An investigation into the burning rate of liquid petroleum gas in a multi-cylinder engine

M. G. Byrne

Wollongong University College

AN INVESTIGATION INTO THE BURNING RATE OF LIQUID PETROLEUM GAS IN A MULTI-CYLINDER ENGINE.

M.G.BYRNE

B.E. N.S.W., Stud.I.E.Aust.

Thesis submitted as part requirement for the Degree of Master of Engineering Science.

School of Civil, Mechanical and Mining Engineering.

Wollongong University College.

University of New South Wales.

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SUMMARY

The burning rates of liquid petroleum gas and petrol fuels in a multicylinder spark ignition engine are examined and compared. Findings are based on the pressure-time traces of cylinder pressure development during combustion.

Pressure tappings were fitted to three of the combustion chambers on the test engine- a Holden six cylinder engine - and a capacitance type pressure transducer together with specialised electronic equipment were used for recording pressure development.This made it possible to depict the pressure-time traces for a particular cylinder on an oscilloscope screen for observation and photographic recording.

The test engine, directly coupled to a dynamometer, was run on both fuels under similar conditions and a comparison of the pressure-time traces is made to establish differences in burning rates. Operating conditions that were varied for the tests included fuel type, ignition timing, air-fuel ratio, and engine speed. Spark plug heat range and gap were not altered.

Results of earlier work conducted to compare the performance of the engine for the two fuels, and which led to the present work, are discussed briefly.

It was found from the investigation that liquid petroleum gas

exhibits a higher rate of pressure rise during combustion and hence has a shorter combined ignition-burning period than that of petrol. To realise full power from spark ignition engines converted to liquid petroleum gas operation, the ignition timing is normally revised to compensate for the differences in the combined ignition-burning periods.

Variation of such parameters as ignition timing, air-fuel ratio and engine speed resulted in similar characteristics being noted for both fuels.

ACKNOWLEDGEMENT

The writer wishes to express his sincere thanks to Professor S.E.Bonamy, Chairman, Department of Mechanical Engineering for the opportunity of working in the Department and for his assistance and guidance in the preparation of this report.

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SECTION I

INTRODUCTION

INTRODUCTION

Although liquified petroleum gas (L.P.G.) offers some advantages over petrol in pollution control and considerable reserves of L.P.G. are available in Australia, the conversion of motor vehicles to L.P.G. operation has been limited by insufficient economic incentives. A recent report[1] issued by the Bureau of Transport Economics estimates that known Australian reserves of L.P.G. would be sufficient for 80 years supply at the present extraction rate and that 14% of Australian motor vehicles could be powered by the supply. This represents a valuable energy source that is currently being exported to other countries.

It is generally agreed upon by those involved with the conversion of spark ignition engines from petrol to L.P.G. operation, that it is necessary to revise the ignition timing to realise full power from the engine. This revised ignition timing required by the L.P.G. indicates that there are differences in the burning rates of the two fuels.

Earlier tests prompted by the increasing awareness of air pollution caused by the automobile engine and the introduction of laws to control exhaust emissions, and carried out by the author[2,3], also indicated these differences. The tests, the results of which will be discussed in Section III, were carried out to compare the performance of a multicylinder engine using petrol and L.P.G. as fuels. Results showed that, for the engine concerned, L.P.G. required the ignition timing to be retarded by between 2 and 4 degrees from that required for petrol operation. Since the peak cylinder pressures for the two fuels should occur at approximately the same crank angle position at a given engine speed for maximum torque, this difference in timing indicated a faster combined ignition-burning period for the L.P.G.

The present work is a continuation of the earlier comparisons and concerns an investigation into the differences in burning rates between L.P.G. and petrol fuels in the same multicylinder engine.

The rate of cylinder pressure rise has been shown to be strongly dependent on the combustion rate [4,5,6,7] and hence the findings presented here are based on pressure-time traces recorded for the two fuels at various operating conditions. Energy liberation in the combustion process is dependent on the flame front area, the concentration of the unburned mixture at the flame front, and the rate of reaction relative to the unburned portion. Variations in these parameters would also result in similar changes in the rate of pressure rise.

For convenience, the liquid petroleum gas is referred to here as Propane, the major constituent of the fuel.

SECTION II

TEST EQUIPMENT AND EXPERIMENTAL PROCEDURE

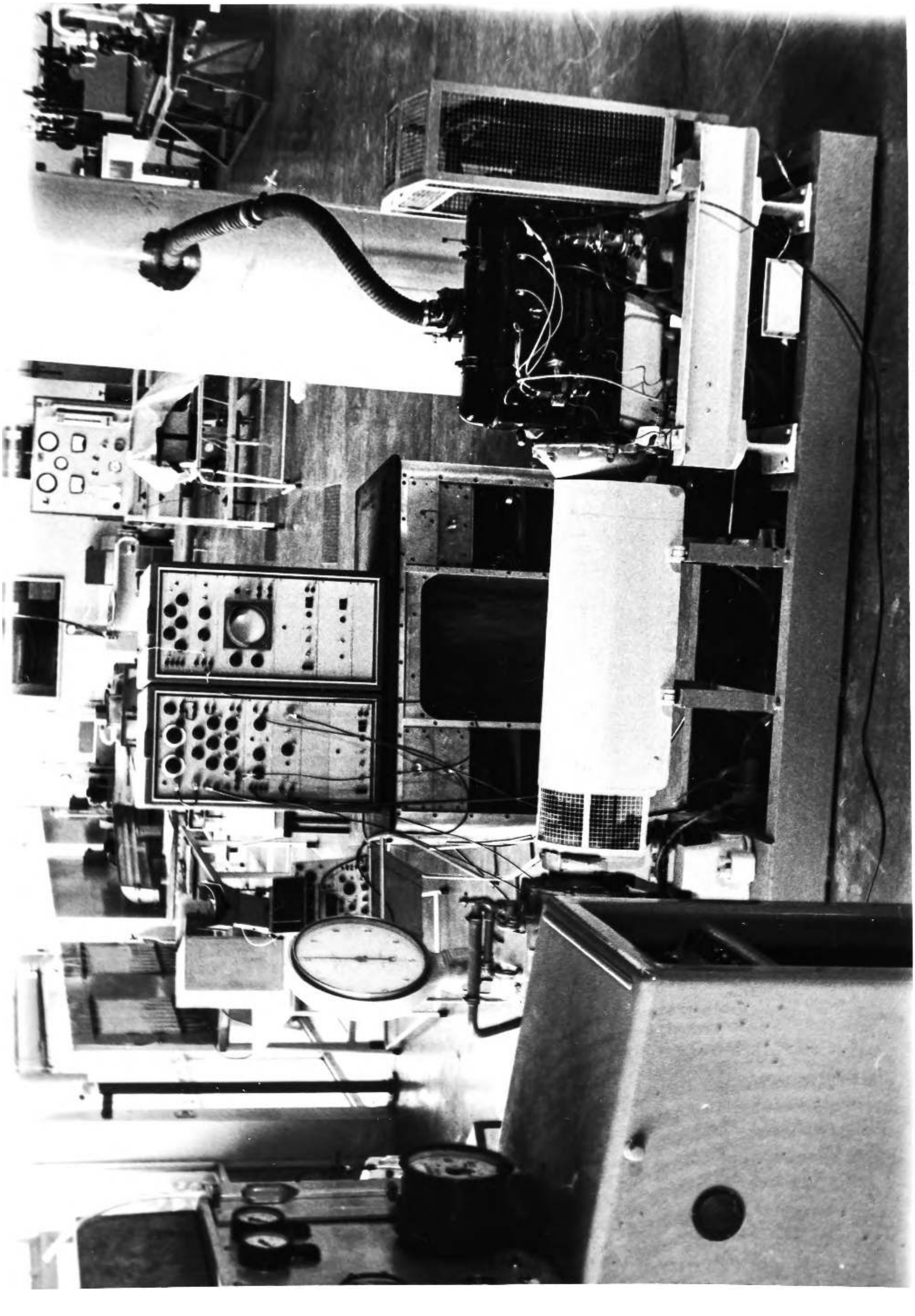

FIGURE 1 : GENERAL VIEW OF TEST ENGINE AND EQUIPMENT

TEST EQUIPMENT AND EXPERIMENTAL PROCEDURE

(a) ENGINE AND DYNAMOMETER

A general view of the test equipment is shown in Figure 1. The test engine was a six cylinder, four stroke 'Holden' engine of 3.062 inch bore and 3.125 inch stroke, having a capacity of 138 cubic inches.

Several modifications were made to the engine. The compression ratio was increased from 6.5:1 to 8.0:1 to make it more comparable with present standards and more suitable for both fuels. The cylinder head face was machined and the volume of the chambers measured by filling with light oil from a burette to ensure that the required ratio was obtained. The vacuum advance mechanism on the 'Bosch' distributor was disconnected and the distributor clamp was modified to allow easy timing adjustment over a wide range.

To measure ignition advance, a coupling on the engine-dynamometer shaft was graduated in crank degrees for a range of 90 degrees and a fixed pointer set at zero degrees to correspond with top dead centre of number 1 cylinder. The top dead centre position was obtained using a dial indicator. A Dawe stroboflood, triggered by an induced current from the high tension lead of number 1 cylinder, illuminated the scale on each firing of the cylinder enabling the ignition advance angle to be read.

The cooling system for the engine consisted of the normal radiator, belt driven water pump and fan with additional 'make-up' water being supplied through the radiator drain plug and withdrawn through the radiator overflow tube. The temperature of the coolant water was measured by a mercury thermometer fitted into the radiator header tank at the point where the water returns from the engine. The coolant temperature was maintained at 80°C during the tests.

The engine was coupled by a drive shaft to a Heenan and Froude 'Dynamatic' dynamometer Mark I G.V.A. having a capacity of 150 BHP at 2400/6000 rpm. The water cooled dynamometer was equipped with speed control which allowed the speed to be set and kept constant for varying engine loadings. The calibration of the engine tachometer on the dynamometer control desk was checked with a Smiths hand tachometer.

(b) <u>OPERATING EQUIPMENT FOR PETROL.</u>

A single barrel downdraught Stromberg BXOV.1 carburettor was used for petrol operation. To allow variation of the air fuel ratio, the main jet was drilled oversize and an adjustable tapered needle was inserted to control the fuel flow rate.

Premium grade petrol was used and was supplied to the carburettor by the normal AC mechanical fuel pump. Petrol consumption rates were metered by the positive displacement method. A two way valve was used to switch from the supply tank to a pipette with a graduated volume of 100 cc's.

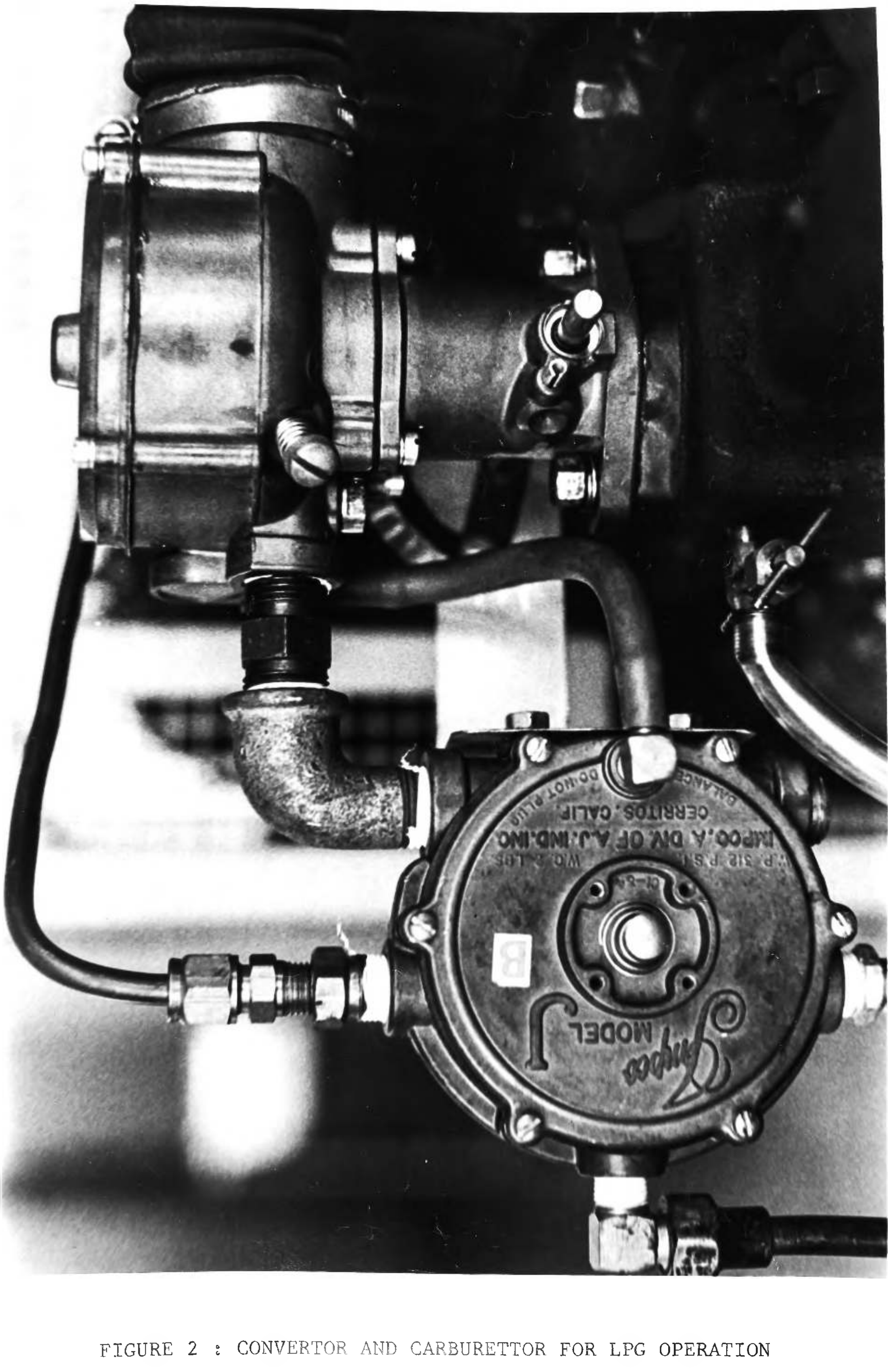

FIGURE 2 : CONVERTOR AND CARBURETTOR FOR LPG OPERATION

(c) OPERATING EQUIPMENT FOR PROPANE

For propane an Impco model 100-8 carburettor was used together with an Impco model JB convertor (Figure 2) which converted the propane from its high pressure liquid phase to a vapour at the low pressure required at the carburettor. Heat for vapourising the propane was supplied by hot water from the cooling system; the inlet temperature being controlled to prevent 'freezing' of the unit.

The propane was supplied from 100 lb liquid withdrawal bottles and from 20 lb bottles which were inverted to ensure liquid withdrawal. The liquid petroleum gas, referred to as propane in this report, was stated by the suppliers to have the following approximate composition: 88% propane, 2% butane and 10% propylene.

The mass flow rate to the engine was measured by switching from the 100 lb bottle to a 20 lb bottle for a period of 30 or 60 seconds. The 20 lb bottle was weighed carefully on a set of Wedderburn precision scales before and after the chosen time interval.

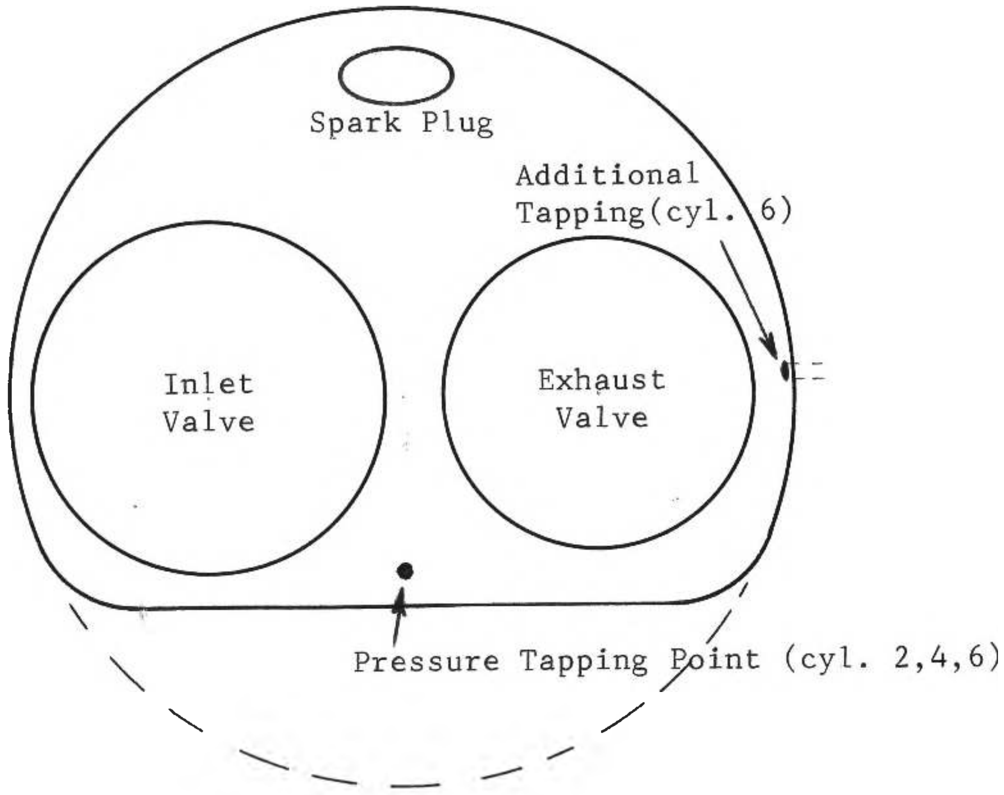

FIGURE 3 : PLAN VIEW OF COMBUSTION CHAMBER SHOWING LOCATION OF PRESSURE TAPPING POINTS.

(d) AIR FLOW MEASURING EQUIPMENT

To measure the air flow rate to the engine the intake was connected by a short length of smooth bore, heavy duty rubber hose of $2\frac{1}{2}$ inch inside diameter to a large surge tank which damped out pulsations prior to metering. Atmospheric air was drawn into the tank through a short pipe containing a metering orifice plate designed to the specifications of B.S. 1042. (Appendix 2) The differential pressure across the orifice plate was measured with an Askania micromanometer model 6-0042/72.

(e) EQUIPMENT FOR CYLINDER PRESSURE MEASUREMENT AND RECORDING.

To allow the cylinder pressure to be recorded, tappings were made into three of the six combustion chambers (cylinders 2, 4 and 6), the direction of entry being from the top of the cylinder head through the water jackets. The point of entry of the tappings into the roof of the combustion chambers was opposite the spark plug location as shown in Figure 3.

To minimise errors in pressure measurement, the length of the passage between the combustion chamber and the pressure transducer was kept to a minimum (approx. 4 inches for each tapping) and copper washers were used to ensure that no additional clearance volumes resulted when the transducer was fitted to the tapping point. The passage hole diameter was 0.047 inches.

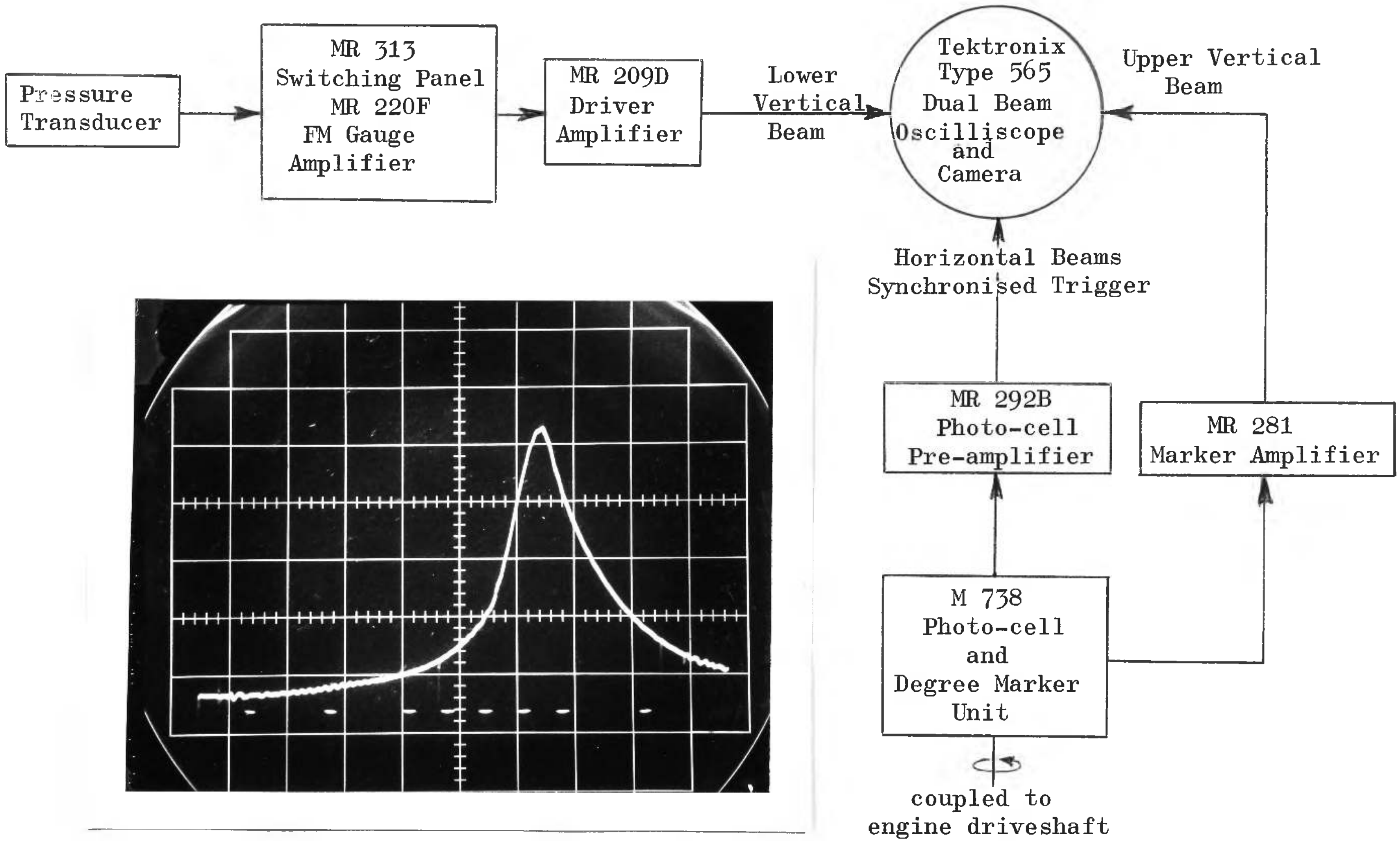

FIGURE 4 BLOCK DIAGRAM OF CYLINDER PRESSURE MEASURING AND RECORDING EQUIPMENT WITH TYPICAL EXAMPLE OF A PHOTOGRAPHIC RECORD OF A SINGLE PRESSURE-TIME TRACE.

A second tapping was fitted to cylinder 6 to allow possible differences in pressure recordings from the two tappings to be examined. The tapping was made horizontally from the back of the head entering the side wall of the combustion chamber near the exhaust valve which resulted in a shorter passage length of approx. two inches. It was later found that the pressure-time traces from this tapping were somewhat erratic, possibly due to the entry position of the tapping in the combustion chamber.

The pressure sensing and recording equipment is represented diagramatically in Figure 4.

The pressure transducer used was a water-cooled Minirack type G202 capacitance unit. The output of the pressure transducer was fed via the coaxial cable through the Minirack MR220F FM gauge amplifier and the MR313 switching panel to a MR209D driver amplifier. The output from the driver amplifier was fed to the lower beam of a Tektronix Type 565 dual beam oscilliscope fitted with Type 3A75 amplifiers on both the lower and upper vertical beam inputs.

The output from the photocell unit was fed via a MR292B photocell pre-amplifier to the horizontal beam trigger circuit of the oscilliscope. The two horizontal sweeps of the oscilliscope were locked together to achieve the synchronisation between the pressure-time trace and the degree marker points.

The output from the degree marker disc and magnetic pick-up was fed via a MR281 marker amplifier to the upper vertical beam of the dual beam oscilliscope.

The results obtained on the oscilliscope were recorded with the special Polaroid camera attachment for the oscilliscope. Controls on the oscilliscope allowed a single trace to be displayed on the screen when required. The camera shutter was held open for a selected period during which the single trace was manually triggered. A typical example of a single pressure-time trace with degree marker points is shown in Figure 4. The centre point of the group of five degree marker points represents top dead centre. These five points are spaced at intervals of 20 crank angle degrees, while the outer points are spaced at 40 degree intervals. On the vertical scale, the distance between grid lines represents 60 psi pressure.

The particular combination of the electrical equipment used did not allow the static calibration of the pressure transducer and equipment against a dead-weight tester to be made. The equipment was dynamically calibrated against a bourden-tube type pressure gauge fitted with a Schrader non-return valve. The gauge and transducer were fitted to the number 6 cylinder and the engine was operated at various conditions to give a range of peak pressures for calibration. The pressure gauge was statically checked against a dead-weight tester.

Although the accuracy of this method might not have matched that of the direct static calibration of the transducer and equipment, it was considered satisfactory as the test results were to be analysed on a comparative basis. Calibration of the transducer showed the response to be essentially linear and, for the equipment settings used for the tests, gave a scale of 60 psi per major vertical division on the oscilliscope screen.

SECTION III

EXPERIMENTAL RESULTS AND DISCUSSION

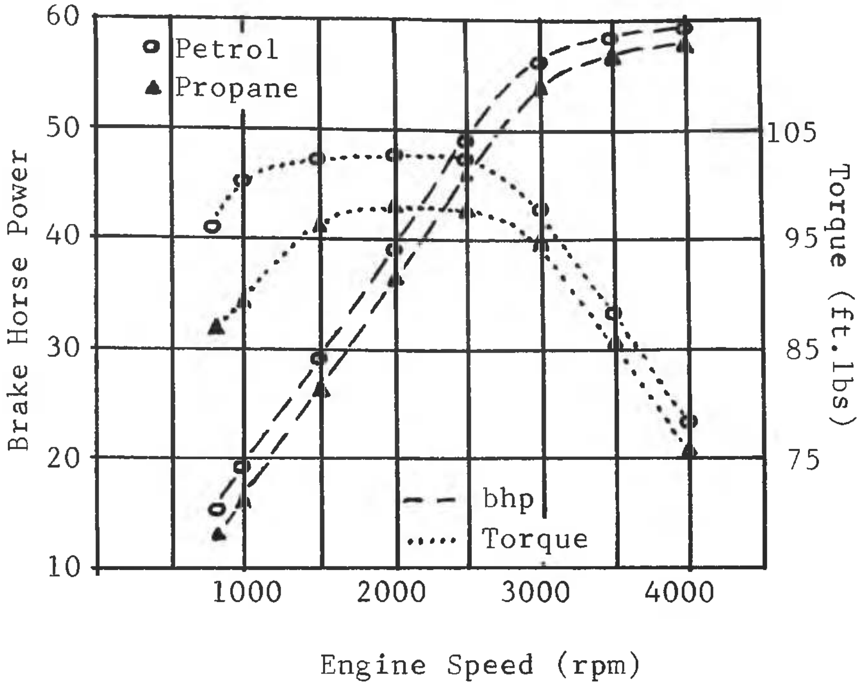

FIGURE 5 : PERFORMANCE CURVES.

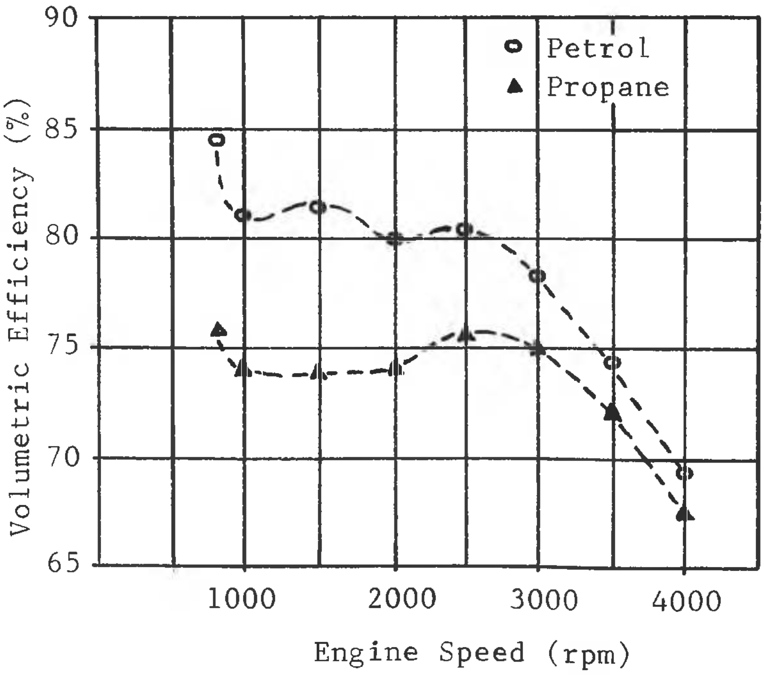

FIGURE 6 : VOLUMETRIC EFFICIENCY.

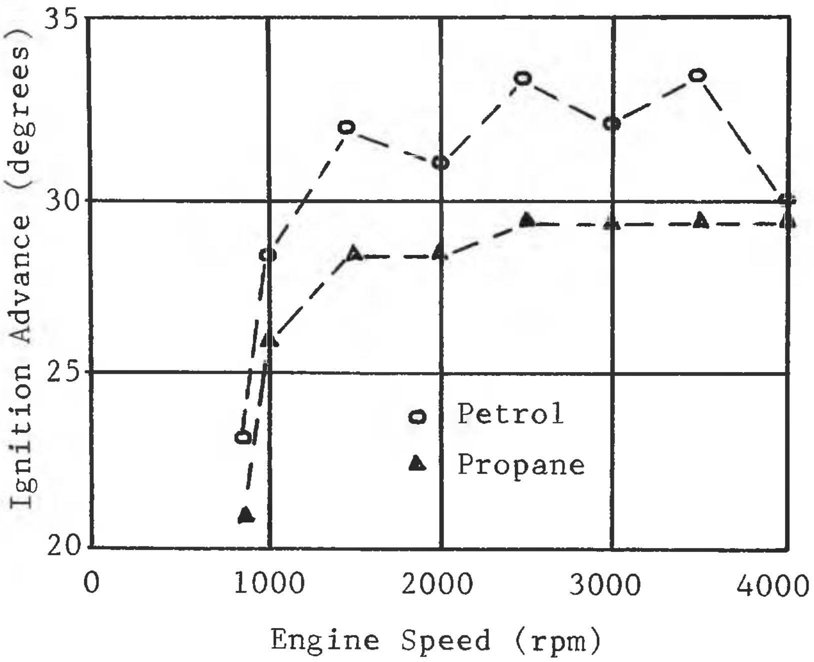

FIGURE 7 : IGNITION ADVANCE FOR OPTIMUM OUTPUT AT FULL LOAD.

EXPERIMENTAL RESULTS AND DISCUSSION

By way of introduction to the discussion of experimental results, three figures showing results of earlier performance tests, carried out on the same engine[2,3], are included.

Figures 5 and 6 show the performance curves and corresponding volumetric efficiencies for the two fuels over the speed range 800-4000 rpm. Conversion to propane resulted in an average decrease in performance of slightly more than 5% over the speed range. This was shown to be attributable mainly to the drop in volumetric efficiency (Fig.6) of the engine when converted to propane operation. The decrease in the volumetric efficiency resulted basically from the displacement of air entering the carburettor by the propane vapour and from the design characteristics of the particular carburettor used.

Figure 7 shows the ignition requirements of the fuels for full load optimum performance. As mentioned earlier, the differences in ignition timing requirements(approx. 2^{o} retard up to 1500rpm and 3^{o} retard above 1500 rpm for propane) indicate a faster combined ignition-burning period for propane, since the peak cylinder pressures for the two fuels should occur at approximately the same crank angle for a given engine speed for maximum torque.

The present experimental results are used to examine the combined ignition-burning periods of the two fuels and the effects of variations in ignition timing, engine speed and air-fuel ratio

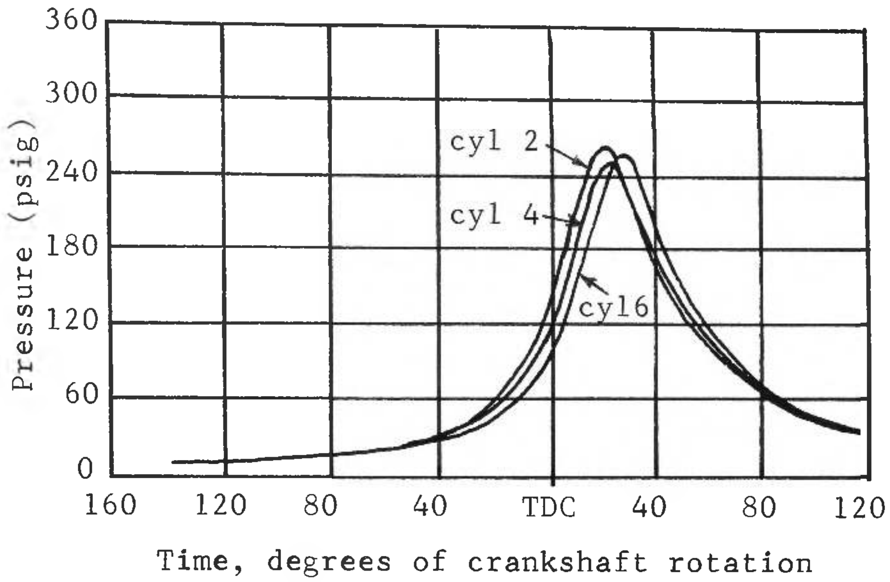

FIGURE 8 : TYPICAL PRESSURE-TIME TRACES SHOWING CYLINDER-TO-CYLINDER VARIATION. CYLINDERS 2, 4, 6.

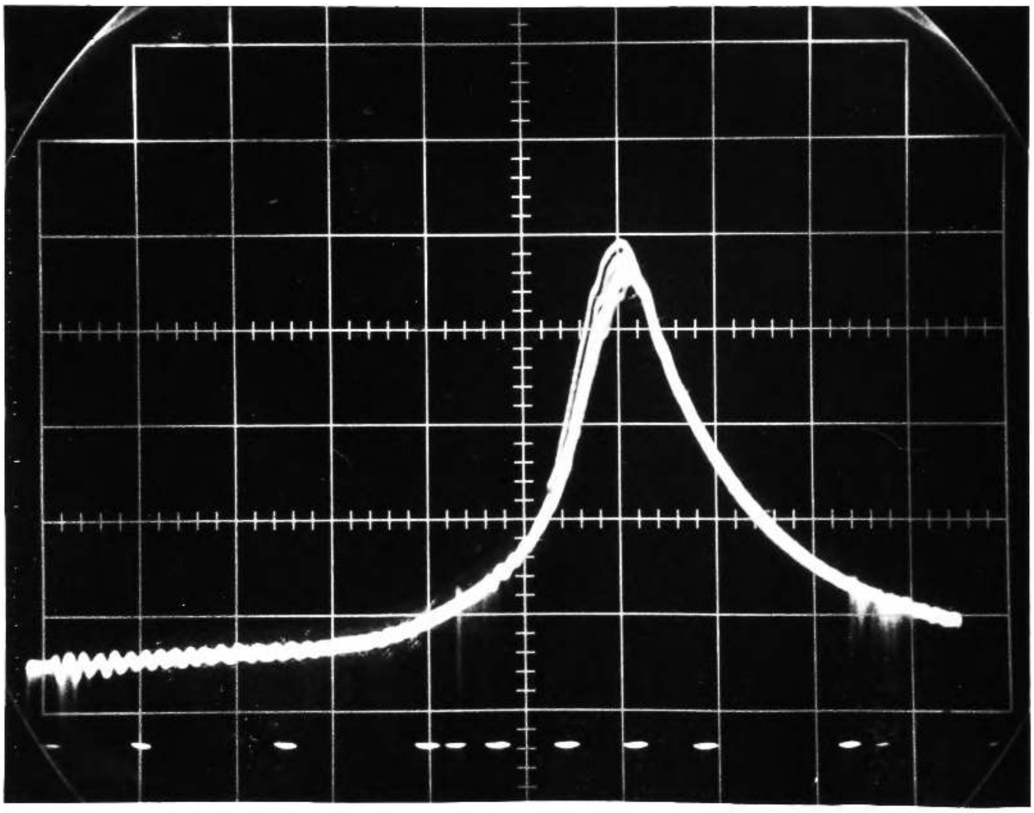

FIGURE 9 : PRESSURE-TIME TRACES SHOWING TYPICAL CYCLE-BY-CYCLE VARIATION. CYLINDER 6.

on the pressure-time curves.

To conduct such a comparison using the pressure-time curves of the two fuels, some point on the curve signifying the completion of combustion must be chosen.

Reports by Rassweiler et al [7] and Marvin et al [8,9] on the examination of flame movement and pressure development using photographic techniques to record flame front movement indicate that, for higher compression engines, the completion of combustion occurs very close to the point of maximum pressure. Variations of the air-fuel ratios and ignition timing away from the optimums reduced the amount of combustion completed at the peak pressure.

Cylinder pressure variation is a fundamental problem in spark ignition engines. Patterson[6] and many others have shown that no two cylinders of a multicylinder engine produce identical average pressure records. For this reason pressure-time traces were recorded for each of the three cylinder tappings. Typical cylinder-to-cylinder variations are illustrated in Figure 8. Because of the variation between cylinders, it was decided to use one particular cylinder (cylinder 6) for the test program.

Cycle-by-cycle variability for a given cylinder can also present problems, particularly where the pressure-time curves are recorded by photographing single traces on the oscilloscope screen. An advantage of having the Polaroid camera was that each photograph of a pressure-time trace could be examined during the tests and judged as to its suitability as an 'average' trace. Figure 9 illustrates the cycle-by-cycle variability recorded over several consecutive cycles. Although it was observed that, for a particular set of operating conditions, cycle-by-cycle variability gave a range of peak pressures, the crank angle position at which the peak pressure occurred remained essentially constant. Thus the point at which peak pressure occurred was used to signify the completion of combustion for the purpose of comparison of ignition -burning periods for optimum pressure development conditions.

Discussion of the experimental work and results is divided into four sections to enable comparisons to be made between the two fuels in relation to the following-

(a) Ignition-burning periods and pressure development.

(b) Effects of engine speed.

(c) Effects of ignition timing.

(d) Effects of air-fuel ratio.

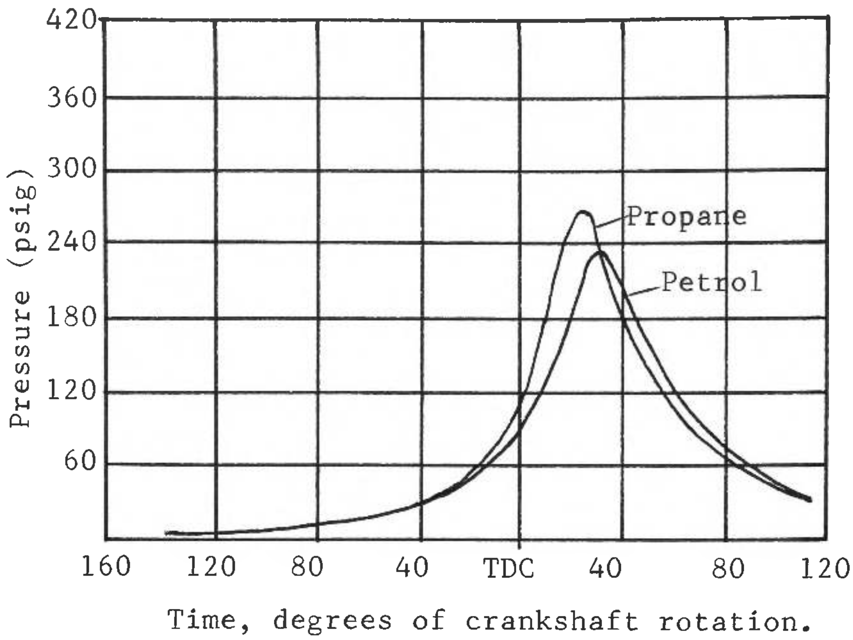

FIGURE 10 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AND PROPANE AT 2000 RPM, OPTIMUM OUTPUT. IGNITION ADVANCE-PETROL 32^{o}BTDC, PROPANE 28^{o}BTDC.

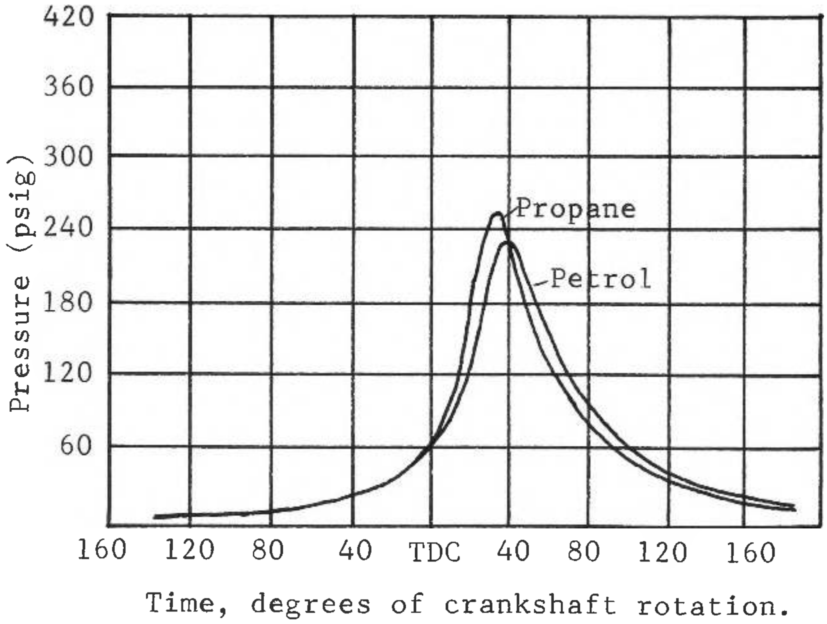

FIGURE 11 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AND PROPANE AT 2500 RPM, OPTIMUM OUTPUT. IGNITION ADVANCE-PETROL 32^{o}BTDC, PROPANE 29^{o}BTDC.

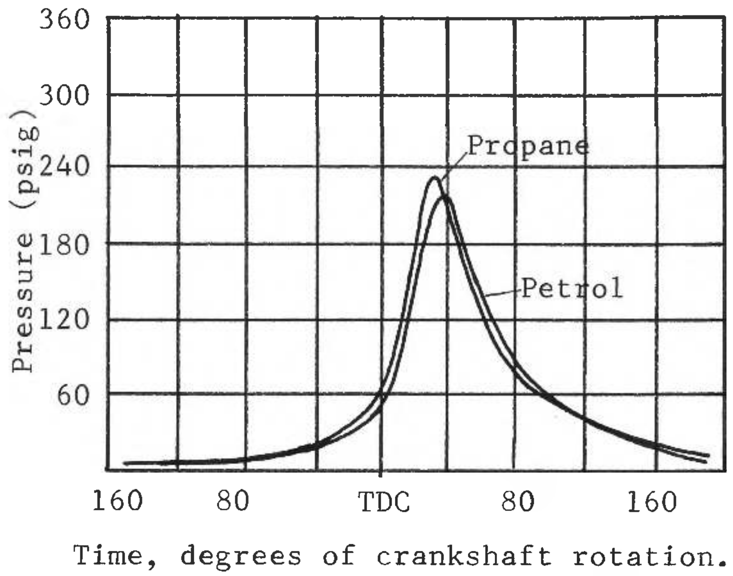

FIGURE 12 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AND PROPANE AT 3000 RPM, OPTIMUM OUTPUT. IGNITION ADVANCE - PETROL 32°BTDC, PROPANE 29°BTDC.

(a) IGNITION-BURNING PEROIDS AND PRESSURE DEVELOPMENT.

To compare the combined ignition-burning peroids of the two fuels, full throttle tests were conducted at a number of engine speeds and at optimum ignition and mixture settings.(ie, leanest mixture for best torque and minimum advance for best torque.) For reasons mentioned earlier, the point at which peak pressure occurred is chosen to signify the completion of combustion for the optimum conditions.

Figures 10, 11 and 12 illustrate typical pressure-time curves for speeds of 2000, 2500 and 3000 rpm respectively. The ignition timings used are noted on each trace and combined ignition-burning periods and maximum rates of pressure rise for the figures are given in Table 1.

From the pressure-time curves it is observed that propane has a shorter combined ignition-burning period. For example, at 2500 rpm (Fig.11) the period for propane was scaled at 64 crank angle degrees (4.26 msec) compared with 72 crank angle degrees (4.80 msec) for petrol; a difference of 8 degrees (0.54 msec).

Peak pressure is seen to occur earlier for the propane fuel for optimum output conditions. For example, seperation of the peaks at 2500 rpm (Fig.11) is approximately 5 degrees (0.33 msec).

The rate of pressure increase on the pressure-time curves is strongly dependent on the combustion rate. From the maximum gradients measured from the pressure-time curves it is found that the propane fuel generally acheived a higher rate of pressure rise, ie, a faster combustion rate. For example, from Figures 10, 11 and 12 the maximum rates of pressure rise are 72 psi/msec, 90 psi/msec and 144 psi/msec respectively for petrol as compared to 108 psi/msec, 135 psi/msec and 144 psi/msec respectively for propane.

When comparing the pressure-time traces for the two fuels, the differences in pressure development are usually evident before top dead centre is reached. This would indicate a slight increase in the negative work (compression) occurring with the propane.

Throughout the tests peak pressures for propane were found to be greater than those for petrol at corresponding operating conditions even though the engine output was less with the propane. The greater rate of pressure rise appears to result in the higher peak pressures. However, because of this steeper rate of pressure increase, the area under the propane pressure-time curve after top dead centre tends to be slightly less than that for petrol. As the difference between this positive area and the negative (compression) work area (found to be slightly greater for propane) is proportional to engine output, the smaller resultant area would agree with the lower engine outputs obtained with propane. Table 1 gives torque values corresponding to the pressure-time curves of Figures 10, 11 and 12.

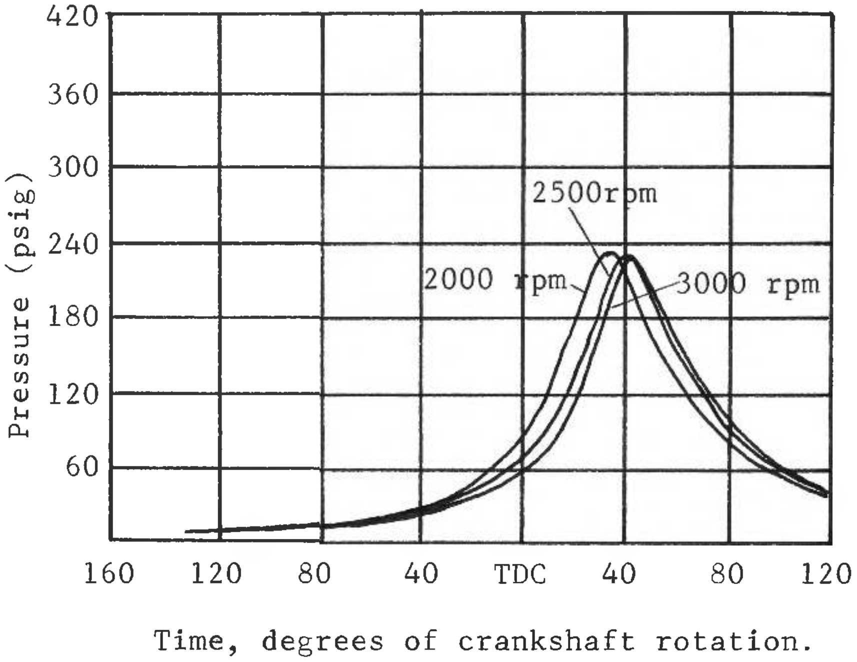

FIGURE 13 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT SPEEDS OF 2000, 2500, AND 3000 RPM. IGNITION ADVANCE 32°BTDC.

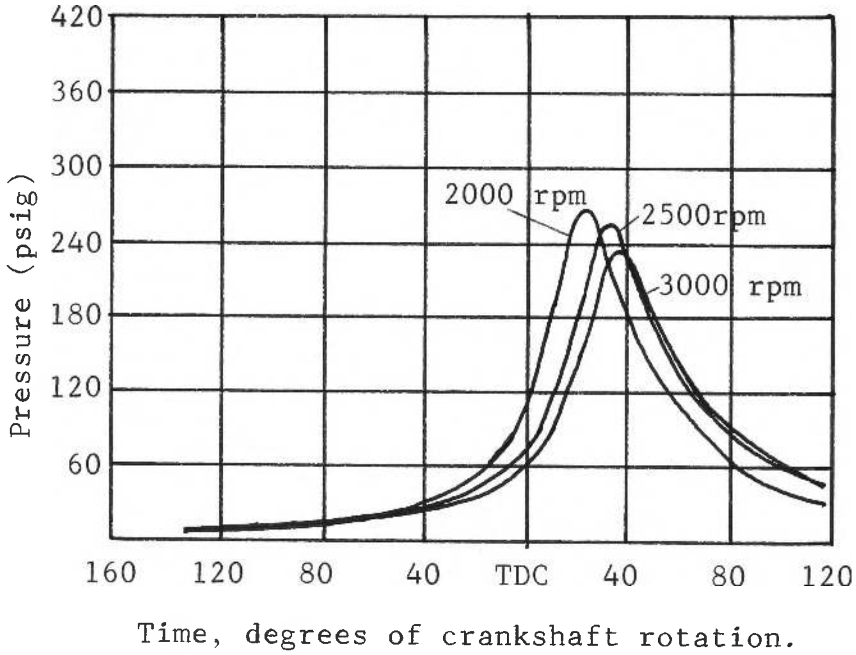

FIGURE 14 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT SPEEDS OF 2000, 2500 AND 3000 RPM. IGNITION ADVANCE - 2000 RPM 28°BTDC, 2500 AND 3000 RPM 29°BTDC.

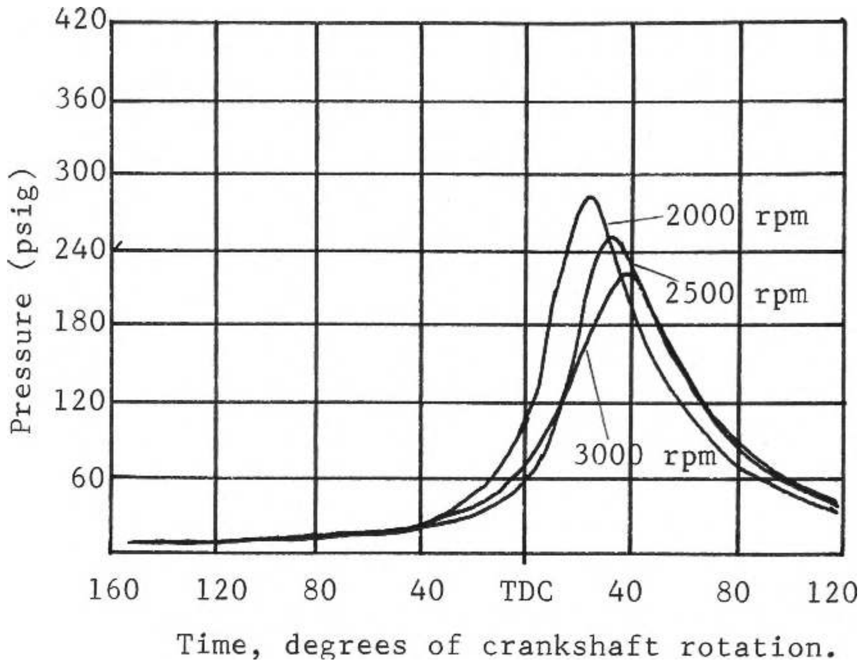

FIGURE 15 : TYPICAL PRESSURE-TIME TRACE FOR PETROL AT SPEEDS OF 2000, 2500 AND 3000 RPM . IGNITION ADVANCE 40^{o}BTDC.

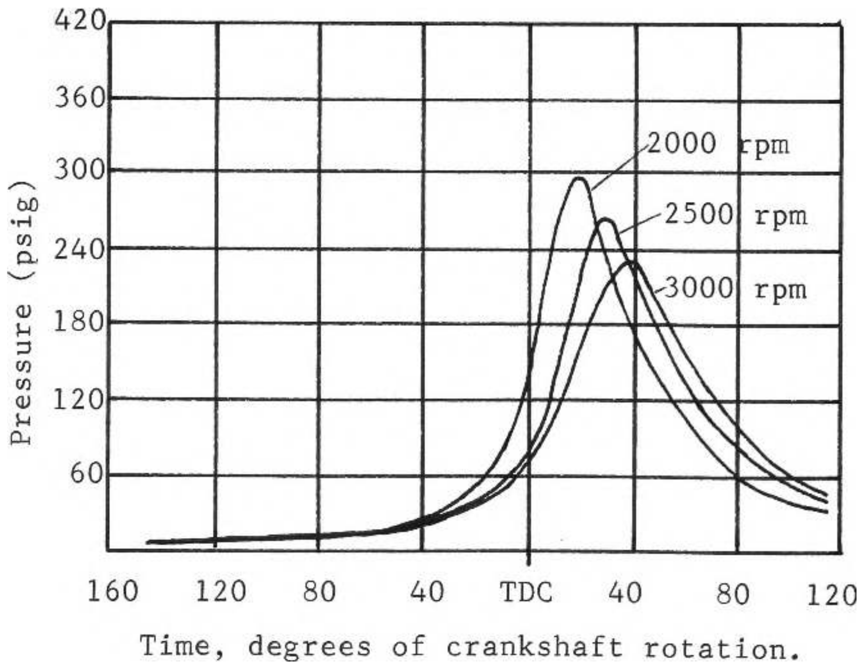

FIGURE 16 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT SPEEDS OF 2000, 2500 AND 3000 RPM. IGNITION ADVANCE 40^{o}BTDC.

(b) EFFECT OF ENGINE SPEED.

As noted in publications by Litchy[5], Marvin and Best[8] and Rassweiler et al[9] it has generally been established that flame speed increases with engine speed. The increase of turbulence of the fuel mixture with engine speed and its effects on flame growth and flame front velocity are considered the main reasons for the increase. Ignition timing is normally advanced as engine speed increases to maintain the desired pressure development and engine output.

Pressure-time traces were recorded at selected engine speeds and ignition timings with full throttle; the air-fuel ratio being adjusted to the leanest mixture for best torque.

The effects of engine speed on pressure development for the two fuels are illustrated in Figures 13, 14, 15 and 16. Figures 13 and 14 are typical pressure-time traces for optimum ignition timing at speeds of 2000, 2500 and 3000 rpm. Figures 15 and 16 are typical pressure-time traces for the same speeds but at a constant ignition timing of 40^{o} BTDC for both fuels.

It can be seen that both fuels exhibit similar characteristics for the engine speed variation. As the engine speed increases the peak pressures are reduced in magnitude and the peak occurs at a later crank angle position. Marvin, Wharton and Roeder[9] found that the flame velocity and the time rate of pressure development increase only a little more slowly than engine speed. This slight difference would explain the above characteristics.

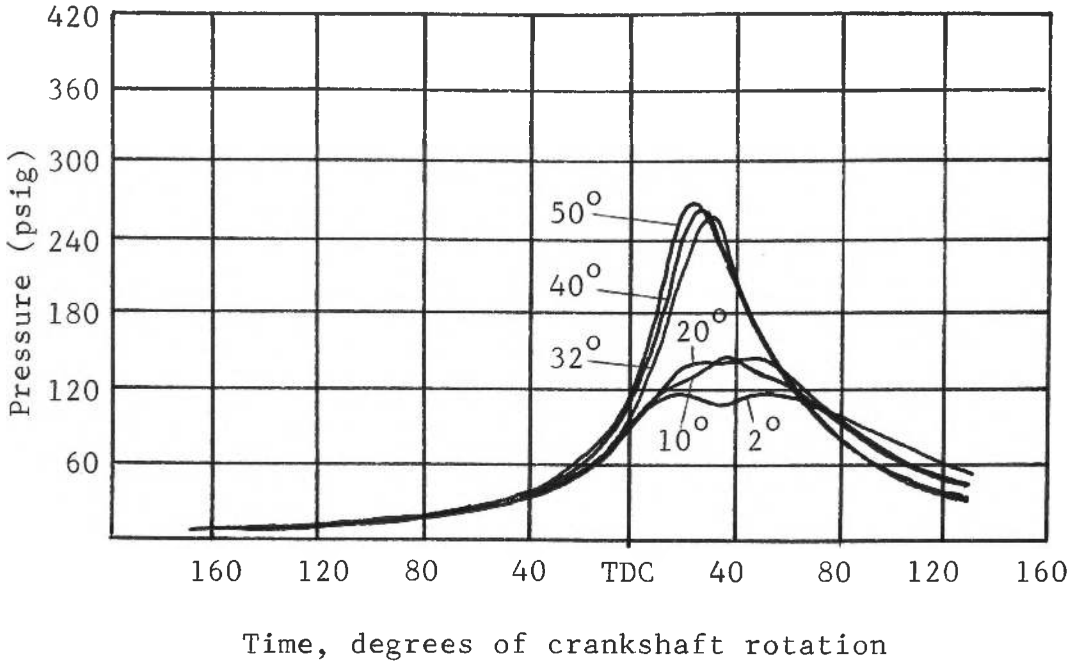

FIGURE 17 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT 2000 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

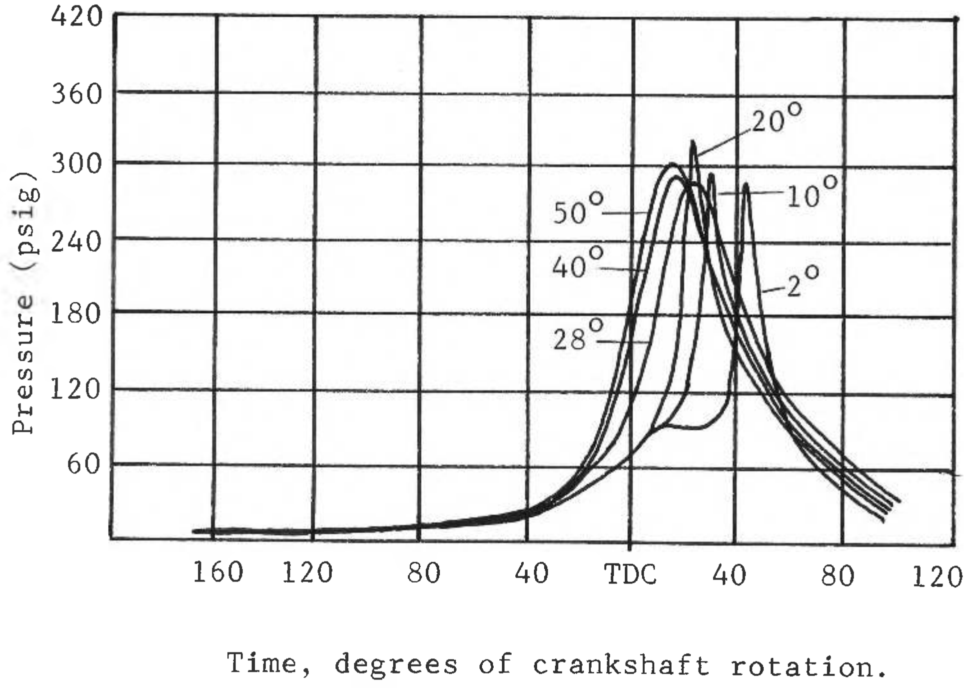

FIGURE 18 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT 2000 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

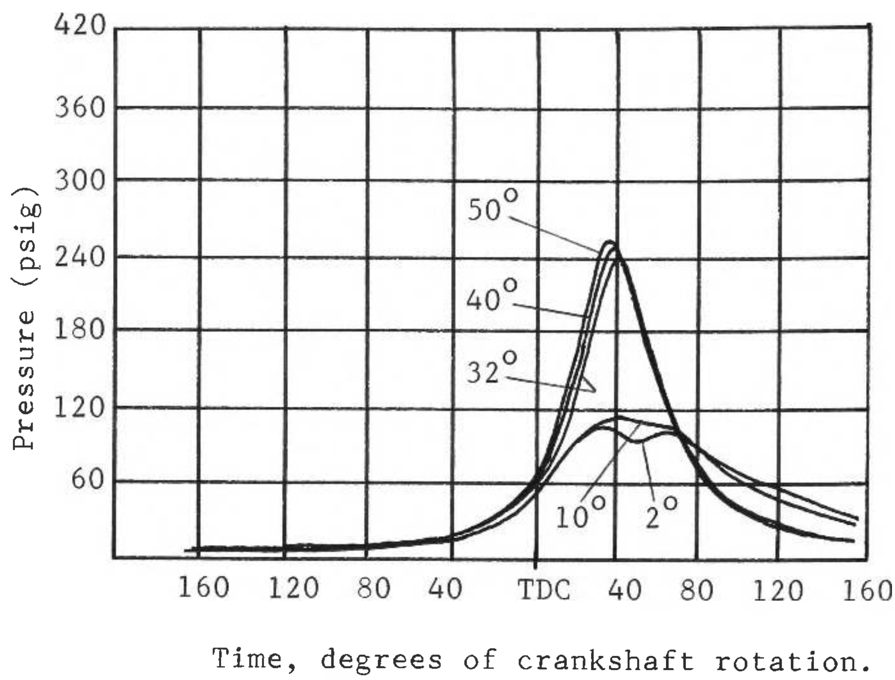

FIGURE 19 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT 2500 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

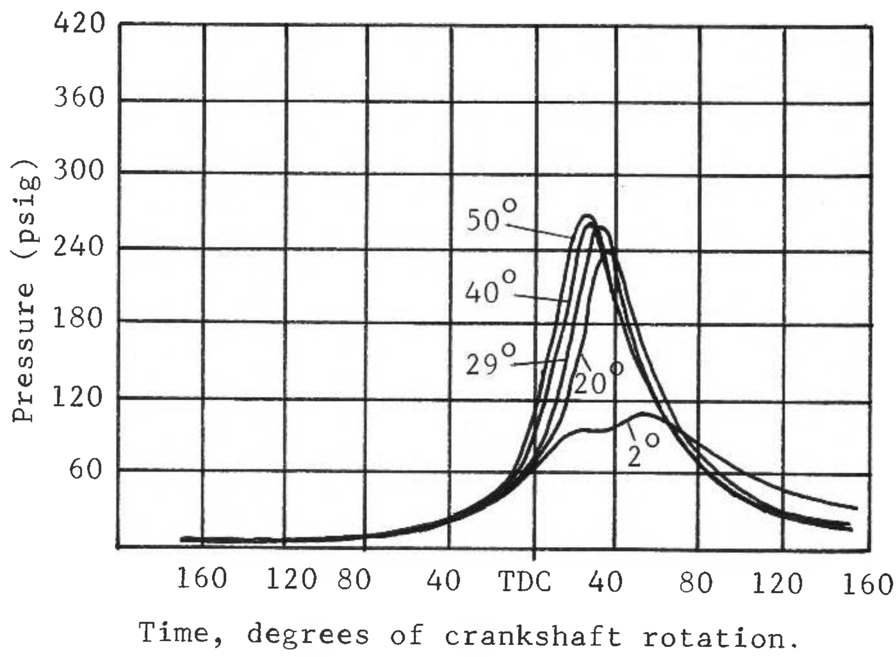

FIGURE 20 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT 2500 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

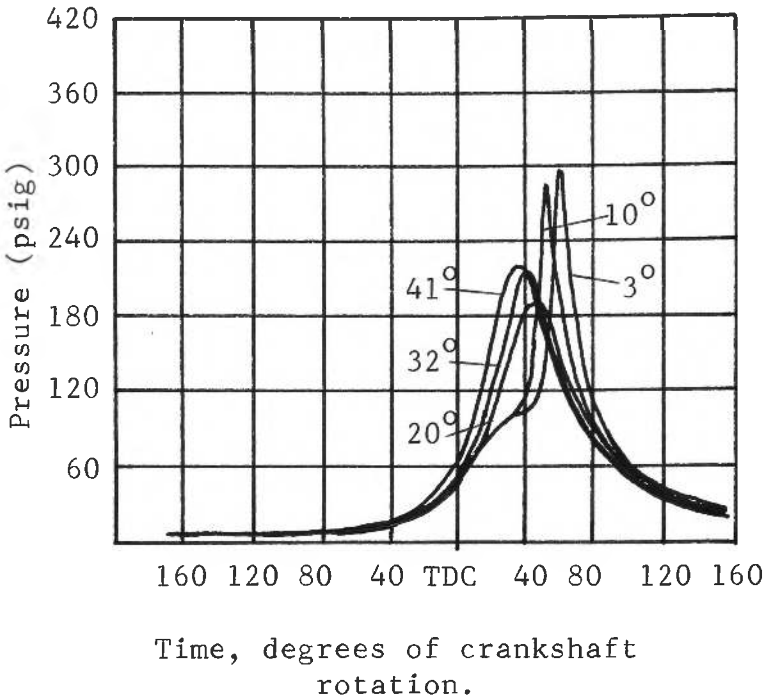

FIGURE 21 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT 3000 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

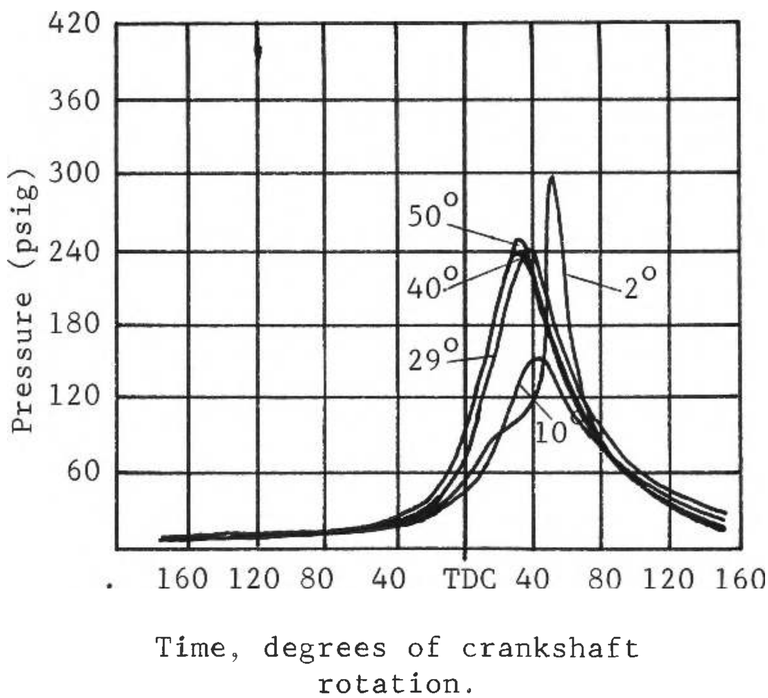

FIGURE 22 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT 3000 RPM AND VARIOUS IGNITION ADVANCE SETTINGS.

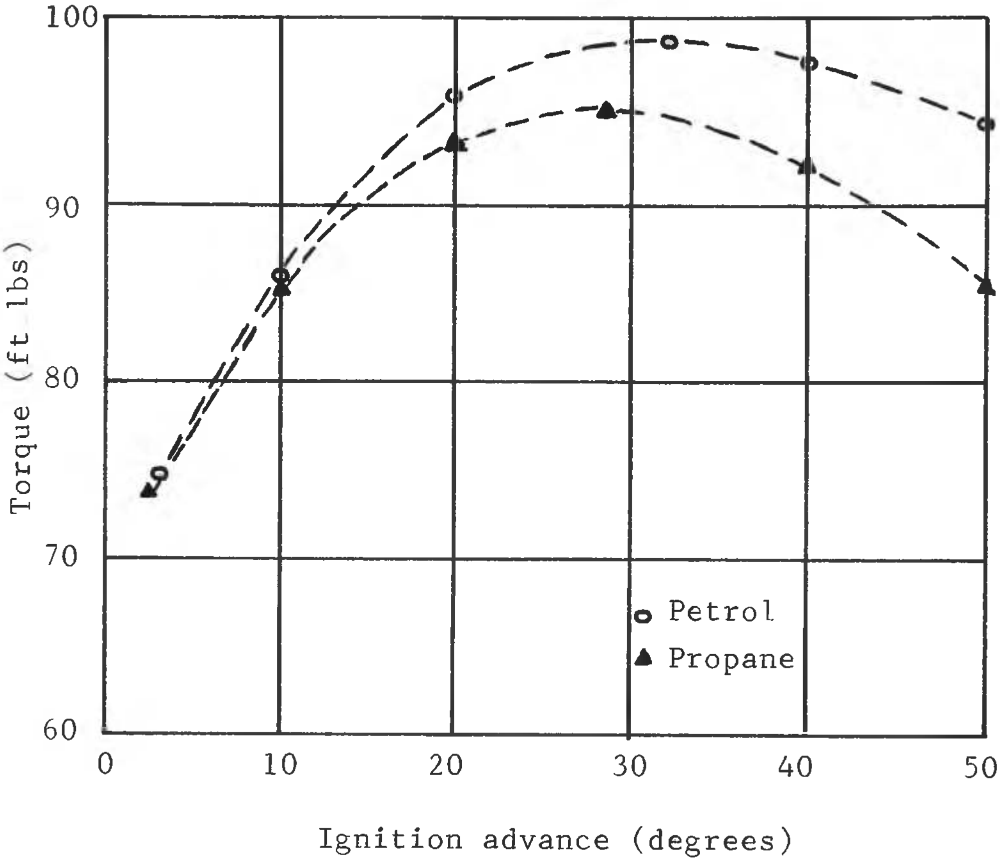

FIGURE 23 : TORQUE VERSUS IGNITION ADVANCE FOR PETROL AND PROPANE AT 2000 RPM.

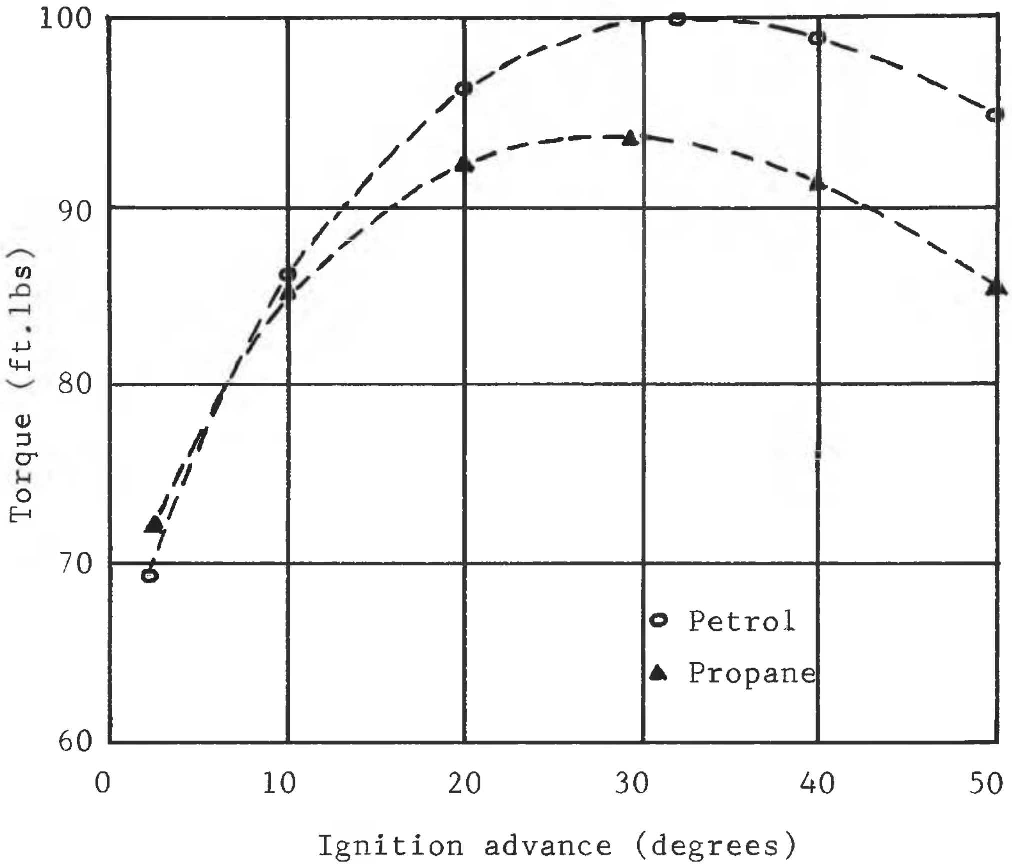

FIGURE 24 : TORQUE VERSUS IGNITION ADVANCE FOR PETROL AND PROPANE AT 2500 RPM

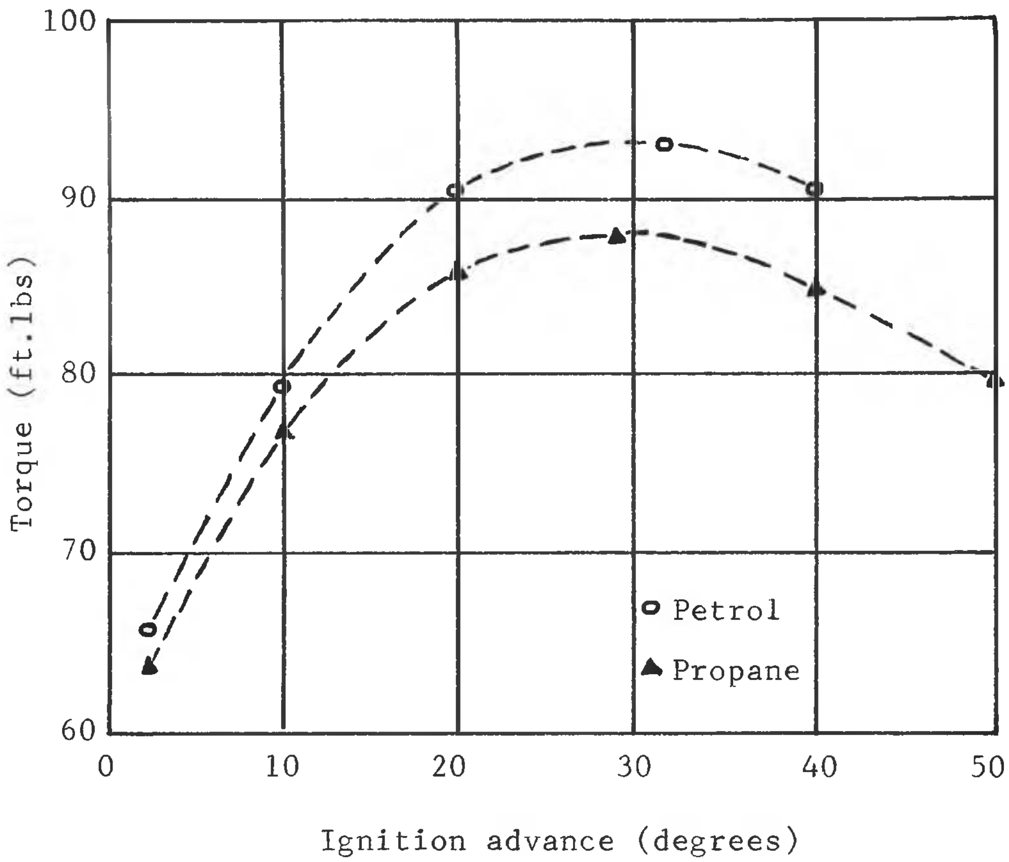

FIGURE 25 : TORQUE VERSUS IGNITION ADVANCE FOR PETROL AND PROPANE AT 3000 RPM.

(c) EFFECT OF IGNITION TIMING.

The timing of ignition is an important factor in the pressure development and power output of the spark ignition engine and the requirements vary with fuel type and engine characteristics.

The optimum performance timing requirements for the test engine, as found in earlier tests[2,3], have already been shown in Figure 7. During these earlier tests it was noted that the propane was more sensitive to changes in ignition timing.

To examine the effects of the variation of ignition timing on pressure development and engine performance at full throttle, tests were conducted at selected engine speeds and with the air-fuel ratio adjusted to leanest mixture for best torque. For each engine speed the ignition timing was varied in steps from 2^{o} BTDC to 50^{o} BTDC. Pressure-time traces and engine output were recorded for each ignition timing, the test results being illustrated in Figures 17 through 25 and tabulated in Tables 2, 3 and 4.

The pressure-time curves obtained indicate that the fuels exhibit similar characteristics with the variation of ignition timing.

As the point of ignition is advanced from 2^{o} BTDC to 50^{o} BTDC the peak pressures increase for both fuels; the propane developing slightly higher peak pressures because of the shorter combined ignition-combustion period for the fuel.

The negative (compression) work area under the pressure-time curves increases with the advance of ignition. As seen from the typical pressure-time curves illustrated in Figures 17 through 22 the rate of increase of this negative work area with ignition advance is greater with propane than with petrol, again as a result of the shorter combined ignition-combustion period of propane.

Considering the differences between the combined ignition-combustion periods of the two fuels, it can be reasoned that for ignition timings further advanced from the optimum, the engine output would be more sensitive with propane; whereas for ignition timings retardedfrom the optimum, the engine output would be more sensitive with petrol.

Actual results obtained are in agreement with the above reasoning. Figures 23, 24 and 25 illustrate the engine torque versus ignition timing for speeds of 2000, 2500 and 3000 rpm and it is seen from these figures that engine output is more sensitive with the propane for timing in advance of the optimum and more sensitive with petrol for timing retarded from the optimum.

It was also found during the tests that, for the more retarded ignition timings (10^o and 2^o BTDC), 'late' pressure peaks occurred. Typical examples are included in Figures 18, 21 and 22. These 'late' pressure peaks were more prevalent with propane at the lower speeds but occurred with similar frequency for both fuels at the higher speeds, eg. 3000 rpm. The late ignition and ' delayed ' combustion would be mainly responsible for these late peaks.

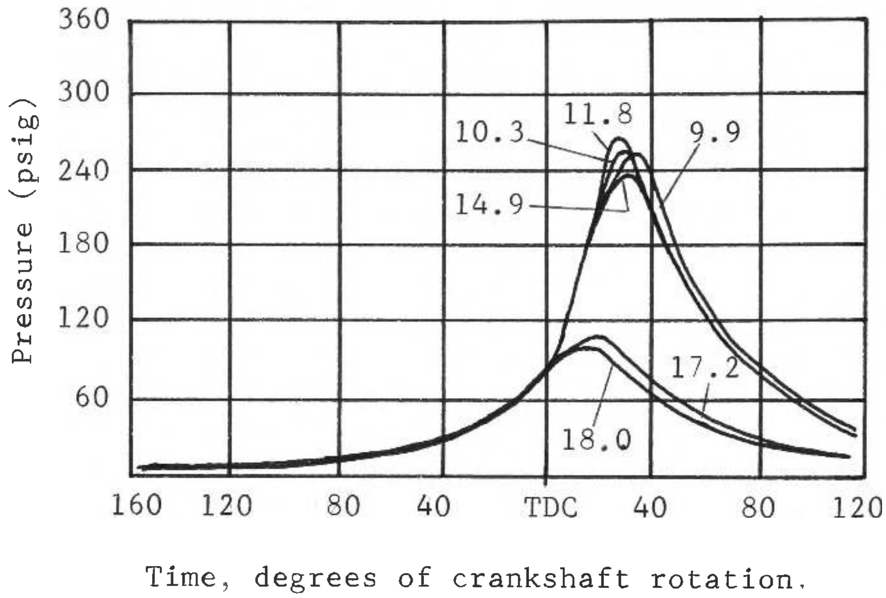

FIGURE 26 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT 2000 RPM AND VARIOUS AIR FUEL RATIOS.

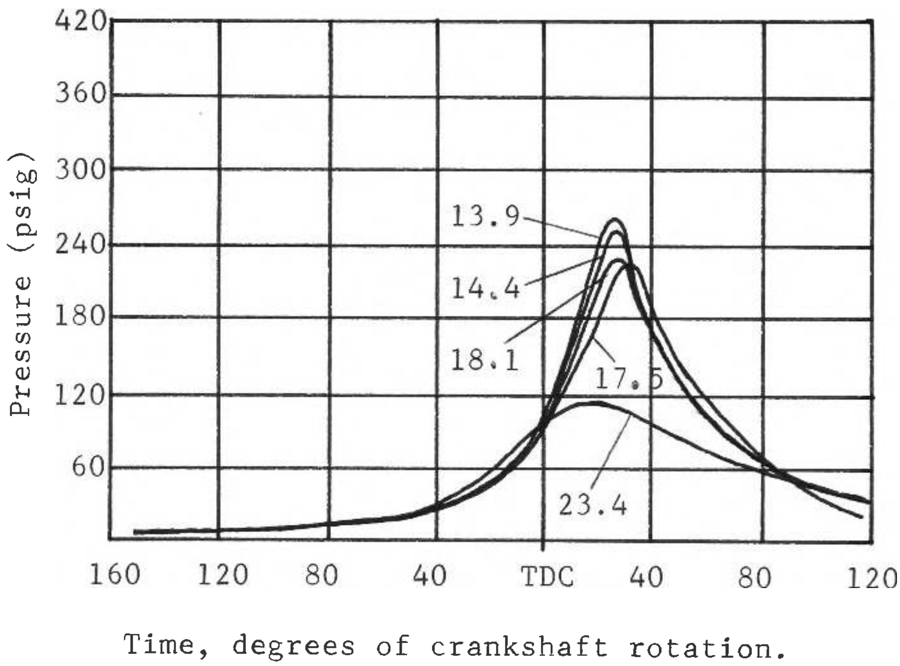

FIGURE 27 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT 2000 RPM AND VARIOUS AIR FUEL RATIOS.

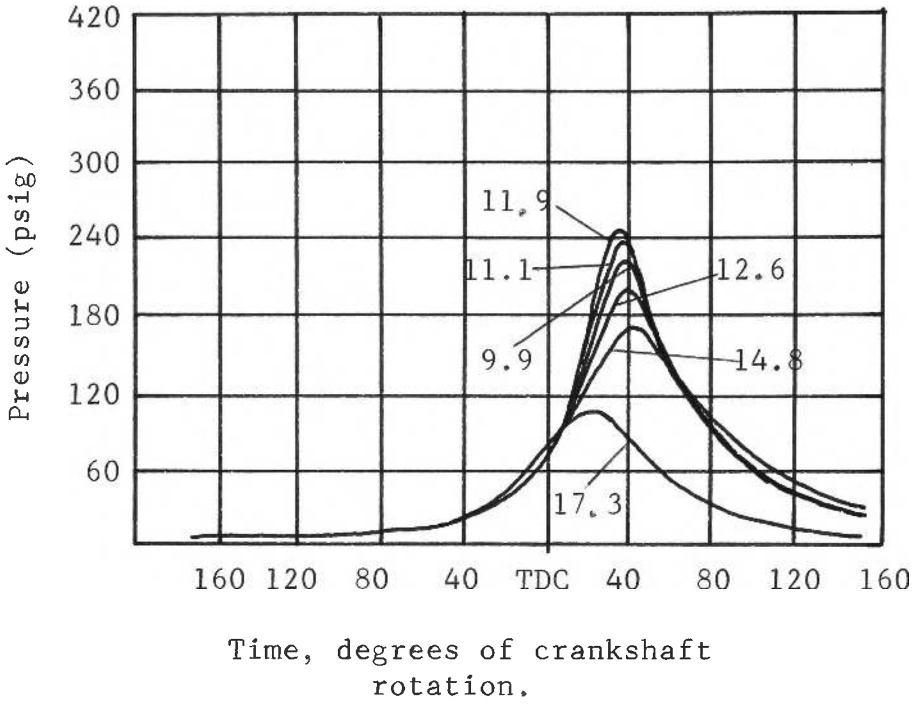

FIGURE 28 : TYPICAL PRESSURE-TIME TRACES FOR PETROL AT 2500 RPM AND VARIOUS AIR FUEL RATIOS.

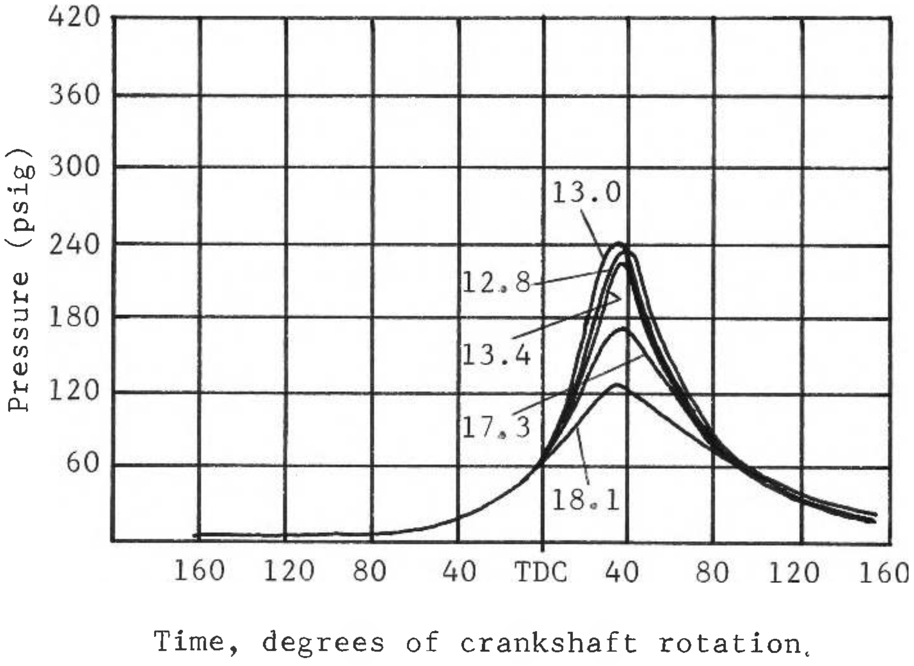

FIGURE 29 : TYPICAL PRESSURE-TIME TRACES FOR PROPANE AT 2500 RPM AND VARIOUS AIR FUEL RATIOS.

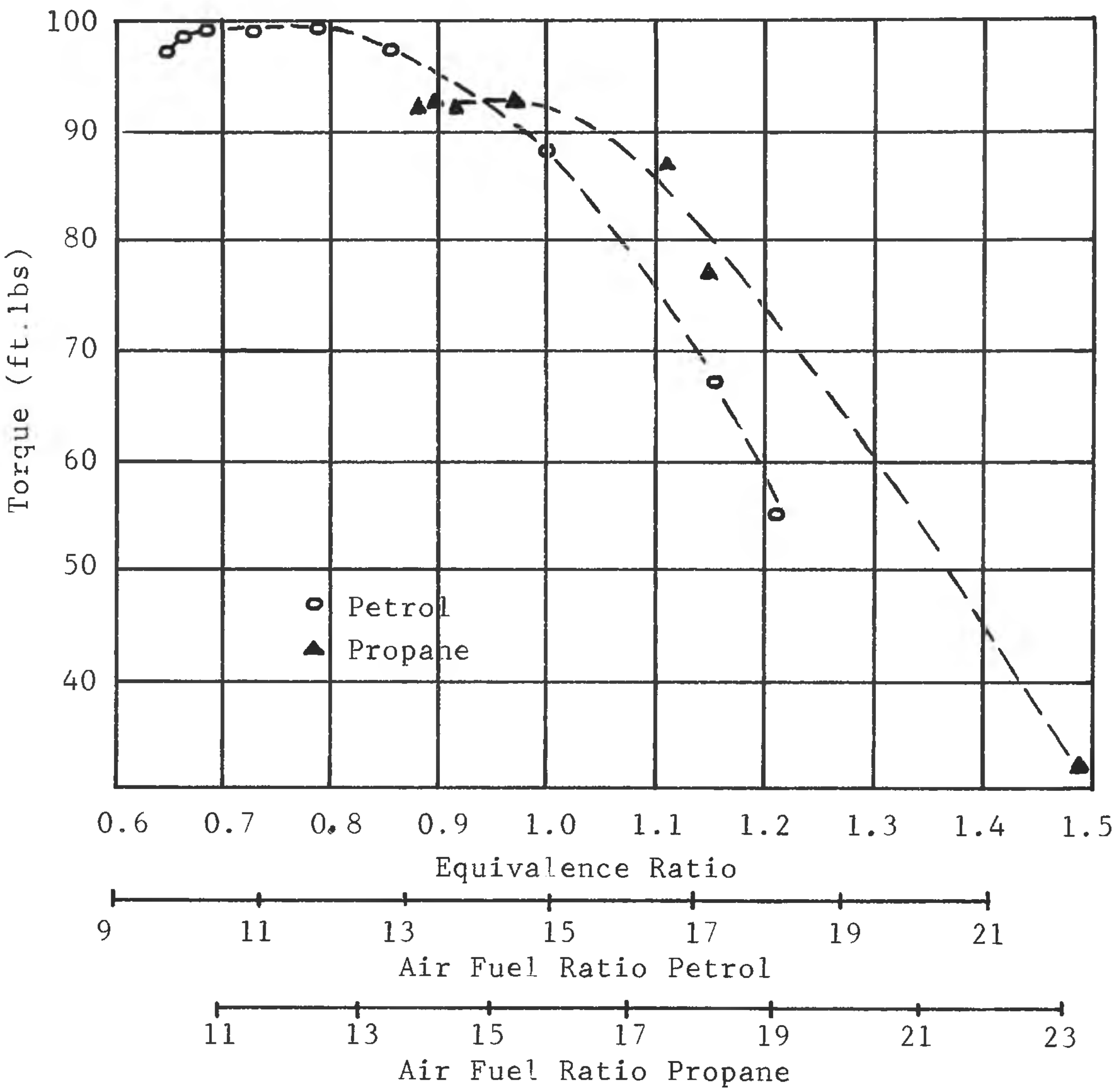

FIGURE 30 : AIR FUEL RATIO VERSUS TORQUE FOR PETROL AND PROPANE AT 2000 RPM.

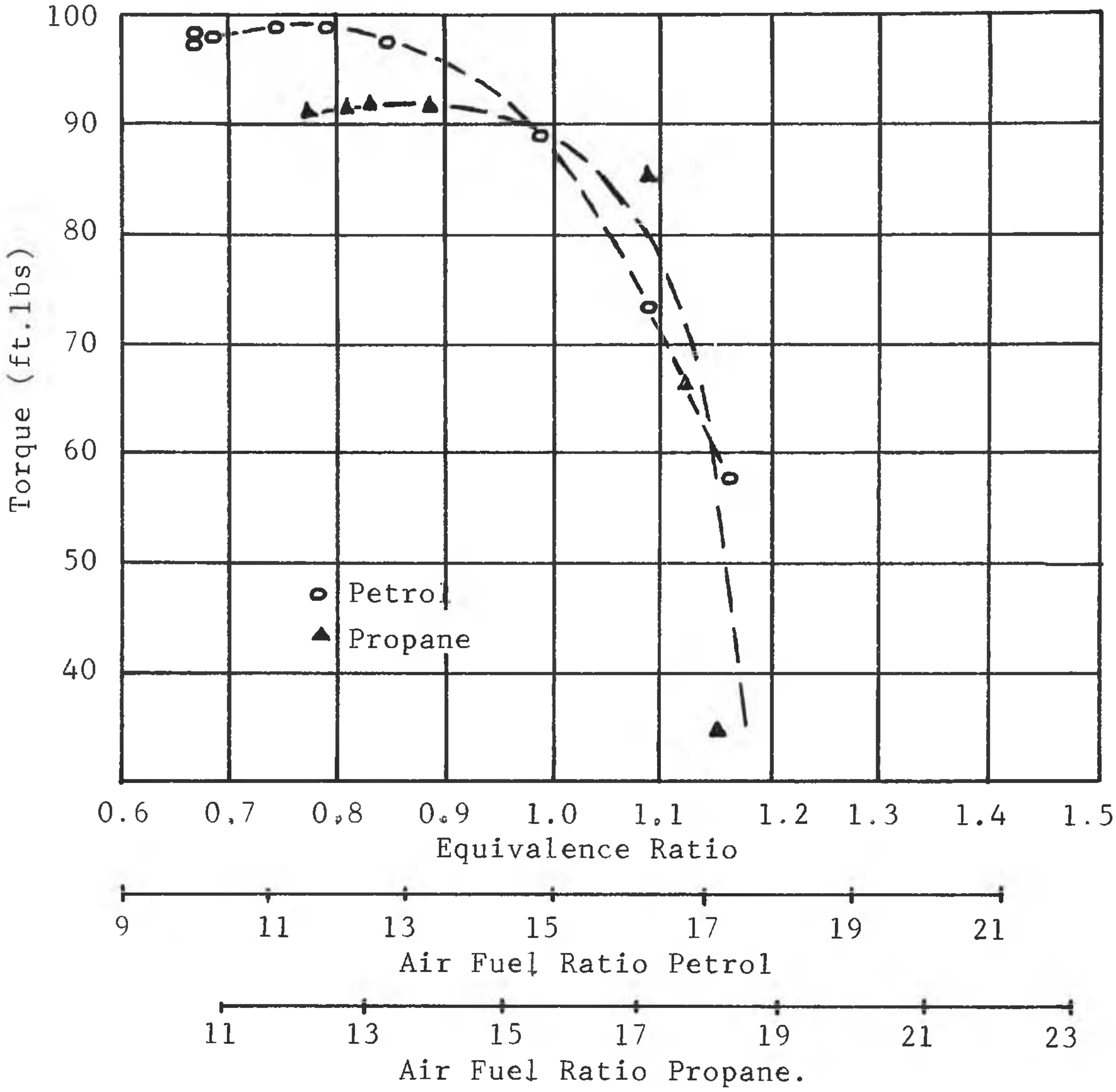

FIGURE 31 : AIR FUEL RATIO VERSUS TORQUE FOR PETROL AND PROPANE AT 2500 RPM.

(d) EFFECT OF AIR-FUEL RATIO.

The effects of air-fuel ratio on pressure development were examined at various engine speeds. The tests were conducted at full throttle with ignition timimg set for maximum engine output at the optimum air-fuel ratio.(ie, leanest mixture for best torque)

Both fuels were found to exhibit similar characteristics with variation of the air fuel ratio. Figures 26 and 27 (2000 rpm) and 28 and 29 (2500 rpm) illustrate pressure-time curves typical of those obtained during the tests. The corresponding torque versus air fuel ratio relationships are shown in Figures 30 and 31 and tabulated in Tables 5 and 6. The equivalence ratio (based on 14.9 stoichiometric for petrol and 15.7 stoichiometric for propane) is included to allow a more meaningful comparison of performance versus ' leanness ' between the two fuels.

Comparison of the pressure-time curves with the corresponding torque curves shows that for both fuels the maximum rates of pressure rise and peak pressures occur for the air fuel ratios in the region of the leanest mixture for best torque. As the air fuel mixture is weakened from this setting, the rate of pressure rise decreases and the peak pressure decreases and occurs at a later angle.

Mixtures 'richer' than those of the optimum region were obtained with the petrol carburettor. However, as mentioned earlier, the design of the propane carburettor prevented the richer mixtures from being achieved.

The pressure-time curves obtained for petrol indicate that rich mixtures result in characteristics similar to those of weak mixtures, ie, the rate of pressure rise decreases, and the peak pressure decreases and occurs at a later angle. Marvin and Best[8] studied flame speed by photographic techniques and found that a decrease in flame speed was caused by using a mixture richer or leaner than that giving maximum power. Similarly Rassweiler, Witwrow and Cornelius[7] studied combustion rates by photographic techniques and found that a weakening of the mixture resulted in an increased combustion time accompanied by the decrease in pressure rise rates and the lower peak pressures occurring at later angles.

SECTION IV

CONCLUSIONS AND BIBLIOGRAPHY

CONCLUSIONS

The test program reported above leads to the following conclusions;

The LPG fuel has a shorter combined ignition-burning period than that of petrol. This results in the revised ignition timings being required to realise full power from an engine converted from petrol to propane operation.

The LPG gives higher rates of cylinder pressure rise during combustion indicating a faster combustion rate than that of petrol.

The LPG generally gives higher peak pressures than those of petrol. However, because of the steeper rate of pressure increase with LPG and the increase in the negative work (compression before TDC) , the overall cycle work under the pressure-time curve is slightly less than that of petrol. This is reflected in the slightly lower torque values obtained with LPG.

For variations in operating parameters (engine speed, ignition timing and air fuel ratio) the two fuels are found to react similarly. As engine speed increases, the peak pressures decrease in magnitude and the peak occurs at a later crank angle position.

As the point of ignition is advanced from top dead centre, the peak pressures increase in magnitude for both fuels; the LPG developing slightly higher peak pressures as a result of the shorter combined ignition-burning period.

Results show that, for ignition timings further advanced from the optimum, the engine output is more sensitive with LPG, whereas, for ignition timings retarded from the optimum, the engine output is more sensitive with petrol. This is explained by the differences between the combined ignition-burning periods of the two fuels and the effect on the areas under the pressure-time curves.

Variation of the air fuel ratios shows that the maximum rate of pressure rise and peak pressure occurs,for both fuels, in the region of leanest mixture for best torque. A leaner or richer mixture results in a decrease in the rate of pressure rise and a decrease in the peak pressures which occur at a later angle.

The particular techniques used for these experiments, ie, photographic recording of single traces of the pressure-time traces from an oscilliscope screen, are limited in the sense that proper statistical evaluation of many samples of the cyclic variations that occur in cylinder pressure development could not be made.

A better understanding of events taking place during combustion in the spark ignition engine is now being obtained with modern techniques such as the use of ionisation gap sensors from which combustion rate data is monitored and statistically analysed by computer. Investigations by such people as Curry[10] and Starkman et al[11] are helping to obtain such an understanding.

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SECTION V

APPENDICES

APPENDIX A

EQUATIONS FOR ENGINE PERFORMANCE FACTORS.

POWER. For the particular dynamometer used the power is given by the equation stated by the manufacturers.

$$BHP = \frac{W \times N}{400} \tag{1}$$

where: BHP = brake horse power.

W = dynamometer balance reading.

N = engine speed, rpm.

TORQUE. The relationship between power and torque is-

$$BHP = \frac{2\pi NT}{33{,}000} \tag{2}$$

where T = torque, ft lbf.

Equating (1) and (2) gives an expression for torque in terms of W-

$$T = 13.13\ W \tag{3}$$

FUEL CONSUMPTION. For petrol, the fuel consumption in lbm/hr. is calculated from the time (t) required to use 100 ccs of petrol of specific gravity 0.78.

$$\text{lbm/hr} = \frac{100 \times 35.315 \times 10^{-6} \times 62.4 \times 0.78 \times 3600}{t}$$

$$= \frac{618.79}{t} \tag{4}$$

For the propane, the fuel consumption was measured in grams over a period of 60 seconds. Thus the consumption in lbm/hr is given by-

$$\text{lbm/hr} = \frac{m \times 60}{453.6} = 0.1322\ m \tag{5}$$

where: m = grams/60 secs.

AIR FUEL RATIO.

$$\text{A/F ratio} = \frac{\text{mass flow rate of air}}{\text{mass flow rate of fuel}} \qquad (6)$$

The mass flow rate of air is obtained from the equation for the orifice plate given at the end of Appendix B. The mass flow rate of fuel is obtained from equation (4) or (5).

VOLUMETRIC EFFICIENCY. The volumetric efficiency (η_v) is given for a particular engine speed as-

$$\eta_v = \frac{M_{act}}{M_{th}} \times 100 \quad (\%)$$

where: M_{act} = actual mass flow rate of air.

M_{th} = theoretical mass flow rate to fill the piston displacement volume at atmospheric conditions.

For the engine used, having a swept volume of 2268 cc and assuming the density of the air as 0.0747 lb/ft^3 the equation becomes-

$$\eta_v = \frac{M_{act}}{0.1797\ N} \times 100 \quad (\%) \qquad (7)$$

where N = engine speed, rpm.

APPENDIX B

ORIFICE PLATE CALCULATIONS USING B.S. 1042 Part I.[14]

Assuming 80% volumetric efficiency of the engine it was found that the mass flow rate of air required at the maximum speed was 575 lbm/hr.

It is desired to have a pressure drop of approximately 2 inches (50 mm) H_2O across the orifice at the maximum flow rate.

An orifice plate with corner tappings (Section 7 of the Code) is chosen as it is suited for measuring flow from a large space into a pipe.

The downstream pipe has an average internal diameter of 6.46 inches. According to Clause 49a of the Code, the orifice diameter can not exceed half the downstream pipe diameter.

The atmospheric conditions for the initial design are taken as-

Temperature,	$T = 20^{o}C$ ($68^{o}F$)
Pressure,	$P = 14.696$ lbf/in^2
Relative humidity,	$\phi = 70\%$

To find the approximate orifice diameter required, the equations 10 and 12a of Clause 14a are used.

$$N = \frac{W}{359.2\ D^2\ h^{\frac{1}{2}} \rho^{\frac{1}{2}}} \tag{10}$$

and

$$CmE = \frac{N}{Z\,\varepsilon} \tag{12a}$$

where: W = mass flow rate. (lbm/hr.)

D = downstream pipe dia. (ins)

h = differential across orifice. (ins H_2O)

ρ = density of air. (lbm/ft^3)

C = basic coefficient.

m = area ratio = $\frac{d^2}{D^2}$

d = orifice dia. (ins)

E = velocity of approach factor.

Z = correction factor for pipe diameter and Reynolds No.

ε = expansibility factor.

To calculate the density at the assumed conditions, Clause 25c gives-

$$\rho = 2.700 \left[\frac{\delta}{KT}(P - p_v) + 0.62\ \frac{p_v}{T} \right]$$

where: δ = specific gravity = 1.000 for air.

K = gas law deviation coeff. = 0.999 (Fig.6)

$T = 528^\circ R$

P = 14.696 lbf/in^2

p_v = partial pressure of water vapour = $\phi\ p_{vs}$

$\phi = 0.7$

$p_{vs} = 0.339$ (Table 5)

Substitution of these values gives the density as 0.07473 lbm/ft^3.

With substitution equation 10 becomes-

$$N = 0.0993$$

To find an approximate value of mE it is assumed that Z and $\mathcal{E}$ in equation 12a are unity and thus-

$$CmE = N = 0.0993$$

and for this application of flow from a large area into a pipe, Clause 56a states that the basic coefficient C = 0.596. Thus-

$$mE = \frac{N}{C} = 0.166$$

Appendix J of the Code gives for mE = 0.166 the ratio $\frac{d}{D}$ =0.404

Therefore-

$$d = (0.404)(6.46) = 2.61 \text{ ins.}$$

For convenience, an orifice diameter of 2.500 inches will be considered.

The more exact calculations for the mass flow rate equation will now be carried out for a 2.500 inch diameter orifice. Equation 7 of Clause 13 of the Code gives the mass flow rate as-

$$W = 359.2 \ CZ\mathcal{E}E \ d^2 h^{\frac{1}{2}} \rho^{\frac{1}{2}} \qquad (7)$$

and

$$Rd = \frac{W}{15.8 \mu d} \qquad (7)$$

where: W = mass flow rate of air (lbm/hr)

d = orifice diameter (ins)

C = basic coeff.

$Z = Z_r Z_d$

Z_r = Reynolds number correction factor.

Z_d = pipe diameter correction factor.

ε = expansibility factor.

E = velocity of approach factor.

h = pressure differential across orifice (ins H_2O)

ρ = air density at working conditions.

μ = absolute viscosity of air (poise)

Rd = Reynolds number.

Now have d = 2.500ins, D = 6.46 ins, C = 0.596(Clause 56e) and for the assumed atmospheric conditions find $\mu = 1.81 \times 10^{-4}$ poise from Fig. 18. Therefore-

$$Rd = \frac{575}{1.81 \times 10^{-4} \times 2.500} = 80,425 \qquad \text{(Eqn.7)}$$

Z_r = 1.000 for Rd = 80,000 (Fig.35b)

Z_d = 1.000 (Clause 56g)

Taking the specific heat ratio as $\gamma = 1.4$ (Fig.21) and for a maximum pressure drop of h = 2 ins H_2O at a pressure of P = 14.696 lbf/in^2 giving $\frac{h}{P} = 0.136$, Figure 36 gives the expansibility factor as $\varepsilon = 0.998$.

For $\frac{d}{D} = 0.387$, Appendix J gives the velocity of approach factor as E = 1.0115.

The density was found to be $\rho = 0.07473$ lbm/ft^3 at the assumed conditions.

When these values are substituted into Equation 7, the mass flow rate equation becomes-

$$W = 369.412\ h^{\frac{1}{2}} \qquad \text{where h = inches } H_2O$$

or

$$W = 73.299\ h'^{\frac{1}{2}} \qquad \text{where h' = mm } H_2O.$$

Checking the differential pressure for the estimated maximum flow rate gives $h' = 61.4$mm H_2O. This is an acceptable differential and hence the 2.500 inch diameter is chosen.

The orifice plate and pressure tappings were made in accordance with the specifications of Clauses 54 and 55 of the Code. The minimum allowable length of downstream pipeline (Clause 48) for the area ratio of $m = 0.150$ is five times the inside diameter of the pipe, ie, 32.3 ins. The actual length used was $36\frac{1}{2}$ ins. thus satisfying the condition.

In actual operation, the volumetric efficiency of the engine at the maximum speed was less than the estimated 80% and the maximum pressure drops were in the order of 50 mm H_2O , the original desired maximum.

Thus, for the calculation of mass flow rates of air to the engine, the equation becomes-

$$W = 268.007 \left\{ 2.700 \left[\frac{(P - p_v)}{0.999\ T} + \frac{0.62\ p_v}{T} \right] \right\}^{\frac{1}{2}} h'^{\frac{1}{2}}$$

where: W = mass flow rate (lbm/hr)

P = atmospheric pressure. ($\mathrm{lbf/in^2}$)

T = atmospheric temperature. (^{o}R)

p_v = partial pressure of water vapour in the air.

h' = differential pressure across orifice. (mm H_2O)

APPENDIX C

TABULATION OF TEST DATA

TABLE 1

COMBINED IGNITION-BURNING PERIODS AND MAXIMUM RATES OF PRESSURE INCREASE

SPEED	FUEL	TIMING	TORQUE	IGNITION-BURNING PERIOD	MAX RATE OF PRESSURE INCREASE
rpm		^{o}BTDC	ft.lbf.	msec (deg)	psi/msec (psi/deg)
2000	propane	28	95	4.25 (51)	108 (9)
	petrol	32	100	5.25 (63)	72 (6)
2500	propane	29	94	4.26 (64)	135 (9)
	petrol	32	99	4.80 (72)	90 (6)
3000	propane	29	88	3.56 (64)	144 (8)
	petrol	32	95	4.0 (72)	144 (8)

TABLE 2

TORQUE VERSUS IGNITION ADVANCE

2000 RPM FULL THROTTLE

PROPANE Air fuel ratio 15 :1
Ambient temperature 23°C
Atmospheric pressure 759.4 mm Hg

IGNITION ADVANCE °BTDC	TORQUE ft.lbf
2	73.5
10	88.0
20	93.3
28	94.9
40	91.8
50	85.1

PETROL Air fuel ratio 12 : 1
Ambient temperature 21°C
Atmospheric pressure 762.4 mm Hg

IGNITION ADVANCE °BTDC	TORQUE ft.lbf
3	74.6
10	85.5
20	95.6
32	98.9
40	97.6
50	94.3

TABLE 3

TORQUE VERSUS IGNITION ADVANCE

2500 RPM FULL THROTTLE

PROPANE Air fuel ratio 14.8 : 1

Ambient temperature $23^{o}C$

Atmospheric pressure 759.4 mm Hg

IGNITION ADVANCE ^{o}BTDC	TORQUE ft.lbf
2	71.9
10	84.7
20	92.3
29	93.6
40	91.4
50	85.1

PETROL Air fuel ratio 12 : 1

Ambient temperature $23^{o}C$

Atmospheric pressure 762.0 mm Hg

IGNITION ADVANCE ^{o}BTDC	TORQUE ft.lbf
2	69.3
10	86.4
20	96.3
32	100.2
40	98.9
50	94.6

TABLE 4

TORQUE VERSUS IGNITION ADVANCE

3000 RPM FULL THROTTLE

PROPANE Air fuel ratio 14.4 : 1

Ambient temperature 23^oC

Atmospheric pressure 759.2 mm Hg

IGNITION ADVANCE ^{o}BTDC	TORQUE ft.lbf
2	63.0
10	76.8
20	85.6
29	87.3
40	84.4
50	79.4

PETROL Air fuel ratio 12.5 : 1

Ambient temperature 23^oC

Atmospheric pressure 762.0 mm Hg

IGNITION ADVANCE ^{o}BTDC	TORQUE ft.lbf
3	65.7
10	79.4
20	90.2
32	92.7
41	90.3

TABLE 5

TORQUE VERSUS AIR FUEL RATIO

2000 RPM FULL THROTTLE

PROPANE Ignition advance 28^{o}

Ambient temperature $24^{o}C$

Atmospheric pressure 759.0 mm Hg

AIR FUEL RATIO	EQUIVALENCE RATIO	TORQUE ft.lbf
13.9	0.886	92.2
14.1	0.896	92.4
14.4	0.92	92.4
15.3	0.972	92.5
17.5	1.111	86.4
18.1	1.15	76.8
23.4	1.49	32.2

PETROL Ignition advance 32^{o}

Ambient temperature $22^{o}C$

Atmospheric pressure 760.0 mm Hg

AIR FUEL RATIO	EQUIVALENCE RATIO	TORQUE ft.lbf
9.6	0.645	97.4
9.9	0.662	98.5
10.3	0.69	99.3
11.0	0.735	99.8
11.8	0.79	99.3
12.6	0.847	97.6
14.9	1.00	88.0
17.2	1.158	67.0
18.0	1.21	55.2

TABLE 6

TORQUE VERSUS AIR FUEL RATIO

2500 RPM FULL THROTTLE

PROPANE Ignition advance 28^o

Ambient temperature 24^oC

Atmospheric pressure 759.0 mm Hg

AIR FUEL RATIO	EQUIVALENCE RATIO	TORQUE ft.lbf
12.2	0.775	91.1
12.8	0.813	91.3
13.0	0.83	91.5
13.9	0.885	91.6
17.2	1.094	85.4
17.5	1.12	65.7
18.1	1.15	34.2

PETROL Ignition advance 32^o

Ambient temperature 22^oC

Atmospheric pressure 760.0 mm Hg

AIR FUEL RATIO	EQUIVALENCE RATIO	TORQUE ft.lbf
9.9	0.666	97.2
10.3	0.69	98.7
11.1	0.746	98.9
11.9	0.795	98.6
12.6	0.845	97.7
14.8	0.995	89.3
16.2	1.09	73.5
17.3	1.16	57.7