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PERFORMANCE EVALUATION OF A FERROUS PICRATE COMBUSTION CATALYST APPLIED TO DIESEL FUEL

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B.Eng. Project Report

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1985

17 Whimbrel Street
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The Head of Department of Mechanical Engineering
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Dear Sir

I submit this report entitled "Performance Evaluation of a Ferrous Picrate Combustion Catalyst Applied to Diesel Fuel, Using the Varimax Engine", based on Project 491/492, undertaken by me as part-requirement for the degree of Bachelor of Engineering in Mechanical Engineering.

Yours faithfully,

John Guld

JOHN GULD

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Mr P Robbins - Technician, School of Process Engineering, WAIT.

ABSTRACT

This report investigated the effects of administering a ferrous picrate combustion catalyst to diesel fuel. Testing of the combustion catalyst involved the use of the Varimax variable compression test and research engine rig (TD35) in its diesel mode.

The feasibility of the combustion catalyst was determined by its effects on three important engine performance parameters, namely power output, fuel consumption and exhaust temperature. The required information was obtained from an extensive testing program.

The results of this testing program indicated that there were benefits gained in using the ferrous picrate combustion catalyst. Increased power output and reduced fuel consumption were observed whenever the combustion catalyst was used.

GLOSSARY OF TERMS

kW	kilowatt
Nm	Newton-metre
rpm	revolutions per minute
cc	cubic centimetres
l	litre
BSFC	brake specific fuel consumption
hr	hour
°C	degrees celsius
kg	kilogram
BTDC	before top dead centre
ml	millilitre
m	metre
s	second
ppm	parts per million. Ratio of quantities
T	torque : Nm
ρ	density : kg/m^3
imep	indicated mean effective pressure
F_R	$\frac{\text{actual fuel-air ratio}}{\text{stoichiometric fuel-air ratio}}$
psia	pounds per square inch absolute
°F	degrees Fahrenheit
deg	degree.

CONTENTS

1.0 INTRODUCTION

During the period March 1985 to July 1985 an extensive engine test programme was successfully completed in conjunction with WAIT and Fuel Technology Pty. Ltd. This test programme involved the detailed evaluation of the effectiveness of a ferrous picrate combustion catalyst which is marketed by Fuel Technology.

Testing of the combustion catalyst involved the use of the Varimax variable compression test and research engine rig (TD35), located in the Thermodynamics Laboratory of the School of Process Engineering, WAIT. For this particular testing programme, diesel fuel was utilised to evaluate the merits of the combustion catalyst for several reasons. Firstly, detailed information of the performance characteristics of the combustion catalyst in diesel fuel as provided by this report was not previously available. Secondly, because large volume users of diesel fuel are the more likely users of this combustion catalyst, there was a need for comprehensive data on the benefits of using diesel fuel treated with the combustion catalyst.

The objective of this project was to analyse the effects of the combustion catalyst on three main aspects of engine performance. These were:

(i) engine brake power

(ii) brake specific fuel consumption

(iii) exhaust temperature.

The above three parameters were instrumental in portraying the advantages or disadvantages of using the combustion catalyst in conjunction with diesel fuel.

Thirty individual test runs were made to gather all the required data. This data was subsequently analysed and graphed.

Generally the results indicated that the introduction of the ferrous picrate combustion catalyst into diesel fuel produced the combined effect of improving engine power output, while simultaneously reducing fuel consumption. Complimenting these gains were marginally higher exhaust gas temperatures, which were in fact evidence of a more complete combustion reaction occurring.

As was expected, the gains in power output and reductions in fuel consumption varied considerably from test run to test run. Both depended on the manner in which the combustion catalyst was administered to the diesel fuel, and on the operational settings and conditions of the engine. These gains are portrayed in the various graphs located in Appendix 2.

Thus the information gathered in this report has served to indicate quite clearly the advantages of using the ferrous picrate combustion catalyst marketed by Fuel Technology Pty. Ltd., with diesel fuel, as a means of reducing running costs.

2.0 BACKGROUND

2.1 Diesel Engine Combustion Theory

2.1.1 The Combustion Process

The four-cycle compression-ignition engine employs the conventional four strokes per cycle of intake, compression, power and exhaust. The air inducted on the intake stroke is either normally aspirated or forced in by a supercharger, while the fuel is injected into the cylinder near the end of the compression stroke. In most current diesel engines, the combustion chamber temperature at the end of the compression stroke is usually in the vicinity of 600°C. This temperature is dependent upon the compression ratio and the initial air temperature.

Near the end of the compression stroke, fuel is sprayed into the combustion chamber at pressures varying from about 1200 psi (8.27 MPa) to over 30 000 psi (207 MPa). The injection pressure is governed by the engine speed and size and by the type of combustion chamber and injection system used.

With the commencement of fuel injection, the combustion process is initiated. Each charge of injected fuel experiences several phases in the reaction as follows:

(1) A delay period which covers the time from the initial injection to the actual ignition of the fuel and air.

(2) A period of rapid combustion of the fuel accumulated during the delay period, characterised by a rapid increase in cylinder pressure.

(3) A period of combustion of the remainder of the fuel charge as it is injected.

(4) An afterburning period in which the unburned fuel may find oxygen and burn, often referred to as the tail of combustion.

In order to understand the paths which may be followed by any fuel particle during the combustion process, an outline is shown in figure 2.1. In following the paths of the fuel particles, it should be understood that after ignition has occurred, many of these steps will be proceeding at the same time, since the mixture is very heterogeneous. It consists of air, precombustion and combustion reaction products in all stages, along with the continuing fuel spray. [Ref 2.]

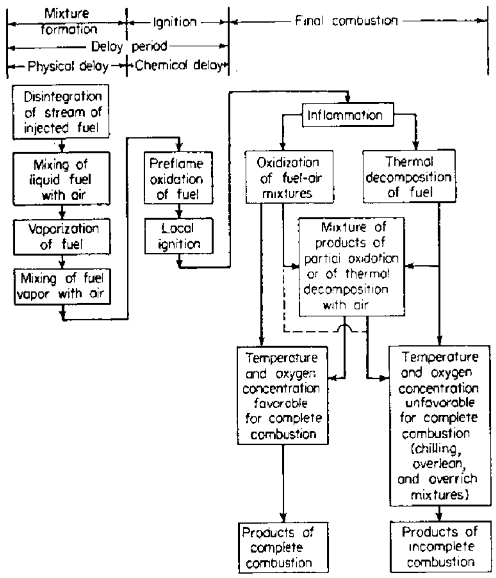


Figure 2.1 The combustion process in a compression ignition engine.

[Reproduced from Figure 8-9. Ref.4]

2.1.2 The Delay Period

The delay period consists of a physical and a chemical delay. The physical delay is required to atomize the fuel, mix it with the air, vapourize it, and produce a mixture of fuel vapour and air. During the chemical delay, preflame oxidation reactions occur in localised regions with a temperature increase of 540°C to 1100°C. These preflame reactions are the side reactions initiated by the catalytic effect of wall surfaces, high temperatures, and miscellaneous particles which form the active chain carriers prior to rapid combustion. As the local temperatures increase, the fuel vapours begin to crack at an accelerating rate and produce materials with high percentages of carbon which become heated to incandescence as local ignition is initiated.

Inflammation develops quickly and proceeds either by rapid and complete oxidation of the fuel and air or by the oxidation of the intermediate products of the chain reactions of the fuel. [Ref. 2]

2.1.3 The Rapid Combustion Period

Combustion in the period of rapid combustion is due chiefly to the burning of fuel that has had time to evaporate and mix with air during the delay period. The rate and extent of burning during this period are therefore closely associated with the length of the delay period and its relation to the injection process. It could be assumed that both the rate and the extent of pressure rise during the second stage of combustion would increase as the delay period increases, since both mixing time and the fraction of fuel taking part in rapid combustion increase. This relation holds true in practice, provided the injection timing is adjusted so that the period of rapid combustion occurs at substantially the same crank angle.

The influence of delay on both rate and extent of pressure rise during this phase is especially strong when the delay period is shorter than the injection period. When the delay is longer than the injection period, the quantity of fuel involved is unaffected by the length of the delay period. [Ref. 2]

2.1.4 The Third Phase of Combustion

The third phase of combustion is the period from maximum pressure to the point where combustion is measurably complete.

When the delay period is longer than the injection period, the third period of combustion will involve only fuel which has not found the necessary oxygen during the period of rapid combustion. In this case, the combustion rate is limited only by the mixing process. This in turn is controlled by the ratio of oxygen to unburned fuel and by the degree to which the two are distributed and mixed at the end of the second period. Even when all the fuel is injected well before the end of the delay period, poor injection characteristics can extend the third period well into the expansion stroke and thus cause low output and poor efficiency.

Under conditions where the second phase of combustion has been completed before the end of injection, some portion of the fuel is injected during the third phase, and the rate of burning will be influenced by the rate of injection as well as by the mixing rate. [Ref. 1].

2.1.5 The Final Phase of Combustion

The final phase of combustion which is often referred to as the tail of combustion, continues after the third phase at a diminishing rate as any remaining fuel and oxygen are each consumed. This last stage and the previous one, are both characterised by diffusion combustion, with production and combustion of carbon particles and a high rate of heat transfer by radiation. This phase occurs well down on the expansion stroke of an engine. Due to the dependence of combustion on the process of fuel finding oxygen, compression ignition engines require excess air if high efficiency and low smoke levels are to be achieved. [Ref. 2].

2.1.6 Combustion Rate

The combustion process described previously may be shown graphically on a combustion rate diagram, as in figure 2.2.

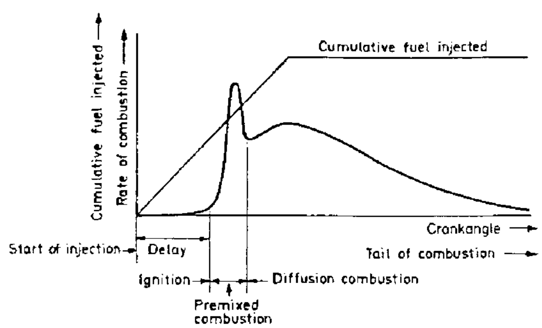


Figure 2.2 Typical combustion rate diagram.

[Reproduced from Ref. 5]

This diagram is typical only. The magnitude of the various parts will depend upon the delay period, the fuel injection characteristics, oxygen availability and the vigour of the mixing process. [Ref. 5]

2.1.7 The Ideal Combustion Process

The thermal efficiency of an internal combustion engine, both spark-ignition and compression-injection, will increase if the combustion time is decreased. A higher temperature can be achieved as the heat release occurs more nearly at constant volume. Cylinder pressure will be higher, and thus more work can be extracted for the same energy input from combustion. The rate of pressure rise during the period corresponding to constant volume combustion should be as rapid as possible without exceeding a certain value. This value is usually determined experimentally for the particular size and design of engine in question. The fuel remaining after the period of rapid pressure rise should be burned at a rate such as to hold the cylinder pressure constant, at the maximum allowable value, until all the fuel is burned.

Figure 2.3 shows both the slow burning and fast burning representations of the Otto engine cycle compared with the ideal engine cycle. Cycle ABCD is the ideal cycle characterised by isentropic processes and constant volume combustion. The point designated 'B' is where ignition must be initiated in order to obtain peak cylinder pressures when the piston is top dead centre. The diagram indicates that the fuel is ignited at the same time for the slow burning and fast burning processes. The fast burning process is more complete at minimum volume and hence better approximates the ideal engine cycle. The control of combustion towards this ideal is one of the major problems in the design and operation of compression-ignition engines. [Ref. 6]

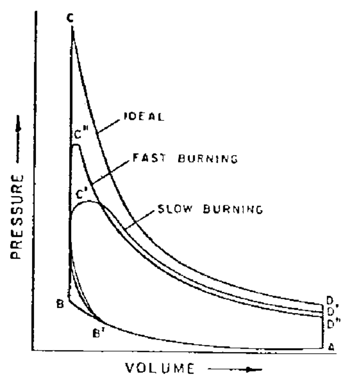


Figure 2.3 Graphical representation of ideal and actual engine cycles.

[Reproduced from Ref. 6]

2.2. The Effects of Operating Conditions on Combustion

With respect to diesel engines, the combustion rate as well as the rate of pressure rise, depends greatly on the design of the combustion chamber and the injection system. In practice, the design of these elements varies widely, and hence it is more difficult to generalise trends in this type of engine than in the case of spark-ignition engines, where there is much less variety in design. In view of these facts, the trends noted for diesel engines of a particular design are not necessarily typical of all types of such engines. [Ref.2]

2.2.1 Definitions

In order to simplify the discussion, the following terms are defined:

Injection time - the time elapsed between the start of spray into cylinder and the end of flow from the injector nozzle.

Injection angle - the crank angle between the start and end of injection.

Delay angle - the crank angle corresponding to the delay period.

2.2.2 Variation of Injection Timing

Pressure-crank angle curves showing the effect of altering injection timing at constant speed are shown in figure 2.4. Maximum indicated mean effective pressure (imep) and therefore maximum efficiency, occurs with peak cylinder pressure at about 15° after top dead centre. This was indicated by run 409.

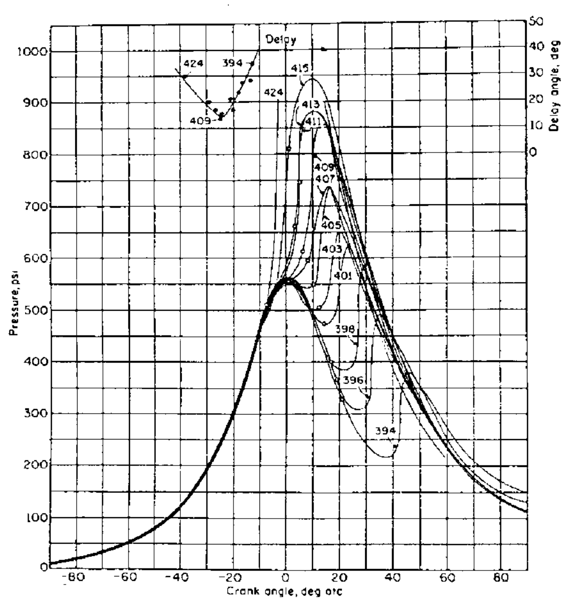


Figure 2.4 The effect of injection timing on the pressure-crank diagram. Numbers indicate test run numbers; 0 indicates 30° after start of injection; • indicates the start of injection. [Reproduced from Ref. 2]

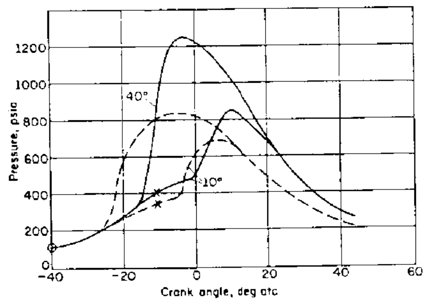
Shortest delays occur when the delay period includes top dead centre, in this case when injection starts about 25° BTDC indicated by run 409. Earlier or later injection results in longer delays. The longest delay is with the latest injection, where average pressure, and therefore average temperature, during the delay period is lowest.

Rates of pressure rise are affected in this case both by the delay angle and by piston motion. Early injection gives a very high rate. Here a long delay occurs, together with inward piston motion during the subsequent rapid combustion. With late injection, rates of pressure rise are again high, owing to long delays, even though the rapid combustion period occurs well after top dead centre, with the piston in rapid descent. Many engines use an injection timing later than that for maximum imep in order to reduce both maximum pressure and rate of pressure rise. Runs 403, 405 and 407 of figure 2.4 illustrate this point. Such late injection timing involves some sacrifice in output and efficiency.

As with spark-injection engines, proper timing of the combustion process is an essential element in diesel engine operation. Hence, the most satisfactory way of considering the influence of other operating variables would be under conditions of best power injection timing, that is, timing to give highest mean effective pressure. As with spark-ignition engines, best power timing brings peak pressure 15° to 20° after dead centre. Under these circumstances the period of rapid combustion will occur while piston motion is small, and the maximum rate of pressure rise will not be greatly affected by piston motion. However diesel engines are usually operated with fixed injection timing, and therefore a change in the delay period means a change in the piston position and piston velocity at which important events occur. [Ref. 2]

2.2.3 The effect of engine speed

With respect to diesel engines the delay time tends to be independent of speed. This implies that the delay angle will vary with speed and that speed may have large effects in the pressure-crank angle diagram. Figures 2.5 and 2.6 together with table 2.2, show that the delay angle increases with increasing speed but not in direct proportion to speed. The two figures disagree on the effect of increasing speed, and consequently increasing delay angle at the second and third stages of combustion. In both cases the delays are generally shorter than the injection angle. Figure 2.5 shows the expected increase in rate of pressure rise and maximum pressure as the delay angle increases. Conversely rather opposite trends are shown in figure 2.6. Many factors other than delay are affected by changes in speed. Among these are spray characteristics, chamber wall temperatures, and volumetric efficiency. Hence it is quite possible to find two very different engines showing different responses to speed changes. Figure 2.5 is less typical of contemporary diesel engines than is figure 2.6. In figure 2.6. it is evident that the second and third stages of combustion respond to engine speed as do spark-injection engines, with combustion rates nearly proportional to speed. Hence diesel engines can be run successfully at high piston speeds. However, to operate satisfactorily at high speeds, fuel of good "ignition quality" (short ignition delay) is required. [Ref. 2]



Symbol	Injection Start	Delay Angle	
		At 570 rpm	At 1500 rpm
O	40° btc	14	24
X	10° btc	5	10

Figure 2.5 The effect of speed on the pressure-crank angle diagram; injection angle 25°; solid line = 1500 rpm, dashed line = 570 rpm.

[Reproduced from Ref. 2]

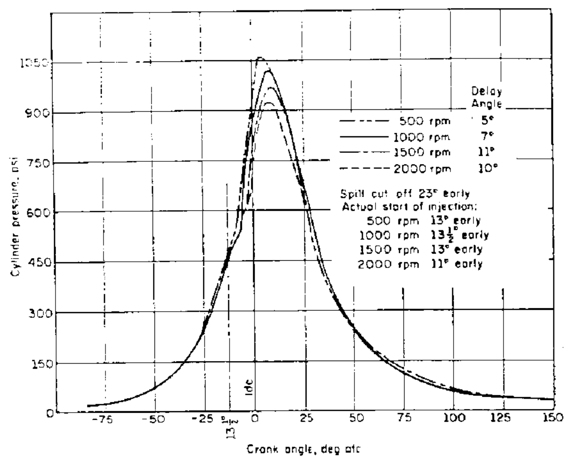


Figure 2.6 The effect of speed on the pressure-crank angle diagram on an automotive engine

[Reproduced from Ref. 2a]

Table 2.1 Delay angles and delay periods for figures 2.5 and 2.6

Figure	Rpm	Start of Injection, deg btc	Delay Angle, deg	Delay, seconds × 1000
	570	40	14	4.10
2 5	1500	40	24	2.67
	570	20	7	2.05
	1500	20	12	1.33
	570	10	5	1.47
	1500	10	10	1.12
	500	13	5	1.67
2 6	1000	13.5	7	1.17
	1500	13	11	1.22
	2000	11	10	0.83

2.2.4 The effect of turbulence

The effect of engine speed on the second and third stages of combustion is probably closely associated with charge turbulence, as in the case of spark-ignition flame propagation. However, in the case of diesel engines the turbulence effect must be associated more closely with the mixing process than with the propagation of chemical reaction. In cases where combustion starts early in the injection process, the use of a strong swirl to give a high air velocity across the spray is very effective in obtaining short second and third combustion stages. Turbulence is obtained to a great extent from the shape of the combustion chamber and also from the velocity and direction of flow of the air entering the cylinder. [Ref. 2]

2.2.5 The effect of compression ratio

The effect of the compression ratio on the pressure-crank angle diagram when injection timing, speed, and fuel quantity are held constant is plotted in figure 2.7. In this case rates of pressure rise are nearly the same. However, if injection timing had been set in a more practical way, that is, so that peak pressure occurred at the same crank angle, it is probable that the rate of pressure rise would have been higher as the compression ratio was reduced and the delay became longer.

Unless maximum cylinder pressures are allowed to be high, increasing the compression ratio in the diesel range of 14 : 1 to 22 : 1 gives only small improvement in efficiency. On the other hand the higher the engine friction, leakage, and torque required for starting. Thus diesel engine designers seek to use the lowest compression ratio consistent with satisfactory starting and operation with the available fuel. [Ref.2]

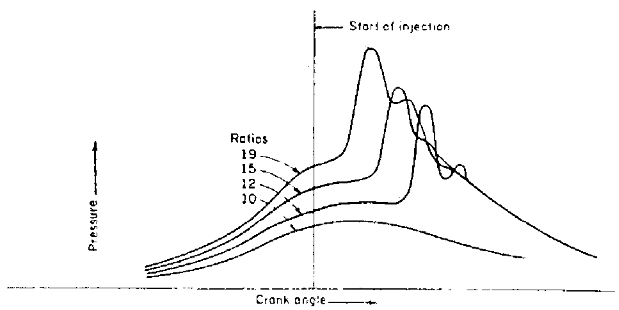


Figure 2.7 The effect of compression ratio on the pressure-crank angle diagram

[Reproduced from Ref.2]

2.2.6 <u>The effects of fuel-air ratios</u>

Some time after fuel is injected into the cylinder there are local fuel-air ratios varying from infinity at the fuel drop surface to zero at points not yet reached by the fuel vapour. Hence, as long as evaporation is not complete before ignition, it would be expected that the quantity of fuel injected would have no direct effect on the delay period. This assumption is illustrated in figure 2.8. An indirect effect of fuel quantity on delay is often noted when fuel-air ratio is decreased. Combustion temperatures are lowered and cylinder wall temperatures reduced. In many diesel engines the length of the delay period increases with decreasing fuel-air ratio because of this reduction in wall temperature. Figure 2.8 also indicates the fact that there is little reduction in the maximum rate of pressure except at very low fuel-air ratios. Also maximum pressure falls steadily with the fuel-air ratio. This tends to show that in this case most, if not all, of the fuel was injected during the delay period. Data from various sources have shown that maximum pressure is little affected by the fuel-air ratio is long as only a fraction of the total fuel is injected during the delay period. [Ref.2.]

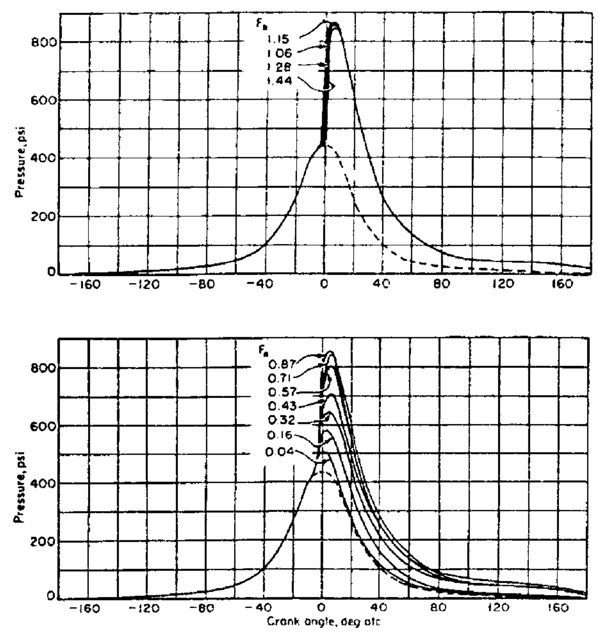


Figure 2.8 The effect of fuel-air ratio on the pressure-crank angle diagram. [Reproduced from Ref.2]

2.2.7 The effects of fuel spray characteristics

The physical characteristics of the fuel spray in relation to the size, and detailed design of the combustion chamber can have enormous effect on the pressure-crank angle diagram of diesel engines. One of the most important and difficult aspects of the development of a new diesel engine is that of determining the optimum spray characteristics from the point of view of power and efficiency without at the same time incurring excessive rates of pressure rise and high maximum cylinder pressures.

If injector nozzles cause poor atomization and/or insufficient penetration of fuel into the combustion chamber, a relatively long delay period will result due to the slow development of very fine droplets. Consequently output and efficiency will not be optimized. Injector nozzles which give very good distribution of fuel through fine atomization and deep penetration throughout the combustion chamber produce the greatest rates of pressure rise and highest maximum cylinder pressures. Thus these two fuel spray characteristics are essential for a diesel engine to achieve maximum output and efficiency. [Ref. 2]

2.2.8 The effects of combustion chamber design

The combustion process in diesel engines should be controlled to avoid both excessive maximum cylinder pressure and an excessive rate of pressure rise, in terms of crank angle. At the same time, the process should be rapid enough so that substantially all of the fuel injected is burned early in the expansion stroke.

Diesel engine combustion chambers are usually divided into two types: open chamber and divided chamber. These two types are defined as follows.

An open combustion chamber is one in which the combustion space incorporates no restrictions that are sufficiently small to cause large differences in pressure between different parts of the chamber during the combustion process. The chambers illustrated in figure 2.9 come under this classification.

A divided combustion chamber is one in which the combustion space is divided into two or more distinct compartments, between which there are restrictions or throats, small enough so that considerable pressure differences occur between them during the combustion process. When the burning starts in a chamber operated from the piston by a throat, the divided chamber is often called a pre-chamber. The chambers illustrated in figure 2.10 are examples of divided chambers.

In the open-chamber type engine, the mixing of fuel and air depends entirely on spray characteristics and on air motion, and it is not really affected by the combustion process itself. With this type of engine, once the compression ratio, maximum operating speed, and operating temperatures are selected, the delay angle is determined chiefly by fuel characteristics. Engines of this type are very sensitive to spray characteristics, which must be carefully determined to secure rapid mixing. The use of multi-orificed injector nozzles and the use of high injection pressures are usually required. With respect to small cylinder high speed engines, mixing is usually assisted by swirl, induced by directing the inlet air tangentially, or by squish, which is air motion caused by a small clearance space over part of the piston.

Divided combustion chambers have been developed chiefly for use in small high speed engines, in an attempt to overcome some of the limitations of the open-chamber type. Divided combustion chambers are characterized by the following properties. They feature extremely high air velocity through the throat during the compression stroke, with resultant intense turbulence and also swirl in the pre-chamber. The first and second stages of combustion can be forced to take place within a space whose structure is so strong that much higher pressures and higher rates of pressure rise can be tolerated within it than can be tolerated in the space over the piston. The mixing process may be greatly accelerated by the early stages of the combustion process itself. At high fuel-air ratios, combustion is incomplete in the prechamber because of insufficient air, and the high pressure developed by the early part of combustion projects the unburned fuel, together with the early combustion products, into the other part of the chamber with very high velocities, thus causing rapid mixing with the air in the space over the piston.

The chief advantages claimed for divided chambers as compared to open chambers in the same engine are as follows:

(1) The ability to use fuels of poorer ignition quality;

(2) The ability to use single-hole injection nozzles and moderate injection pressures and to tolerate greater degrees of nozzle fouling;

(3) The ability to employ higher fuel-air ratios without smoke.

Against these advantages the following disadvantages are usually assumed:

(1) The need for more expensive cylinder construction;

(2) More difficult cold starting because of greater heat loss through the throat;

(3) Poorer fuel economy due to greater heat losses and pressure losses through the throat, which result in lower thermal efficiency and higher pumping loss.

Open chambers tend to be used for engines designed to run at high piston speeds, as in road vehicle applications. The reason for this is that the heat and friction losses in divided-chamber engines become more important as speed increases.

Open chambers are also generally used in the case of large cylinders, which are limited to low speeds. At low speeds, delay angles are relatively short even with fuels of poor ignition quality, and there is no need for the advantages offered by the divided chambers in respect to fuel toleration.

The divided chamber is used primarily in the industrial-engine field, where fuel quality may be hard to control and where reliability under adverse situations is more important than fuel economy or operation at high speeds. The fact that divided chamber engines are less sensitive to spray variations due to nozzle fouling, and that nozzle fouling is less likely because single-hole nozzles of relatively large bore can be used, apparently are important considerations in the choice of divided-chamber engines for this type of service. [Ref. 2]

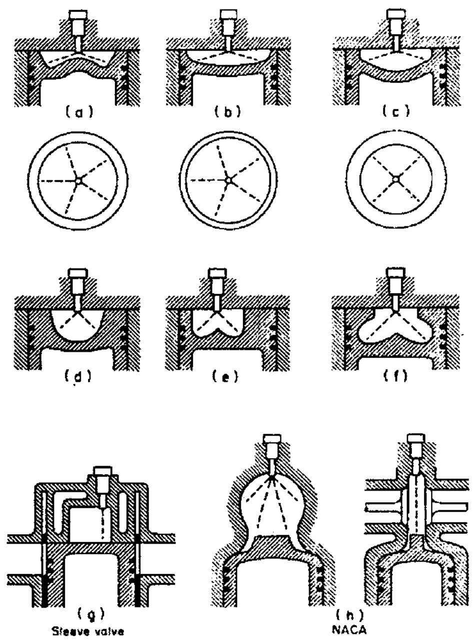


Figure 2.9 Examples of open combustion chambers

[Reproduced from Ref. 2]

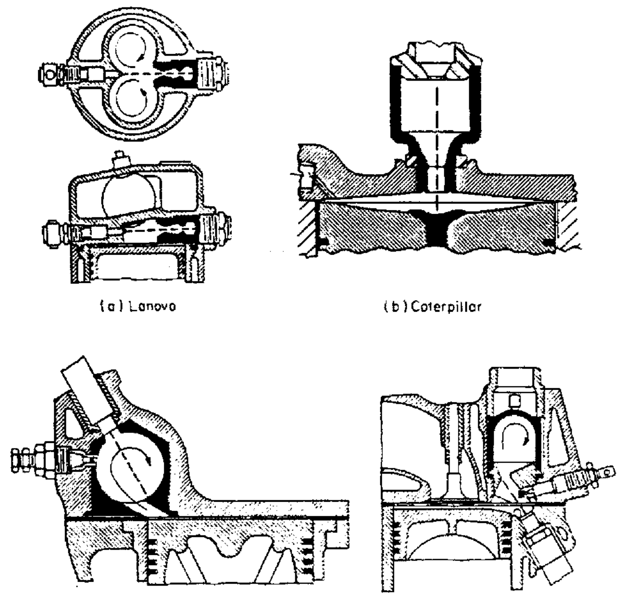


Figure 2.10 Examples of divided combustion chambers.

[Reproduced from Ref. 2]

2.2.9 The effects of diesel fuel cetane rating

The current method of rating diesel fuels in respect to ignition quality depends on engine-test comparisons with reference fuels, as in the case of octane rating. The primary reference fuels are normal cetane, $C_{16}H_{34}$, a straight-chain paraffin having excellent ignition quality, and alpha - methylnapthalene, $C_{10}H_7CH_3$, a napthenic compound having poor ignition quality. Good ignition quality implies a short delay angle at a given speed, compression ratio etc. The percentage of cetane in a blend of these two fuels giving the same delay angle as the fuel under test is taken as the cetane number of the test fuel.

The cetane rating of diesel fuel correlates best with starting ability and engine roughness. High cetane ratings result in improvements of cold-starting ability in compression ignition engines. The term engine roughness applies to the intensity of vibration of various engine parts, caused by high rates of pressure rise in the cylinders. Engine roughness decreases with increases in cetane rating in most cases. However, since the degree of roughness with a given rate of pressure rise varies greatly with the stiffness of the various engine parts, the effect of cetane number on roughness will vary with engine design.

Figure 2.11 shows the effects of running a diesel engine on three different fuels having widely varying cetane numbers. The engine was operated at constant throttle and a constant injection timing of 20° BTDC. The crank angle required for each of the three combustion stages shown was approximately constant for all engine speeds. It also indicates that cetane number does not affect the total combustion time, but lower cetane numbers cause longer delay periods coupled with increased speed of combustion after auto-ignition. The faster rate of burning of the low cetane fuels results in undesirable engine roughness. [Ref. 4]

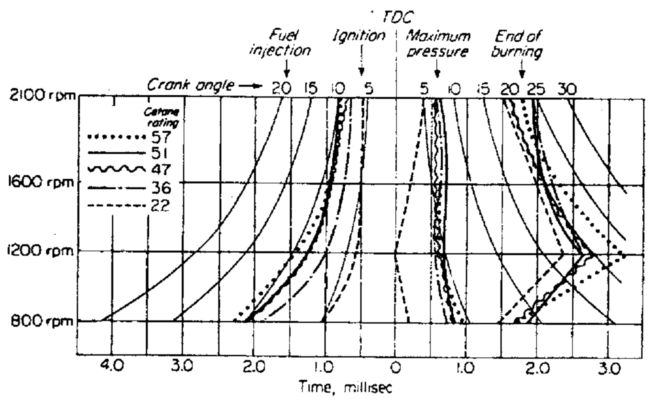


Figure 2.11 The effect of diesel fuel cetane rating on the three stages of combustion at various speeds.
[Reproduced from Ref. 4]

2.2.10 <u>The effect of inlet temperature and pressure</u>

Increasing the inlet temperature reduces the delay period because the temperature of the air in the cylinder at the time of injection is increased. This effect is illustrated in figure 2.12. At optimum ignition timing, increased inlet temperature tends to reduce the rate of pressure rise in the cylinder.

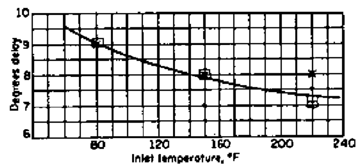


Figure 2.12 The effect of inlet temperature on delay angle at three different values of F_R; • F_R = 0.72, x F_r = 0.61, [] F_R = 0.55 [Reproduced from Ref. 2]

Increasing the inlet pressure decreases the delay period because of the large influence pressure has on temperature. The tests shown in figure 2.13 show the effect of inlet pressure variation on the delay period, with constant fuel quantity and injection timing. As the delay period increases, piston motion causes the average temperature during this period to fall. The effect of pressure should be much less with best-power injection timing, where the temperature during the delay period would be more uniform.

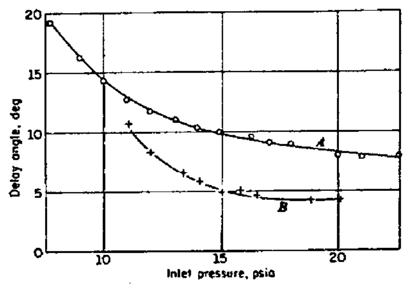


Figure 2.13 The effect of inlet pressure variation on the delay angle; A = open combustion chamber, B = divided combustion chamber [Reproduced from Ref. 2]

2.3 Combustion Catalyst Theory

2.3.1 Flame propagation

The speed with which the combustion process occurs influences the efficiency with which the heat released by the chemical reaction can be used. With greater rates of heat release, higher peak temperatures can be obtained. The rate of heat transfer varies with the temperature difference between the heat source and the body receiving the heat. If higher temperatures can be achieved through a more rapid combustion process, it is often possible to transfer a greater protion of the heat released into useful work. The burning rate will dictate the volume of combustion chamber required to burn a given quantity of fuel.

The flame front and reaction zone for normal rapid combustion of a fuel and air mixture may be represented by a diagram such as figure

2.14. Figure 2.14 indicates that unburned gas must be heated from the initial temperature T_o to some elevated temperature T_i, the ignition temperature, before the reaction can start at the reaction zone. Within the very narrow reaction zone, the chain reaction proceeds to chemical equilibrium. In the region directly behind the reaction zone, called the luminous flame zone, the radiations which appear as the visible portion of the flame are emitted. The reaction zone and the luminous flame zone are almost superimposed, the total width of both regions being no more than 0.5 mm for rapid combustion of premixed fuel and air. [Ref. 4]

2.3.2 Factors affecting flame speed

Considerable research work has indicated that at least four factors influence the rate of flame propagation in varying degrees. These are:

(1) The mechanism of the combustion reaction.

(2) The kinetics of the individual reactions in the mechanism.

(3) The diffusion of chain carriers, or propagating centres, from the reaction zone into the unburned gases.

(4) The rate of heat transfer from the reaction zone to the adjacent heating zone of the unburned gases.

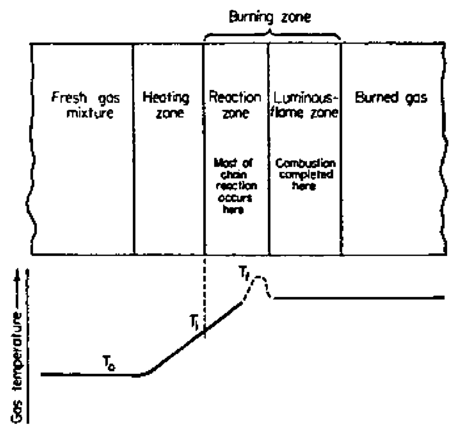


Figure 2.14 Temperature variation within the vicinity of the flame front. T_o = internal temperature, T_i = ignition temperature, T_f = maximum flame temperature [Reproduced from Ref. 2]

No present theory of combustion is sufficiently complete to establish which of the above factors are most important in controlling the flame speed. The mechanisms of the chemical reactions followed in rapid burning are still not fully understood. It is a complex problem, not only because of the speed of the burning process, but also because so many physical variables, such as temperature, pressure, humidity, and fuel-air ratio, effect the mechanism of reaction so much.

Chemical kinetics deals with the manner and speed with which a given chemical reaction will progress. Studies of kinetics establish the rates of burning by estimating the speed of each individual reaction in the chain mechanism. The general form of the equation used in this type of analysis is the classical Arrhenius equation:

$$W = Ke^{-E/RT} \qquad (2.1)$$

where W = reaction rate

K = a constant, depending upon the concentration of reacting substances, and collision frequency of molecules.

E = activation energy, or energy required to initiate reaction

R = universal gas constant

T = absolute temperature

Another equation proposed by W Z Nusselt and P J Daniell based on thermal theories of flame propagation is:

$$V_T \propto \sqrt{\frac{K}{C}\left[\frac{T_f - T_i}{T_i - T_o}\right]} \qquad (2.2)$$

where V_T = transformation velocity, the speed, relative to the unburned gases, with which the flame front moves from the burned to the unburned gases. It is measured in a direction normal to the surface of the flame front.

K = thermal conductivity

C = mean specific heat

T_f = maximum flame temperature

T_i = ignition temperature

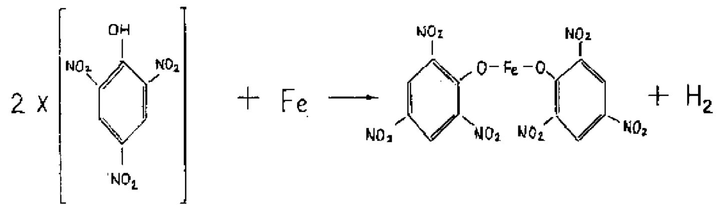
T_o = initial temperature.

With regard to the physical and chemical properties of the fuel-air mixture, equation (2.1) indicates that if the activation energy E of the fuel-air mixture can be decreased, the reaction rate will tend to increase. Similarly, if the constant K is increased by increasing the concentration of reacting substances and the collision frequency of molecules during combustion, the reaction rate will increase. Equation (2.2) indicates that if maximum flame temperature can be increased, the transformation velocity or flame speed will increase. Also, a similar effect can be achieved by increasing the thermal conductivity of the fuel-air mixture, or by decreasing its ignition temperature T_i.

Section 2.1.7 indicated that the thermal efficiency of an internal combustion engine will increase if the combustion time is decreased. A shorter combustion time implies a faster flame speed. Thus if a proposed combustion catalyst is to be of any benefit in terms of improving power output and/or decreasing fuel consumption, it must increase flame speed in some way. Although a combustion catalyst may function to promote a more complete combustion reaction, this does not necessarily result in power and fuel economy gains as would be expected. Even though the combustion reaction may be more complete, if it proceeds too slowly or begins either too early or too late in the expansion cycle, efficiency will not be maximised.

2.3.3 Mode of action of the ferrous picrate combustion catalyst

The combustion catalyst distributed by Fuel Technology Pty Ltd is essentially a complex organo-metallic compound consisting of ferrous picrate salt suspended in a toluene carrier solution. Ferrous picrate is produced when iron (Fe) is dissolved in picric acid (trinitrophenol). This chemical process is illustrated by equation (2.3).



(2.3)

picric acid (trinitrophenol) + iron ⟶ ferrous picrate complex + hydrogen gas

When the combustion catalyst is introduced to any liquid hydrocarbon fuel, the ferrous picrate forms flat crystalline particles which become dispersed with appropriate mixing. With the commencement of combustion, these particles act as propagating centres, initiating multiple flame fronts after absorbing radiant energy from the original source of ignition. These propagating centres are in effect increasing the thermal conductivity of the fuel-air mixture since the transmission of heat through it is much more rapid with their presence. Once combustion has been initiated the ferrous pircrate dissociates into ferrous ions and picrate ions. The ferrous ions will promote the formation of free hydrocarbon radicals for the combustion process, due to their electron configuration. The picrate ions undergo further decomposition which provides additional free hydrocarbon radicals for the combustion process as well as providing kinetic energy to local fuel molecules in excess of their normal

activation energy.

The various decomposition processes occuring during combustion have the effect of increasing the concentration of reacting substances due to the production of additional molecules and free radicals. These smaller, more volatile hydrocarbon molecules and radicals ignite more easily and burn more quickly than the larger molecules from which they were produced.

Since the number of molecules undergoing combustion is increased substantially, then the collision frequency between them will also increase. A larger number of individual particles in a given volume of fuel-air mixture has the effect of increasing its thermal conductivity. If combustion is more complete, more energy is liberated while the flame front traverses through the fuel-air mixture, hence maximum flame temperature would tend to increase.

The introduction of ferrous picrate into a liquid hydrocarbon fuel thus produces effects which equations (2.1) and (2.2) suggest would tend to increase flame speed and reduce combustion time. A faster, more complete combustion reaction is conducive to greater power output, reduced fuel consumption, and less unwanted exhaust emissions.

3.0 THE VARIMAX ENGINE TEST PROGRAM

The testing of internal combustion engines invariably involves a large number of operational parameters. To obtain a comprehensive data base from which the required information can be extracted, a considerable amount of testing has to be performed. Thus is it imperative to limit the number of parameters which will be altered during testing to avoid obtaining an excessive amount of data, and a long testing schedule.

In relation to the evaluation of the combustion catalyst, it was necessary to choose parameters, which when altered, would best illustrate the effects of the combustion catalyst. In addition to this, these chosen parameters should also relate to the most commonly altered settings and conditions during normal operation of the engine. This would ensure that the information gathered would be representative of the effects of using the combustion catalyst under normal everyday conditions.

The objective of this report was to analyse the effects of the combustion catalyst in engine brake power, brake specific fuel consumption, and exhaust temperature. In order to considerably broaden the scope of the test program in terms of relevance to simulating true commercial and industrial operating conditions, the following parameters were introduced to be varied accordingly:

(1) Engine speed

(2) Throttle setting

(3) Fuel injection timing

(4) The concentration of the combustion catalyst in the diesel fuel.

The manner in which each of these paramaters was altered is described below.

Engine speed in all tests was varied from 1600 rpm to 2400 rpm, in increments of 200 rpm.

Throttle settings were altered alternatively from half throttle to full throttle in the majority of tests.

Fuel injection timing was varied from 18° BTDC to 42° BTDC in increments of 6°, in specific tests. The standard injection timing was 30° BTDC.

The concentration of the combustion catalyst in the diesel fuel was altered in a number of ways. Three different mixing ratios were employed. These were:

(i) 1 : 1200

(ii) 1 : 1600 (standard recommended ratio)

(iii) 1 : 2000.

A similar effect was achieved by utilizing seven samples of the combustion catalyst, each of which had a different ionic concentration of the active ingredient ferrous picrate. The actual ionic concentration of the ferrous picrate was indicated by the presence of ferrous (Fe^{2+}) ions in each sample. The seven samples used are listed below with their respective ionic concentration of ferrous (Fe^{2+}) ions.

(i) Sample FPC1 - 11 ppm

(ii) Sample FPC2 - 21 ppm

(iii) Sample FPC3 - 40 ppm

(iv) Sample FPC4 - 72 ppm

(v) Sample FCP5 - 147 ppm

(vi) Sample FCP6 - 2 ppm

(vii) Sample FPC7 - 37 ppm.

Figure 3.1 below illustrates the seven FPC combustion catalyst samples. It is quite apparent that, increasing the concentration of ferrous picrate in a given amount of toluene carrier solution, results in the progressive darkening of the green colour of the solution.

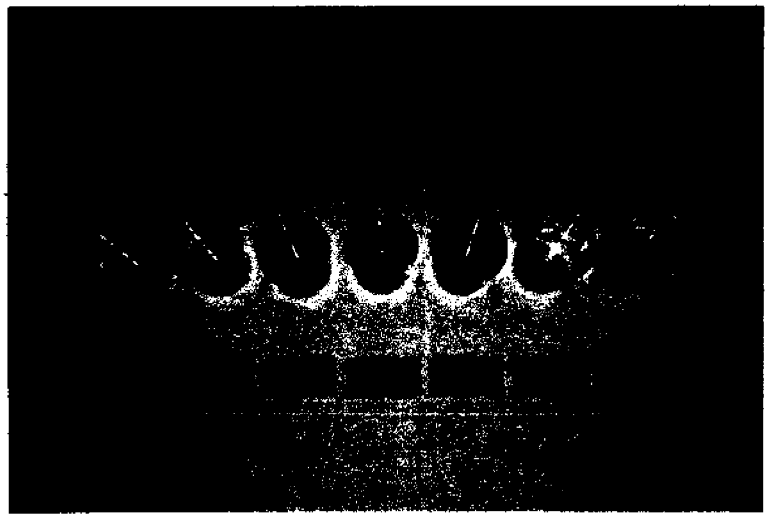


Figure 3.1 The seven FPC samples

Since the ferrous picrate combustion catalyst is normally dissolved in toluene, which is relatively expensive compared to other commercially available solvents, it was decided to test different types and blends of solvent. This would help determine whether or not a more economical solvent formulation could be substituted for the expensive toluene. If this could be done, it would reduce both the manufacturing cost of the combustion catalyst and the running costs of current and potential diesel fuel consumers. The alternate solvent formulations used for testing purposes were:

(i) 100% refined toluene as standard

(ii) 100% "Shellsol" crude toluene

(iii) 50% toluene, 50% heating oil (kerosine)

All blends were administered to the diesel fuel at the standard recommended mixing ratio of 1 : 1600.

Engine based parameters which were held constant during the entire test programme were compression ratio and valve timing. The compression ratio used during all tests was 18 : 1. Valve timing was set to the engine manufacturers' recommended values for diesel fuel and is listed below.

INTAKE VALVE	OPENS	10.8° BTDC
	CLOSES	42.6° ABDC
EXHAUST VALVE	OPENS	7.6° BBDC
	CLOSES	21.6° ATDC

VALVE OVERLAP = 32.4°

Baseline tests using untreated diesel fuel were conducted at the beginning, middle and end of the test programme to check whether any "drift" in engine performance had occurred, due to the introduction of the combustion catalyst.

For all tests conducted in the Varimax engine test program, full details of which parameters were altered in each particular test, are given on each page of tabulated results in APPENDIX 1.0

3.1 Testing Method

The commencement of a new test, with the engine in a cold state, involved a set procedure. This procedure was strictly adhered to in order to ensure the validity, accuracy and repeatability of the Varimax test program.

From initial start up the engine was run at part throttle for five minutes and then slowly brought up to full throttle in thirty seconds. This ensured a gradual warm-up of the engine. The warm-up period was continued until the engine temperature reached 65°C.

With the baseline tests, testing commenced once this temperature was reached and remained stable. Testing of diesel fuel treated with the combustion catalyst commenced two hours after normal operating temperature was reached. The reason for this was that an average delay of two hours was noted before actual gains in power and fuel economy were witnessed. This delay period is often referred to as the conditioning period. Possible reasons leading to the existence of this conditioning period are given in section 4.0

Once testing was commenced, the following readings were recorded during all tests:

(i) Brake torque

(ii) The time required for the engine to consume the fuel contained in a 48 ml pipette

(iii) Exhaust temperature

(iv) Ambient air temperature.

Five readings of brake torque and the elapsed time for the consumption of 48 ml of diesel fuel were recorded at the various speeds specified in section 3.0. This applied to all tests except those dealing with the seven FPC samples and the variation of injection timing. In those tests, only four different readings of the two preceding parameters were recorded. All readings were subsequently averaged and a mean value was recorded.

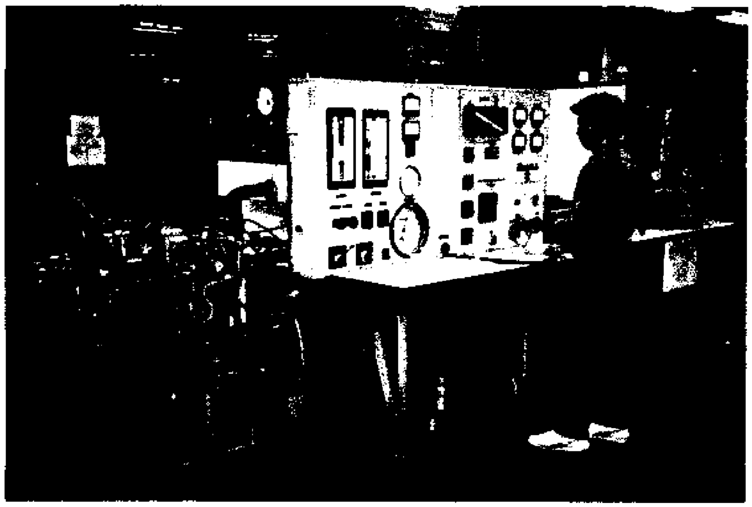
The data gathered throughout the test program was graphed accordingly, with the aid of an IBM computer graphics facility. In all graphs, a curve giving best parabolic fit to the plotted data was drawn to represent the general trend in each test. The resultant graphs located in APPENDIX 2.0, served to display the effects of introducing the combustion catalyst into diesel fuel.

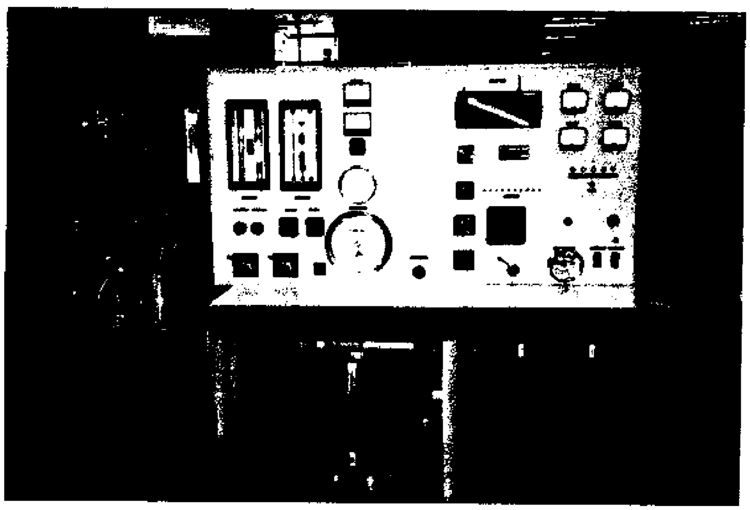
3.2 Equipment Used

3.2.1 The Varimax engine test rig

The Varimax engine test program was conducted on the Varimax variable compression test and research engine rig (TD35), located in the Thermodynamics Laboratory of the School of Process Engineering; see figures 3.2 and 3.3. For specifications and data on this test rig see APPENDIX 3.0. A general description of the test rig and associated equipment used, is given below.

The Varimax test engine is a single cylinder, four stroke, variable compression ratio engine, adaptable to either spark ignition or compression ignition operation. The engine features an open combustion chamber design. It is coupled to an electric dynamometer loaded by a resistance bank.





Figures 3.2 and 3.3 The Techquipment Varimax test and research engine rig (TD35), School of Process Engineering, WAIT

The engine is fitted with an instrumentation panel which provides information such as dynamometer brake torque, engine speed, fuel flowrate, exhaust temperature, cooling water flowrates and temperatures, air intake orifice pressure differential, dynamometer field current, and generated voltage and current.

During operation, cooling water from the engine was mixed with water from the mains inlet in a tank before recycling into the engine. A portion of the warm water mixture was drained at a set level in the tank thereby maintaining a temperature in the range of 65° C to 70°C.

Figure 3.4 shows the rotameter and pipette arrangements used to measure cooling water and diesel fuel flowrates during all tests.

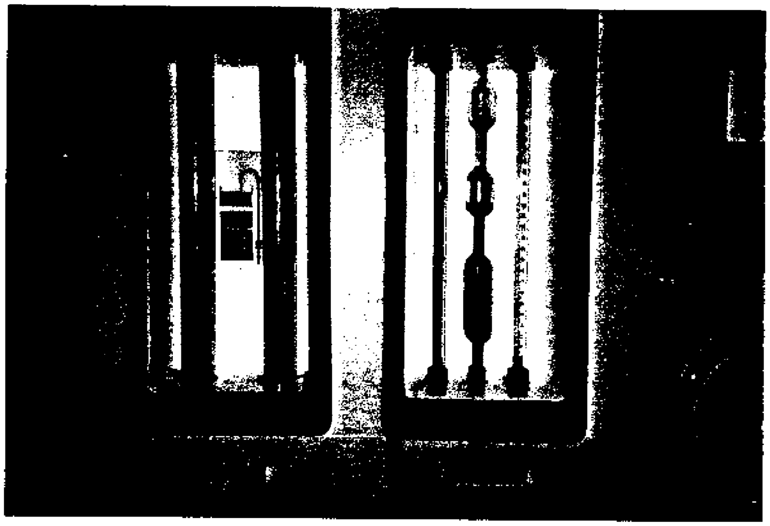


Figure 3.4 Rotameter and pipette arrangement for measuring cooling water and diesel fuel flowrates

Engine speed during all tests was displayed by an analogue tachometer. Similarly, engine brake torque was displayed by an analogue "torque meter". Both of these instruments are shown in figure 3.5.

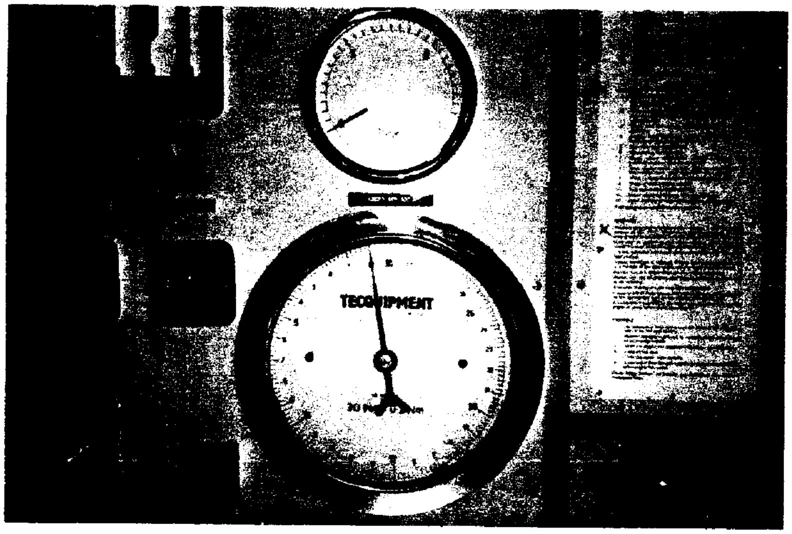


Figure 3.5 Tachometer and "torque-meter" located on instrument panel

Figure 3.6 indicates the controls used in altering the various cooling water and fuel flowrates, as well as the two throttle controls that are used when the engine is run in either compression ignition or spark ignition modes.

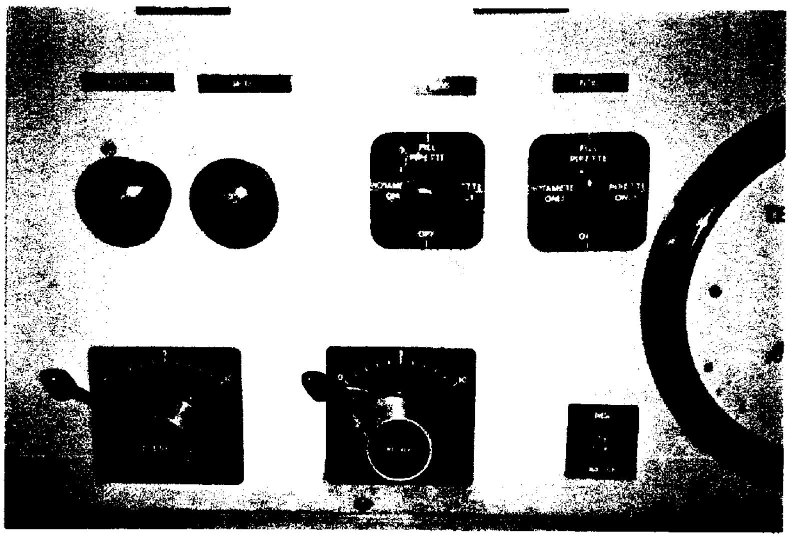


Figure 3.6 Throttle, rotameter and pipette controls

3.2.2 <u>Fuel storge and supply facilities</u>

All of the diesel fuel used for establishing the baseline data and all the treated fuel tests were drawn from the same source at the same time. This would ensure that fuel quality was constant for all tests. Figure 3.7 illustrates the two clean 200 litre steels drums used to store all of the diesel fuel for the test program.

Figure 3.7 Fuel storage facility

The Varimax engine test rig is fitted with four ten litre fuel tanks; two for gasoline use and two for diesel. The fuel gravitates through two independent C.A.V. fuel filters, each of approximately 500 ml capacity, to the injector pump. The separate fuel tanks allowed various fuel blends to be tested without contamination or stopping the engine. Fuel temperature at the point of volume measurement was held constant. Figure 3.8 illustrates the four fuel storage tanks located above the engine.

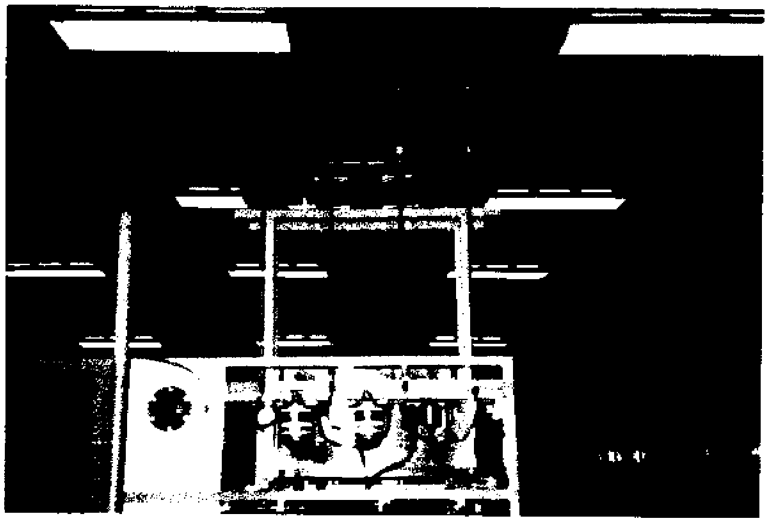


Figure 3.8 Fuel storage tanks located above the engine

3.2.3 Applying the combustion catalyst to the fuel

The combustion catalyst was administered to the diesel fuel with the aid of a burette. The required quantity of combustion catalyst was added to two litres of diesel fuel held in an appropriate graduated glass cylinder under the burette. The mixture was then poured into one of the two ten litre overhead fuel tanks. This procedure was repeated until the tank was filled. Figure 3.9 shows the actual process of applying the combustion catalyst to the diesel fuel.

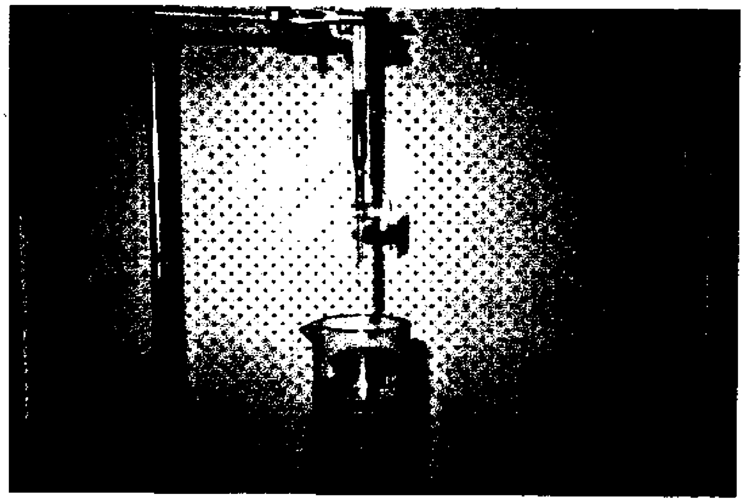


Figure 3.9 Applying the combustion catalyst to the diesel fuel

3.3 Sample Calculations

3.3.1 Brake power

The brake power output of the Varimax engine was calculated by obtaining the torque exerted by it, at any given speed. This torque value was obtained by reading the appropriate figure from the "torque-meter" dial face.

The brake power output was then given by:

$$\text{brake power output} = \frac{2\pi\, N\, T}{60\ 000} \quad \text{(kw)}$$

where N = engine rpm

T = torque Nm

Using a typical torque value from test 1.0(a) at 1600 rpm as an example:

$$\text{brake power} = \frac{2\pi \times 1600 \times 37.40}{60\ 000}$$

therefore brake power = 6.27 kW.

3.3.2 <u>Brake specific fuel consumption</u>

The brake specific fuel consumption of the engine was calculated as follows. Five readings of the time required for the engine to consume 48 ml of diesel fuel were recorded. The average value was then calculated.

Using values from test 1.0(a) at 1600 rpm as an example:

Time 1 = 63.75 seconds

Time 2 = 63.59 seconds

Time 3 = 63.64 seconds

Time 4 = 63.45 seconds

Time 5 = 63.67 seconds

Average time = 63.62 seconds

The density of the diesel fuel used throughout the test program was taken as 920 kg/m^3.

Therefore 48 ml of diesel fuel is equivalent to:

$$48 \text{ ml} \times 10^{-6} \ \frac{m^3}{ml} \times 920 \ \frac{kg}{m^3} = 0.04416 \text{ kg}$$

Thus fuel consumption in kg/hr is:

$$\text{fuel consumption} = \frac{0.04416 \times 3600}{63.62}$$

$$\text{fuel consumption} = 2.499 \text{ kg/hr}$$

Brake specific fuel consumption is calculated as follows:

$$BSFC = \frac{\text{fuel consumption kg/hr}}{\text{brake power kW}}$$

$$= \frac{2.499}{6.270}$$

therefore BSFC = 0.399 kg/kW hr

4.0 DISCUSSION OF RESULTS

From the commencement of testing program, a major problem was encountered with the torque-meter display. It was observed that at speeds of 1600, 1800, and 2000 rpm, the dial-pointer began to oscillate rapidly back and forth over a scale distance of about 0.6 Nm, on average.

Although the dynamometer was fitted with an oil filled damper, it was obviously inadequate in damping out the higher harmonic vibrations of the single cylinder Varimax engine. The damper was subsequently examined and found to have an excessive clearance between the piston and the cylinder. Thus it offered little resistance to movement at high velocity which is characteristic of a damper having a small damping coefficient.

To correct the problem, a rubber washer was incorporated into the piston to reduce the piston-cylinder clearance and effectively increase the damping ratio of the damper. The engine was then run at the speeds at which the vibration problem was encountered to observe whether the modification was successful or not. The modification proved to be a success, reducing the oscillation of the dial-pointer down to a scale distance of 0.2 Nm, or one scale division. The final damper configuration is shown in figure 4.1. Having corrected the damper problem, testing proceeded normally.

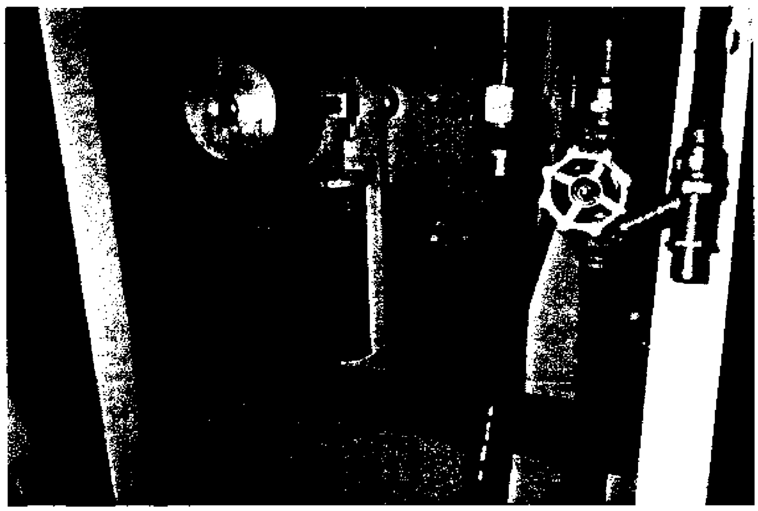


Figure 4.1 The dynamometer damper configuration

An interesting anomaly was noted at the start of tests involving the introduction of the combustion catalyst into the diesel fuel. It was assumed that any gains in power output and fuel economy would be realised immediately. However, a delay of approximately two hours was noted between the time normal operating temperature was reached and testing commenced, and when gains were realised. This anomaly, which has occured in previous test programs, is often called the conditioning period. Its cause is not yet fully understood, however a possible explanation will be outlined here.

The conditioning period may relate to the time required for the combustion catalyst to react with, and slowly remove carbon deposits from the combustion chamber surfaces. The lack of improvement in power output and fuel economy is probably due to the reaction between the ferrous picrate and the carbon deposits proceeding instead of the intended reaction between the ferrous picrate and the diesel fuel. It appears that the ferrous picrate may have a stronger affinity for pure carbon particles than it does for hydrocarbon fuel molecules and radicals. Once most of the carbon deposits have been removed from an engines' combustion chamber surfaces, the ferrous picrate is then free to react with the hydrocarbon fuel molecules and radicals in its normal manner as detailed in section 2.3.3. Gains in power output and fuel economy follow accordingly.

Throughout the Varimax test program; engine speed, throttle setting, fuel injection timing, and the concentration of the combustion catalyst in the fuel, were all varied to examine the effects of the combustion catalyst on the combustion process. Power output, fuel economy and exhaust temperature were the means used for illustrating these effects.

Since the mode of action of the ferrous picrate was to increase flame speed, confirmation of this was required in the test results. Section 2.2 describes in detail the effects of engine speed, injection timing, and fuel-air ratio on flame speed.

Increasing engine speed in theory results in a smaller delay angle; i.e. a shorter delay period. Thus the ferrous picrate should have shown greater effect at lower engine speeds than at higher engine speeds. An examination of all test results revealed that gains in torque were relatively constant across the test speed range. Hence the Varimax engine did not follow the expected trend. However this did not indicate that flame speed was not increased.

Fuel injection timing has a very pronounced effect on flame speed. If injection timing has been optimised in terms of power output, injecting the fuel early or later than this results in longer delay periods and slower flame speeds. The standard injection timing of the Varimax engine was 30° BTDC. However testing revealed that the optimum setting was 36° BTDC. Hence the combustion catalyst should have shown greatest gains in torque at injection timings earlier and later than this. The results did in fact show this trend, therefore confirming that flame speeds were increased by the ferrous picrate.

With respect to compression-ignition engines, engine speed and power output are governed by the quantity of fuel injected into the combustion chamber. Since these engines are not throttled in any way, the quantity of air inducted into the engine is governed by its volumetric efficiency at any particular speed. In effect altering the "throttle" setting of a compression ignition engine alters the fuel-air ratio also. Section 2.2.6 indicated that as fuel-air ratio is decreased, the delay period increases and flame speed decreases because combustion temperatures are reduced.

The part throttle tests carried out at half maximum power, indicated power increase with the use of ferrous picrate, although they were smaller than those seen at full throttle. Hence flame speed was increased with its use, as expected.

The quantity of ferrous picrate administered to the fuel would be expected to influence the magnitude of resultant power gains. The greater the quantity administered to the fuel the greater the expected power increase and vice versa. Tests 2, 3, 4 and 10 showed that this thinking was correct. With respect to tests 2, 3 and 4, increasing the mixing ratio from 1 : 1600 to 1 : 1200 resulted in an extra gain in power and fuel economy, over that witnessed at a 1 : 1600 mixing ratio. Alternatively, when the mixing ratio was decreased from 1 : 1600 to 1 : 2000 1200 the power increase was smaller than that seen at a 1 : 1600 ratio. Although the various graphs show that increasing the quantity of ferrous picrate in the diesel fuel, does increase power and fuel economy gains, they also revealed a case of diminishing returns in terms of power and fuel economy for increases in mixing ratio. An economic study would be required to determine the optimum mixing ratio in a given situation.

Exhaust temperature readings were recorded in order to provide additional proof of the beneficial effects of the ferrous picrate. In all cases where power gains were observed, higher temperatures were also witnessed. Higher exhaust temperatures are indicative of higher maximum flame temperatures and a greater energy output from a given quantity of fuel-air mixture. Higher maximum flame temperatures result in increased flame speeds, which in turn are responsible for increases in combustion efficiency and therefore power output.

Tests 6, 7 and 8 involving the use of the standard solvent formulation, and two others, proved to be very informative. The three formulations were all administered to the fuel at the standard mixing ratio of 1 : 1600. In all cases, virtually identical power, fuel economy, and exhaust temperature increases were noted. This indicated that the most economical formulation to use was the blend containing 50% toluene, 50% heating oil (kerosine). This is because kerosine is undoubtedly less costly than toluene. Future testing programs may reveal that toluene could be eliminated altogether as a solvent and that 100% kerosine could be used in its place. However, ferrous picrate may not dissolve properly in pure kerosine and a small amount of toluene may need to be added. The lack of difference in output etc., experienced with the alternate solvent formulations is most certainly due to the mixing ratio employed. The quantity of toluene and kerosine entering the engine per unit time is very small, thereby having no measurable effect on power output etc.

5.0 CONCLUSIONS

The Varimax engine test program has served to show quite convincingly the benefits of using the ferrous picrate combustion catalyst, marketed by Fuel Technology Pty. Ltd, in conjunction with diesel fuel. Power increases witnessed ranged from 0.5% to 2.7%, with an average increase of about 2.0%. Gains in fuel economy ranged from 1.0% to 4.1%, with an average gain of about 2.5%. The altering of engine based parameters such as injection timing, throttle setting, and engine speed served to show that expected gains will vary from engine to engine, since detailed designs will invariably differ. Different engines will respond in different ways to the ferrous picrate combustion catalyst. If an engine has had every facet of its operation optimised for maximum power output, the ferrous picrate combustion catalyst will only give a small improvement in power output. The small benefit realised may be outweighed by the increased running costs incurred by using the combustion catalyst. On the other hand, if an engine has not been set up for maximum power output, the ferrous picrate should show gains in power output of at least the same magnitude seen here, if not greater. The gains in efficiency observed in this program are economically feasible since they outweigh the increased running costs.

APPENDIX 1

TABULATED TEST RESULTS

6.0 REFERENCES

1. The Internal-Combustion Engine in Theory and Practice, Volume 1 by Taylor.
2. The Internal-Combustion Engine in Theory and Practice, Volume 2 by Taylor.
3. The High Speed Internal-Combustion Engine by Ricardo and Hempson.
4. Fuels and Combustion by Smith and Stinson.
5. Internal Combustion Engines, Volume 1 by Benson and Whitehouse.
6. The effects of CV100 on Fleet Operation, SAE Technical Paper series.

Test 1(a) Full throttle test - clean diesel fuel

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.40	63.75	550	T_{AVE} = 63.62 sec
1600	37.40	63.59		
1600	37.40	63.64		P_{AVE} = 6.27 kW
1600	37.40	63.45		
1600	37.40	63.67		BSFC = 0.399 kg/kW hr
1800	37.00	56.19	580	T_{AVE} = 56.17 sec
1800	37.00	56.15		
1800	37.00	56.19		P_{AVE} = 6.97 kW
1800	37.00	56.11		
1800	37.00	56.20		BSFC = 0.406 kg/kW hr
2000	35.80	50.86	590	T_{AVE} = 50.82 sec
2000	35.80	50.91		
2000	35.80	50.73		P_{AVE} = 7.50 kW
2000	35.80	50.71		
2000	35.80	50.90		BSFC = 0.417 kg/kW hr
2200	33.80	46.84	605	T_{AVE} = 46.70 sec
2200	33.80	46.60		
2200	33.80	46.68		P_{AVE} = 7.79 kW
2200	33.80	46.66		
2200	33.80	46.71		BSFC = 0.437 kg/kW hr
2400	31.50	42.21	620	T_{AVE} = 42.17 sec
2400	31.50	42.16		
2400	31.50	42.12		P_{AVE} = 7.92 kW
2400	31.50	42.21		
2400	31.50	42.14		BSFC = 0.475 kg/kW hr

Ambient air temperature = 23°C

Test 1 (b) Part throttle test - clean diesel fuel - half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.70	115.29	350	T_{AVE} = 115.17 secs
1600	18.70	114.95		
1600	18.70	115.16		P_{AVE} = 3.12 kW
1600	18.70	115.21		
1600	18.70	115.24		BSFC = 0.441 kg/kW hr
1800	18.50	103.26	355	T_{AVE} = 103.33 sec
1800	18.50	103.39		
1800	18.50	103.33		P_{AVE} = 3.49 kW
1800	18.50	103.28		
1800	18.50	103.41		BSFC = 0.441 kg/kW hr
2000	17.90	93.85	375	T_{AVE} = 98.32 sec
2000	17.90	93.94		
2000	17.90	93.66		P_{AVE} = 3.75 kW
2000	17.90	93.90		
2000	17.90	93.77		BSFC = 0.452 kg/kW hr
2200	16.90	81.80	400	T_{AVE} = 81.87 sec
2200	16.90	81.69		
2200	16.90	81.91		P_{AVE} = 3.89 kW
2200	16.90	81.73		
2200	16.90	81.78		BSFC = 0.500 kg/kW hr
2400	15.80	72.15	440	T_{AVE} = 72.19 sec
2400	15.80	72.19		
2400	15.80	72.22		P_{AVE} = 3.97 kW
2400	15.80	72.20		
2400	15.80	72.17		BSFC = 0.555 kg/kW hr

Ambient air temperature = 23°C

Test 1(c) Variation in injection timing from the standard setting of 30° BTDC at full throttle and 2400 rpm. Clean diesel fuel

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.40	43.83	610	T_{AVE} = 43.64 sec
42°	32.40	43.49		P_{AVE} = 8.14 kW
42°	32.40	43.47		
42°	32.40	43.78		BSFC = 0.448 kg/kW hr
36°	33.30	42.96	620	T_{AVE} = 42.97 sec
36°	33.30	42.97		P_{AVE} = 8.37 kW
36°	33.30	42.97		
36°	33.30	42.97		BSFC = 0.442 kg/kW hr
30°	32.00	42.21	625	T_{AVE} = 42.18 sec
30°	32.00	42.16		P_{AVE} = 8.37 kW
30°	32.00	42.12		
30°	32.00	42.21		BSFC = 0.469 kg/kW hr
24°	29.50	42.43	655	T_{AVE} = 42.47 sec
24°	29.50	42.38		P_{AVE} = 7.39 kW
24°	29.50	42.58		
24°	29.50	42.50		BSFC = 0.506 kg/kW hr
18°	24.00	41.93	680	T_{AVE} = 41.88 sec
18°	24.00	41.88		P_{AVE} = 6.03 kW
18°	24.00	41.92		
18°	24.00	41.80		BSFC = 0.630 kg/kW hr

Ambient air temperature = 23°C

Test 2(a) Full throttle test - combustion catalyst introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.70	63.98	570	T_{AVE} = 63.90 sec
1600	37.70	63.96		
1600	37.70	63.95		P_{AVE} = 6.32 kW
1600	37.70	63.83		
1600	37.70	63.80		BSFC = 0.394 kg/kW hr
1800	37.30	56.37	595	T_{AVE} = 56.34 sec
1800	37.30	56.24		
1800	37.30	56.39		P_{AVE} = 7.03 kW
1800	37.30	56.36		
1800	37.30	56.33		BSFC = 0.401 kg/kW hr
2000	36.50	51.14	605	T_{AVE} = 51.18 sec
2000	36.50	51.21		
2000	36.50	51.17		P_{AVE} = 7.64 kW
2000	36.50	51.24		
2000	36.50	51.13		BSFC = 0.410 kg/kW hr
2200	34.50	47.20	620	T_{AVE} = 77.19 sec
2200	34.50	47.22		
2200	34.50	47.18		P_{AVE} = 7.95 kW
2200	34.50	47.16		
2200	34.50	47.19		BSFC = 0.424 kg/kW hr
2400	32.20	42.35	630	T_{AVE} = 42.36 sec
2400	32.20	42.39		
2400	32.20	42.46		P_{AVE} = 8.09 kW
2400	32.20	42.28		
2400	32.20	42.31		BSFC = 0.464 kg/kW hr

ambient air temperature = 22°C

Test 2(b) Part throttle test - combustion catalyst introduced at 1 : 1600 ratio. Half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.80	118.38	355	T_{AVE} = 118.27 sec
1600	18.80	117.37		
1600	18.80	118.78		P_{AVE} = 3.16 kW
1600	18.80	118.20		
1600	18.80	118.61		BSFC = 0.425 kg/kW hr
1800	18.70	105.51	375	T_{AVE} = 106.03 sec
1800	18.70	105.43		
1800	18.70	106.23		P_{AVE} = 3.52 kW
1800	18.70	106.59		
1800	18.70	106.41		BSFC = 0.426 kg/kW hr
2000	18.30	94.11	590	T_{AVE} = 94.40 sec
2000	18.30	94.38		
2000	18.30	94.65		P_{AVE} = 3.83 kW
2000	18.30	94.36		
2000	18.30	94.48		BSFC = 0.440 kg/kW hr
2200	17.30	82.33	440	T_{AVE} = 82.40 sec
2200	17.30	82.33		
2200	17.30	82.63		P_{AVE} = 3.99 kW
2200	17.30	82.37		
2200	17.30	82.33		BSFC = 0.483 kg/kW hr
2400	16.10	72.71	475	T_{AVE} = 73.16 sec
2400	16.10	73.39		
2400	16.10	73.29		P_{AVE} = 7.92 kW
2400	16.10	72.91		
2400	16.10	73.49		BSFC = 0.536 kg/kW hr

ambient air temperature = 22°C

Test 2(c) Variation in injection timing from the standard setting of 30° BTDC at full throttle and 2400 RPM. Combustion catalyst introduced at 1 : 1600 ratio.

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.80	43.84	610	T_{AVE} = 43.86 sec
42°	32.80	43.91		P_{AVE} = 8.24 kW
42°	32.80	43.79		
42°	32.80	43.89		BSFC = 0.44 kg/kW hr
36°	33.70	43.09	625	T_{AVE} = 43.10 sec
36°	33.70	43.11		P_{AVE} = 8.47 kW
36°	33.70	43.16		
36°	33.70	43.06		BSFC = 0.435 kg/kW hr
30°	32.20	42.35	630	T_{AVE} = 42.37 sec
30°	32.20	42.39		P_{AVE} = 8.09 kW
30°	32.20	42.28		
30°	32.20	42.28		BSFC = 0.464 kg/kW hr
24°	30.00	42.65	665	T_{AVE} = 42.66 sec
24°	30.00	42.67		P_{AVE} = 7.39 kW
24°	30.00	42.73		
24°	30.00	42.59		BSFC = 0.494 kg/kW hr
18°	24.40	42.12	700	T_{AVE} = 42.11 sec
18°	24.40	42.06		P_{AVE} = 6.13 kW
18°	24.40	42.09		
18°	24.40	42.15		BSFC = 0.616 kg/kW hr

ambient air temperature = 22°C

Test 3(a) Full throttle test - combustion catalyst introduced at 1 : 1200 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.9	64.25	575	T_{AVE} = 64.20 sec
1600	37.9	64.21		
1600	37.9	64.29		P_{AVE} = 6.35 kW
1600	37.9	64.33		
1600	37.9	64.38		BSFC = 0.389 kg/kW hr
1800	37.5	56.41	600	T_{AVE} = 56.48 sec
1800	37.5	56.38		
1800	37.5	56.53		P_{AVE} = 7.07 kW
1800	37.5	56.49		
1800	37.5	56.61		BSFC = 0.398 kg/kW hr
2000	36.6	51.23	610	T_{AVE} = 51.25 sec
2000	36.6	51.19		
2000	36.6	51.24		P_{AVE} = 7.66 kW
2000	36.6	51.29		
2000	36.6	51.32		BSFC = 0.405 kg/kW hr
2200	34.7	47.35	625	T_{AVE} = 47.28 sec
2200	34.7	47.28		
2200	34.7	47.24		P_{AVE} = 8.00 kW
2200	34.7	47.26		
2200	34.7	47.29		BSFC = 0.420 kg/kW hr
2400	32.3	42.43	635	T_{AVE} = 42.48 sec
2400	32.3	42.44		
2400	32.3	42.51		P_{AVE} = 8.12 kW
2400	32.3	42.56		
2400	32.3	42.48		BSFC = 0.460 kg/kW hr

ambient air temperature = 22.5°C

Test 3(b) Part throttle test - combustion catalyst introduced at 1 : 1200 ratio. Half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	19.00	119.27	365	T_{AVE} = 119.21 sec
1600	19.00	119.20		
1600	19.00	119.19		P_{AVE} = 3.18 kW
1600	19.00	119.24		
1600	19.00	119.16		BSFC = 0.419 kg/kW hr
1800	18.80	106.61	380	T_{AVE} = 106.62 sec
1800	18.80	106.50		
1800	18.80	106.67		P_{AVE} = 3.54 kW
1800	18.80	106.71		
1800	18.80	106.63		BSFC = 0.421 kg/kW hr
2000	18.30	95.06	410	T_{AVE} = 95.02 sec
2000	18.30	95.09		
2000	18.30	95.00		P_{AVE} = 3.83 kW
2000	18.30	94.98		
2000	18.30	94.95		BSFC = 0.437 kg/kW hr
2200	17.40	82.48	450	T_{AVE} = 82.45 sec
2200	17.40	82.46		
2200	17.40	82.40		P_{AVE} = 4.01 kW
2200	17.40	82.42		
2200	17.40	82.49		BSFC = 0.481 kg/kW hr
2400	16.20	73.77	620	T_{AVE} = 73.70 sec
2400	16.20	73.79		
2400	16.20	73.61		P_{AVE} = 4.07 kW
2400	16.20	73.65		
2400	16.20	73.67		BSFC = 0.530 kg/kW hr

ambient air temperature = 22.5°C

Test 3(c) Variation in injection timing from the standard setting of 30° BTDC at full throttle and 2400 RPM. Combustion catalyst introduced at 1 : 1200 ratio

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.80	43.86	610	T_{AVE} = 43.86 sec
42°	32.80	43.95		P_{AVE} = 8.24 kW
42°	32.80	43.76		
42°	32.80	43.88		BSFC = 0.440 kg/kW hr
36°	33.80	43.15	625	T_{AVE} = 43.11 sec
36°	33.80	43.08		P_{AVE} = 8.50 kW
36°	33.80	43.12		
36°	33.80	43.10		BSFC = 0.434 kg/kW hr
30°	32.30	42.43	635	T_{AVE} = 42.99 sec
30°	32.30	42.49		P_{AVE} = 8.12 kW
30°	32.30	42.51		
30°	32.30	42.56		BSFC = 0.461 kg/kW hr
24°	30.30	42.61	675	T_{AVE} = 42.63 sec
24°	30.30	42.64		P_{AVE} = 7.61 kW
24°	30.30	42.70		
24°	30.30	42.56		BSFC = 0.490 kg/kW hr
18°	24.70	42.19	710	T_{AVE} = 42.12 sec
18°	24.70	42.09		P_{AVE} = 6.21 kW
18°	24.70	42.07		
18°	24.70	42.11		BSFC = 0.608 kg/kW hr

ambient air temperature = 21°C

Test 4(a) Full throttle test - combustion catalyst introduced at 1 : 2000 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.60	63.81	560	T_{AVE} = 63.83 sec
1600	37.60	63.84		
1600	37.60	63.93		P_{AVE} = 6.30 kW
1600	37.60	63.78		
1600	37.60	63.80		BSFC = 0.395 kg/kW hr
1800	37.20	56.25	590	T_{AVE} = 56.22 sec
1800	37.20	56.20		
1800	37.20	56.21		P_{AVE} = 7.02 kW
1800	37.20	56.19		
1800	37.20	56.27		BSFC = 0.403 kg/kW hr
2000	36.20	51.11	600	T_{AVE} = 51.09 sec
2000	36.20	51.06		
2000	36.20	51.03		P_{AVE} = 7.58 kW
2000	36.20	51.15		
2000	36.20	51.10		BSFC = 0.410 kg/kW hr
2200	34.20	47.03	610	T_{AVE} = 47.01 sec
2200	34.20	46.98		
2200	34.20	46.96		P_{AVE} = 7.88 kW
2200	34.20	47.06		
2200	34.20	47.02		BSFC = 0.429 kg/kW hr
2400	32.00	42.23	625	T_{AVE} = 42.25 sec
2400	32.00	42.25		
2400	32.00	42.28		P_{AVE} = 8.04 kW
2400	32.00	42.29		
2400	32.00	42.19		BSFC = 0.468 kg/kW hr

ambient air temperature = 21°C

Test 4(b) Part throttle test - combustion catalyst at 1 : 2000 ratio.
Half power at each speed

RPM	Torque (Nm	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.80	116.15	355	T_{AVE} = 116.20 sec
1600	18.80	116.23		
1600	18.80	116.20		P_{AVE} = 3.15 kW
1600	18.80	116.18		
1600	18.80	116.22		BSFC = 0.434 kg/kW hr
1800	18.60	104.47	370	T_{AVE} = 104.47 sec
1800	18.60	104.40		
1800	18.60	104.51		P_{AVE} = 3.50 kW
1800	18.60	104.53		
1800	18.60	104.45		BSFC = 0.435 kg/kW hr
2000	18.10	94.11	385	T_{AVE} = 94.12 sec
2000	18.10	94.17		
2000	18.10	94.08		P_{AVE} = 3.79 kW
2000	18.10	94.13		
2000	18.10	94.10		BSFC = 0.446 kg/kW hr
2200	17.10	82.06	410	T_{AVE} = 82.09 sec
2200	17.10	82.09		
2200	17.10	82.10		P_{AVE} = 3.94 kW
2200	17.10	82.11		
2200	17.10	82.08		BSFC = 0.491 kg/kW hr
2400	16.00	72.76	460	T_{AVE} = 72.72 sec
2400	16.00	72.70		
2400	16.00	72.68		P_{AVE} = 4.02 kW
2400	16.00	72.71		
2400	16.00	72.75		BSFC = 0.544 kg/kW hr

ambient air temperature = 21.5°C

Test 4(c) Variation in injection timing from standard setting of 30° BTDC at full throttle and 2400 RPM. Combustion catalyst introduced at 1 : 2000 ratio

BTDC TIMING	Torque (Nm	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.70	43.76	610	T_{AVE} = 43.76 sec
42°	32.70	43.71		P_{AVE} = 8.22 kW
42°	32.70	43.74		
42°	32.70	43.81		BSFC = 0.442 kg/kW hr
36°	33.60	43.06	625	T_{AVE} = 43.06 sec
36°	33.60	43.05		P_{AVE} = 8.44 kW
36°	33.60	43.10		
36°	33.60	43.01		BSFC = 0.437 kg/kW hr
30°	32.20	42.24	630	T_{AVE} = 42.23 sec
30°	32.20	42.28		P_{AVE} = 8.09 kW
30°	32.20	42.19		
30°	32.20	42.21		BSFC = 0.465 kg/kW hr
24°	29.80	42.58	660	T_{AVE} = 42.57 sec
24°	29.80	42.51		P_{AVE} = 7.49 kW
24°	29.80	42.55		
24°	29.80	42.62		BSFC = 0.499 kg/kW hr
18°	24.30	42.01	695	T_{AVE} = 42.00 sec
18°	24.30	42.04		P_{AVE} = 6.11 kW
18°	24.30	41.95		
18°	24.30	41.99		BSFC = 0.620 kg/kW hr

ambient air temperature = 21.5°C

Test 5 Engine baseline test on clean diesel fuel at full throttle

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.30	63.64	550	T_{AVE} = 63.60 sec
1600	37.30	63.58		
1600	37.30	63.73		P_{AVE} = 6.25 kW
1600	37.30	63.49		
1600	37.30	63.55		BSFC = 0.400 kg/kW hr
1800	36.80	56.23	580	T_{AVE} = 56.17 sec
1800	36.80	56.20		
1800	36.80	56.12		P_{AVE} = 6.94 kW
1800	36.80	56.14		
1800	36.80	56.18		BSFC = 0.408 kg/kW hr
2000	35.90	50.83	590	T_{AVE} = 50.84 sec
2000	35.90	50.77		
2000	35.90	50.80		P_{AVE} = 7.52 kW
2000	35.90	50.92		
2000	35.90	50.86		BSFC = 0.416 kg/kW hr
2200	33.80	46.69	605	T_{AVE} = 46.69 sec
2200	33.80	46.73		
2200	33.80	46.78		P_{AVE} = 7.79 kW
2200	33.80	46.65		
2200	33.80	46.62		BSFC = 0.437 kg/kW hr
2400	31.40	42.24	620	T_{AVE} = 42.19 sec
2400	31.40	42.14		
2400	31.40	42.16		P_{AVE} = 7.89 kW
2400	31.40	42.19		
2400	31.40	42.22		BSFC = 0.477 kg/kW hr

ambient air temperature = 21.5°C

Test 6(a) Full throttle test - Introducing combustion catalyst dissolved in refined WA toluene at a 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.70	63.89	570	T_{AVE} = 63.86 sec
1600	37.70	63.86		
1600	37.70	63.83		P_{AVE} = 6.32 kW
1600	37.70	63.81		
1600	37.70	63.91		BSFC = 0.394 kg/kW hr
1800	37.30	56.26	595	T_{AVE} = 56.32 sec
1800	37.30	56.28		
1800	37.30	56.36		P_{AVE} = 7.03 kW
1800	37.30	56.40		
1800	37.30	56.30		BSFC = 0.402 kg/kW hr
2000	36.50	51.13	605	T_{AVE} = 51.13 sec
2000	36.50	51.11		
2000	36.50	51.16		P_{AVE} = 7.64 kW
2000	36.50	51.21		
2000	36.50	51.05		BSFC = 0.407 kg/kW hr
2200	34.40	47.23	620	T_{AVE} = 47.22 sec
2200	34.40	47.11		
2200	34.40	47.30		P_{AVE} = 7.92 kW
2200	34.40	47.25		
2200	34.40	47.20		BSFC = 0.425 kg/kW hr
2400	32.30	42.25	630	T_{AVE} = 42.32 sec
2400	32.30	42.41		
2400	32.30	42.36		P_{AVE} = 8.12 kW
2400	32.30	42.29		
2400	32.30	42.27		BSFC = 0.463 kg/kW hr

ambient air temperature = 22°C

Test 6(b) Part throttle test - Introducing combustion catalyst dissolved in refined WA toluene at a 1 : 1600 ratio. Half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.90	118.56	355	T_{AVE} = 118.55 sec
1600	18.90	118.65		
1600	18.90	118.72		P_{AVE} = 3.17 kW
1600	18.90	118.49		
1600	18.90	118.33		BSFC = 0.423 kg/kW hr
1800	18.70	105.44	375	T_{AVE} = 105.43 sec
1800	18.70	105.25		
1800	18.70	105.56		P_{AVE} = 3.52 kW
1800	18.70	105.50		
1800	18.70	105.40		BSFC = 0.428 kg/kW hr
2000	18.30	94.18	395	T_{AVE} = 94.41 sec
2000	18.30	94.40		
2000	18.30	94.57		P_{AVE} = 3.83 kW
2000	18.30	94.37		
2000	18.30	94.51		BSFC = 0.449 kg/kW hr
2200	17.20	82.38	440	T_{AVE} = 82.44 sec
2200	17.20	82.41		
2200	17.20	82.49		P_{AVE} = 3.96 kW
2200	17.20	82.61		
2200	17.20	82.30		BSFC = 0.487 kg/kW hr
2400	16.10	72.82	475	T_{AVE} = 73.04 sec
2400	16.10	72.72		
2400	16.10	73.45		P_{AVE} = 4.05 kW
2400	16.10	72.98		
2400	16.10	73.24		BSFC = 0.537 kg/kW hr

ambient air temperature = 22°C

Test 6(c) Variation in injection timing - Introducing combustion catalyst dissolved in refined WA toluene at a 1 : 1600 ratio. Test done at full throttle 2400 RPM

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.70	43.87	610	T_{AVE} = 43.89 sec
42°	32.70	43.95		P_{AVE} = 8.22 kW
42°	32.70	43.83		
42°	32.70	43.92		BSFC = 0.441 kg/kW hr
36°	33.70	43.16	625	T_{AVE} = 43.15 sec
36°	33.70	43.19		P_{AVE} = 8.47 kW
36°	33.70	43.14		
36°	33.70	43.09		BSFC = 0.435 kg/kW hr
30°	32.30	42.33	630	T_{AVE} = 42.36 sec
30°	32.30	42.36		P_{AVE} = 8.12 kW
30°	32.30	42.44		
30°	32.30	42.30		BSFC = 0.462 kg/kW hr
24°	30.00	42.70	665	T_{AVE} = 42.63 sec
24°	30.00	42.66		P_{AVE} = 7.54 kW
24°	30.00	42.61		
24°	30.00	42.55		BSFC = 0.495 kg/kW hr
18°	24.40	42.17	700	T_{AVE} = 42.13 sec
18°	24.40	42.08		P_{AVE} = 6.13 kW
18°	24.40	42.13		
18°	24.40	42.13		BSFC = 0.616 kg/kW hr

ambient air temperature = 22°C

Test 7(a) Full throttle test - Introducing combustion catalyst dissolved in "Shellsol" crude toluene at a 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.80	63.92	570	T_{AVE} = 63.90 sec
1600	37.80	63.90		
1600	37.80	63.88		P_{AVE} = 6.33 kW
1600	37.80	63.94		
1600	37.80	63.86		BSFC = 0.393 kg/kW hr
1800	37.30	56.29	595	T_{AVE} = 56.30 sec
1800	37.30	56.30		
1800	37.30	56.27		P_{AVE} = 7.03 kW
1800	37.30	56.33		
1800	37.30	56.32		BSFC = 0.402 kg/kW hr
2000	36.40	51.12	605	T_{AVE} = 51.13 sec
2000	36.40	51.09		
2000	36.40	51.10		P_{AVE} = 7.62 kW
2000	36.40	51.15		
2000	36.40	51.17		BSFC = 0.408 kg/kW hr
2200	34.50	47.20	620	T_{AVE} = 47.22 sec
2200	34.50	47.17		
2200	34.50	47.25		P_{AVE} = 7.95 kW
2200	34.50	47.24		
2200	34.50	47.22		BSFC = 0.423 kg/kW hr
2400	32.30	42.38	630	T_{AVE} = 42.34 sec
2400	32.30	42.34		
2400	32.30	42.31		P_{AVE} = 8.12 kW
2400	32.30	42.29		
2400	32.30	42.37		BSFC = 0.462 kg/kW hr

ambient air temperature = 22.5°C

Table 7(b) Part throttle test - Introducing combustion catalyst dissolved in "Shellsol" crude toluene at a 1 : 1600 ratio. Half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.90	118.59	355	T_{AVE} = 118.47 sec
1600	18.90	118.47		
1600	18.90	118.27		P_{AVE} = 3.17 kW
1600	18.90	118.61		
1600	18.90	118.43		BSFC = 0.423 kg/kW hr
1800	18.70	105.55	375	T_{AVE} = 105.41sec
1800	18.70	105.48		
1800	18.70	105.25		P_{AVE} = 3.52 kW
1800	18.70	105.35		
1800	18.70	105.40		BSFC = 0.428 kg/kW hr
2000	18.20	94.00	395	T_{AVE} = 94.41 sec
2000	18.20	94.51		
2000	18.20	94.39		P_{AVE} = 3.81 kW
2000	18.20	94.48		
2000	18.20	94.65		BSFC = 0.442 kg/kW hr
2200	17.30	82.59	440	T_{AVE} = 82.45 sec
2200	17.30	82.42		
2200	17.30	82.52		P_{AVE} = 3.39 kW
2200	17.30	82.30		
2200	17.30	82.44		BSFC = 0.483 kg/kW hr
2400	16.10	72.72	475	T_{AVE} = 72.96 sec
2400	16.10	72.98		
2400	16.10	73.10		P_{AVE} = 4.05 kW
2400	16.10	73.02		
2400	16.10	73.00		BSFC = 0.538 kg/kW hr

ambient air temperature = 22.5°C

Test 7(c) Variation in injection timing - Introducing combustion catalyst dissolved in "Shellsol" crude toluene at a 1 : 1600 ratio.
Test done at full throttle 2400 rpm

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.80	43.99	610	T_{AVE} = 43.88 sec
42°	32.80	43.80		P_{AVE} = 8.24 kW
42°	32.80	43.90		
42°	32.80	43.84		BSFC = 0.440 kg/kW hr
36°	33.60	43.22	625	T_{AVE} = 43.16 sec
36°	33.60	43.09		P_{AVE} = 8.44 kW
36°	33.60	43.19		
36°	33.60	43.15		BSFC = 0.436 kg/kW hr
30°	32.40	42.37	630	T_{AVE} = 42.35 sec
30°	32.40	42.48		P_{AVE} = 8.14 kW
30°	32.40	42.25		
30°	32.40	42.30		BSFC = 0.461 kg/kW hr
24°	29.90	42.64	665	T_{AVE} = 42.66 sec
24°	29.90	42.70		P_{AVE} = 7.51 kW
24°	29.90	42.55		
24°	29.90	42.77		BSFC = 0.496 kg/kW hr
18°	24.40	42.20	600	T_{AVE} = 42.11 sec
18°	24.40	42.02		P_{AVE} = 6.13 kW
18°	24.40	42.07		
18°	24.40	42.16		BSFC = 0.616 kg/kW hr

ambient air temperature = 22°C

Test 8(a) Full throttle test - Introducing combustion catalyst dissolved in 50% toluene, 50% kerosine at a 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.70	63.91	570	T_{AVE} = 63.89 sec
1600	37.70	63.95		
1600	37.70	63.87		P_{AVE} = 6.32 kW
1600	37.70	63.84		
1600	37.70	63.89		BSFC = 0.394 kg/kW hr
1800	37.20	56.31	595	T_{AVE} = 56.32 sec
1800	37.20	56.33		
1800	37.20	56.28		P_{AVE} = 7.01 kW
1800	37.20	56.36		
1800	37.20	56.30		BSFC = 0.403 kg/kW hr
2000	36.50	51.18	605	T_{AVE} = 51.15 sec
2000	36.50	51.15		
2000	36.50	51.11		P_{AVE} = 7.64 kW
2000	36.50	51.17		
2000	36.50	51.14		BSFC = 0.407 kg/kW hr
2200	34.60	47.23	620	T_{AVE} = 47.21 sec
2200	34.60	47.21		
2200	34.60	47.24		P_{AVE} = 7.97 kW
2200	34.60	47.17		
2200	34.60	47.18		BSFC = 0.422 kg/kW hr
2400	32.20	42.30	630	T_{AVE} = 42.32 sec
2400	32.20	42.33		
2400	32.20	42.36		P_{AVE} = 8.09 kW
2400	32.20	42.28		
2400	32.20	42.35		BSFC = 0.464 kg/kW hr

ambient air temperature = 21.5°C

Test 8(b) Part throttle test - Introducing combustion catalyst dissolved in 50% Toluene, 50% kerosine at a 1 : 1600 ratio. Half power at each speed

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	18.80	118.29	355	T_{AVE} = 118.44 sec
1600	18.80	118.39		
1600	18.80	118.30		P_{AVE} = 3.15 kW
1600	18.80	118.57		
1600	18.80	118.66		BSFC = 1.342 kg/kW hr
1800	18.60	105.67	375	T_{AVE} = 105.98 sec
1800	18.60	105.78		
1800	18.60	106.03		P_{AVE} = 3.51 kW
1800	18.60	106.08		
1800	18.60	105.91		BSFC = 0.428 kg/kW hr
2000	18.20	94.25	395	T_{AVE} = 94.95 sec
2000	18.20	94.63		
2000	18.20	94.52		P_{AVE} = 3.81 kW
2000	18.20	94.47		
2000	18.20	94.40		BSFC = 0.442 kg/kW hr
2200	17.30	82.55	440	T_{AVE} = 82.48 sec
2200	17.30	82.34		
2200	17.30	82.42		P_{AVE} = 3.99 kW
2200	17.30	82.44		
2200	17.30	82.63		BSFC = 0.483 kg/kW hr
2400	16.10	72.40	475	T_{AVE} = 72.80 sec
2400	16.10	72.36		
2400	16.10	73.01		P_{AVE} = 4.05 kW
2400	16.10	73.28		
2400	16.10	72.94		BSFC = 0.539 kg/kW hr

ambient air temperature = 21°C

Test 8(c) Variation of injection timing - Introducing combustion catalyst dissolved in 50% Toluene, 50% kerosine at a 1 : 1600 ratio.
Test done at full throttle 2400 RPM

BTDC TIMING	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
42°	32.90	43.92	610	T_{AVE} = 43.84 sec
42°	32.90	43.80		P_{AVE} = 8.27 kW
42°	32.90	43.77		
42°	32.90	43.85		BSFC = 0.438 kg/kW hr
36°	33.60	43.10	625	T_{AVE} = 43.16 sec
36°	33.60	43.14		P_{AVE} = 8.44 kW
36°	33.60	43.18		
36°	33.60	43.22		BSFC = 0.436 kg/kW hr
30°	32.40	42.36	630	T_{AVE} = 42.36 sec
30°	32.40	42.38		P_{AVE} = 8.14 kW
30°	32.40	42.27		
30°	32.40	42.41		BSFC = 0.461 kg/kW hr
24°	30.20	42.63	665	T_{AVE} = 42.69 sec
24°	30.20	42.77		P_{AVE} = 7.59 kW
24°	30.20	42.80		
24°	30.20	42.54		BSFC = 0.491 kg/kW hr
18°	24.60	41.90	700	T_{AVE} = 42.02 sec
18°	24.60	41.92		P_{AVE} = 6.18 kW
18°	24.60	42.20		
18°	24.60	42.06		BSFC = 0.612 kg/kW hr

ambient air temperature = 21.5°C

Test 9 Final baseline test at full throttle using clean diesel fuel

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.5	63.72	550	T_{AVE} = 63.65 sec
1600	37.5	63.60		
1600	37.5	63.65		P_{AVE} = 6.28 kW
1600	37.5	63.70		
1600	37.5	63.58		BSFC = 0.398 kg/kW hr
1800	37.1	56.21	580	T_{AVE} = 56.20 sec
1800	37.1	56.16		
1800	37.1	56.22		P_{AVE} = 6.99 kW
1800	37.1	56.28		
1800	37.1	56.13		BSFC = 0.405 kg/kW hr
2000	35.70	50.88	590	T_{AVE} = 50.87 sec
2000	35.70	50.92		
2000	35.70	50.83		P_{AVE} = 7.48 kW
2000	35.70	50.80		
2000	35.70	50.90		BSFC = 0.418 kg/kW hr
2200	33.90	46.61	605	T_{AVE} = 46.71 sec
2200	33.90	46.67		
2200	33.90	46.73		P_{AVE} = 7.81 kW
2200	33.90	46.76		
2200	33.90	46.80		BSFC = 0.436 kg/kW hr
2400	31.70	42.15	620	T_{AVE} = 42.16 sec
2400	31.70	42.25		
2400	31.70	42.10		P_{AVE} = 7.97 kW
2400	31.70	42.13		
2400	31.70	42.18		BSFC = 0.473 kg/kW hr

ambient air temperature = 21.5°C

Test 10(a) Full throttle test - Sample FPC1 introduced at 1 : 1600 RPM

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.50	63.46	550	T_{AVE} = 63.57 sec
1600	37.50	63.57		P_{AVE} = 6.28 kW
1600	37.50	63.59		
1600	37.50	63.66		BSFC = 0.398 kg/kW hr
1800	37.10	56.24	580	T_{AVE} = 56.20 sec
1800	37.10	56.16		P_{AVE} = 7.00 kW
1800	37.10	56.18		
1800	37.10	56.20		BSFC = 0.404 kg/kW hr
2000	36.00	50.86	590	T_{AVE} = 50.83 sec
2000	36.00	50.91		P_{AVE} = 7.54 kW
2000	36.00	50.73		
2000	36.00	50.80		BSFC = 0.415 kg/kW hr
2200	33.80	46.80	600	T_{AVE} = 46.74 sec
2200	33.80	46.65		P_{AVE} = 7.79 kW
2200	33.80	46.77		
2200	33.80	46.73		BSFC = 0.437 kg/kW hr
2400	31.70	42.22	620	T_{AVE} = 42.19 sec
2400	31.70	42.15		P_{AVE} = 7.97 kW
2400	31.70	42.17		
2400	31.70	42.20		BSFC = 0.473 kg/kW hr

ambient air temperature = 22°C

Test 10(a) Full throttle test - Sample FPC2 introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.60	63.68	550	T_{AVE} = 63.74 sec
1600	37.60	63.75		P_{AVE} = 6.30 kW
1600	37.60	63.81		
1600	37.60	63.70		BSFC = 0.396 kg/kW hr
1800	37.20	56.21	580	T_{AVE} = 56.22 sec
1800	37.20	56.28		P_{AVE} = 7.01 kW
1800	37.20	56.17		
1800	37.20	56.23		BSFC = 0.403 kg/kW hr
2000	36.10	50.87	590	T_{AVE} = 50.89 sec
2000	36.10	50.93		P_{AVE} = 7.56 kW
2000	36.10	50.86		
2000	36.10	50.90		BSFC = 0.413 kg/kW hr
2200	34.00	46.72	605	T_{AVE} = 46.81 sec
2200	34.00	46.80		P_{AVE} = 7.83 kW
2200	34.00	46.71		
2200	34.00	46.79		BSFC = 0.434 kg/kW hr
2400	31.80	42.23	620	T_{AVE} = 42.25 sec
2400	31.80	42.30		P_{AVE} = 7.99 kW
2400	31.80	42.27		
2400	31.80	42.18		BSFC = 0.471 kg/kW hr

ambient air temperature = 22°C

Test 10(c) Full throttle test - Sample FPC3 introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.50	63.99	570	T_{AVE} = 63.98 sec
1600	37.50	64.01		P_{AVE} = 6.28 kW
1600	37.50	64.04		
1600	37.50	63.87		BSFC = 0.396 kg/kW hr
1800	37.10	56.45	595	T_{AVE} = 56.43 sec
1800	37.10	56.40		P_{AVE} = 6.99 kW
1800	37.10	56.51		
1800	37.10	56.36		BSFC = 0.403 kg/kW hr
2000	36.30	51.23	600	T_{AVE} = 51.25 sec
2000	36.30	51.19		P_{AVE} = 7.60 kW
2000	36.30	51.26		
2000	36.30	51.30		BSFC = 0.406 kg/kW hr
2200	34.40	47.25	615	T_{AVE} = 47.21 sec
2200	34.40	47.21		P_{AVE} = 7.93 kW
2200	34.40	47.22		
2200	34.40	47.17		BSFC = 0.425 kg/kW hr
2400	32.00	42.47	630	T_{AVE} = 42.41 sec
2400	32.00	42.36		P_{AVE} = 8.04 kW
2400	32.00	42.28		
2400	32.00	42.51		BSFC = 0.466 kg/kW hr

ambient air temperature = 23°C

Test 10(d) Full throttle test - Sample FPC4 introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.70	63.90	570	T_{AVE} = 63.90 sec
1600	37.70	63.95		P_{AVE} = 6.32 kW
1600	37.70	63.93		
1600	37.70	63.80		BSFC = 0.394 kg/kW hr
1800	37.20	56.30	595	T_{AVE} = 56.35 sec
1800	37.20	56.29		P_{AVE} = 7.01 kW
1800	37.20	56.44		
1800	37.20	56.36		BSFC = 0.402 kg/kW hr
2000	36.50	51.26	605	T_{AVE} = 51.22 sec
2000	36.50	51.29		P_{AVE} = 7.64 kW
2000	36.50	51.17		
2000	36.50	51.14		BSFC = 0.406 kg/kW hr
2200	34.60	47.27	620	T_{AVE} = 47.16 sec
2200	34.60	47.08		P_{AVE} = 7.97 kW
2200	34.60	47.19		
2200	34.60	47.11		BSFC = 0.423 kg/kW hr
2400	32.20	42.39	630	T_{AVE} = 42.38 sec
2400	32.20	42.43		P_{AVE} = 8.09 kW
2400	32.20	42.31		
2400	32.20	42.37		BSFC = 0.464 kg/kW hr

ambient air temperature = 23°C

Test 10(e) Full throttle test - Sample FPC5 introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	38.00	64.27	580	T_{AVE} = 64.28 sec
1600	38.00	64.23		P_{AVE} = 6.37 kW
1600	38.00	64.28		
1600	38.00	64.32		BSFC = 0.388 kg/kW hr
1800	37.40	56.42	600	T_{AVE} = 56.51 sec
1800	37.40	56.44		P_{AVE} = 7.05 kW
1800	37.40	56.50		
1800	37.40	56.66		BSFC = 0.399 kg/kW hr
2000	36.30	51.18	605	T_{AVE} = 51.24 sec
2000	36.30	51.26		P_{AVE} = 7.60 kW
2000	36.30	51.22		
2000	36.30	51.30		BSFC = 0.408 kg/kW hr
2200	34.30	47.23	620	T_{AVE} = 47.20 sec
2200	34.30	47.17		P_{AVE} = 7.90 kW
2200	34.30	47.21		
2200	34.30	47.19		BSFC = 0.426 kg/kW hr
2400	32.00	42.50	625	T_{AVE} = 42.57 sec
2400	32.00	42.55		P_{AVE} = 8.04 kW
2400	32.00	42.60		
2400	32.00	42.62		BSFC = 0.464 kg/kW hr

ambient air temperature = 22.5°C

Test 10(f) Full throttle test - Sample FPC6 introduced at 1 : 1600 ratio

RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.30	63.47	550	T_{AVE} = 63.64 sec
1600	37.30	63.63		P_{AVE} = 6.25 kW
1600	37.30	63.71		
1600	37.30	63.73		BSFC = 0.400 kg/kW hr
1800	37.00	56.20	580	T_{AVE} = 56.19 sec
1800	37.00	56.22		P_{AVE} = 6.97 kW
1800	37.00	56.18		
1800	37.00	56.16		BSFC = 0.406 kg/kW hr
2000	35.80	50.90	590	T_{AVE} = 50.86 sec
2000	35.80	50.78		P_{AVE} = 7.50 kW
2000	35.80	50.85		
2000	35.80	50.92		BSFC = 0.417 kg/kW hr
2200	33.90	46.66	600	T_{AVE} = 46.76 sec
2200	33.90	46.78		P_{AVE} = 7.81 kW
2200	33.90	46.89		
2200	33.90	46.72		BSFC = 0.435 kg/kW hr
2400	31.40	42.11	620	T_{AVE} = 42.12 sec
2400	31.40	42.13		P_{AVE} = 7.89 kW
2400	31.40	42.16		
2400	31.40	42.08		BSFC = 0.478 kg/kW hr

ambient air temperature = 22.5°C

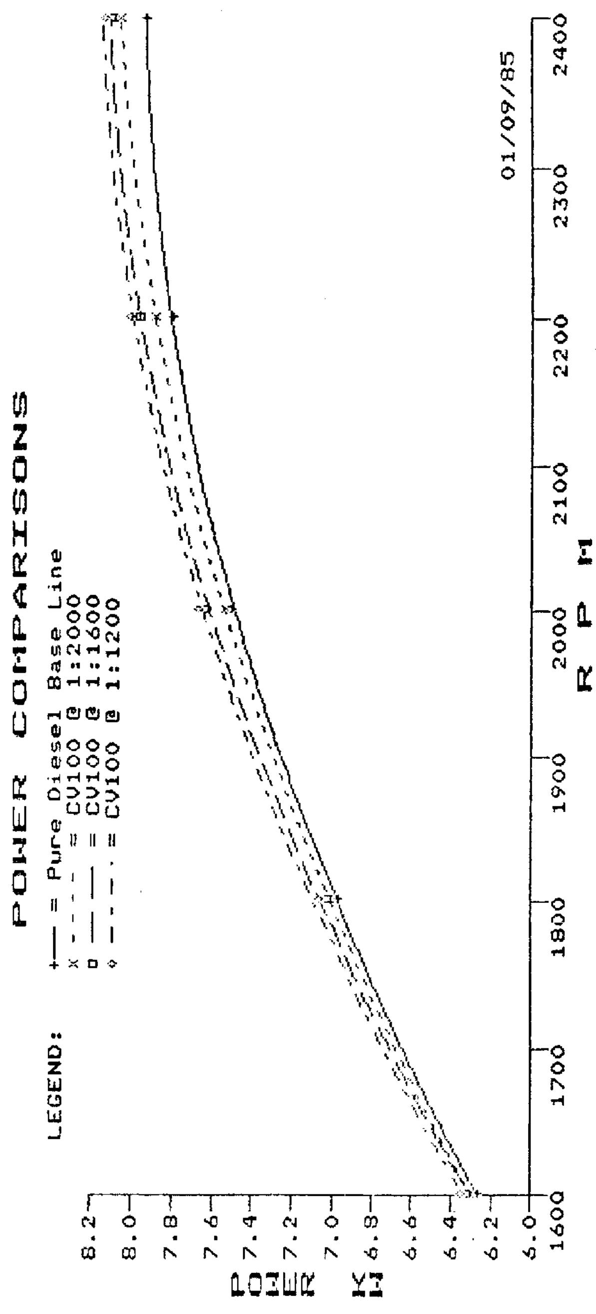
Test 10(g) Full throttle test - Sample FPC7 introduced at 1 : 1600 ratio

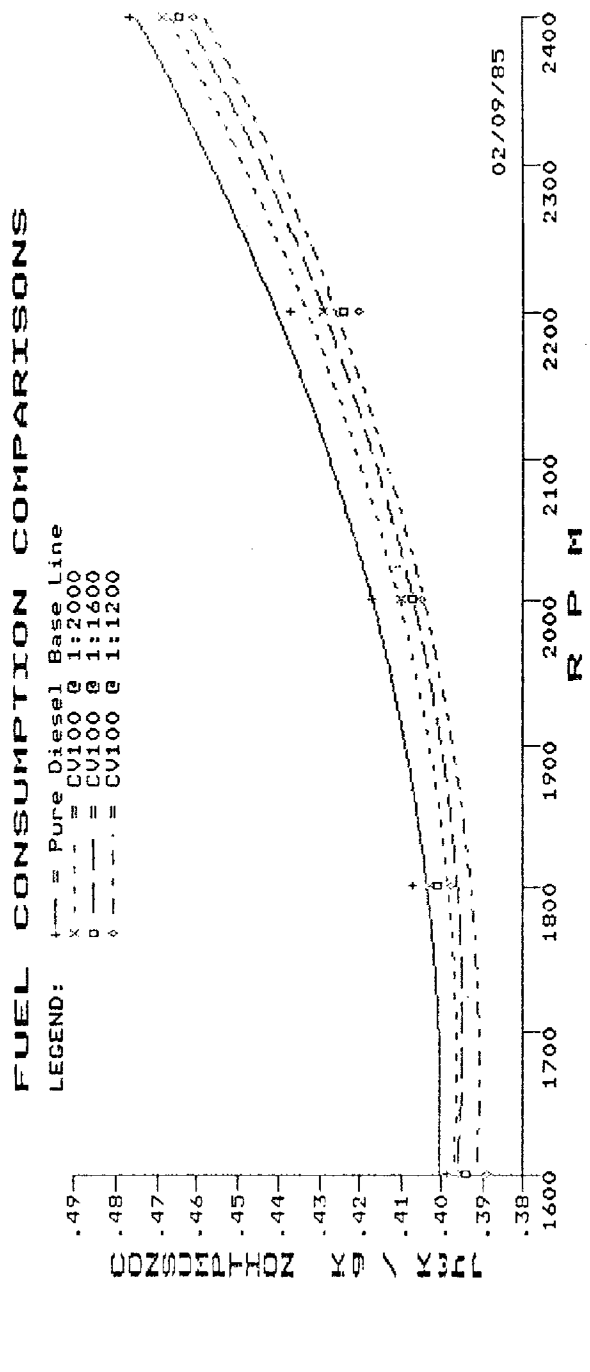
RPM	Torque (Nm)	Time for 48 ml fuel (sec)	Exhaust Temp (°C)	
1600	37.60	63.70	550	T_{AVE} = 63.75 sec
1600	37.60	63.78		P_{AVE} = 6.30 kW
1600	37.60	63.84		
1600	37.60	63.68		BSFC = 0.396 kg/kW hr
1800	37.10	56.23	580	T_{AVE} = 56.25 sec
1800	37.10	56.31		P_{AVE} = 6.99 kW
1800	37.10	56.20		
1800	37.10	56.26		BSFC = 0.404 kg/kW hr
2000	36.20	50.83	590	T_{AVE} = 50.84 sec
2000	36.20	50.90		P_{AVE} = 7.58 kW
2000	36.20	50.85		
2000	36.20	50.79		BSFC = 0.413 kg/kW hr
2200	33.90	46.75	605	T_{AVE} = 46.85 sec
2200	33.90	46.81		P_{AVE} = 7.81 kW
2200	33.90	46.94		
2200	33.90	46.91		BSFC = 0.434 kg/kW hr
2400	31.80	42.29	620	T_{AVE} = 42.24 sec
2400	31.80	42.33		P_{AVE} = 7.99 kW
2400	31.80	42.15		
2400	31.80	42.20		BSFC = 0.475 kg/kW hr

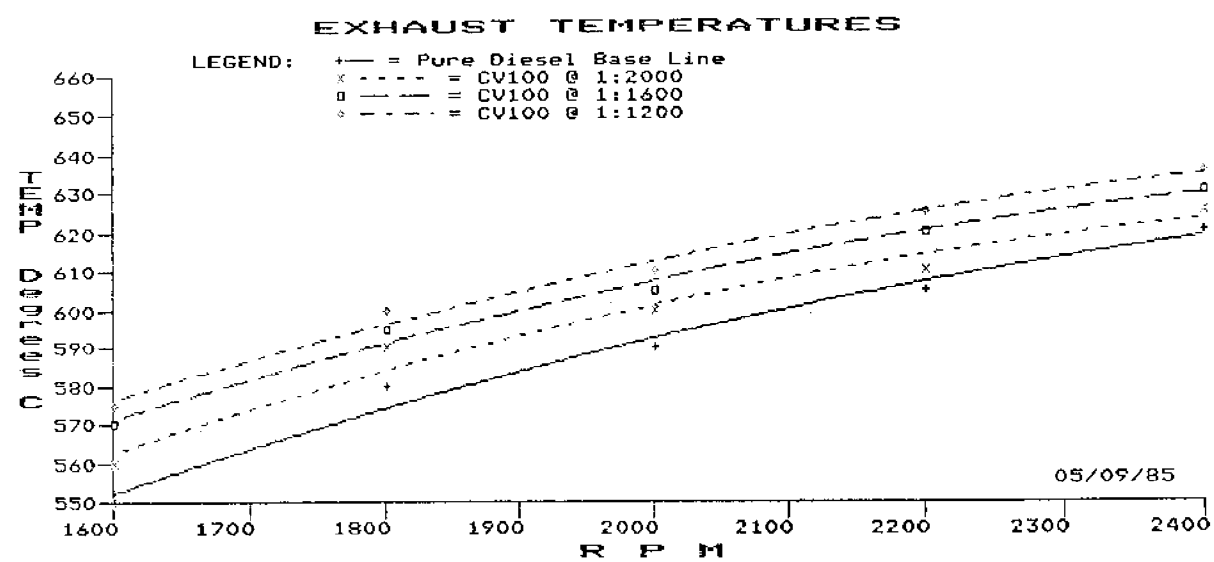
ambient air temperature = 23°C

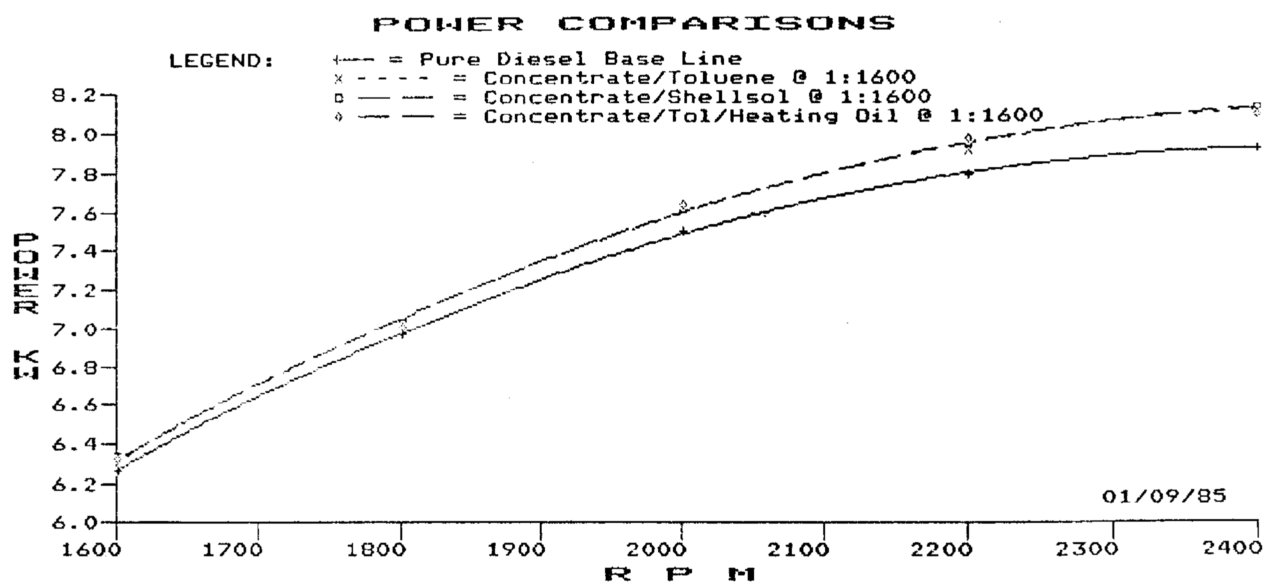
APPENDIX 2

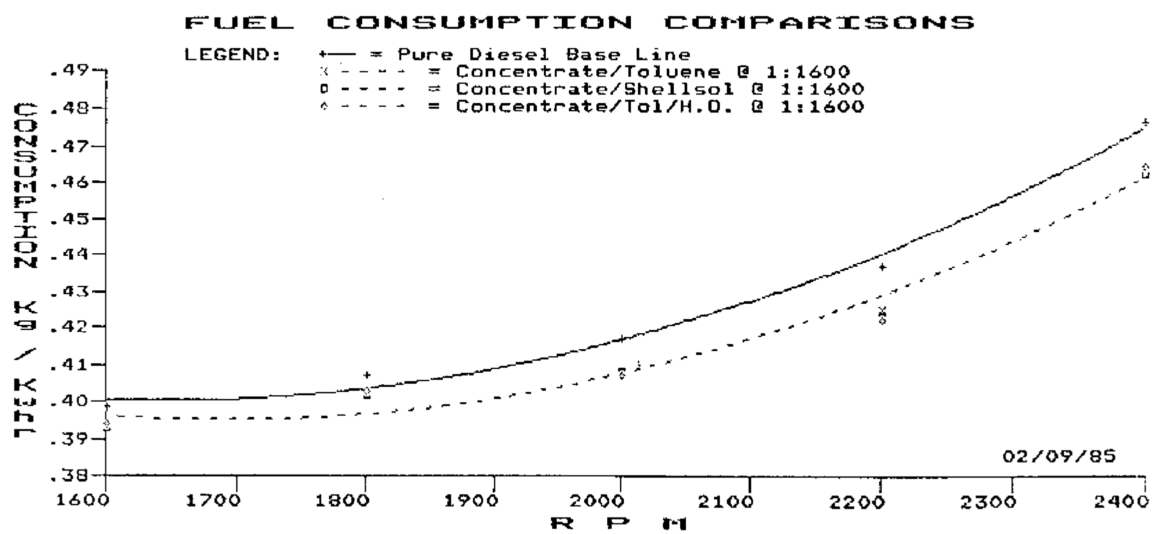
GRAPHS OF TEST RESULTS


POWER COMPARISONS
LEGEND:
+—— = Pure Diesel Base Line
x - - - = CV100 @ 1:2000
□ —— = CV100 @ 1:1600
◇ — · — · = CV100 @ 1:1200
01/09/85
POWER KW
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
6.4
6.2
6.0
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M

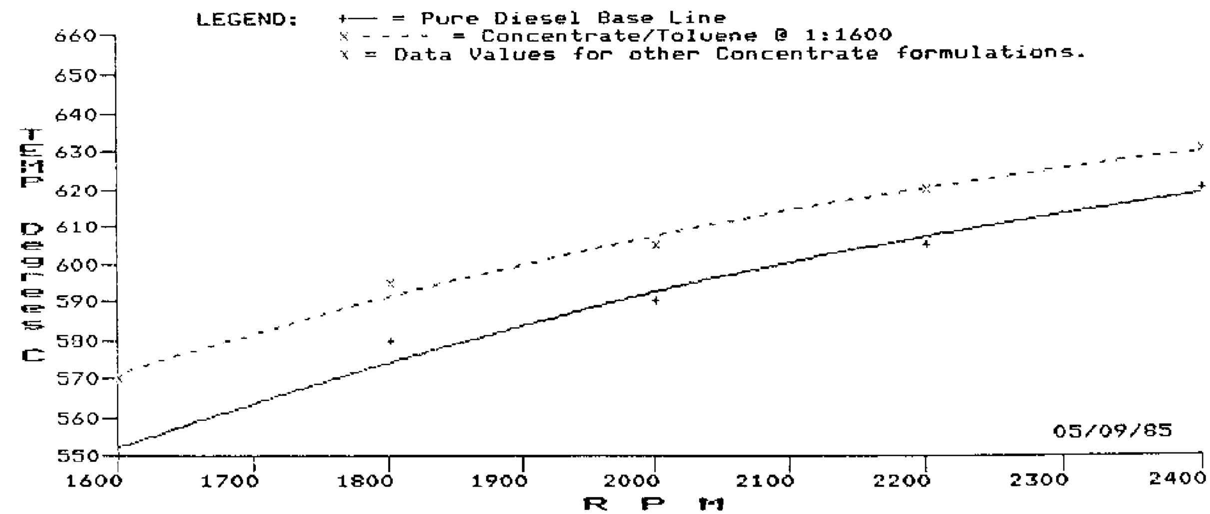

FUEL CONSUMPTION COMPARISONS
LEGEND:
+ = Pure Diesel Base Line
× = CV100 @ 1:2000
□ = CV100 @ 1:1600
◇ = CV100 @ 1:1200
.49
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
02/09/85

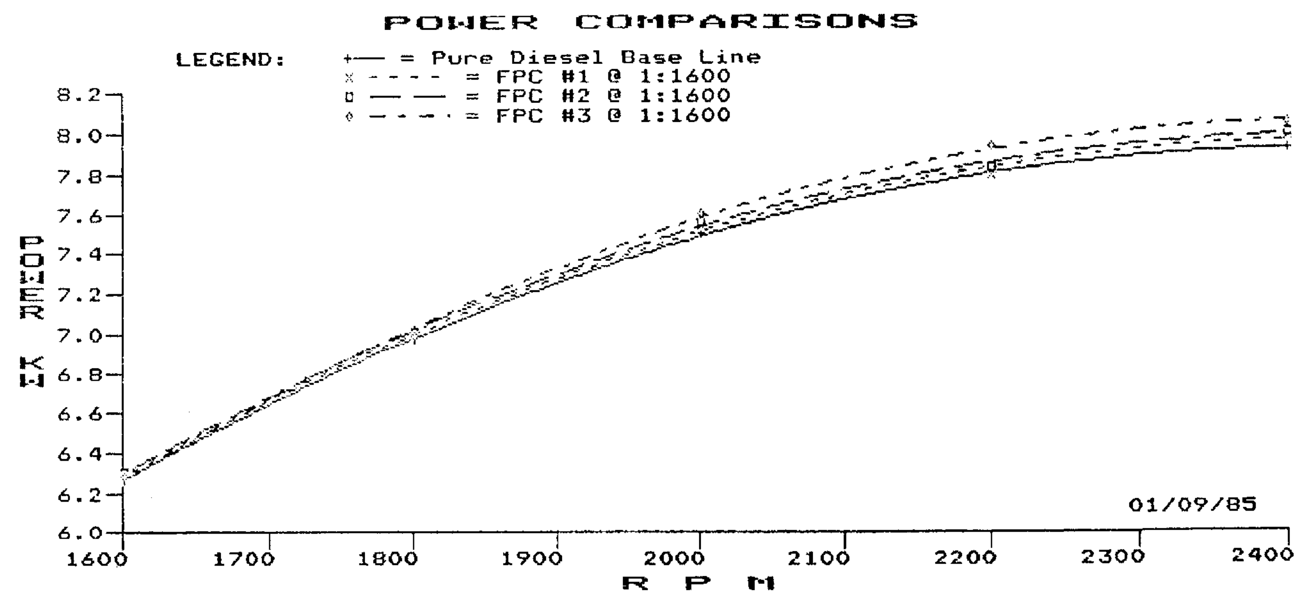

EXHAUST TEMPERATURES
LEGEND:
+—— = Pure Diesel Base Line
x - - - - = CV100 @ 1:2000
□ —— —— = CV100 @ 1:1600
◇ — - — - = CV100 @ 1:1200
TEMP Degrees C
660
650
640
630
620
610
600
590
580
570
560
550
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
05/09/85

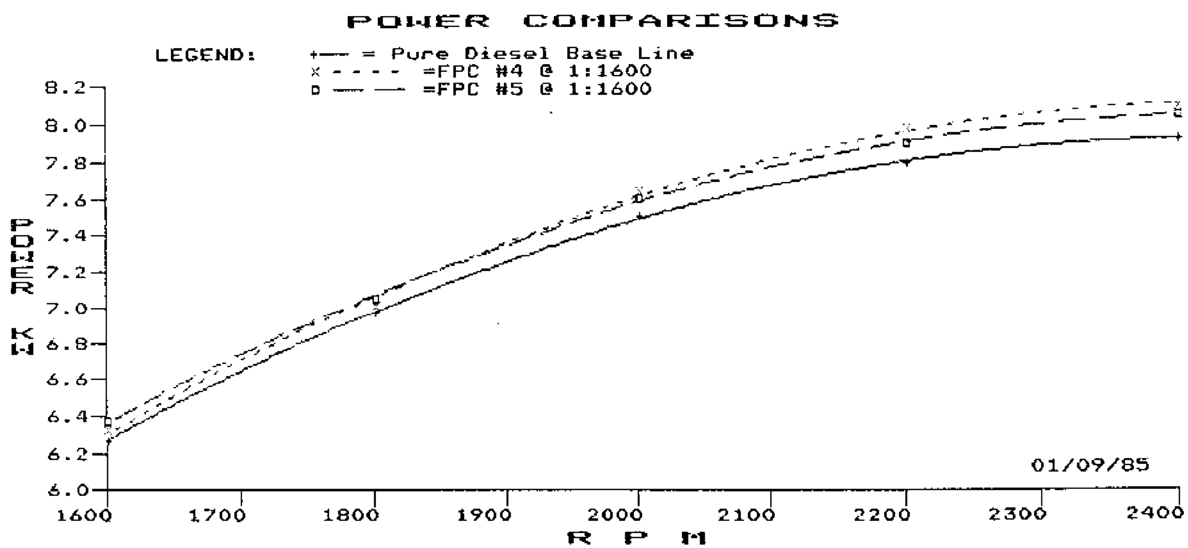

POWER COMPARISONS
LEGEND:
= Pure Diesel Base Line
= Concentrate/Toluene @ 1:1600
= Concentrate/Shellsol @ 1:1600
= Concentrate/Tol/Heating Oil @ 1:1600
POWER KW
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
6.4
6.2
6.0
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85

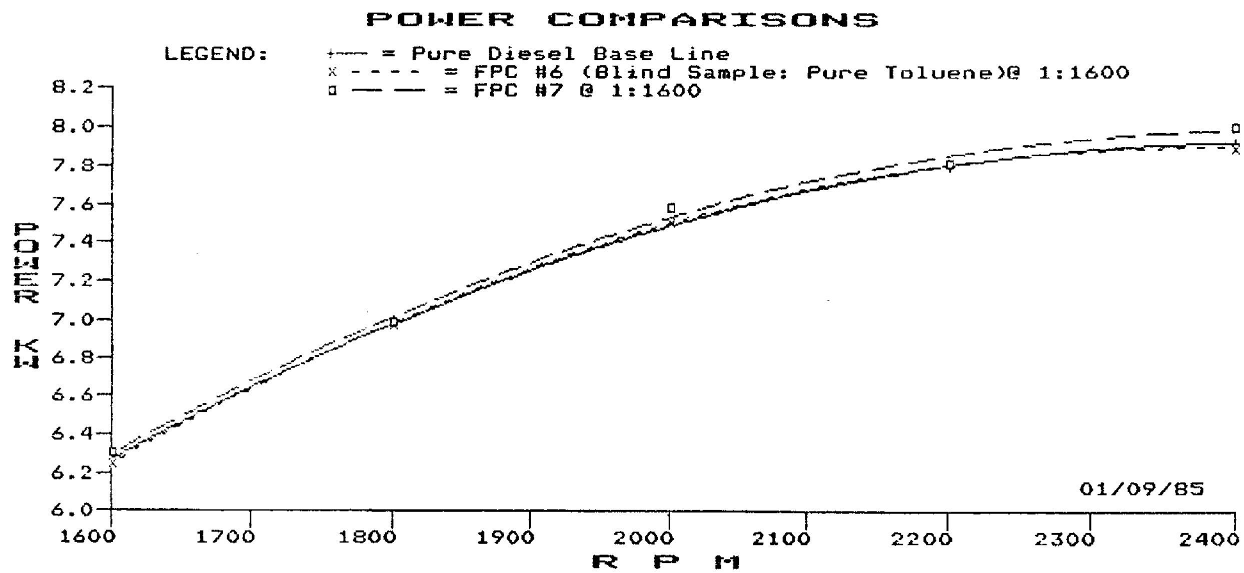

FUEL CONSUMPTION COMPARISONS
LEGEND:
= Pure Diesel Base Line
= Concentrate/Toluene @ 1:1600
= Concentrate/Shellsol @ 1:1600
= Concentrate/Tol/H.O. @ 1:1600
.49
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
02/09/85

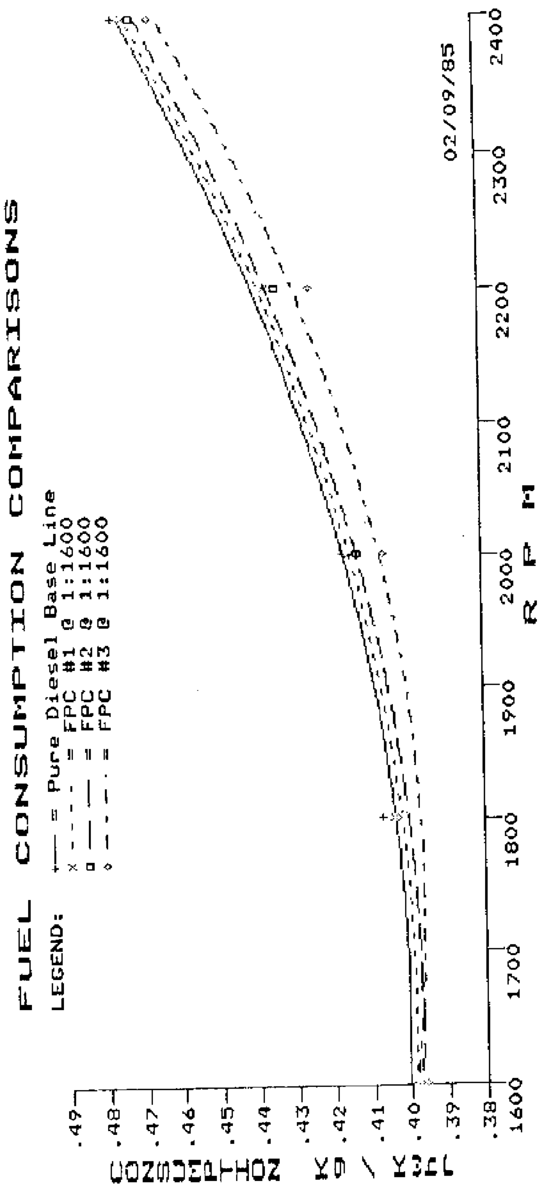
EXHAUST TEMPERATURES

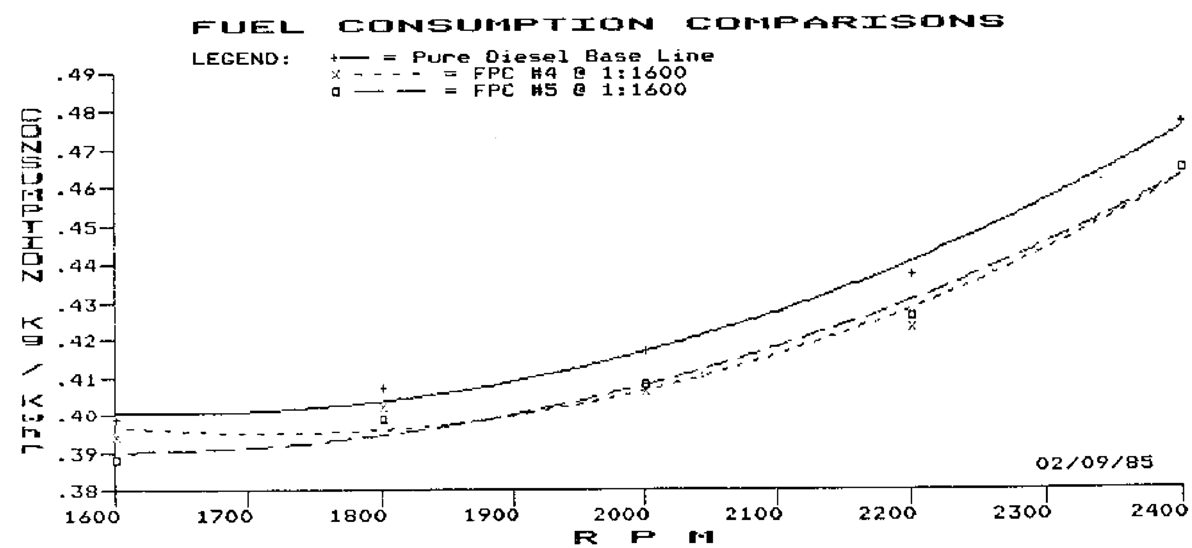


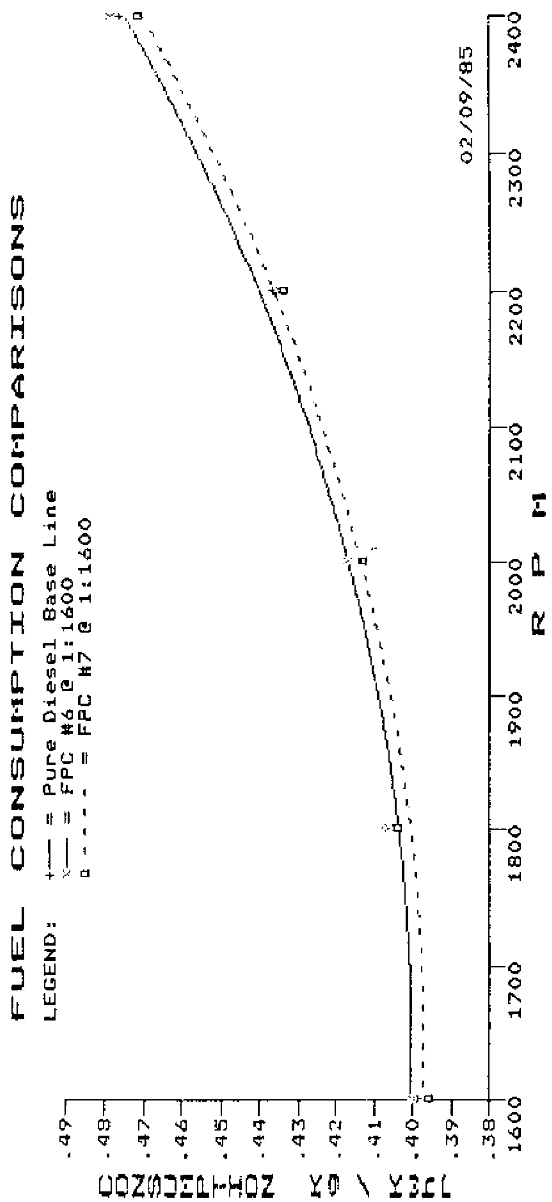

POWER COMPARISONS
LEGEND:
+—— = Pure Diesel Base Line
x - - - - = FPC #1 @ 1:1600
□ —— —— = FPC #2 @ 1:1600
◊ — - — - = FPC #3 @ 1:1600
POWER KW
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
6.4
6.2
6.0
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85

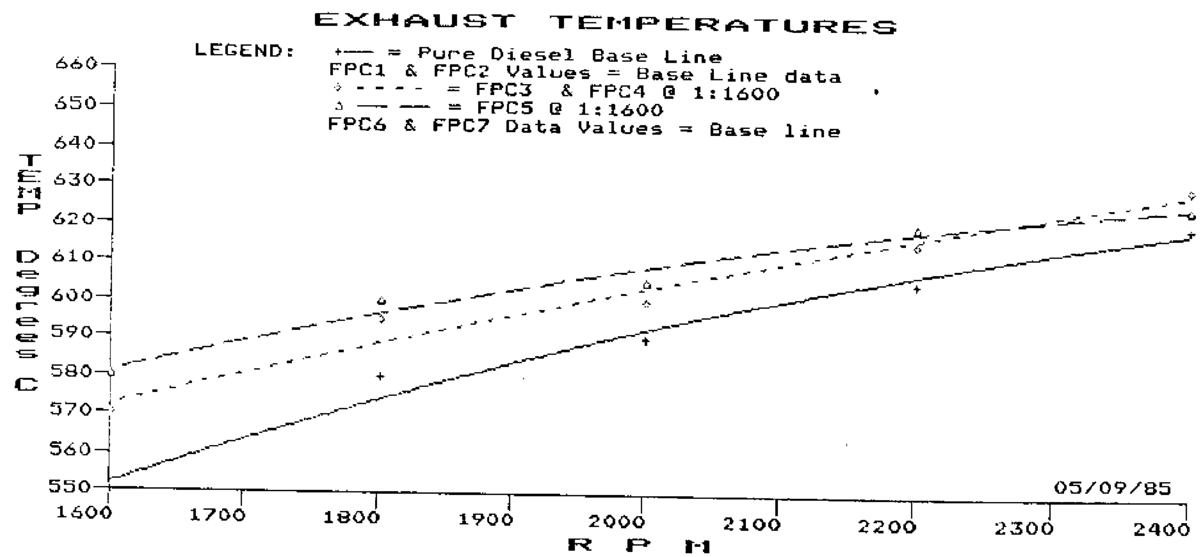

POWER COMPARISONS
LEGEND:
= Pure Diesel Base Line
=FPC #4 @ 1:1600
=FPC #5 @ 1:1600
POWER KW
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
6.4
6.2
6.0
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85

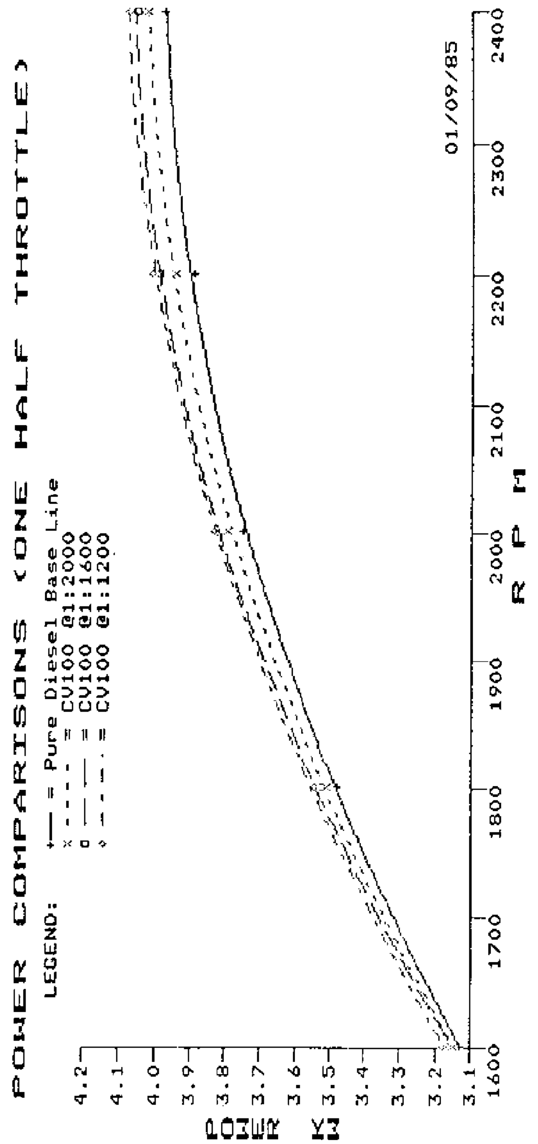

POWER COMPARISONS
LEGEND:
= Pure Diesel Base Line
= FPC #6 (Blind Sample: Pure Toluene)@ 1:1600
= FPC #7 @ 1:1600
POWER KW
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
6.6
6.4
6.2
6.0
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85


FUEL CONSUMPTION COMPARISONS
LEGEND:
+—— = Pure Diesel Base Line
x - - - = FPC #1 @ 1:1600
□ — — = FPC #2 @ 1:1600
◇ — - = FPC #3 @ 1:1600
.49
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
02/09/85

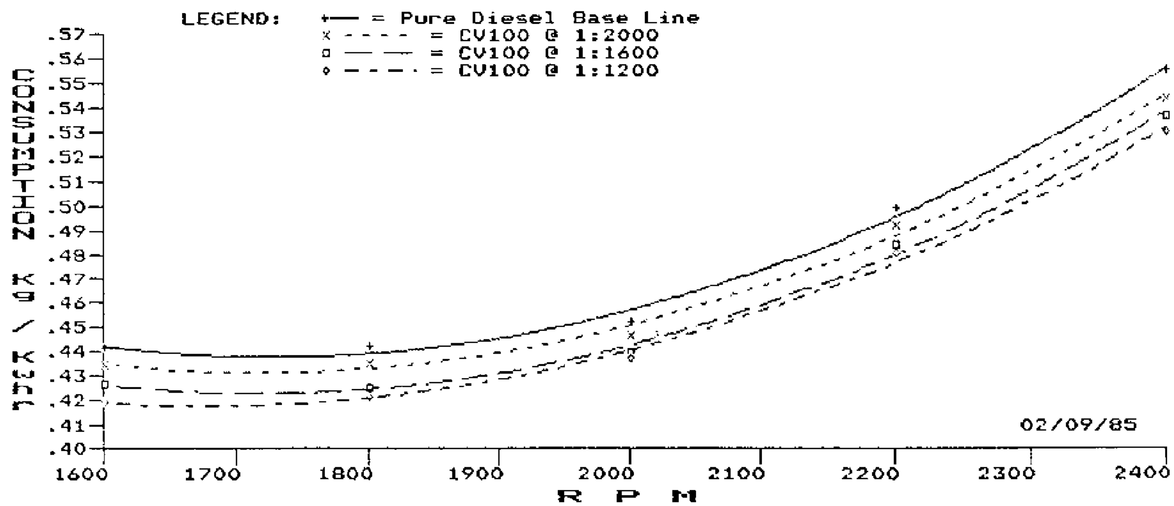

FUEL CONSUMPTION COMPARISONS
LEGEND:
= Pure Diesel Base Line
= FPC #4 @ 1:1600
= FPC #5 @ 1:1600
.49
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
02/09/85

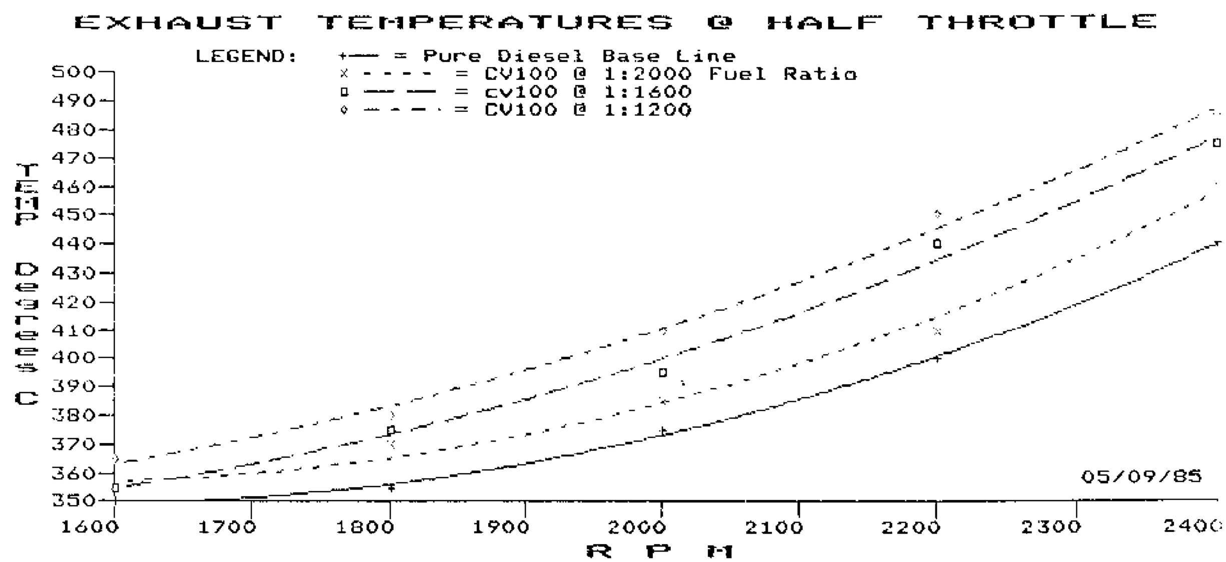

FUEL CONSUMPTION COMPARISONS
LEGEND:
+—— = Pure Diesel Base Line
x—— = FPC #6 @ 1:1600
□ - - - - = FPC #7 @ 1:1600
02/09/85
R P M
1600
1700
1800
1900
2000
2100
2200
2300
2400
.38
.39
.40
.41
.42
.43
.44
.45
.46
.47
.48
.49

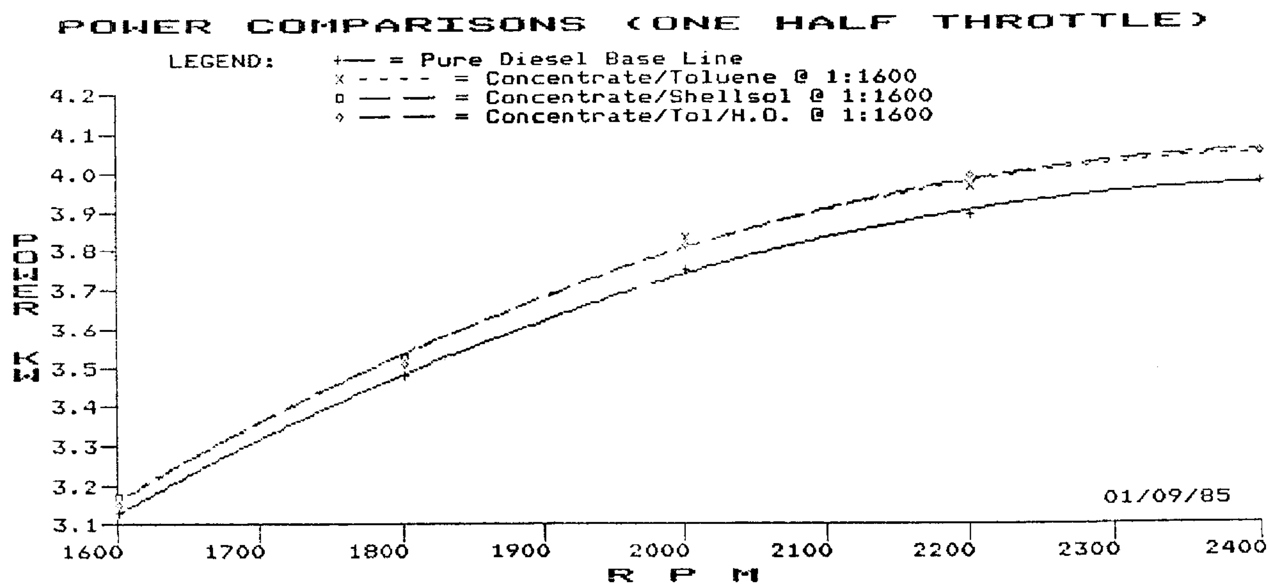

EXHAUST TEMPERATURES
LEGEND:
+—— = Pure Diesel Base Line
FPC1 & FPC2 Values = Base Line data
◊ - - - - = FPC3 & FPC4 @ 1:1600
Δ —— —— = FPC5 @ 1:1600
FPC6 & FPC7 Data Values = Base line
TEMP Degrees C
660
650
640
630
620
610
600
590
580
570
560
550
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
05/09/85


POWER COMPARISONS (ONE HALF THROTTLE)
LEGEND:
+—— = Pure Diesel Base Line
x - - - - = CV100 @1:2000
□ —— —— = CV100 @1:1600
◊ — - — - = CV100 @1:1200
4.2
4.1
4.0
3.9
3.8
3.7
3.6
3.5
3.4
3.3
3.2
3.1
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85

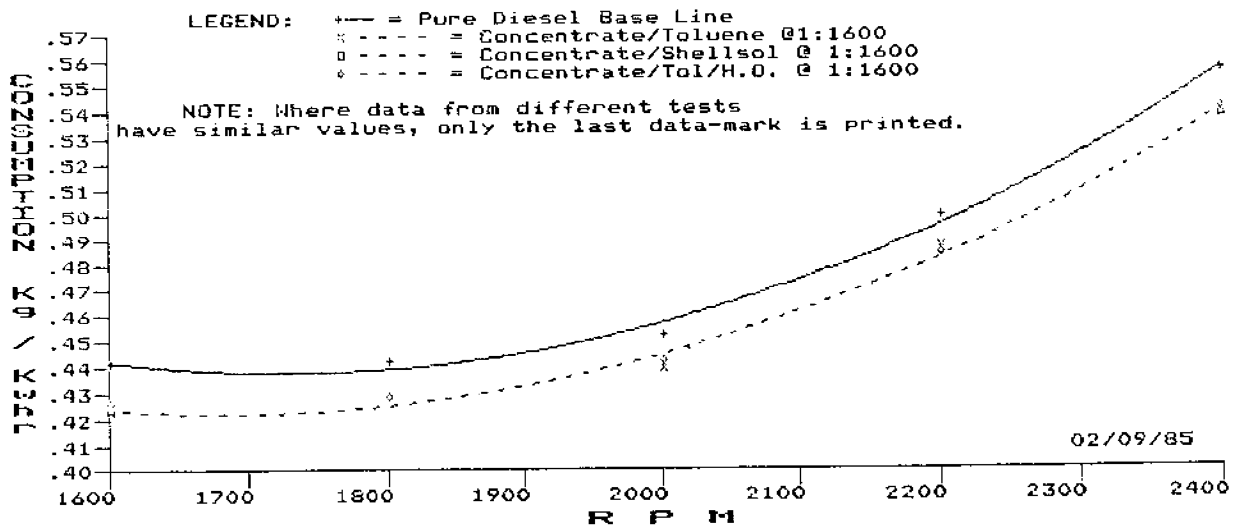
FUEL CONSUMPTIONS AT HALF THROTTLE

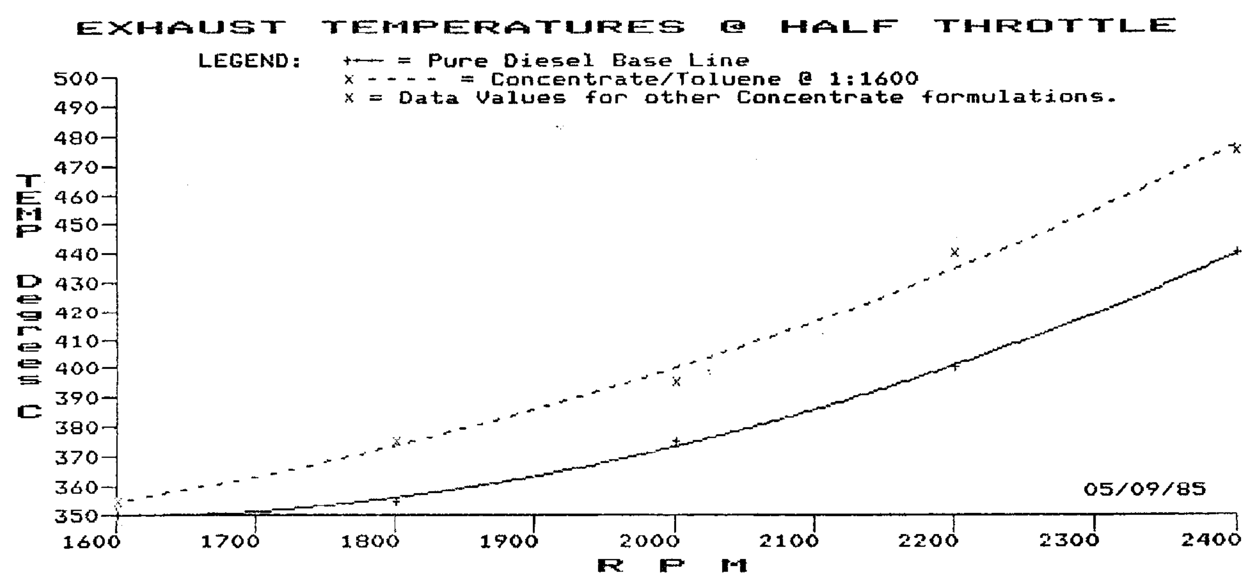


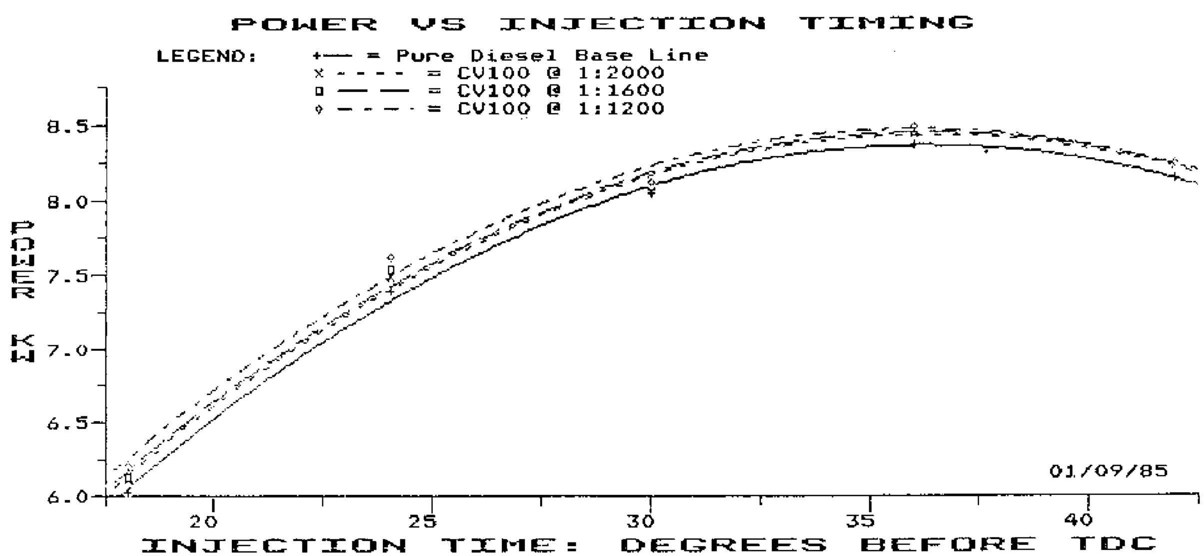

EXHAUST TEMPERATURES @ HALF THROTTLE
LEGEND:
= Pure Diesel Base Line
= CV100 @ 1:2000 Fuel Ratio
= cv100 @ 1:1600
= CV100 @ 1:1200
TEMP DEGREES C
500
490
480
470
460
450
440
430
420
410
400
390
380
370
360
350
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
05/09/85

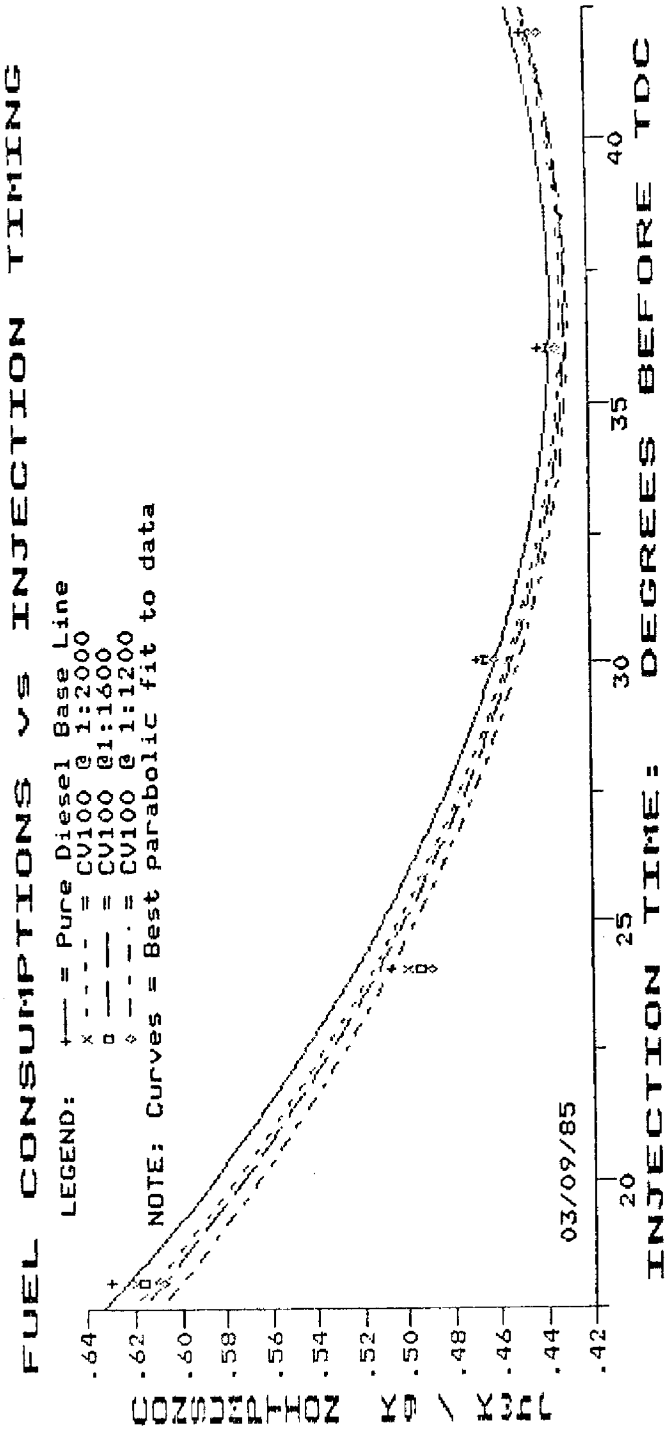

POWER COMPARISONS (ONE HALF THROTTLE)
LEGEND:
= Pure Diesel Base Line
= Concentrate/Toluene @ 1:1600
= Concentrate/Shellsol @ 1:1600
= Concentrate/Tol/H.O. @ 1:1600
POWER KW
4.2
4.1
4.0
3.9
3.8
3.7
3.6
3.5
3.4
3.3
3.2
3.1
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
01/09/85

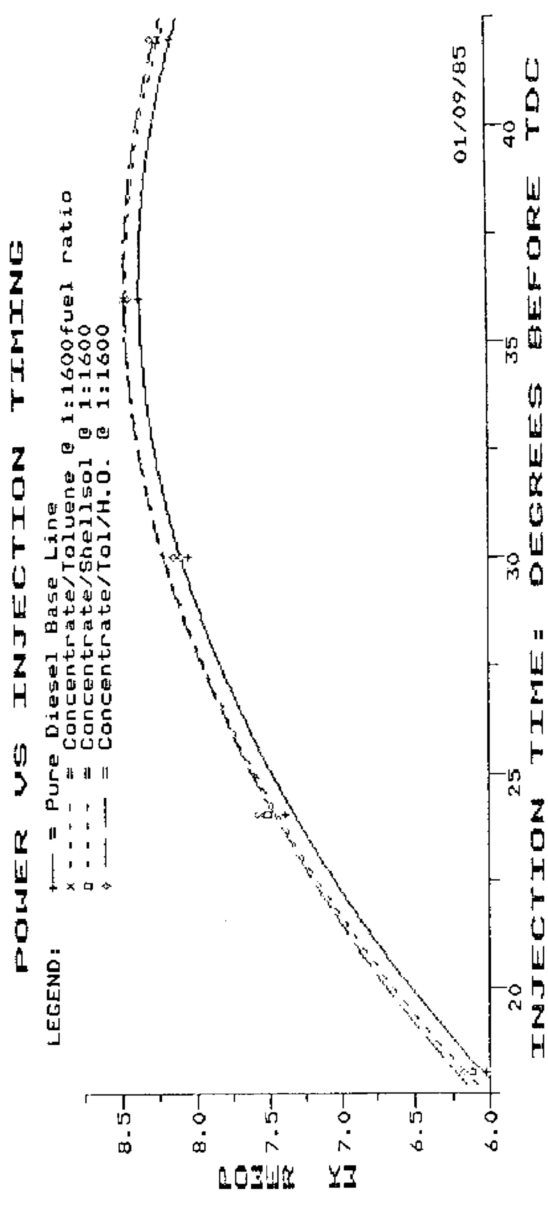
FUEL CONSUMPTIONS AT HALF THROTTLE

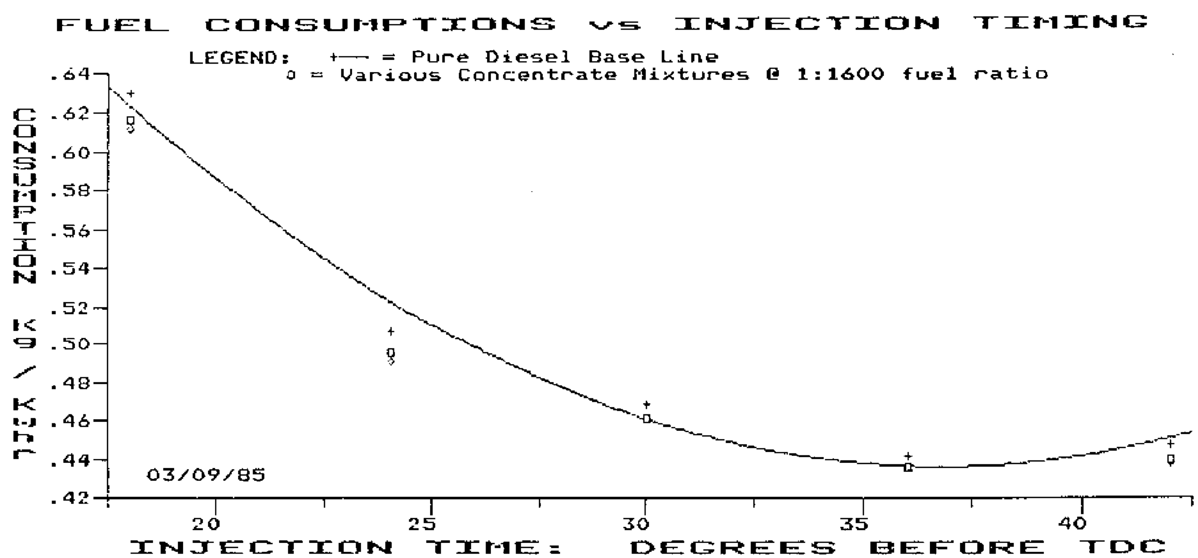



EXHAUST TEMPERATURES @ HALF THROTTLE
LEGEND: +—— = Pure Diesel Base Line
x - - - - = Concentrate/Toluene @ 1:1600
x = Data Values for other Concentrate formulations.
TEMP Degrees C
500
490
480
470
460
450
440
430
420
410
400
390
380
370
360
350
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
05/09/85

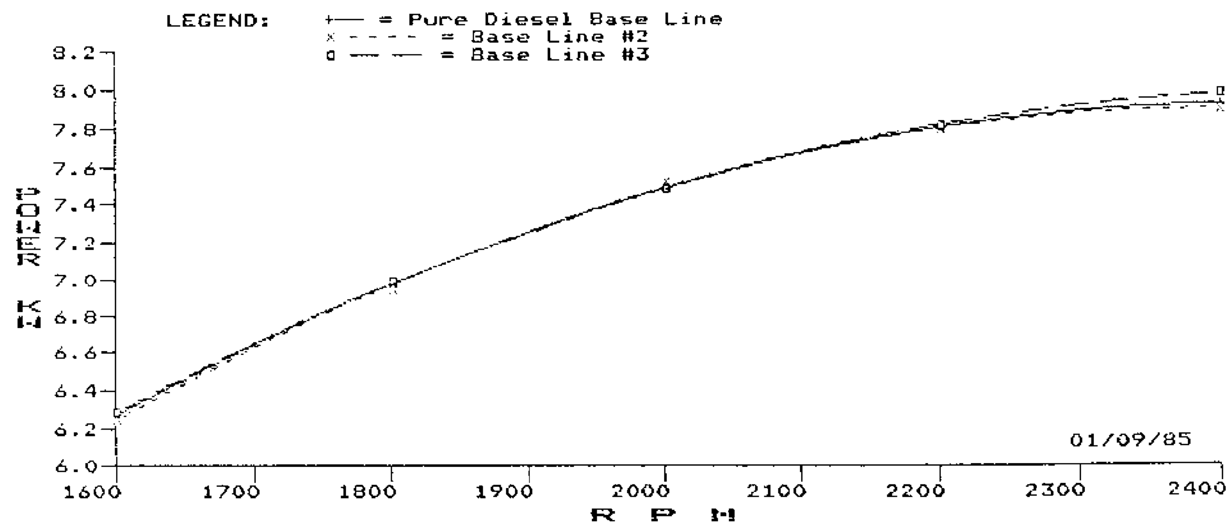

POWER VS INJECTION TIMING
LEGEND:
+ = Pure Diesel Base Line
x = CV100 @ 1:2000
□ = CV100 @ 1:1600
◇ = CV100 @ 1:1200
POWER KW
8.5
8.0
7.5
7.0
6.5
6.0
20
25
30
35
40
INJECTION TIME: DEGREES BEFORE TDC
01/09/85

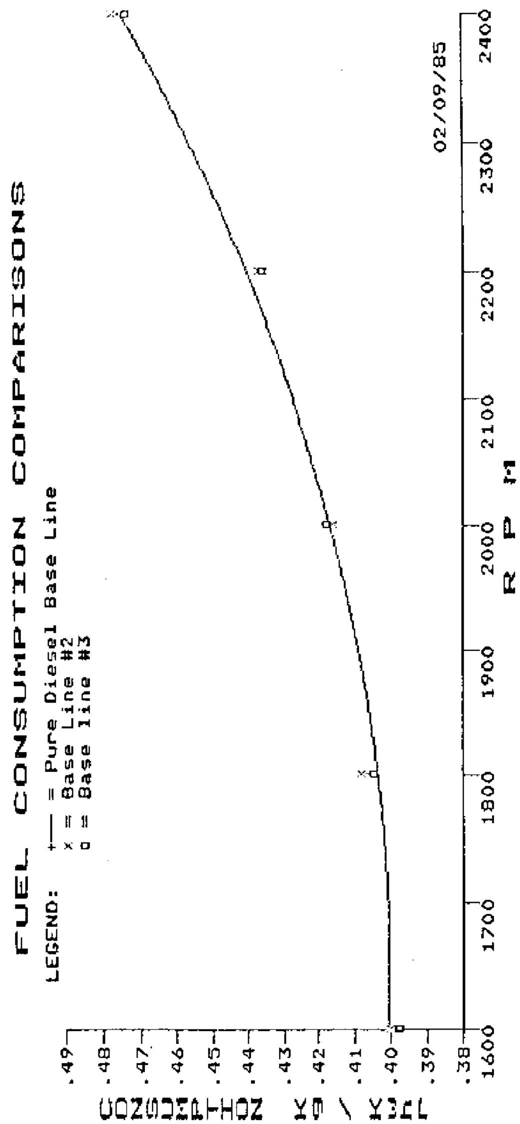

FUEL CONSUMPTIONS VS INJECTION TIMING
LEGEND:
+—— = Pure Diesel Base Line
x - - - - = CV100 @ 1:2000
□ —— —— = CV100 @1:1600
◇ — - — - = CV100 @ 1:1200
NOTE: Curves = Best parabolic fit to data
03/09/85
INJECTION TIME: DEGREES BEFORE TDC
.64
.62
.60
.58
.56
.54
.52
.50
.48
.46
.44
.42
20
25
30
35
40


POWER VS INJECTION TIMING
LEGEND:
= Pure Diesel Base Line
= Concentrate/Toluene @ 1:1600fuel ratio
= Concentrate/Shellsol @ 1:1600
= Concentrate/Tol/H.O. @ 1:1600
8.5
8.0
7.5
7.0
6.5
6.0
POWER KW
20
25
30
35
40
INJECTION TIME= DEGREES BEFORE TDC
01/09/85


FUEL CONSUMPTIONS vs INJECTION TIMING
LEGEND: +— = Pure Diesel Base Line
o = Various Concentrate Mixtures @ 1:1600 fuel ratio
CONSUMPTION Kg / KWH
.64
.62
.60
.58
.56
.54
.52
.50
.48
.46
.44
.42
03/09/85
20
25
30
35
40
INJECTION TIME: DEGREES BEFORE TDC

PURE DIESEL RUNS: POWER COMPARISONS

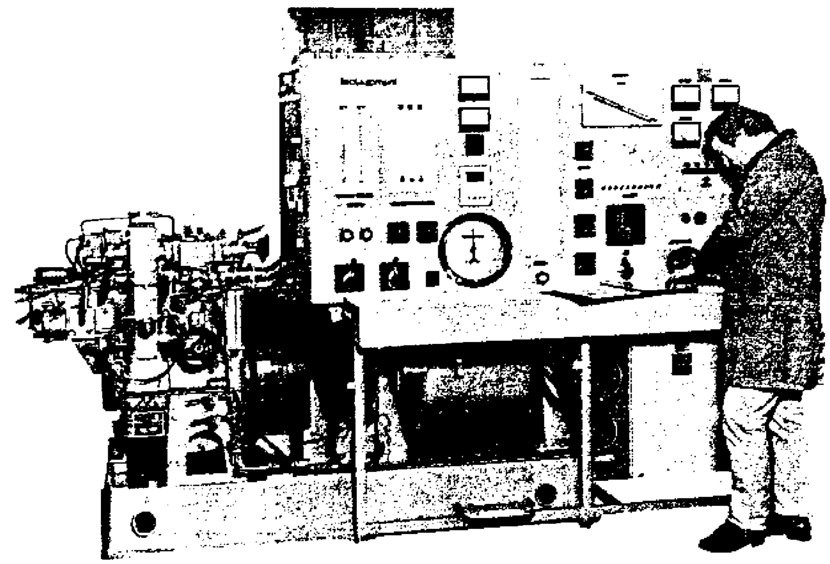



FUEL CONSUMPTION COMPARISONS
LEGEND: +—— = Pure Diesel Base Line
x = Base Line #2
□ = Base line #3
.49
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
1600
1700
1800
1900
2000
2100
2200
2300
2400
R P M
02/09/85

APPENDIX 3

ENGINE AND EQUIPMENT SPECIFICATION

TD35 Varimax Test and Research Engine Rig



Features

Designed specially for teaching and research purposes
Simultaneous study of dynamics and thermodynamics of the internal combustion engine
A robust engine with advanced and unique features
Variable Compression Ratio 4,5:1 to 20:1 whilst engine is running
Petrol/diesel operation with minimal changeover time
Valve timing and opening period adjustable whilst the engine is running
Strain gauged crankshaft suspension system allows analysis of gas and dynamic forces
Transducers indicate cylinder pressure, diesel fuel line pressure, injector needle lift, and flywheel cyclic variations
Mixture strength can be controlled manually with different carburettor chokes supplied
Diesel injection timing may be varied
Spark timing may be varied
Mass of the flywheel can be altered by inertia ring addition
Basic engine, dynamometer motor, and electrical loading unit designed to accept supercharging
Fully integrated test rig complete with instrumentation
Separate cooling circuits for cylinder head and cylinder jacket
Package units for supercharging, petrol injection and operation on natural gas and LPG fuel are available as optional extras

Description

The Varimax Engine Test Rig was designed and developed specially as a teaching unit and for the evaluation of the effects on engine performance of certain fundamental variables. This makes it an invaluable tool for research workers, university lecturers and students.

The engine is a four-stroke, vertical single cylinder, water cooled diesel/petrol unit, nominally rated at 7,6kW (10bhp) with speed variation between 500 and 3000rev/min. The compression ratio can be adjusted between 4,5:1 and 20:1 by raising or lowering the complete crankshaft assembly which is carried in a cradle pivoted on an axis parallel with the crankshaft. Suspension members for carrying this cradle project through the crankcase to anchor points. These members have prepared surfaces to which are attached strain gauges for determining the vertical and horizontal components of the forces acting on the main bearings.

The cast iron cylinder head houses overhead camshafts which are chain driven from the crankshaft. A compensating linkage ensures that there is no phase shift of the camshafts as the crankshaft is raised and lowered.

It is possible to vary the timing and the period for both inlet and exhaust valves whilst the engine is running. These variables are quite independent of each other. Alternatively the valve mechanism may be locked at set values.

The cylinder head is provided with three similar apertures suitable for receiving sparking plugs or pressure transducers, or a fuel injector when operating as a compression ignition engine.

The aluminium piston has one oil scraper and three compression rings.

Ignition is by means of a magneto, driven from one of the half-speed drives. Spark timing is fully adjustable.

A separate cold water make up supply to the mixing tank enables temperatures across the engine to be stabilized during test.

A limit switch short circuits the magneto at compression ratios in excess of 13:1 and, in the event of the engine overspeeding, actuates a solenoid to automatically cut off the fuel supply when running as a diesel engine.

The normal carburettor fitted is a down-draught type supplied with a variety of chokes and an adjustable metering jet.

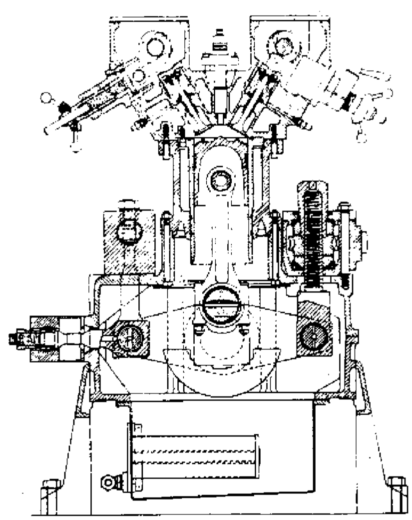
A motor driven pump circulates cooling water through the cylinder head and the cylinder liner jacket, as two separate systems. The flow through each system can be controlled independently and measured.

Crankshaft, camshaft and cradle bearings are all pressure lubricated from a motor driven oil pump. All pipes are BSS colour coded, but a section is in semi-transparent nylon to reveal oil flow.

A crank angle timing disc is provided.

The moment of inertia of the flywheel can be increased by the addition of an inertia ring.

The engine is connected to a trunnion mounted dc swinging field dynamometer through a shaft with flexible universal couplings at each end.



SECTION THROUGH ENGINE

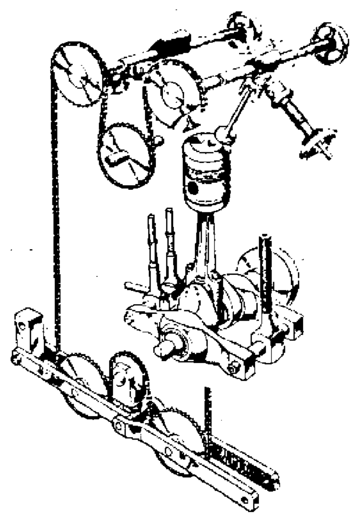
Electrical loading unit

An adjacent framework houses the electrical loading unit. A number of resistance mats in parallel provide 40 equal increments of load. A field regulator provides fine adjustment between each step.

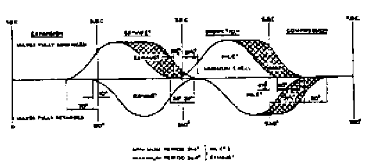
A separate circuit enables the engine to be motored for both starting and determination of friction horse power.

A dc supply of 50 amps at 220V is required for starting purposes, and also provides the generator field excitation. If no such supply is available suitable rectifier cubicle can be quoted for as an extra on request. (Item TD35e).

A single phase ac supply of 15 amps at 200/250V 50/60Hz is also required.



CAMSHAFT AND DRIVE



VALVE TIMING DIAGRAM
NOMINAL VALUES ONLY

Engine services

A framework over the dynamometer incorporates the following engine services:

(a) A large capacity air intake box with a pulsation damper and an air flow measuring orifice.
(b) A cooling water mixing tank. A supply of cold water is required together with an arrangement for disposal of the overflow to drain.
(c) Four fuel tanks (2 diesel, 2 petrol).

Engine control panel

A panel in front of the services and loading frames carries the following instruments and controls:

A three bulb pipette for fuel consumption measurements and two direct reading rotameters for continuously monitoring diesel fuel and petrol flow.
Twin flowmeters for measuring cooling water flow through the engine jacket and the cylinder head respectively. Valves for controlling the cooling water in each of these two circuits.
Master thermoelectric temperature indicator and selector switch for all water temperatures, oil temperature and air intake temperature
A separate thermoelectric indicator for exhaust temperature
Inclined-scale manometer for air orifice pressure drop
Exhaust gas sampling point
Engine rev/min Indicator
Fuel Selector taps
Throttle controls – Petrol and Diesel
Load control switches
Full/half speed load switch
Torque indicating unit
Ignition switch
Dynamometer field regulator
Decompressor lever
Lubricating oil pressure gauge
Emergency STOP button
Motor/Generator change-over switch
Starting/motoring rheostat
Dynamometer armature voltmeter
Dynamometer armature ammeter-Motoring
Dynamometer armature – Generating
Dynamometer field voltmeter
HRC Fuses to protect both Motoring and Generating circuits. Starters for oil and water pumps
Warning panel
Additional instrumentation includes a compression ratio indicator mounted on the engine.
A hand-held stroboscope is also provided to indicate both ignition and valve timing, from degree markings on the flywheel and shutters actuated by the valve tappets
The engine may be converted from diesel to petrol and vice versa in only a few minutes, without removing the cylinder head.

Electronic instrumentation

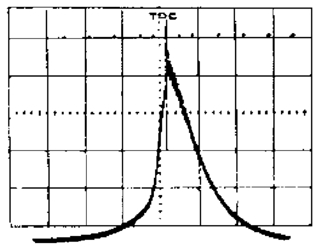
The following transducers are fitted to or incorporated in the engine:
Cylinder pressure (piezo-electric type)
Diesel fuel line pressure (resistance type)
Diesel needle lift (differential transformer)
Vertical and horizontal forces in members supporting the crankshaft (strain gauges)
Flywheel cyclic irregularity (inductive pick-up)
Crank angle timing (inductive pick-up)

These transducers are connected to a multi-point socket from which a cable leads to a separate free standing Electronic Control Panel TD35a Mk II.

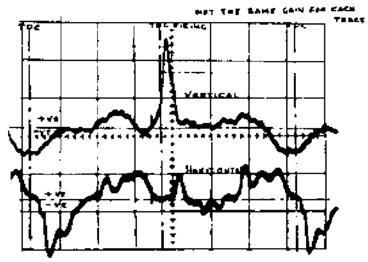
CYLINDER PRESSURE TRACES

1550 rev/min Full throttle 10.5:1 Compression ratio Petrol 98 octane 20° BTDC static ignition 10.7 BHP 87.5 BMEP.

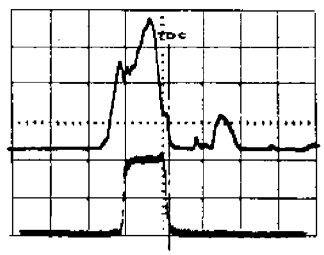
Heavy detonation



VERTICAL AND HORIZONTAL FORCES



DIESEL FUEL LINE PRESSURE AND NEEDLE LIFT



Range of experiments and performance graphs

Suggested experiments and investigations which may be conducted with this engine and for which complete controls and instrumentation are provided are listed below. This list is by no means exhaustive and serves only as a guide for carrying out a number of experiments pertinent to reciprocating internal combustion engine thermodynamics and dynamics. The facilities incorporated in the design enable an extensive range of projects to be carried out.

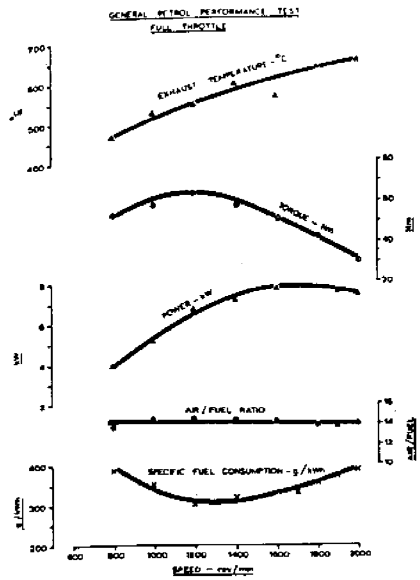
1. Volumetric efficiency – the effect of valve timing. Independent variation of timing and period for inlet and exhaust valves.
2. Measurement of gas and dynamic forces – polar load diagram.
3. Analaysis of cyclic irregularity.
4. Exhaust emission.
5. Measurement of friction and fluid pumping losses.

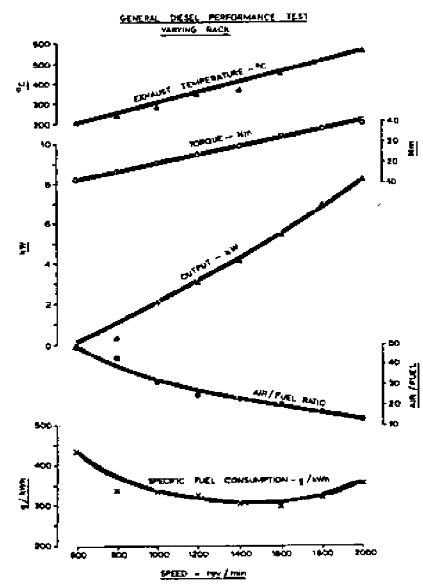
As a spark ignition engine:

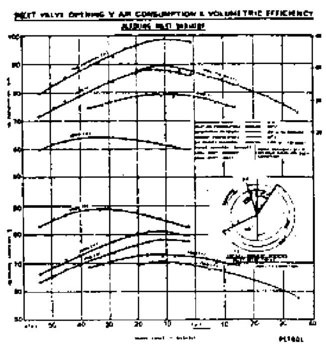
6. Performance characteristic curves of power, specific fuel consumption, etc., over the full speed range.
7. Mixture strength test v thermal efficiency and torque. Also power against air fuel ratio, specific fuel consumption, exhaust temperature.
8. Effect of variable compression ratio on power and thermal efficiency. Also detonation and pre-ignition.
9. Variation of ignition timing – relationship with speed for maximum power developed.
10. Detonation and Octane rating.

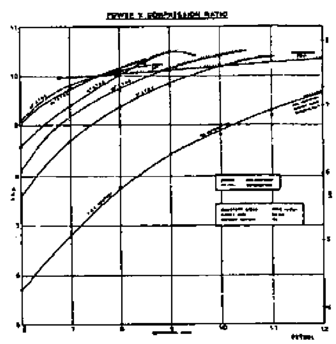
As a compression ignition engine:

11. Performance characteristic curves
12. Effect of variable compression ratio at selected injection timings.
13. Variation of injection timing.
14. Fuel injection equipment studies – needle lift and fuel line pressure can be displayed.









Engine specification

Bore: 95.25mm(3.75in).
Stroke: 114.3mm(4.5in).
rev/min: 500-3000.
Nominal power: 7.5kW(10bhp).
Compression ratio: 4.5:1 to 20:1.

Optional extra

TD35a Mk II Electronics Control Panel.
TD35b Supercharger and associated equipment.
TD35c Petrol Injection Equipment.
TD35d Natural gas and LPG fuel equipment.
TD35e Rectifier Unit to provide 220V 50A dc supply from 3 phase ac supply (voltage and frequency to be specified by the customer at the enquiry stage).

Space required

For free access around the engine test bed an area of 3960mm(156in) by 2540mm(100in) is required. The electronic control panel is a separate unit which should be positioned conveniently close to the engine.