EFFECT OF ENGINE SPEED ON PERFORMANCE AND EMISSIONS OF A DUAL CHAMBER SPARK IGNITION ENGINE

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ABSTRACT

The performance, fuel economy and emissions from a dual chamber, stratified charge, spark ignition engine with different engine speed were investigated. A previous study at British Leyland, using the same engine returned a good fuel economy and showed promise for meeting most proposed emissions standards. It was noted in this study that higher engine speed resulted in improved fuel consumption, UHC and NO_x emissions were reduced significantly in all throttle settings; CO emissions were higher probably due to post flame and exhaust port reaction effects. Engine combustion pressures were generally lower, a result of reduced volumetric efficiency with increased engine speed. The high UHC noted in the Leyland tests might be associated with partial quenching and it was understood that excessive in-cylinder turbulence can lead to partial quenching of flame propagation, while the turbulence is a function of engine speed.

1. INTRODUCTION

In their earlier investigations using the same engine, British Leyland Technology Ltd.¹ 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 6% of total air inlet and pre-chamber AFR of 6 : 1. This condition was reported to give the best emissions compromise for the engine in the previous Leyland experiments.

The tests at the reference running condition revealed some differences in engine performance vis a vis those conducted previously at British Leyland¹. In particular, unburned hydrocarbon emissions were significantly lower in the Leeds tests. Unburned Hydrocarbon (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.

It has been suggested that excessive in-cylinder turbulence can lead to partial quenching of flame propagation². As the divided chamber engine generates rather

more turbulence than a conventional engine, it was considered that the high UHC noted in the Leyland tests might be associated with such partial quenching. Since the turbulence generation is a function of engine speed, one would expect the problem to be worse at high engine speed. It was therefore decided to look at the influence of engine speed on performance, particularly UHC emissions.

Partial quench as a result of excessive turbulence has been suggested to be a cause of UHC emissions². It was thought that the dual chamber engine might be particularly prone to this effect, as a result of the high degree of turbulence generated by the jet flows induced through the orifice between the two chambers. These jet flows and induced turbulence would increase with engine speed³. The current engine was therefore operated over a range of engine speed, with particular interest in the UHC emissions behaviour at high engine speeds. In practice the maximum obtainable engine speed proved to be 2500 rpm (c.f. the reference test speed of 2000 rpm)⁴. Above this speed the engine proved subject to extreme vibration, due to the engine's poor dynamic balancing - it will be recalled that the engine had a single operating cylinder in a converted four cylinder engine block. To give a range of engine speeds, a further test series was conducted at the lower speed of 1500 rpm.

Reported in this paper are experiments designed to examine the effects of engine speed on engine performance and emissions.

2. EQUIPMENT

2.1 Engine

The engine was based on a 4-cylinder Triumph Slant-4 water cooled cylinder block. The cylinder bore was 90.3 mm and it had a stroke of 78.0 mm. A piston, of the flat crown type was fitted to number one cylinder only. A three valve, pre-chamber, cylinder head was fitted to number one cylinder. The valves (intake, exhaust and pre-chamber) were actuated by cams on a single overhead camshaft, driven by the crankshaft by means of a chain.

Shown in Figure 1 is a cross section of the cylinder head fitted to the working cylinder. The engine crankshaft was fitted with an extension to drive, via a flexible coupling, a shaft encoder; the latter drove the data acquisition system as outlined in a subsequent section of this section.

The main chamber carburettor was an 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 used 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, made in accordance with the guidelines of British Standard Institution (BSI: 1042 (1981)). 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, as this amount proved to be the optimum in the earlier British Leyland study¹.

The cooling water outlet temperature was controlled using a commercial thermostat, fitted to the cooling water outlet on the engine block. This controlled the cooling water flow-rate, to give a constant 90°C water outlet temperature as monitored by a thermocouple. This temperature was maintained the same in all tests.

The engine was connected, by a flexible coupling, to a D.C. motor type dynamometer. This was of 0 - 40 KW capacity, supplied by David McClure Ltd. The dynamometer was governed using a Safronic control system. If run at a set speed in the motoring mode, the dynamometer would automatically switch to the absorption mode when the engine's ignition switch was triggered and the engine generated sufficient power to maintain the set speed.

The torque developed by the firing engine was measured using a load cell supplied by Transducer (CEL) Limited. This had a range of 0 - 100 Newtons, this load was converted to an electric signal and displayed as torque on a meter supplied by Hottinger Baldwin Messtechnik. Dependent upon the direction of the rotation, the force on the load cell would be positive or negative. A switch at the control panel allowed the polarity of the signal to be changed. This allowed measurement of either firing or motoring torque for the engine.

Engine Details:

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

2.1.1 Engine instrumentation

The engine is coupled to the dynamometer by a flexible coupling; the control panel, consisting of cycle and timing selector, pressure transducer charge amplifiers, speed dial and torque meter dial. A commercial shaft encoder was used; this was of Type 320D/360, supplied by Hohner Automation Ltd. The pulses created 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 Mobelec Magnum contactless electronic ignition unit, a standard ignition coil, a 12V battery and a commercial spark plug. The control system counted 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 (Before Top Dead Centre) and 99°ATDC (After Top Dead Centre). A switch allowed either 2 or 4 stroke engine operation to be selected; ignition could be restricted to alternate or every 3rd, 4th or 5th cycle if required. The system provided spark and TDC signals to feed to the on-line computer; it also “gated” the shaft encoder pulse which triggered the computer’s data acquisition system.

Engine cylinder pressure is clearly an important parameter; it provides information to permit calculation of the indicated power output and can be used as a basis for estimation of fuel mass burning rate. In the study reported here, the pressure in each chamber was measured using a piezo electric transducer; these were of type 6121, supplied by Kistler Instrumente A.G. They were capable of measuring rapidly varying pressure in the range of 0 - 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 type 5007 Kilter universal electrostatic charge amplifier; this converted the electrostatic signals into a voltage. The charge amplifier was allowed to warm up and stabilize for a period of at least half an hour, before use either in calibration or experimental work.

2.1.2 On-line data acquisition

A Micro-Consultants 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. This unit allowed a maximum data acquisition of 14 bits at 200 kHz. The ADC unit was interfaced to the VAX-8600 minicomputer via a direct memory access interface (DR-11B). The ADC system also allowed an external “gate” input, this only allowed sampling to occur when an external supply of +5 volts was supplied to the gate channel. 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 program used for this work was based largely on that developed by Hynes⁵.

2.2 Gas analyzers

The system was designed to sample and measure the concentration of total unburned hydrocarbons, carbon monoxide, carbon dioxide, oxygen and NO_x in the engine exhaust. The system is set out diagrammatically in Figure 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.

2.2.1 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 calibration (span) gas was supplied by the Special Gases Division of the British Oxygen Company (BOC). The analyser ion current output is a function of the carbon number, i.e. twice the concentration of propane (C_3H_8) would be required to give the same response as that of n-hexane (C_6H_{14}). The instrument therefore yielded the total unburned hydrocarbon concentration in terms of the equivalent concentration of hexane. The manufacture's claimed accuracy for the unit was $\pm 1.5\%$ of full scale deflection (FSD); it had ranges 0-10, 0-100, 0-1000 and 0-10,000 ppm by volume.

2.2.2 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 $\pm 1\%$ FSD.

2.2.3 NO_x Analyser

A Thermo Electron Corporation chemiluminescent NO_x analyser was used to measure NO_x concentrations. The principle of the chemiluminescent analyser is based upon the chemiluminescent gas phase reaction between ozone and nitric oxide. The reaction of ozone with nitric oxide, when heated under vacuum at $650^\circ C$, produces electronically excited molecules of nitrous oxide. When the electronically excited molecules decay, they emit light which can be measured using a photo multiplier. In the instrument this photo multiplier output was amplified and displayed. The light intensity was proportional to the nitric oxide concentration in the sample. Calibration was again performed by standardising with a known gas mixture. The instruments quoted accuracy was $\pm 1\%$ FSD and the unit had ranges of 0 - 25, 0 - 100, 0 - 250, 0 - 1000, 0 - 2500 and 0 - 10,000 ppm by volume.

3. TEST PROCEDURES

In the 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.

4. EXPERIMENTAL RESULTS

4.1 Engine Performance

The effects of engine speed on power output and fuel economy are shown in Figure 3. The corresponding values of optimum ignition advance are shown in Figure 4; a pressure-crank angle diagram, for wide-open throttle at the mixture strength yielding optimum specific fuel consumption (sfc), is shown in Figure 5. It can be seen that the indicated mean effective pressure (imep) at a given air fuel ratio (AFR) falls with increasing engine speed. This is largely the result of the falling volumetric efficiency with engine speed, shown in Figure 6; this is also demonstrated by the lower pressure prior to ignition, shown in Figure 4.

In contrast, the isfc improves with increasing engine speed, Figure 3. This is because of reduced time available (per cycle) for heat transfer to the walls and because the combustion duration, in terms of crank-angle, decreases with increased engine speed.

The increased turbulence generation with higher engine speed has resulted in an even greater increase in turbulent burning rate (even though the combustion pressure and hence temperature were lower, Figure 5). This is in line the findings of Bradley et al², who showed that the initial effect of increased turbulence intensity is to produce an even greater increase in turbulent burning velocity. Bradley et al² suggests that further increase in turbulence intensity would result in diminishing returns, with no commensurate increase in burning velocity - because of partial quenching by excessive turbulence, especially for less chemically reactive lean mixtures. Unfortunately it was not possible to further increase engine speed to explore this effect. As explained previously, this maximum was imposed by engine vibration problems.

Similar trends were noted at the 65% throttle setting; at the 40% throttle setting it was difficult to obtain good steady running of the engine at 2500 rpm; this was possibly due to very high residual gas effects and to low engine volumetric efficiency (Figure 6).

4.2 Unburned Hydrocarbon

The effects of engine speed on UHC emissions are shown in Figure 7. At a given power output (imep) at each throttle setting, UHC emissions fell with increasing engine speed. At a given AFR, UHC levels declined marginally with engine speed at a given throttle setting. As discussed when considering fuel economy, there was no evidence of partial quench by excessive turbulence as the engine speed increased. (This is not to say it may not happen at still higher engine speeds). This suggests that the UHC observed in the current tests was more likely to have been associated with wall quench and/or crevice effects. At the lower imep's (at a given AFR), with increasing engine speed, the density of unburned fuel in quench and crevice regions would be reduced. Also, at a higher engine speed, the engine piston and cylinder wall might be expected to run hotter; this could result in a small reduction in ring pack crevice volume⁶, associated with differential expansion effects of the aluminium piston relative to the cast iron cylinder wall. Higher engine speed can also be expected to result in increased exhaust gas temperature⁸ leading to enhanced oxidation of UHC late in the cycle and in the exhaust. Both the above effects might contribute to the noted reduction in UHC emissions with engine speed.

4.3 Carbon Monoxide

The CO emission patterns shown in Figure 8 follow those noted previously. At a given throttle setting and engine speed the CO concentration can be seen to be fairly constant with AFR increase, rising sharply with oxygen deficiency for rich mixtures. At each throttle setting CO concentration generally increased with engine speed. As before, this is thought to be partially associated with oxidation of hydrocarbons late in the expansion and exhaust processes - as observed above, the gas temperature (and hence oxidation) might be expected to increase with engine speed. However this cannot explain the increase in CO with increased throttling, evident in Figure 8 and especially apparent at the highest engine speed. This is more likely to be associated with quenching of CO oxidation due to the rapid fall in temperature consequent upon expansion due to piston motion⁷. This might be expected to be more marked at the lower pressures and temperatures of the throttled (and high engine speed) cases, especially given the reduced time available for the recombination reactions at the higher engine speed.

4.4 Nitrogen Oxides

It can be seen in Figure 9 that NO_x concentration fell with increasing engine speed and with greater throttling of the engine. This is because combustion peak pressure falls with engine speed⁹ and degree of throttling. In association with this reduced peak pressure one would expect lower peak temperatures and hence NO formation rate^{10, 11}.

5. CONCLUSION

Reported in this paper are the effects of engine speed on engine performance and emissions. Higher engine speed resulted in improved fuel consumption UHC and NO_x emissions were reduced significantly in all throttle settings; CO emissions were higher probably due to post flame and exhaust port reaction effects. Engine combustion pressures were generally lower, a result of reduced volumetric efficiency with increased engine speed.

6. REFERENCES

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Figure 1. Cross Section of the cylinder head

Figure 2. General schematic diagram of equipments system.

Figure 3. Effect of Engine Speed on Engine Performance.

Figure 4. Effect of Engine Speed on MBT Ignition Timing for the Three Throttle Settings.

Figure 5. Pressure-crank angle diagrams for wide-open throttle setting and optimum sfc., at 2500, 2000 and 1500 rpm

Figure 6. Volumetric efficiency of the engine at the reference test condition for different speed and throttle setting.

Figure 7. Effect of engine speed on total UHC emissions.

Figure 8. Effect of Engine Speed on CO Emissions

Figure 9. Effect of Engine Speed on NOx Emissions.