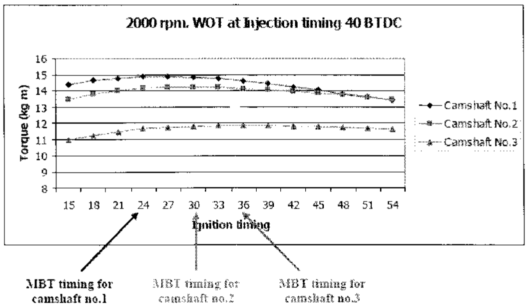


Fig. 6. MBT at 2000 rpm and WOT versus Torque.

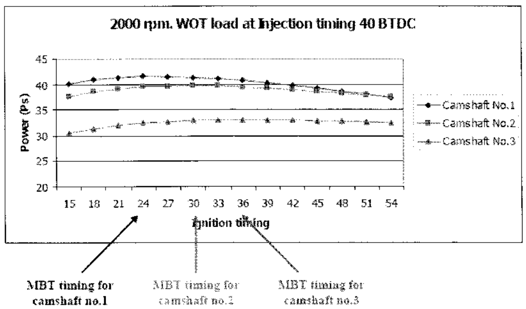


Fig. 7. MBT at 2000 rpm and WOT versus Power.

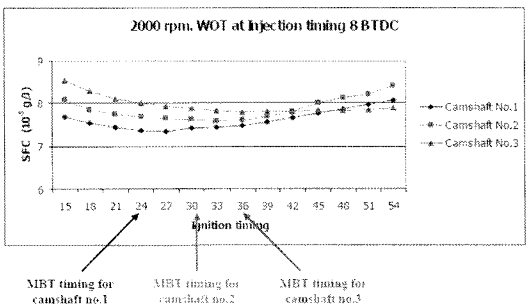


Fig. 8. MBT at 2000 rpm and WOT versus SFC.

BTDC.

This section is to show the benefit of choosing a little retard ignition timing from MBT timing based on the data in figures 6 to 11. Tables 5 to 7 show the output of using MBT ignition timing comparing to 3-degree and 6-degree retard.

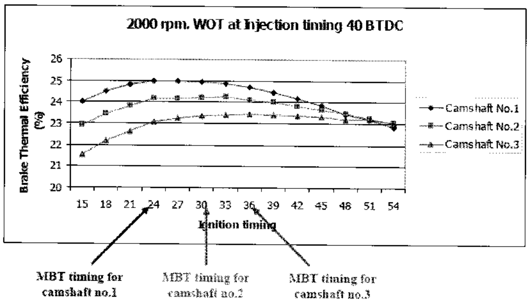


Fig. 9. MBT at 2000 rpm and WOT versus Brake Thermal Efficiency.

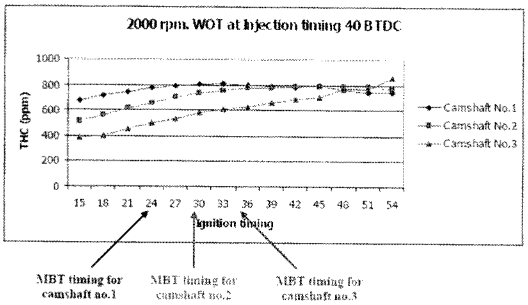


Fig. 10. MBT at 2000 rpm and WOT versus THC.

Tables 5 shows that the ignition timing for camshaft no.1 at WOT and 2000 rpm should be 21° BTDC because brake thermal efficiency reduces around 0.6% but the THC and NOx emissions reduce 4.12% and 11.165% respectively, while the ignition timing of 18° BTDC is not recommended because the brake thermal efficiency reduces up to 2%.

Table 6 recommends the ignition timing of 24° BTDC because the loss in the efficiency is very similar to the ignition timing of 27° BTDC, while it can reduce THC and NO_x emissions 10.84% and 14.49% For camshaft no.3, table 4-6 shows a big difference between two

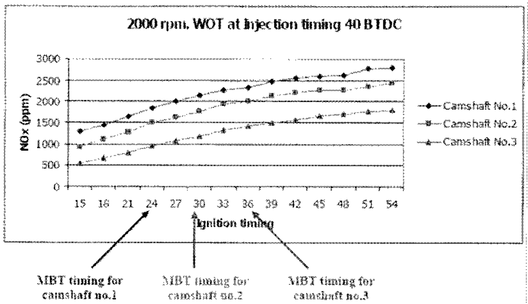


Fig. 11. MBT at 2000 rpm and WOT versus NO_x.

Table 5. Comparison between MBT ignition timing and retard ignition timing for camshaft no.1.

Camshaft no.1	24° BTDC (MBT timing)	21° BTDC	Difference percentage (%)	18° BTDC	Difference percentage (%)
Torque (kg-m)	14.88	14.77	- 0.74	14.63	- 1.67
Power (Ps)	41.56	41.26	- 0.73	40.87	- 1.66
Thermal Eff.	24.99	24.83	- 0.63	24.48	- 2.02
SFC(E-05) (g/J)	7.34	7.39	0.64	7.49	2.06
THC (ppm)	775.00	743.00	- 4.13	714.00	- 7.87
NO_x (ppm)	1845.00	1639.00	-11.17	1446.00	-21.63

Table 6. Comparison between MBT ignition timing and retard ignition timing for camshaft no.2.

Camshaft no.2	30° BTDC (MBT timing)	27° BTDC	Difference percentage (%)	24° BTDC	Difference percentage (%)
Torque (kg-m)	14.22	14.21	- 0.08	14.19	- 0.21
Power (Ps)	39.71	39.67	- 0.10	39.62	- 0.23
Thermal Eff.	24.21	24.16	- 0.24	24.16	- 0.23
SFC(E-05) (g/J)	7.58	7.60	0.24	7.60	0.23
THC (ppm)	738.00	703.00	- 4.74	658.00	- 10.84
NO_x (ppm)	1759.00	1613.00	-8.30	1504.00	-14.50

Table 7. Comparison between MBT ignition timing and retard ignition timing for camshaft no.3.

Camshaft no.3	33° BTDC (MBT timing)	30° BTDC	Difference percentage (%)	27° BTDC	Difference percentage (%)
Torque (kg-m)	11.83	11.81	- 0.16	11.73	- 0.79
Power (Ps)	33.02	32.93	- 0.28	32.71	- 0.93
Thermal Eff.	23.42	23.37	- 0.22	23.25	- 0.71
SFC(E-05) (g/J)	7.84	7.85	0.22	7.89	0.72
THC (ppm)	606.00	582.00	- 4.00	537.00	- 11.39
NO_x (ppm)	1308.00	1174.00	-10.24	1081.00	-17.35

ignition timings. If the ignition timing of 30° BTDC is chosen, brake thermal efficiency reduces 0.277% while THC and NO_x emissions reduce 4.00% and 10.24% respectively. Otherwise, the brake thermal efficiency reduces 0.71% while THC and NO_x emissions reduce up to 11.39% and 17.35% respectively. However, The levels of emissions from camshaft no.3 are relatively low. Therefore, the MBT timing (33° BTDC) or 30° BTDC should be appropriate.

4. Conclusions

This study provides results the of spark ignition natural gas diesel engine (2.2 liters, 4-stroke-4-cylinder engine). The following conclusions have been reached:

1) For gaseous indirect injection system, the injection timing has less influence on the engine performance than load, speed, valve timing and ignition timing.
2) The ignition timing of camshaft no.1 and 2 should be retarded around 3°CA (Crank Angle) and 6°CA respectively. Since the THC and NO_x emissions can decrease up to 10.84% and 14.5% respectively while camshaft no.3 should go with MBT timing to maintain high brake thermal efficiency.
3) This can compare the fuel flow rate, efficiencies and emissions, which can show another view about economics. Moreover, it may lead to show the possibility that natural gas diesel engine can give out more power than the unmodified engine.

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