Final Report

Control of Exhaust Emissions from Small Engines Using E-10 and E-85 Fuels

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Executive Summary

A single-cylinder, air cooled, four-stroke, spark ignition engine with a carburetor was used to compare exhaust emissions produced by gasoline and blends of gasoline and ethanol. Three fuels were used in the tests: 87 Octane unleaded gasoline (E-0), 90% gasoline and 10% ethanol (E-10) and 15% gasoline and 85% ethanol (E-85). The engine fuel system was modified to handle flow rates of the three fuels. A variable size-metering orifice was used to control mixture air-to-fuel ratio. The ignition system was modified to allow the operator to change ignition timing. Tests were conducted at several engine speeds and engine loads ranging from 100% to light load. The exhaust emissions were measured by a Nicolet FT-IR emissions system calibrated for raw exhaust gases from spark ignition engines.

The results showed that the E-10 and E-85 fuels improved energy conversion efficiency but the specific fuel consumption increased when the engine was run on E-85. At stoichiometric air-to-fuel ratio the CO emissions improved slightly when E-10 and E-85 fuels but the improvement in specific CO levels was small. Even the modified, controlled fuel system experienced variations in air-to-fuel ratio which impacted CO emissions. Compared to E-0 the specific HC emissions increased when the engine was run on E-85 at the OEM spark timing. The E-10 fueled showed a smaller increase.

Relative to E-0 the E-85 and E-10 fuels produced lower specific emission levels of NO at the OEM spark timing, the better results were from the E-85 fuel. An examination of the engine revealed combustion chamber deposits after the engine was run on E-10 and E-85 fuels for a prolonged period of time. The use of IR Fuel-conditioning device in the fuel system improved energy conversion efficiency of the engine but had adverse effects on HC and NO emissions at the near rated speed. The CO emissions were not much affected by the device.

Both, the E-10 and E-85, fuels improved CO and NO emissions when the engine was subjected to cycle B tests but the HC emissions increased in relation to the base fuel, E-0. Cycle A tests showed somewhat similar qualitative results.

1. Introduction

Alcohol based fuels have received renewed consideration for use in internal combustion engines because of their oxygenated characteristics and their potential to reduce exhaust emissions. Ethanol has long been considered a good spark ignition engine fuel and engines were run on ethanol very early in their development process. Pure ethanol has very low vapor pressure (about 17 kPa compared to 60 to 90 kPa for gasoline) but when mixed with gasoline the vapor pressure of the blend increases until the volume percent of ethanol increases beyond 30% or so. Ethanol has other physical and chemical characteristics that are important when considering it as a fuel for spark ignition (SI) engines. Its heating value is lower than that of gasoline, both on mass basis (27 MJ/kg Vs 44 MJ/kg) as well as on volume basis (21.3 MJ/l Vs 32 MJ/l). This would require higher fuel flow rate into the engine to produce the same power output as a spark ignition engine assuming the fuel conversion efficiencies are identical with the two fuels. The heat of vaporization of ethanol is higher, 920 kJ/kg Vs approximately 350 kJ/kg for a typical gasoline, meaning that a fuel-air mixture inducted into the engine will have significant charge cooling effect and may impede fuel vaporization. Ethanol is generally blended with gasoline to assist in cold start of spark ignition engines. Ethanol blended fuels have been claimed to reduce exhaust emissions, more specifically of carbon monoxide because of the inherent presence of oxygen in the fuel molecule. Most of these claims have been made for vehicle engines where they are tested on a specified cycle that includes continuous changes in engine load and speed. Even though the fuel systems in vehicles are better controlled they still experiences frequent changes in air-to-fuel ratio during the test cycle. The impact of ethanol blends on exhaust emissions from vehicle engines may not be the same as that on off-highway engines. In the latter case the fuel system is not well controlled and may use carburetion type of fuel system. This certainly is the case in small off-highway engines that used in domestic and industrial applications. These engines may operate at constant or variable speed and at constant or variable load. However, in most of the cases the engines are not subjected to rapid changes as those experienced by vehicle engines.

Very little is known about the effects of ethanol blends on exhaust emissions produced by small engines with carburetion fuel system. This project was undertaken to address the lack of information in this arena.

2. Experimental Set Up

The experimental work was conducted on a Honda single cylinder, air-cooled spark ignition engine. The engine had a horizontal shaft, overhead valve design with a carburetor type fuel system.

2.1 The Engine and Dynamometer

The engine specifications are given in Table 1. A water-brake dynamometer, with a precision water pressure control, was used to load the engine. The water flow rate was changed by a fine metering valve in order to change the engine load or speed. A direct coupling was used between the engine and the dynamometer. The engine had its own ignition system that comprised of CDI. However, the operator had no control over ignition timing and the timing variation was limited to 18- 22 deg. (before top dead center) based on engine speed. The system also proved to be unreliable when the engine was operated on E-85 fuel. For these reasons the OEM ignition system was replaced by an externally triggered system operated from the shaft connecting the engine and the dynamometer. The new system incorporated the traditional ignition coil and breaker point arrangement similar to that found in off-highway spark ignition engines.

Table 1
Engine Specifications

Bore	82 mm
Stroke	64 mm
Compression Ratio (modified)	8.2 : 1
No of Cylinders	1
Maximum Recommended Speed	3400 rpm
Cooling System	Air-cooled
Ignition Timing (modified)	variable

The engine-dynamometer test rig was instrumented with a load cell, crankshaft encoder, temperature sensors and actuators. A volumetric fuel flow measurement system was employed to measure the fuel flow rate into the engine. The original air induction system was modified to incorporate a hot film airflow sensor and a surge tank was used between the sensor and the intake manifold to reduce flow measurement variations caused by pressure pulsations.

2.2 Exhaust Gas Analysis

The original engine exhaust system was modified to incorporate positive exhaust flow using the test cell exhaust tunnel system. A wide-range exhaust oxygen sensor and gas temperature probes were installed in the exhaust system. The oxygen sensor was calibrated for E-0, E-10 and E85 fuel using standard exhaust has analysis equipment. Samples of exhaust gas were withdrawn from the engine exhaust pipe at about 150-mm downstream of the exhaust valve. A Nicolet FT-IR exhaust emissions apparatus was used to analyze concentrations of carbon monoxide (CO), nitric oxide (NO) and total hydrocarbons (THC) in the engine exhaust. A heated sample line, maintained at 165 C, was used to withdraw samples on exhaust on a continuous basis. A 2-m sample cell was used to analyze several gaseous species including those of interest in this work. The cell, the inlet manifold and the inlet lines were all kept at the recommended temperature of 165 C. The samples were scanned at a rapid rate to get almost a continuous output of the concentrations of the three exhaust species.

The Nicolet system was calibrated with known concentration of CO, NO and methane and propane hydrocarbons. Unlike the flame ionization detection (FID) system the FT-IR analyzer detects concentrations of individual hydrocarbons. Total hydrocarbon level in the engine exhaust is determined by summing up the hydrocarbons detected by the system. It is quite possible for the concentrations of hydrocarbons detected by the FT-IR system to be different than those measured by a FID analyzer. In this report we have compared the relative levels of hydrocarbons produced by the engine using the same base hydrocarbon species. This yields relative hydrocarbon emissions when the engine was fueled by gasoline (E-0), 10% ethanol and 90% gasoline (E-10) and 85% ethanol and 15% gasoline (E-85).

2.3 The Fuel System

The original fuel system was made up of a single barrel, fixed metering jet carburetor with a manual choke, a fuel tank and a fuel valve. Since the energy content of E-10 and E-85 are different than the baseline fuel (gasoline or E-0) it was necessary to modify the carburetor so that it can supply appropriate fuel flow rate to the engine. The modifications included the replacement of the main carburetor jet by a variable orifice jet, a stainless steel fuel cell with a fuel filter and a fuel flow measuring system. To avoid swelling of seals and fuel leaks the fuel lines, connectors and carburetor seals were made up of materials that were compatible with gasoline and ethanol fuels.

Fuel blends with appropriate concentrations of ethanol were prepared in the fuels laboratory using standard grade gasoline and laboratory grade ethanol. Care was taken to insure homogeneity of the mixture before and during the tests.

The last modification of the fuel system involved the use of near-IR fuel conditioner. The conditioner is claimed to alter fuel molecule configuration thereby improving combustion and exhaust emissions. The fuel conditioner is a small filter like element that sits very close to the carburetor. Fuel delivery lines capable of transmitting infrared radiation to fuel were used to make the conditioner work. The fuel conditioner was installed about 100 mm upstream of the carburetor.

3. Testing Procedure

All tests were conducted at steady state conditions after the engine had reached its operating temperature. The engine was set at the desired speed and the load was varied from light to near full load and vice-versa. The full load was limited to the lowest possible full load that could be achieved with the three fuels. Since the carburetor system cannot maintain identical air-to-fuel (A/F) ratio for a given set of metering orifice size it was decided to test the engine for emissions when the A/F ratio was stoichiometric (or chemically correct A/F ratio). The wide range oxygen sensor together with its digital display was used to adjust the carburetor main jet to set the A/F ratio. Even with this precision system the A/F ratio did vary, the variation was dependent on load and, to a lesser extent, on the engine speed. The results discussed in this report pertain to stoichiometric A/F ratio. Comparison of exhaust emissions with the three fuels can only be done at this specific A/F ratio. The test data was collected for one minute and averaged over the time. To improve confidence level the tests were repeated three times for each speed, fuel and original ignition timing.

The engine's original ignition timing was18 – 22 CA degrees before top dead center (BTDC), a retarded setting compared to MBT timing for this engine. MBT timing provides improved torque and better fuel conversion efficiency but can impact exhaust emissions. Tests were conducted to evaluate exhaust emissions at MBT timing with the three fuels.

The use of near-IR fuel conditioner requires that the fuel lines and system be conditioned for several hours before tests. The engine was run at different loads and speeds for over 20 hours before tests were conducted with the fuel conditioner. The testing procedure was no different than that was used without the conditioner and all tests were conducted at the original ignition timing.

4. Results

The off-highway engine testing procedure specifies testing at different engine speeds and loads depending on application of the engine. To cover two major engine applications tests were conducted at about 85% maximum engine speed (approximately 2800 rpm in this case) and two intermediate speeds, 2400 rpm and 1800 rpm. The owner's manual recommends that the engine be used for applications at or above 2000 rpm. However, the engine is capable of operating at 1800 rpm; it was found to be less stable at low speeds and low loads.

4.1 OEM Spark Timing Operation for Cycle B

Figure 1 shows specific energy consumption of the engine on three different fuels. Since ethanol blends have different energy contents than gasoline it is more convenient to compare engine performance on an energy basis than on mass basis. Over most of the engine load the difference in energy consumption between pure gasoline (E-0) and gasoline with 10% ethanol (E-10) is very small. But the 85% ethanol blend (E-85) does produce improved specific energy consumption, as shown in the figure.

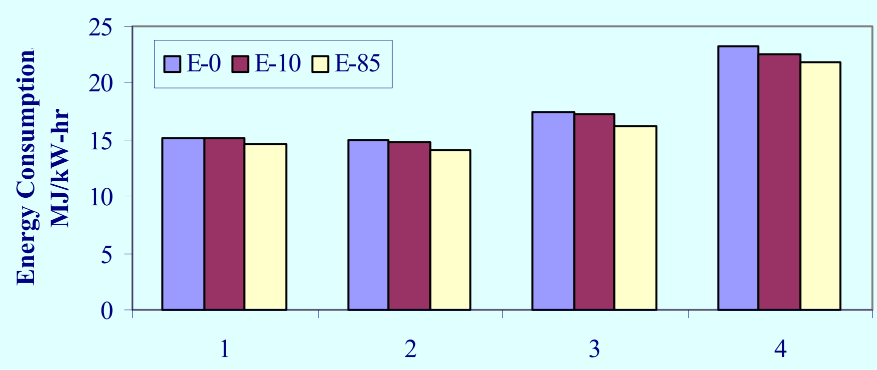

Fig. 1 Energy Consumption Comparison at 2800 rpm (OEM Spark Timing)

The effects of base fuel and ethanol blends on carbon monoxide (CO) emissions are shown in Figure 2. Carbon monoxide emissions are very sensitive to mixture A/F ratio and its variations. Unlike fuel injection systems, the mixture produced by carburetors can have large variations in A/F ratio. In the present fuel system, which comprised of a modified carburetor, the A/F ratio was maintained at near constant value (stoichiometric value corresponding to the fuel used) by a fine adjustment of the main metering jet. Even then, the A/F ratio was found to vary by as much as +/- 4%, depending on engine load and speed. This produced measurable changes in exhaust CO levels. The comparative levels shown in Figure 2 are based on averages of CO emissions over at least one minute of scan. E-85 fuel showed a

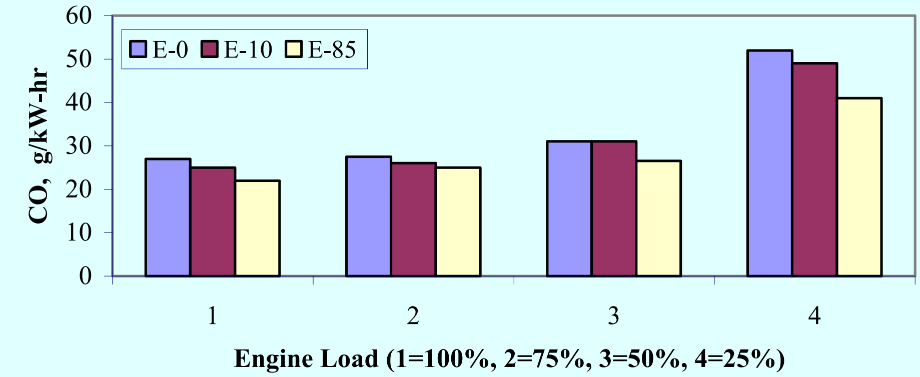

Fig. 2. CO Exhaust Emissions Comparison at 2800 rpm (OEM Spark Timing)

small improvement in CO levels over most of the operating range. Unlike an uncontrolled fuel system the present test method accounted for the presence of oxygenated fuel in the blend. The tight control on A/F ratio in this experimental set up resulted in lower CO levels than those realized in a typical off-highway, small SI engine. The test engine, with its original unmodified fuel system, produced about 250 to 400% more CO than those shown in Figure 2 depending on engine load and operating speed.

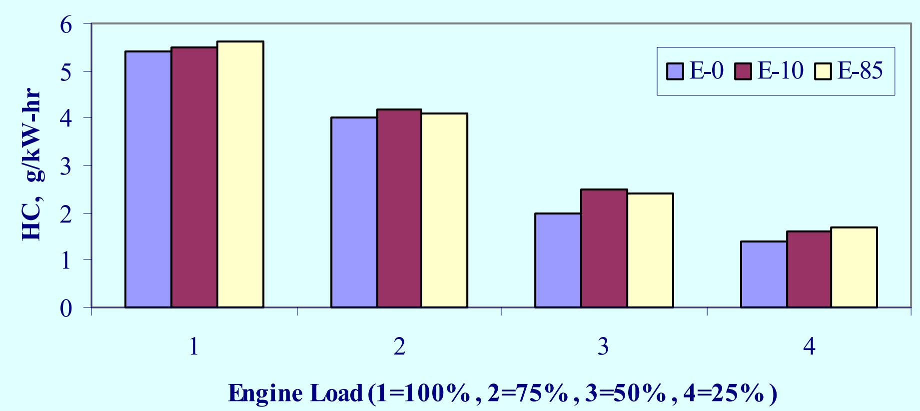

Fig 3. HC Exhaust Emissions Comparison at 2800 rpm (OEM Spark Timing)

The hydrocarbon (HC) levels, in general, showed an opposite trend with the alcohol blends, as shown in Figure 3. The specific emissions of HC, in general, were higher with either E-10 or E-85 compared to E-0. Some tests, conducted at low ambient temperatures, showed a much higher increase in specific HC emissions. Ethanol and its E-10 and E-85 blends have higher heat of vaporization than gasoline. Consequently, vaporization of ethanol blends requires more heat input than needed to vaporize the same mass of gasoline. Inadequate vaporization of the fuel can lead to an increase in HC emissions. A portion of the fuel that remains in liquid form as it enters the cylinder can lead to increased emission of HC.

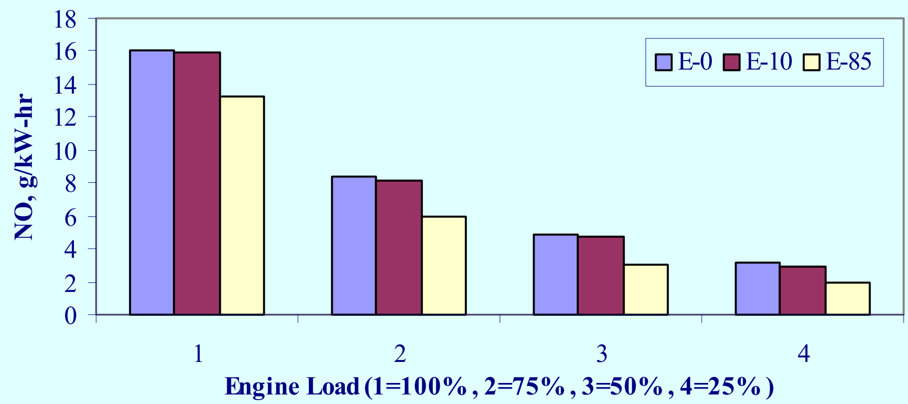

Fig. 4. NO Exhaust Emissions Comparison at 2800 rpm (OEM Spark Timing)

Figure 4 shows exhaust emissions of nitric oxide (NO) in the engine exhaust. The NO levels decreased as the ethanol content in the blend increased. The reduction in NO showed a consistent pattern at all loads and engine speeds when the original equipment manufacturer's (OEM) ignition timing was employed. Two factors would contribute to NO reduction in this SI engine: (a) charge cooling resulting in lower inlet mixture temperature, and (b) slower flame spread of E-85 fuel compared to E-0. The higher heat of vaporization of E-85 produced a reduction in inlet charge temperature of as much as 14 C compared to the mixture formed by E-0 at the identical load and speed.

4.2 Exhaust Emissions on Cycle B

The exhaust concentrations of CO, HC and NO measured at the continuous rated speed of 2800 rpm was used to estimate exhaust emission as specified under cycle B. This cycle is specified for small, non hand-held engines that power mechanical devices operating near constant speed. Since the fuels used in the comparative study are different than gasoline, a modified scale was used to estimate exhaust emissions on cycle. A weight value was assigned to different loads, as suggested and used by several other investigators. The weight assigned was as shown in Table 2. Figure 5 shows exhaust emissions of the three chemical species over this cycle. It shows that the engine operating on E-85 would emit lower NO and a slightly lower CO than the base gasoline fuel but the HC emissions would be higher. The current regulations require NO + HC levels to be at or below the set value. The results show that the engine would emit lower NO + HC when operated on E-85 compared to E-0 operation. However, there was hardly any difference in NO + HC emissions between E-10 and E-0 fuels.

Table 2
Weight Factors for Different Loads

Load, %	Modified Weight Factor
100	10%
75	22%
50	30%
25	32%
10	6%

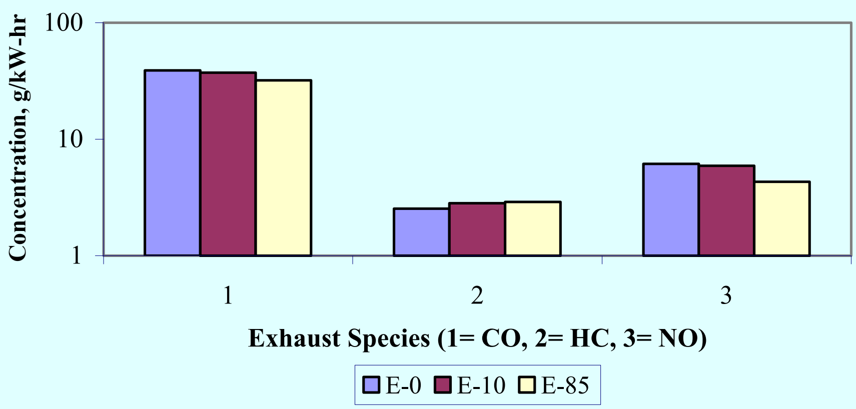

Fig. 5 Exhaust Emissions on Cycle B

The fuel consumption, on energy and volume basis, is shown in Table 3 if the engine is operated on cycle B and OEM spark timing.

Table 3
Energy Consumption on Cycle B

	E-0	E-10	E-85
Energy Consumption, MH/kW-hr	19.0	18.65	17.86
Energy Consumption, liter/kW-hr	0.598	0.609	0.77

The engine was more efficient when operated on E-85 fuel but on volumetric basis the fuel consumption was higher due to the lower energy content of E-85 compared to E-0 or E-10 fuels.

4.3 Operation at Intermediate Speeds

The intermediate speed testing is specified for non hand-held engines that may power devices such as mowers, tillers, etc. that operate at varying speeds and loads. The cycle A emissions testing is considered to be appropriate for such engines with weight factor same as those shown in Table 2. Two intermediate speeds were selected for tests: 2400 rpm and 1800 rpm. The selection of these speeds was based on application of the test engine. The results of exhaust emissions and energy consumption at different loads are shown in Appendix A when the engine was operated at intermediate speeds at OEM spark timing.

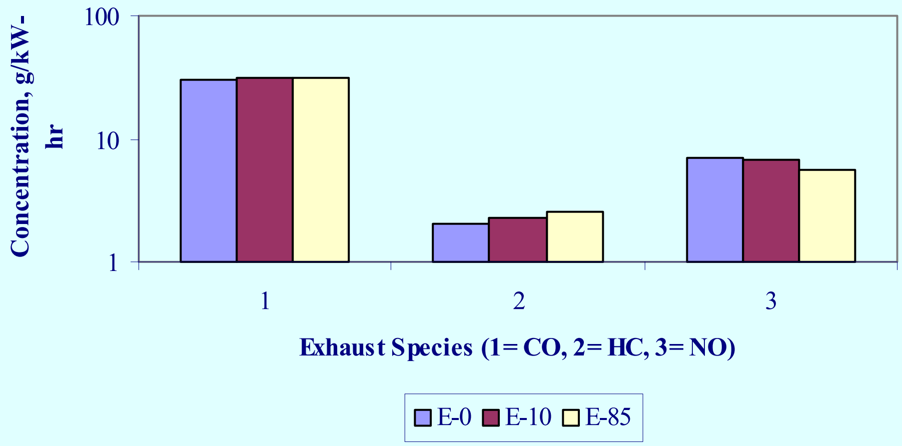

Fig. 6 Exhaust Emissions on Cycle A, 2400 rpm

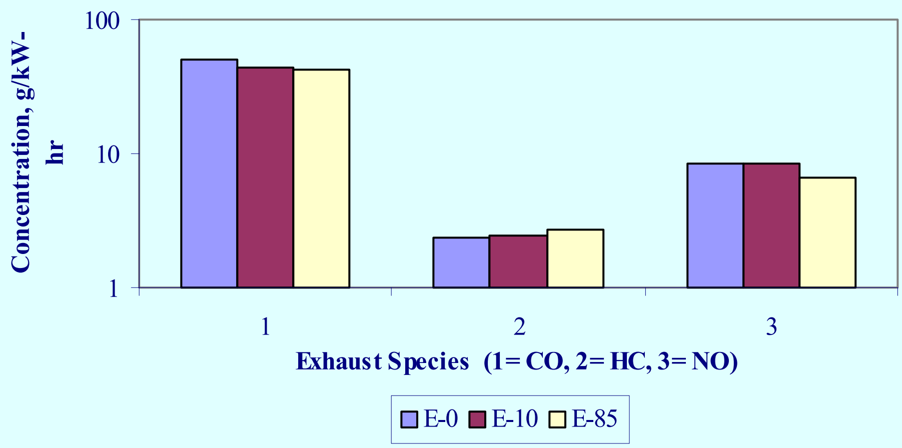

Fig. 7 Exhaust Emissions on Cycle A, 1800 rpm

A comparison of exhaust emissions for cycle A are shown in Figures 6 and 7 for engine operation at 2400 rpm and 1800 rpm, respectively. The use of E-85 reduced NO emissions over the cycle at both the engine speeds. Not much difference was found in CO levels between the three fuels. As stated before, the mixture A/F ratio was regulated except for the inherent variations encountered in a carburetion system. Depending on the extent of these variations the engine could produce more or less CO in the exhaust. Figures 6 and 7 show that the differences in CO levels between the three fuels over the cycle were small.

The HC levels increased when the engine was fueled on E-85. The increase is evident in Figure 6 and 7 when comparisons are made with the corresponding emissions levels with E-0 and E-10. The trend in exhaust HC emissions on cycle A at the intermediate speeds is very similar to that observed on cycle B at the near maximum continuous rated speed.

4.4 Engine Tests at MBT Spark Timing

The test engine had near fixed spark timing that varied between 18 and 22 deg. BTDC. The OEM ignition system was changed to allow the operator to vary ignition timing. Tests were

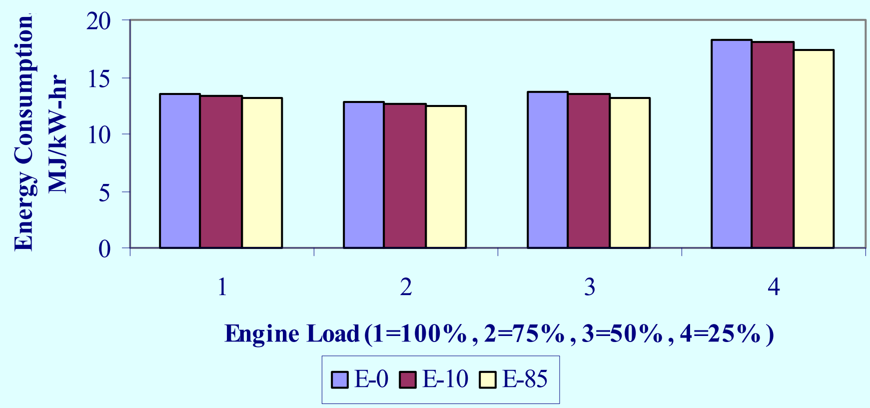

Fig. 8 Specific Energy Consumption, 2800 rpm (MBT or Adv Spark)

Conducted to assess the impact of MBT ignition on fuel economy and exhaust emissions. The spark timing was varied until the least advanced timing was achieved for the maximum torque for a given setting of the engine throttle. The A/F ratio was adjusted to achieve near stoichiometric operation. When the engine was operated on E-0 and E-10 fuels the MBT timing produced a low intensity knock. In such cases the timing was retarded until the audible knock was eliminated. This is discussed later in the report under combustion chamber deposits.

Figure 8 shows the specific energy consumption at different loads and fuels at 2800 rpm. Comparison of engine energy consumption at OEM spark and MBT spark, figures 1 and 8 respectively, shows that MBT timing produced better engine fuel economy over most of the operating range. The difference is more prominent at lighter loads. Since knocking was not experienced on E-85 fuel it would be possible to improve on the fuel economy values shown in Figure 8 by increasing the engine compression ratio. The MBT timing for E-85 fuel was a few crank angle degrees advanced compared to E-0 fuel when the engine was delivering high loads, typically 100 to 75%. Not much difference in MBT timing was observed at lower engine loads.

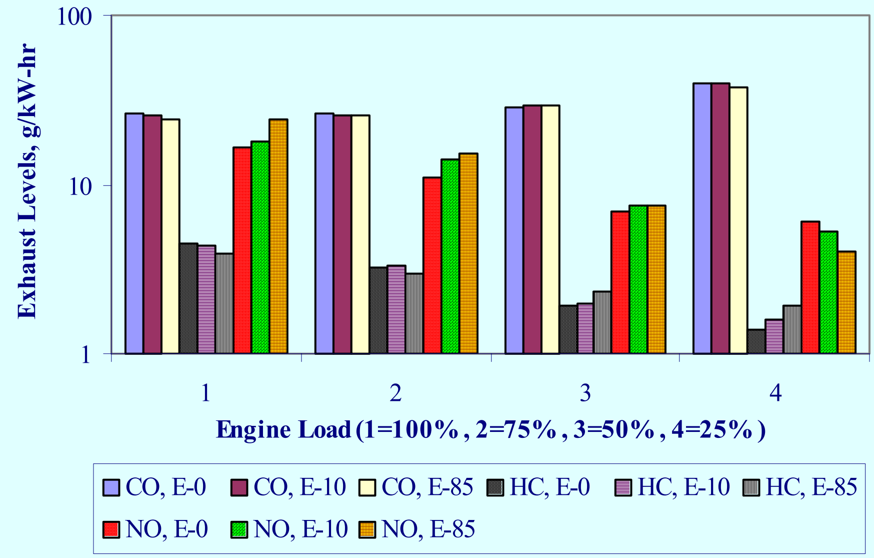

Fig. 9 CO, HC and NO Emissions at 2800 rpm (MBT or Adv Spark)

The results of exhaust emissions are shown in Figure 9 when the spark was set at MBT or advanced relative to the OEM setting. The trend in CO emissions is similar to that observed in previous tests. However, the HC emissions decreased with E-85 fuel in relation to E-0 fuel at high engine loads. This is believed to be the result to increased residence time for the hydrocarbons to oxidize in the cylinder prior to the opening of the exhaust valve. At high loads the spark timing for E-85 was at MBT while those for the E-0 and E-10 fuels had to be retarded in relation to their MBT values because of knock (their timing was still advanced relative to the OEM setting). The difference in MBT timings between the three fuels at lighter loads was insignificant, particularly when the engine load was below 50%. This situation did not help in reducing HC levels with E-85 fuel at part throttle operation, as shown in Figure 9.

The combustion of E-85 fuel in the engine at MBT timing and high loads increased NO levels in relation to those of E-0 and E-10 fuel. E-85 operation had more advanced timing while the E-0 and E-10 had relatively less advanced setting due to the low level knock experienced at high loads. NO levels in SI engines are sensitive to temperature, A/F ratio and spark timing. Since the average A/F ratio was kept about the same the advanced spark timing produced higher NO levels at high loads. The trend in NO emissions at lower loads are similar to those observed at OEM spark timing since the variation in MBT spark timing for the three fuels was very small (knock-free operation). However, the specific values of NO in Figure 9 are higher than those shown in Figure 4 due to different setting for ignition timings.

5 Exhaust Emissions with IR Fuel Conditioner

The fuel economy and exhaust emissions were measured with the IR fuel-conditioning device in the fuel line upstream of the carburetor. The engine was tested at several speeds and loads, as described earlier. Figure 10 shows the effect of the device on energy consumption when E-0, E-10 and E-85 were used to run the engine.

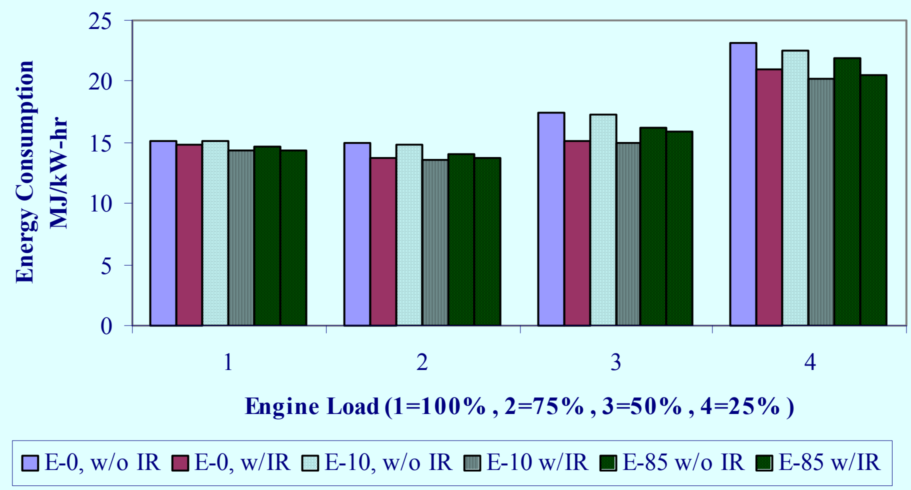

Fig. 10 Specific Energy Consumption with and w/o IR Fuel Device, 2800 rpm (OEM Spark Timing)

For a given load and fuel the device improved energy consumption of the engine. This pattern is evident for all three fuels and at all engine loads although the extent of improvement varied with fuel and load. Similar patterns were observed at 2400 rpm.

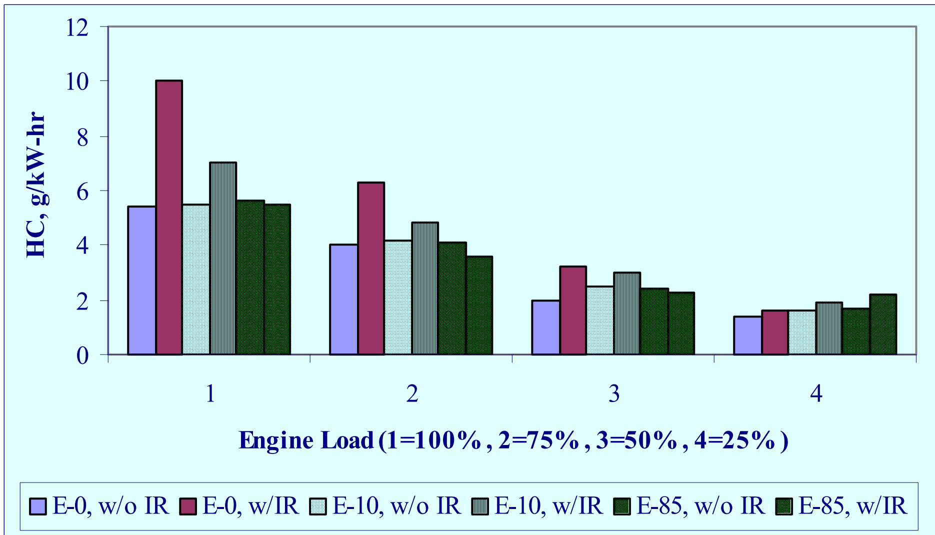

Fig. 11 HC Emissions with and w/o IR Fuel Device, 2800 rpm (OEM Spark Timing)

Figure 11 shows HC emissions with and without the fuel-conditioning device in the engine fuel system. The device adversely influenced HC emissions at all loads and fuels. The impact of the device is more pronounced at high loads and E-0 and E-85 fuels. The HC emissions with E-85 generally increased but the increase was lower than those realized with the E-0 and E-10 fuels.

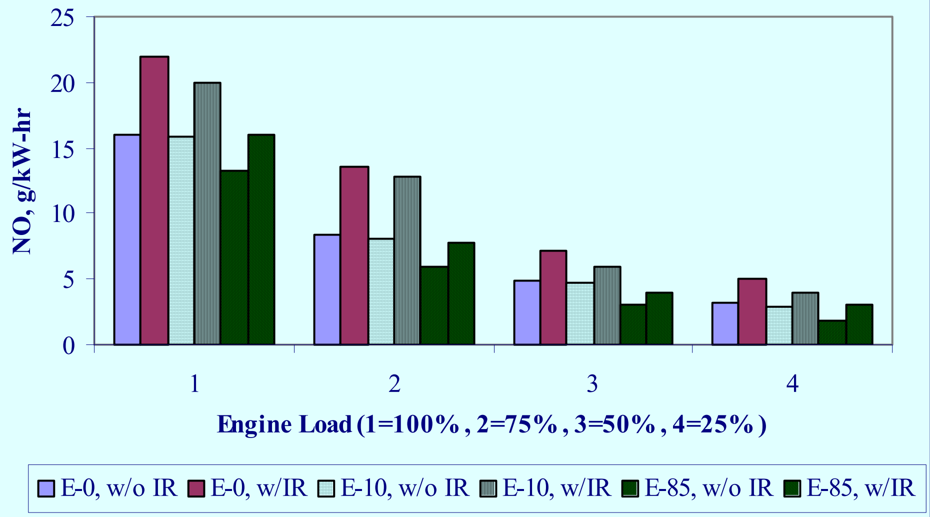

Fig. 12 NO Emissions with and w/o IR Fuel Device, 2800 rpm (OEM Spark Timing)

The presence of the fuel-conditioning device in the system increased NO emissions, as shown in Figure 12. Irrespective of the type of fuel the device contributed to an increase in specific NO emission. Similar differences were found at 2400 rpm.

The IR device did not seem to impact CO emissions much when compared to the same fuel but without the use of the device. Any differences in CO levels with and without the IR device could be attributed to A/F ratio variations in the fuel-air mixture. It is claimed that the IR fuel-conditioner influences fuel molecules thereby improving combustion process. The limited number of tests conducted in the project indicates that the IR fuel-conditioning device improved energy conversion efficiency of the single-cylinder, small air-cooled engine but had adverse effects on HC and NO emissions. More controlled experiments are needed to evaluate the impact of the device on exhaust emissions and fuel economy of E-10 and E-85 fuels.

6 Effects on Combustion Chamber

The engine combustion chamber was examined after the engine break-in but before the use of alcohol blend fuels. Examination of the combustion chamber volume showed a layer of spongy deposits on walls including the piston crown surface. No chemical analysis of the deposits was made to determine its composition. However, combustion chamber deposits are known to affect combustion process and exhaust emissions. The test schedule did not permit sufficient time to determine if the deposits would burn or get destroyed if the engine was reverted back to E-0 operation after it had been run on E-10 and E-85 fuels for a prolonged period of time. Nevertheless, it appears that the use of E-85 (and may be even E-10) in small, air-cooled engines produce combustion chamber deposits that may impact engine performance.

Combustion chamber deposit build up is known to increase HC emissions. The absorption and desorption of hydrocarbons by the surface deposits is believed to be the major cause of increased HC emissions from SI engines. In addition, deposits adversely affect heat transfer through the cylinder walls resulting in increased mixture temperature during combustion. This can promote knocking tendency of the engine.

Engine knock was detected when tests were conducted at MBT spark timing with E-0 (87 Octane unleaded) and E-10 fuels. The spark timing was advanced by as much as 14 degrees with respect to the OEM setting. No knock was detected when the engine was run on E-85 and advanced timing. It is not clear if the knock was the result of advanced timing alone or a combination of timing and combustion chamber deposits. Likewise, the HC emissions increased with E-10 and E-85 fuels. Again, the test program did not allow us to investigate the cause of this increase and whether it is attributable to deposits, poor vaporization, chemical process, etc.

7 Summary

The results of the project can be summarized as follows:

- E-10 and E-85 fuels improved energy conversion efficiency of the small test engine. However, the specific fuel consumption, on mass or volume basis, increased when the engine was run on E-85. This occurs because of the lower energy content of the E-85, which is not compensated by the improved energy conversion efficiency.

- CO emissions improved slightly when E-10 and E-85 fuels were used. However, the improvement in specific CO levels was small. CO levels are very sensitive to A/F ratio and its variations. Although efforts were made to maintain the average A/F ratio at stoichiometric value the inherent variations in A/F ratio in a carbureted engine resulted in large variations in exhaust CO. A small, air-cooled SI engine can produce increased CO emission with E-85 compared to E-0 if the fuel system produces larger variations in mixture A/F ratio.

- Compared to E-0 the specific HC emissions increased when the engine was run on E-85 at the OEM spark timing. The E-10 fueled showed a smaller increase. The higher heat of vaporization of E-85 (and also E-10, to some extent) reduced temperature of the mixture before it entered the cylinder. The lower mixture temperature, the possibility of inadequate vaporization and combustion chamber deposits could have contributed to higher HC emissions from the engine.

- Relative to E-0 the E-85 and E-10 fuels produced lower specific emission levels of NO at the OEM spark timing, the better results were from the E-85 fuel. The mixture temperature decreased as the alcohol portion of the fuel increased. The reduction in NO emissions with E-10 and E-85 could be attributed to several factors, some of them being the reduction in charge temperature and the effect of alcohol on combustion duration.

- Examination of the engine revealed combustion chamber deposits after the engine was run on E-10 and E-85 fuels for over 150 hours. These deposits could have contributed to higher HC emissions as well as the slight engine knock experienced when the engine was operated on E-0.

- The use of IR Fuel-conditioning device in the fuel system improved energy conversion efficiency of the engine but had adverse effects on HC and NO emissions at the near rated speed. The CO emissions were not much affected by the device. More work is needed to evaluate the application of the device to small SI engines.

- Both, the E-10 and E-85, fuels improved CO and NO emissions when the engine was subjected to cycle B tests but the HC emissions increased in relation to the base fuel, E-0. Cycle A showed somewhat similar results but the reduction in CO was limited.

Appendix

Energy Consumption and Exhaust Emissions Results
2400 rpm and 1800 rpm

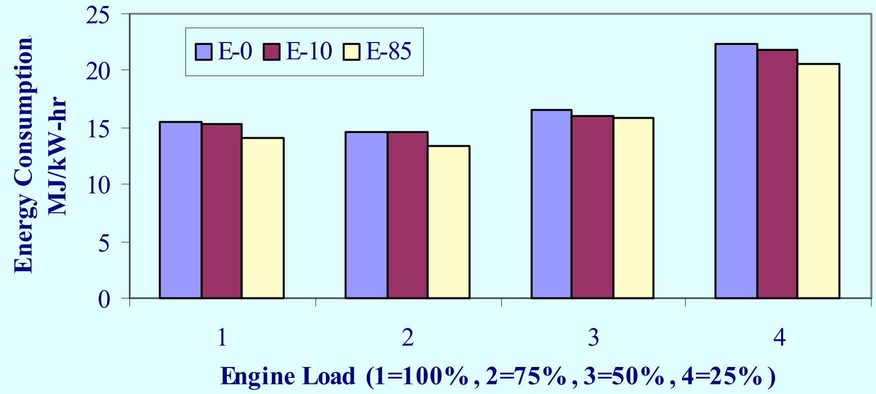

Fig. 13 Energy Consumption Comparison at 2400 rpm (OEM Spark Timing)

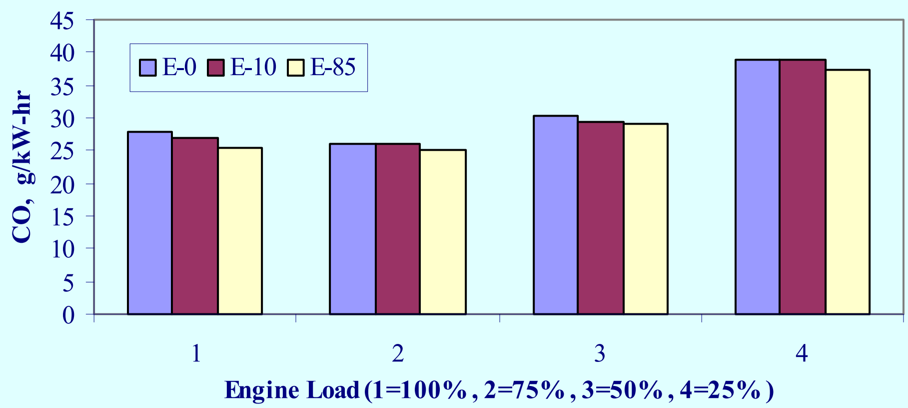

Fig. 14 CO Exhaust Emissions Comparison at 2400 rpm (OEM Spark Timing)

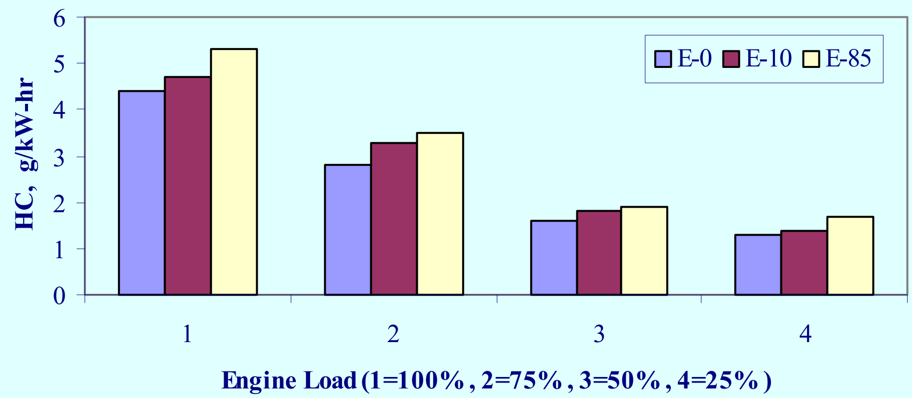

Fig. 15 HC Exhaust Emissions Comparison at 2400 rpm (OEM Spark Timing)

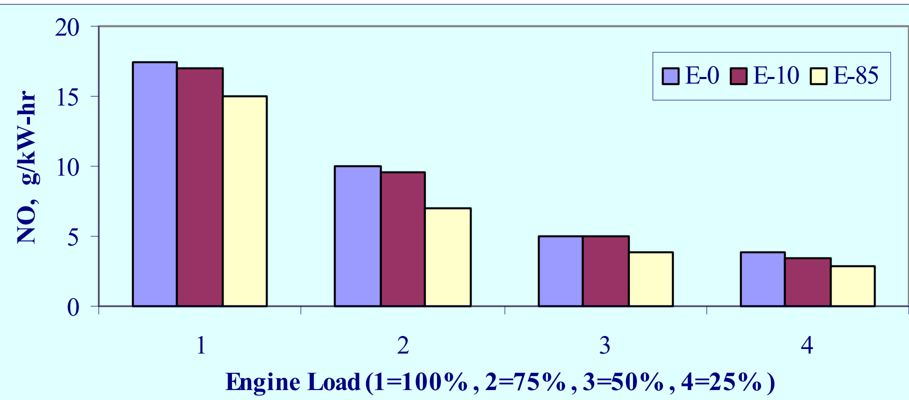

Fig. 16 NO Exhaust Emissions Comparison at 2400 rpm (OEM Spark Timing)

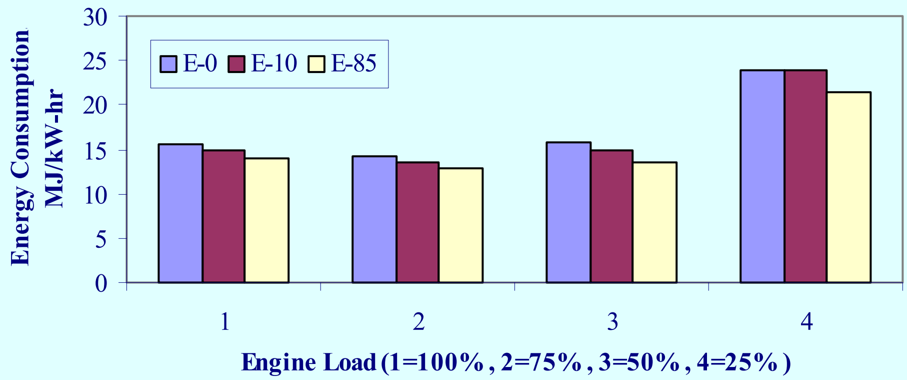

Fig. 17 Energy Consumption Comparison at 1800 rpm (OEM Spark Timing)

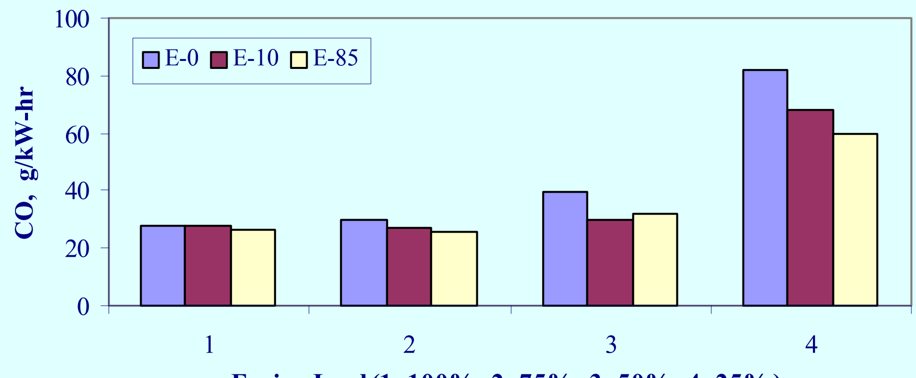

Fig. 18 CO Exhaust Emissions Comparison at 1800 rpm (OEM Spark Timing)

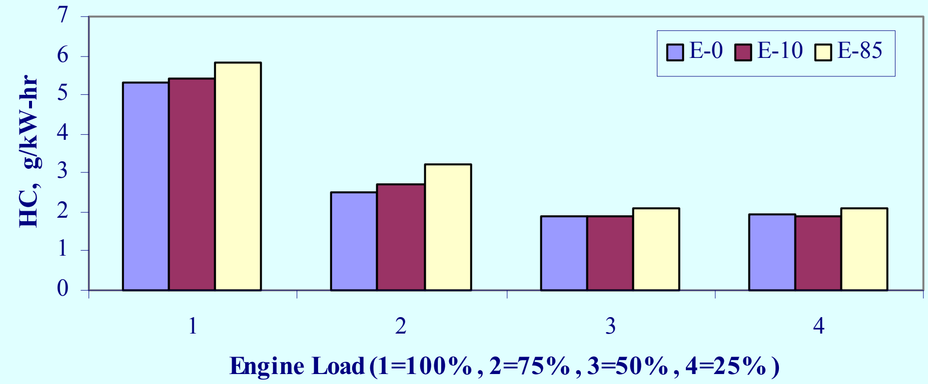

Fig. 19 HC Exhaust Emissions Comparion at 1800 rpm (OEM Spark Timing)

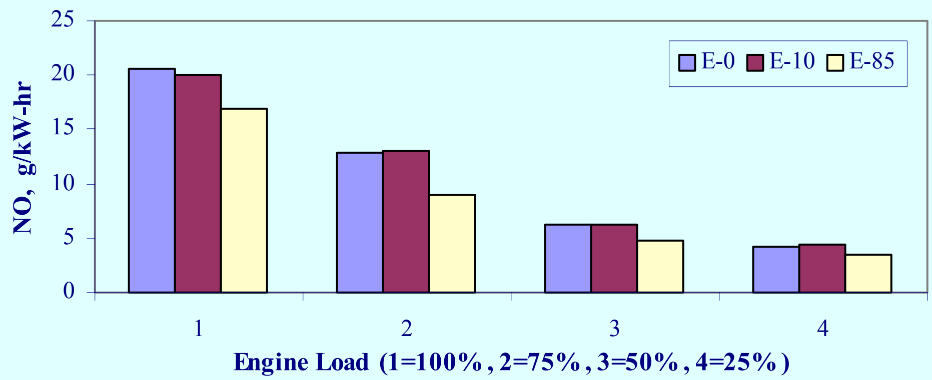

Fig. 20 NO Exhaust Emissions Comparison at 1800 rpm (OEM Spark Timing)