Pertanika J. Sci. & Technol. 8(2): 229-239 (2000)

ISSN: 0128-7680

Effect of Ignition Timing on Fuel Consumption and Emissions of a Dual Chamber Spark Ignition Engine

Ch. Rangkuti
Mechanical Engineering Department
Univ. of North Sumatera
Medan, Indonesia

Received: 12 March 1998

ABSTRAK

Kajian telah dijalankan terhadap penggunaan dan penyebaran bahan api daripada bilik berkembar, cas berstrata, enjin nyalaan pencucuh dengan masa nyalaan yang berbeza. Pembakaran terlambat yang berkait dengan pemasaan MBT, menghasilkan penggunaan bahan bakar kurang baik, terutamanya dengan campuran yang sedikit. Pembakaran lemah yang berkait dengan pembakaran lambat tidak akan meningkatkan penyebaran UHC – sebaliknya penurunan penyebaran CO lebih tinggi (untuk campuran yang sedikit); paras NO_x tidak banyak bezanya apabila terbuka luas dan 65% kecepatan dan kerendahannya signifkan pada 40% kecepatan.

ABSTRACT

Fuel consumption and emissions from a dual chamber, stratified charge, spark ignition engine with different ignition timing were investigated.
Retarded ignition, relative to MBT timing, yielded poorer fuel consumption, especially with lean mixtures. The poorer combustion associated with the late burning did not result in increased UHC emissions - these in fact is reduced. Emissions of CO were higher (for lean mixtures); NO_x levels were much the same at wide-open and 65% throttle settings and significantly lower at the 40% throttle setting.

Keywords : ignition time, combustion speed, emissions, lean mixture

INTRODUCTION

In their earlier investigations using the same engine, British Leyland Technology Ltd. (Weaving 1982) specified a "base-line" test condition which they used as a standard/reference for comparing performance at other test conditions. The base-line conditions were at engine speed of 2000 rpm., pre-chamber air flow at 6 % of total air inlet and pre-chamber air fuel ratio (AFR) of 6 : 1. This condition was reported to give the best compromised emissions for the engine in the previous Leyland experiments.

Tests conducted in this study at the reference running condition revealed some differences in engine performance compared with those conducted previously at British Leyland (Weaving 1982). In particular, unburned hydrocarbon (UHC) emissions were significantly lower in the tests conducted.

The UHC emissions were considered to be the major problem with the engine. Hence further consideration was given to the differences between the two sets of results.

In an attempt to assess the relative contributions of the main combustion event and crevices to UHC emissions the "reference test" was repeated with a fixed ignition advance instead of the maximum best torque (MBT) timing adopted in all other tests. The fixed advance was selected to be 22° Before Top Dead Centre (BTDC), the optimum ignition timing at the wide-open throttle setting for the richest mixture was used in the reference test. This resulted in retarded timing compared with MBT at other AFR's, ignition being particularly "late" for lean mixtures. This was expected to result in poor combustion taking place late in the cycle, with incomplete combustion and increased UHC emissions.

In this paper the experiments are designed to explore this difference and report particularly the effects of ignition timing on fuel consumption and emissions of a dual chamber stratified charge engine.

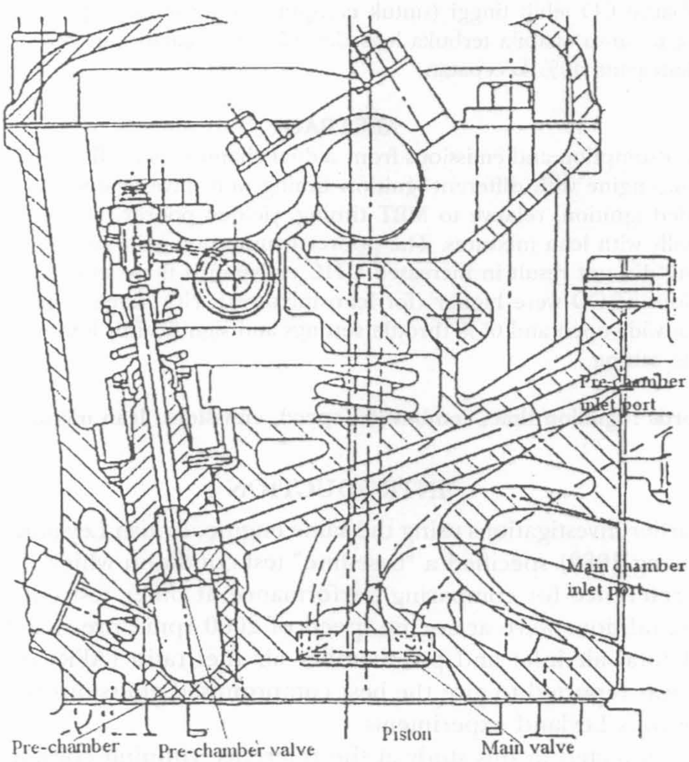

Fig 1. Schematic drawing of engine cylinder head

MATERIALS AND METHODS

Engine

The engine was based on a 4 cylinder Triumph Slant engine with water cooled cylinder block. The cylinder bore was 90.3 mm and stroke of 78.0 mm. A cross section of the cylinder head fitted to the working cylinder is shown in *Fig. 1.* The engine crankshaft was fitted with an extension to drive, via a flexible coupling, a shaft encoder. At the other end, the engine was connected to a D.C. motor type dynamometer by a flexible coupling.

The main chamber carburettor was of SU type AUB9203, and the fuel to the pre-chamber was supplied from the tank via a fine needle control valve, to a rotameter. The pre and main chamber air flows use separate intake systems; the air supply to the main chamber was drawn in via a large surge tank fitted with a 16 mm diameter metering nozzle.

The pre-chamber air flow-rate was measured by an air rotameter and this was fixed at 6 % of the air flow-rate into the engine in all experiments, because this amount proved to be the optimum in the earlier study (Weaving 1982).

The torque developed by the engine was measured using a load cell. A switch at the control panel allowed the polarity of the signal to be changed. This allowed measurement of either firing or motoring torque of the engine.

Engine Instrumentation

The engine instrumentation mainly consists of a dynamometer and the control panel, which includes cycle and timing selector, pressure transducer charge amplifiers, speed dial and torque meter dial.

Pulses created by the commercial shaft encoder were sent to an external clock, incorporated in a VAX-8600 computer, to instruct the analogue to digital converter (ADC) to take samples each time a pulse was generated by the encoder.

The ignition system used comprised a contactless electronic ignition unit, a standard ignition coil, a 12 V battery and a commercial spark plug.

The control system counted the pulses from the shaft encoder and triggered the spark at the required angle. The spark timing could be set to any crank-angle between 99°BTDC and 99° After Top Dead Centre (ATDC). A switch on the panel allowed either 2 or 4 stroke engine operation to be selected. The ignition could be restricted to alternate or every 3rd, 4th or 5th cycle if required. The system provided spark and Top Dead Centre (TDC) signals to be fed to the on-line computer; it also "gated" the shaft encoder pulse which triggered the computer's data acquisition system.

The pressure in each chamber inside the engine cylinder was measured using a piezo electric transducer. They were capable of measuring rapidly varying pressure in the range of 0 to 250 bar, while maintaining good linearity and having a very good frequency response. The signal from each transducer was transmitted, via two balanced leads, to a universal electrostatic charge amplifier; which converted the electrostatic signals into a voltage.

On-line Data Acquisition

A very high speed ADC unit was used to convert the pressure signals (from both chambers), as well as spark and TDC signals to a digital form. The ADC unit was interfaced to the VAX-8600 minicomputer via a direct memory access interface.

Once a signal was sampled, the information could be stored in the computer and immediately processed to yield output such as pressure-crank angle diagram, pressure-volume diagram and indicated mean effective pressure (imep), the data to be used in the figures presented in this paper. The computer programme used for this work was based largely on that developed by Hynes (1986).

Gas Analysers

The system was designed to sample and measure the concentration of total unburned hydrocarbons (UHC), carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2) and oxides of nitrogen (NO_x) in the engine exhaust. The system is set out diagrammatically in *Fig.* 2. It includes sample probe, stop valves, heated filter, water traps, drying agents, three way valves and heated line with temperature control. The sample was fed to the hydrocarbon analyser via a continuously heated sampling line which kept the sample temperature at 150°C throughout, in order to prevent any condensation of the higher hydrocarbons. The gas samples fed to the other analysers were led via a water trap and tubes containing drying agents, as it was important to avoid water condensation in the instruments. The oxygen analyser sample was fed from the high range CO analyser, as the former analyser did not have a pump of its own.

Total Hydrocarbon Analyser

The total unburned hydrocarbon concentrations were measured using an Analysis Automation Ltd. Series 520 Hydrocarbon Analyser; this incorporates a flame ionization detector (FID) for total unburned hydrocarbon measurement.

In the current work calibration was effected using a 400 ppm concentration of normal-hexane in nitrogen. The manufacture's claimed accuracy for the unit was ± 1.5% of full scale deflection (FSD); it had ranges 0-10, 0-100, 0-1000 and 0-10,000 ppm by volume.

Infra Red Analysers

Carbon monoxide concentrations were measured using two Grub-Parsons Series 20 infra red gas analysers, one with ranges of 0 - 0.1 % and 0 - 0.5 % the other having ranges of 0 - 3.0 % and 0 - 15.0 % by volume. Carbon dioxide concentrations were measured using a similar type of analyser, with ranges of 0 - 15.0 % by volume. The quoted accuracy of both instruments was ± 1 % FSD.

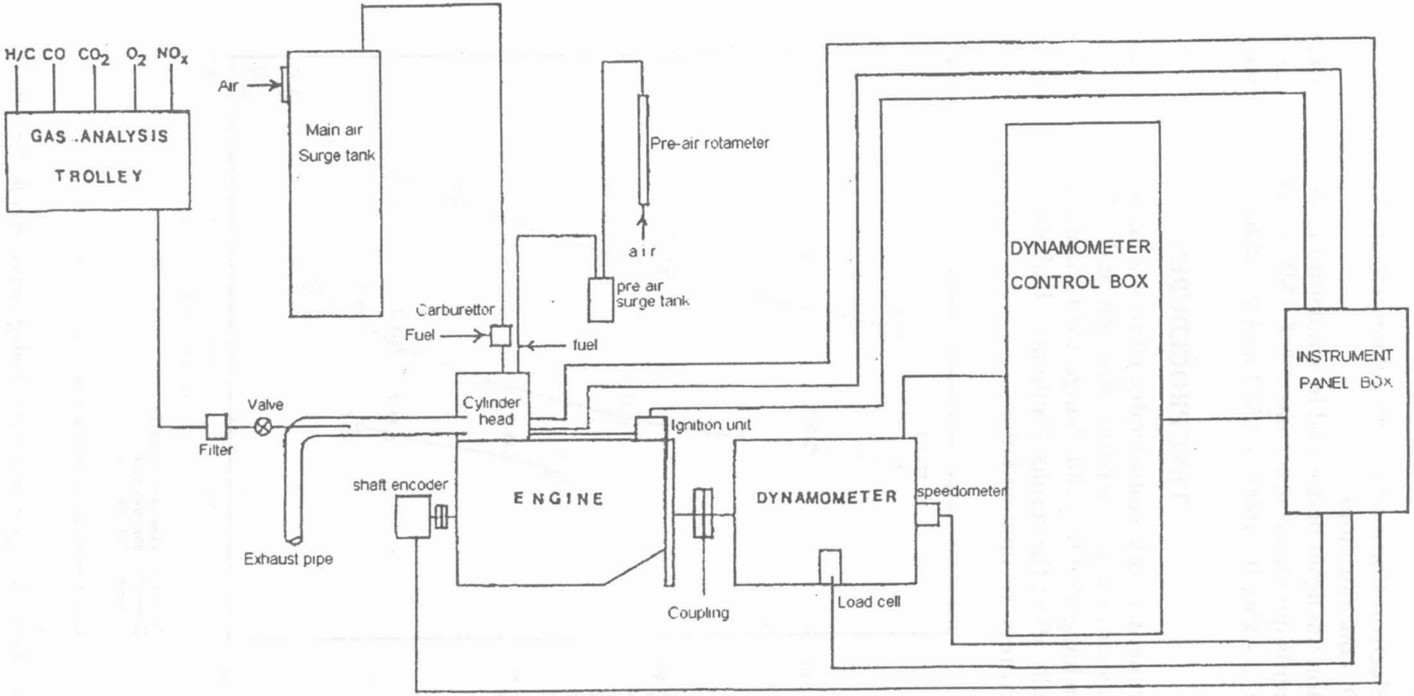

Fig 2. General schematic diagram of equipments system

NO_x Analyser

A Thermo Electron Corporation chemiluminescent NO_x analyser was used to measure NO_x concentrations.

Calibration was again performed by standardising with a known gas mixture. The instruments quoted accuracy was ± 1 % FSD and the unit had ranges of 0 - 25, 0 - 100, 0 - 250, 0 - 1000, 0- 2500 and 0 - 10,000 ppm by volume.

TEST PROCEDURES

In this experimental work, considerable effort was made to ensure that variables assumed constant, such as mixture strength and inlet mixture temperature remained unchanged; if they did change, the variation was not sufficiently great as to materially affect the results. Equipment had also to be used according to the manufacturers' recommendations.

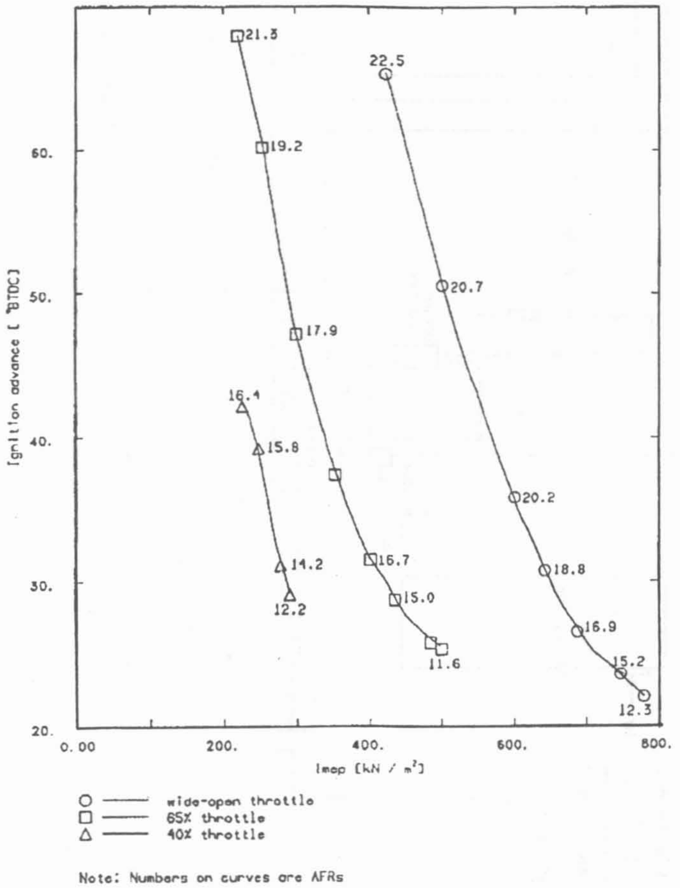

Fig 3. The reference test ignition advance timing versus imep for three throttle settings

RESULTS AND DISCUSSION

Engine Fuel Consumption Performance

The ignition advance timing set to give MBT for wide-open, 65 % and 40 % throttle setting with various AFRÆs for reference tests are shown in *Fig. 3.* It can be seen that for a leaner mixture, the ignition advance required is higher.

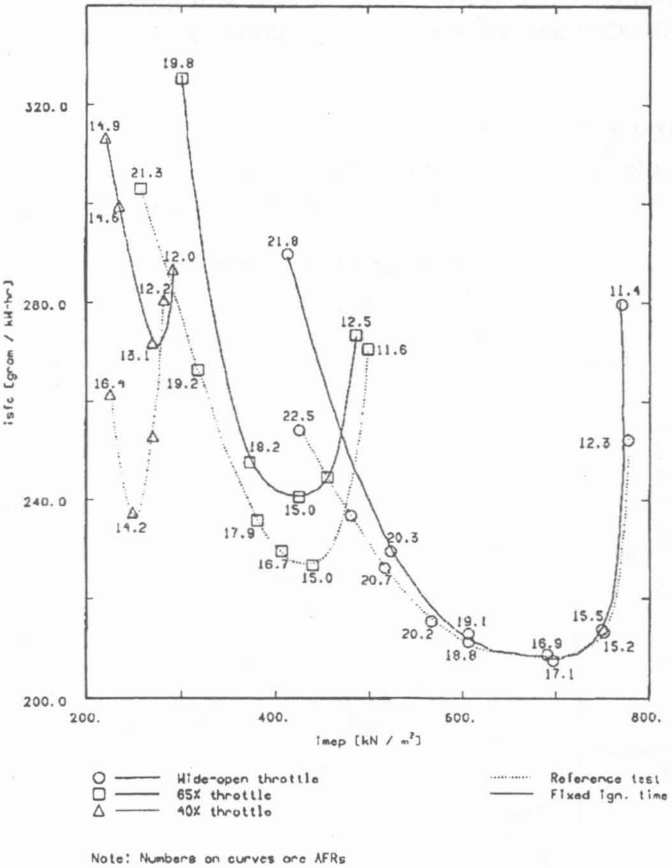

Fig 4. Engine isfc vs. imep for reference and fixed ignition timing

Similarly for 65% and 40% throttle settings, the ignition timings for MBT are also higher.

The effects of the fixed ignition time (22°BTDC) compared with the reference tests (ignition timing set to give MBT) on engine specifc fuel consumption are shown in *Fig. 4.* The specific fuel consumptions for wide-open throttle, at the AFR of optimum sfc and for fuel rich mixtures, were almost identical with those obtained with MBT timing. The sfc progressively deteriorated with increasingly lean mixtures for the each fixed ignition timing. This was expected, due to the progressively later ignition with respect to MBT timing.

For the 65% and 40% throttle settings these effects were even more obvious because of the relatively greater retardation of the fixed ignition timing as compared with the time ignition giving MBT.

When ignition was retarded, a secondary effect was to produce a weaker mixture in the pre-chamber at ignition since there is time for a greater amount of weak main chamber mixture to be pushed into the pre-chamber by piston motion. This effect was also observed to re-inforce the slowing of the combustion event.

Unburned Hydrocarbons

At full throttle, the fixed ignition timing resulted in a very marginal reduction in UHC, compared to the reference tests results, *Figure 5.*

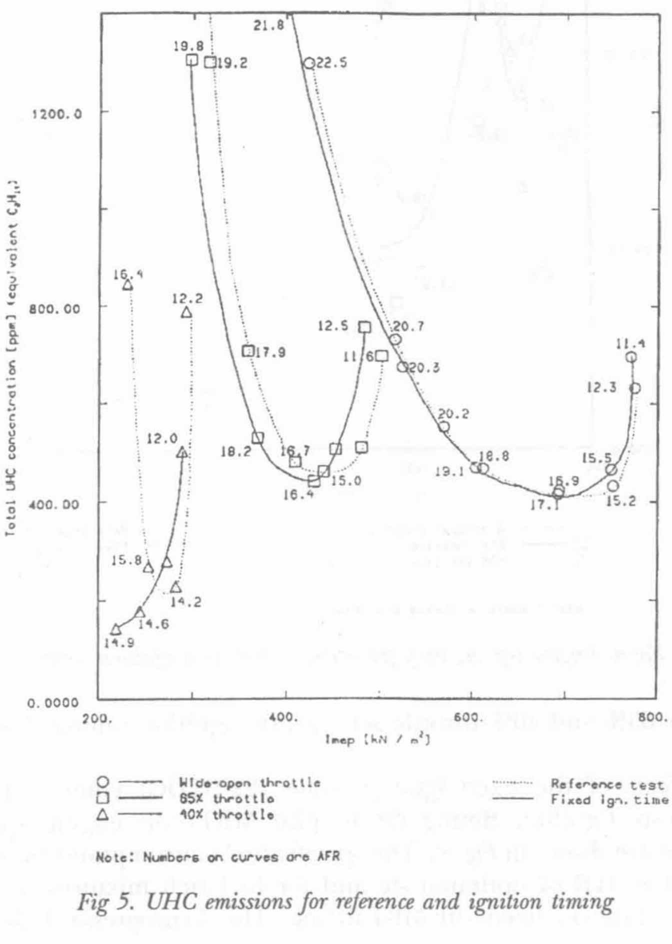

Fig 5. UHC emissions for reference and ignition timing

The UHC concentrations for optimum engine performance at wide-open throttle setting, were almost identical with the reference test, as one would expect, and marginally lower at lean mixtures as in *Fig. 5.* The same trend was even more evident for the 65% and 40% throttle settings. The UHC concentrations were generally lower than for the corresponding MBT timing. The effects were more marked for the heavily throttled lean case, at the time when ignition was most retarded.

With retarded ignition, one would expect increased average post flame and exhaust port temperatures (Kaiser *et al. 1983*). This should result in increased post flame and exhaust port burn-up of the UHC stored in the crevices and oil films. In addition, the peak pressure in the cylinder would be lower and the amount of unburned material stored in crevice volumes should be reduced (Lavoie *et al.* 1980 and Rangkuti 1990). These factors might explain the observed reduction in UHC, which occurred in spite of the marked deterioration in sfc.

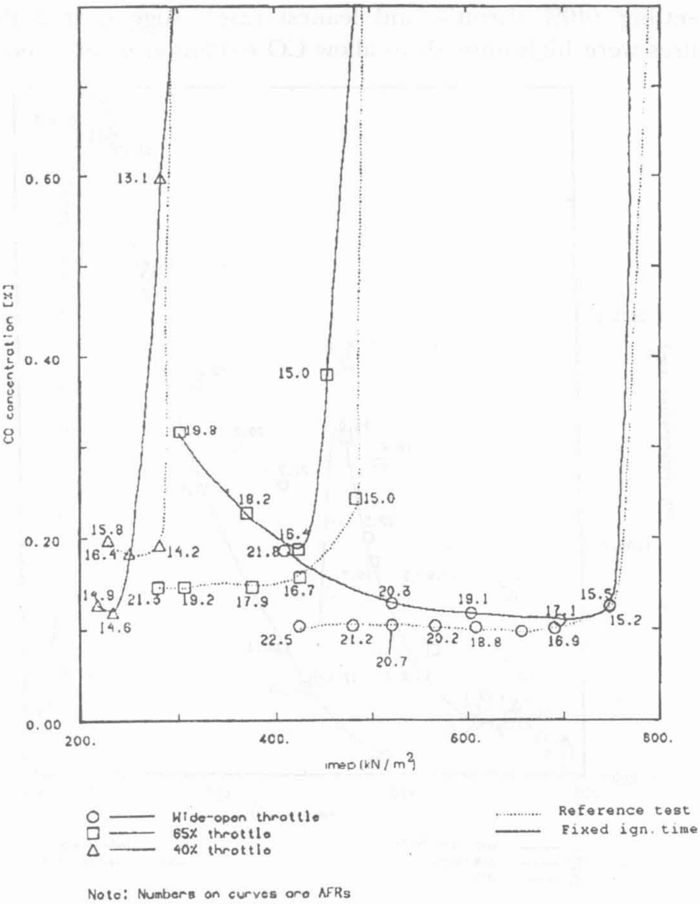

Fig 6. CO emissions for reference and fixed ignition timing

Carbon Monoxide

It can be seen from *Fig. 6* that for rich mixture, CO concentrations at wide-open throttle were essentially the same for fixed and MBT timing. As the mixture became leaner, and the difference in ignition timing more marked, the CO level increased a great deal. At the more retarded ignition conditions at the 65% throttle setting, these effects were even more marked. At the 40% throttle setting, CO levels were similarly higher - except at very lean (late burn and cool) conditions *(Fig. 6)*. With retarded ignition, it was expected that there would be an increase of the average post flame and the exhaust port temperature. This resulted in increased post flame and exhaust port reaction, with some of the UHC (emerging from the crevices late in the cycle) converted to CO. This was particularly so for lean mixtures, with plenty of oxygen available. However, the exhaust temperatures are generally expected to be too low to allow rapid further oxidation of this CO into CO_2 (Lavoie *et al.* 1980). The simultaneously low UHC and CO (with high specific fuel consumption), at the most retarded ignition setting (40% throttle and leanest case), suggest that the exhaust temperatures were high enough to allow CO oxidation to proceed.

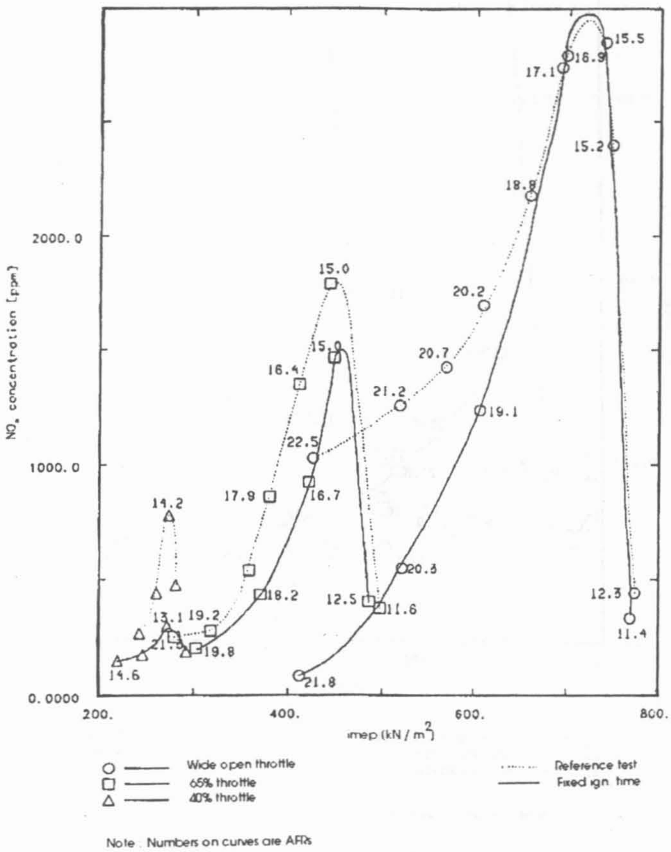

Fig 7. NO_x emissions for reference and fixed ignition timing

Oxides of Nitrogen

The retarded ignition settings resulted in reduction of cylinder pressure and temperature which led to reduced NO_x emissions. Those conditions for most retarded by MBT timing, generally resulted in the most marked fall in NO_x output (see *Fig. 7*).

CONCLUSION

This paper reported the effects of ignition timing on engine fuel consumptions and emissions. The retarded ignition, relative to MBT timing, gave poor fuel consumption, especially with lean mixtures. The poor combustion associated with the late burning did not result in increased UHC emissions - but in fact it reduces. Emissions of CO were high (for lean mixtures), NO_x levels were the same at wide-open and 65% throttle settings but significantly lower at the 40% throttle setting.

REFERENCES

WEAVING, J.H. 1982. A fundamental investigation of the combustion process in a stratified charge engine of the pre-chamber type. British Leyland Technology Report No. ETR 3271.

HYNES, J. 1986. Turbulence effects on combustion in spark ignition engiines. Ph.D. Thesis, Department of Mechanical Engineering, University of Leeds.

KAISER, E.W., W.G. ROTHSCHILD and G.A. LAVOIE 1983. The effect of fuel on operating variables on hydrocarbon species distribution in the exhaust from multicylinder engine. *Combustion Science and Technology*: 245-260.

LAVOIE, G.A. and P.N. BLUMBERG. 1980. A fundamental model for predicting fuel consumption, NOx and HC emissions of the conventional spark ignition engine. *Combustion Science and Technology*: 225-258.

RANGKUTI, C. 1990, Performance and emissions from a dual chamber stratified charge engine. Ph.D. Thesis, University of Leeds.

APPENDIX

Engine Details :

Number of cylinder	: 1	Valve timing,	
Type	: S.I. dual combustion chamber	*main chamber:*	
		inlet-open	: 16°BTDC
Cycle	: 4	inlet-close	: 56°ATDC
Bore	: 90.3 mm	exhaust-open	: 56°BBDC
Stroke	: 78.0 mm	exhaust-close	: 16°ATDC
Volume displacement	: 499.5 cc	*pre-chamber:*	
Connecting rod length	: 129.5 mm	inlet-open	: 20°ATDC
Pre-chamber volume	: 5 cc (nominally 10%)	inlet-close	: 20°ABDC
Throat size	: 7.94 mm		
Compression ratio	: 9.33 : 1		