CHAPTER 4

DUAL FUEL ENGINE EXPERIMENTAL INVESTIGATIONS

4.1 INTRODUCTION

The various methods of successfully introducing ethanol as a supplementary fuel in C.I. engines and their limitations were adequately covered in an earlier section. It has been suggested there that among the various methods available, the carburetion of ethanol into the system as a retrofit arrangement seem to be the most viable and simple, as it requires only marginal changes in the engine configuration. However, problems associated with dual fuel operation beyond the threshold limit of ethanol admission are: increase of ignition delay, diesel knock, misfiring, incomplete combustion, high rate of pressure rise and peak pressure. In order to understand the factors influencing the varied performance under different operating and design conditions extensive engine tests have been carried out on a variety of diesel engines, over a wide range of loads and speeds. The test programme included, dual fuel operation of an automotive naturally aspirated multicylinder diesel engine, a multicylinder turbocharged engine, studies on a single cylinder engine with different combustion chamber configurations and long duration wear studies on a single cylinder engine. The experience gained and data generated helped in finalising and executing detailed experimental
investigations on a single cylinder C.I. engine for correlating the dual fuel combustion model developed. In addition to the normally observed engine parameters such as thermal efficiency, delay, peak pressure and the like, considerable emphasis was placed on the development of a dual fuel combustion modelling and ignition delay correlation in the present work.

A complete description of the experimental setups and the associated instrumentations used in these investigations are presented in this chapter.

4.2 DUAL FUEL OPERATION OF A MULTICYLINDER AUTOMOTIVE DIESEL ENGINE

A direct injection, water cooled, six cylinder, automotive diesel engine of 6200 CC cubic capacity fitted with a centrifugal governor was used to examine the performance of the engine at a wide range of speeds and loads. The engine was directly coupled to a hydraulic dynamometer for loading. To facilitate the admission and atomization of ethanol and mixing it with air, a suitable carburettor was installed in the air intake system. While all the jets were made inoperative, the main jet of the carburettor was replaced by a variable area opening jet for monitoring the flow of ethanol. Provision was also made to measure the temperature of the cylinder liner at 72 locations around the cylinders. In order to heat the intake air to facilitate the complete
evaporation of ethanol, an electrical air heater was also introduced suitably ahead of the carburettor.

The following investigations were carried out on the engine.

(a) Base line testing of the engine
(b) Dual fuel running with ambient air admission
(c) Dual fuel running with preheated air admission

The test matrix consisted of running the engine from 800 to 2200 RPM at intervals of 200 RPM with various loads. The mechanical governor incorporated in the fuel injection pump caused the appropriate reduction in the quantity of diesel injection as the quantum of ethanol induction was increased. The admission of ethanol was limited up to the point at which the engine developed occasional missing or hunting or knocking.

4.3 DUAL FUEL OPERATION OF A MULTICYLINDER TURBO-CHARGED COMPRESSION IGNITION ENGINE

Experimental investigations, to examine the suitability and adaptability of ethanol as a supplementary fuel in a turbocharged compression ignition engine, were carried out on a stationary four cylinder, direct injection engine. The engine was coupled to a hydraulic dynamometer and the set up was instrumented for the measurements of air flow, fuel flow, mixture temperature at inlet to the compressor and manifold, exhaust temperature, boost and back pressures. Induction of
ethanol was facilitated with a suitable carburettor having a variable area opening main jet and was installed ahead of the compressor and downstream of the air measuring device.

Cylinder pressure vs crank angle diagrams were observed and recorded with the help of a piezoelectric pressure transducer introduced in one of the cylinders, a crank degree marker fitted on to the flywheel and a dual beam oscilloscope. Provision was also made to measure the peak cylinder pressure with the help of a peak pressure indicator. The dynamic injection point, prerequisite for the determination of ignition delay, was indicated by a suitable sensor which was introduced in the high pressure diesel pipe.

In order to have a reference data for comparing the dual fuel operations, a thorough investigation of the engine performance on a baseline diesel fuel was first completed over a wide range of speeds and loads. For all the test runs, the mean temperature of the engine cooling water was kept at 75°C ±5°C and the oil temperature at 90°C. For dual fuel operation, the engine was started first on straight diesel and after the attainment of the steady state conditions, ethanol was admitted in increasing amounts until the engine started to miss or its running became knock limited. No attempt was made to optimize the injection timing and control the inlet air temperature and pressure. The schematic of the experimental setup and the general view of the engine with instrumentation are shown in Fig. 4.1 and Plate 4.1 respectively.
FIG. 4.1. SCHEMATIC DIAGRAM OF EXPERIMENTAL SETUP

TURBO CHARGED C.I. ENGINE

1. ORIFICE PLATE 7. PIEZO ELECTRIC PRESSURE PICK UP 12. STORAGE OSCILLOSCOPE
2. SURGE TANK 8. DYNAMIC INJECTION PROBE 13. HYDRAULIC DYNAMOMETER
3. CARBURETOR 9. CRANK DEGREE PICK UP 14. MANOMETER
4. DIESEL TANK 10. CHARGE AMPLIFIER 15. DIGITAL TACHOMETER
5. ETHANOL TANK 11. PRESSURE AND CRANK ANGLE SIGNALS TC. THERMO COUPLES
6. DIESEL PUMP 12. STORAGE OSCILLOSCOPE
SUPER IMPOSING CIRCUIT

FIG. 4.1. SCHEMATIC DIAGRAM OF EXPERIMENTAL SETUP
TURBO CHARGED C.I. ENGINE
4.4 STUDIES ON THE EFFECT OF COMBUSTION CHAMBER CONFIGURATIONS

The results of the earlier studies revealed that at higher percentages of diesel substitution, owing to the extension of delay, ignition starts well after the termination of diesel injection. Under such conditions of operation, it was believed that the organised movement of air to have only very little influence on the performance of the dual fuel engine. It was also thought that a reduction in the amount of air-ethanol mixture enveloping the pilot spray could minimize the rapid and violent heat release and result in a smooth operation and also permit a higher percentage of diesel substitution.

In order to draw some conclusions with regard to the influence of combustion chamber geometry on dual fuel combustion processes, studies were made on a single cylinder direct injection, four stroke, compression ignition engine of 80 mm bore and 110 mm stroke having a compression ratio of 16.5:1 with different combustion chamber configurations namely oblong, smaller hemispherical than the standard and toroidal chambers as shown in Fig.4.2. Performance studies were also made with a single hole injector to examine whether the engine could be operated smoothly with a single hole injector.
FIG. 4. PISTONS USED IN THE INVESTIGATION TO STUDY THE EFFECT OF COMBUSTION CHAMBER CONFIGURATIONS.
4.5 WEAR STUDIES IN DIESEL ENGINES OPERATING ON DUAL FUEL SYSTEMS

Fuel influences the lubrication behaviour of any internal combustion engine either by direct contact with the cylinder walls or by blowby into the crankcase. In dual fuel operation the possibilities of droplets of ethanol entering the combustion chamber are not ruled out. Unevaporated ethanol and the associated water reaching the crankcase is likely to form an emulsion with the lubricating oil, which could flash to vapour upon contact with hot surfaces leaving insufficient lubricant at certain locations. Besides the corrosive action of contaminated water in alcohol and products of combustion like acetaldehyde, formaldehyde, formic acid and the like may contribute to corrosion and wear of engine parts.

In order to critically examine the effect of simultaneous combustion of aqueous ethanol and diesel fuels on the wear and deposits on the engine components, 500 hours endurance test was conducted on two similar engines, one running on dual fuel and other on straight diesel.

The study was conducted on two new, single cylinder, four stroke, direct injection engines developing 3.7 kW at 1500 RPM. Both the engines were allowed to run for a short running-in period of 50 hours. After the running-in period the engines were dismantled and the cylinder, piston and rings were measured and the engines reassembled.
For the 500 hours durability test, both the engines were allowed to run on full load, one on straight diesel and the other on dual fuel at 60 percent of diesel substitution by 190 proof ethanol, for a duration of 10 hours each day. Consumptions of lubricating oil and fuels were recorded throughout the test at regular intervals. After the endurance test the critical dimensions were measured for quantifying the wear pattern of the components.

4.6 EXPERIMENTAL INVESTIGATION FOR CORRELATION STUDIES

The test engine employed for this investigation was a single cylinder, 4 stroke, water cooled and direct injection compression ignition engine, capable of developing 3.7 kW at 1500 RPM with a nominal compression ratio of 16.5:1. The piston had a hemispherical combustion chamber with a volume of 28.6 CC. The detailed specifications of the engine are given in Appendix 2. The fuel injection system consisted of a 3 hole closed type injector and a MICO plunger pump with a plunger dia of 8 mm operated by the camshaft and an injection timing adjuster capable of changing the injection timing within ±10° crank angle from the optimum value as recommended by the manufacturers. Plate 4.2 shows the general view and the details of the experimental setup used in these investigations. The engine speed, sensed by an inductive pickup through an interruptor wheel installed on to the extension of the camshaft was indicated by a digital panel meter. Lubricating oil temperature was measured by a
copper-constantan thermocouple. The schematic of the experimental setup is given in Fig. 4.3.

4.6.1 DYNAMOMETER

The test engine was directly coupled to a General Electric, cradle type electrical dynamometer with suitable switching and controls which facilitate its use either as a primemover for hot motoring tests or a power absorbing generator for loading the engine. The torque arm of the dynamometer is 0.5334 m. Output measurements were made from a spring balance having a resolution of 0.1 kg.

4.6.2 MODIFIED COOLING SYSTEM

In order to maintain the inlet and outlet temperatures of the engine cooling water at any desired level so as to eliminate the large variations of temperature difference between the inlet and outlet conditions, a closed-loop cooling system, incorporating a circulating pump, air to water heat exchanger and suitable by-pass arrangements, was installed. The inlet and outlet temperatures of the coolant were recorded by Copper-constantan thermocouples.

4.6.3 AIR INDUCTION AND FUMIGATION SYSTEMS

The air before reaching the carburettor was allowed to pass through a calibrated orifice plate of 12.75 mm diameter which was used to monitor the air consumption. A large surge tank, having a size approximately 150 times the swept volume
FIG. 4.3. SCHEMATIC OF THE EXPERIMENTAL SETUP
SINGLE CYLINDER C.I-ENGINE
of the engine was used for damping out the pressure waves of intake air flow. The pressure drop across the orifice plate was measured by a water column U-tube manometer. To the inlet system was connected a Zenith updraught carburettor, fitted with a manually operated variable opening main jet. The throttle was locked in the full open position and all the additional components such as choke and idling jet were made inoperative.

To assist the evaporation of ethanol, the carburettor was followed by a mixer heater of 5 cm diameter and 20 cm length with a centrally located 1500 Watt capacity CFR air heater and a short length of inlet manifold of 3 cm diameter and 18 cm length. In order to provide uniform heating to the air-ethanol mixture, another 1000 Watt band heater was also installed on to the exterior surface of the inlet manifold. The temperature of the mixture could be maintained at any desired level by controlling the voltage through two variable voltage transformers. The temperature of the mixture immediately after the carburettor and at the beginning of the inlet port of the engine were determined with the help of calibrated copper-constantan thermocouple probes, a selector switch and a digital millivolt meter.

4.6.4 FUEL SYSTEMS

The original injection system, supplied with the engine was retained for the admission of diesel fuel. Diesel and
ethanol fuels were supplied to the engine from the respective fuel tanks. The carburettor throttle was kept always fully open and the rate of flow of ethanol was controlled by varying the area of the main jet by means of a fine threaded needle valve. Fuel consumption rates were monitored by taking the average of four timing measurements of known volumes of fuels.

4.6.5 PRESSURE MEASUREMENT

The measurement of cylinder pressure was made using a KISTLER Quartz (piezo-electric) pressure transducer, model 601, having a range of 0-200 bar. The transducer, mounted on a suitable adaptor, was made to communicate with the combustion chamber through a 1.5 mm diameter and 45 mm long communicating passage, drilled in the cylinder head. The change in the value of the compression ratio of the engine as a result of the additional volume formed by the connecting passage was negligible. It was also estimated that an error of only 0.5° crank angle or less could result in the time base due to the propagation of pressure wave through the communication passage. The charge output of the transducer, which is proportional to the rate of pressure change with time, was fed to a KISTLER charge amplifier (Model 5007) to provide a voltage signal proportional to the cylinder pressure. This pressure signal was superimposed with the crank degree marker signals and displayed on a ECIL dual beam storage oscilloscope. Due to the limitations in the number of
channels in the oscilloscope, the superimposed pressure signal together with either the injector needle lift signal or the differentiated pressure signal was fed to the oscilloscope for the estimation of the duration of injection and the dynamic injection point or the point of ignition respectively. Using the storage feature of the oscilloscope typical cylinder pressure traces were photographed one at a time using a ROBOT camera.

4.6.6 MEASUREMENT OF DYNAMIC INJECTION POINT AND DURATION OF INJECTION

A proximity measuring device of a non-contact electromagnetic sensor, having a high frequency response, was used for the determination of the dynamic injection point and the duration of injection. The sensor was installed on top of the injector with the help of a suitable adaptor so as to maintain a small clearance between the top of the injector spindle and the sensing element. The burst of electric pulses produced by the pick up during the period of diesel injection as a result of the movement of the spindle were converted into a single trace by the signal shaper. The circuit diagram of the wave shaper and the wave forms at different stages of the processor are shown in Fig.4.4.

4.6.7 CRANK DEGREE MARKER (CDM)

For the measurement of cylinder pressure with respect to the crank position, a crank degree marker, an electromagnetic
NEEDLE LIFT SIGNAL SHAPER

FIG. 4.4. NEEDLE LIFT SIGNAL SHAPER
pick-up and a signal processing unit were used. The degree marker, consisting of a 12 mm thick perspex ring with 5° interval slots on the periphery to receive thin (0.8 mm) iron strips, was fitted concentrically with the flywheel. Electrical signals produced by the combined effect of the electromagnetic pick up and the iron strips were fed to the signal processor for suitable modifications and finally superimposed on the cylinder pressure trace. The signal processor consists of a number of stages such as pre-amplifier, comparator, differentiator, clipper and differential amplifier. The detailed circuit diagram and the wave forms corresponding to different stages of the processor are presented in Fig.4.5.

4.6.8 DIFFERENTIATOR

The ignition point was determined with the help of an electronic differentiator, the output of which was proportional to the rate of pressure change with respect to time (crank angle). This point, referred to as the point of inflection of the cylinder pressure trace, was clearly identified on the screen of the oscilloscope as a sharp rise from the differentiator output. The electronic circuit of the differentiator is shown in Fig.4.6. Owing to the objectionably high noise level, observed in the output of this unit, it was used only for the identification of the ignition point.
4.6.9 TRIGGER AMPLIFIER

In order to obtain a sharp, flicker free and stable signal of any desired portion of the cylinder pressure trace, the external trigger mode of the oscilloscope was used. The signal derived from the electromagnetic pick up which was suitably positioned with respect to a single pulse generator installed onto the extension of the camshaft, was fed to a suitable amplifier. The output of the amplifier was used to trigger the scope. The relevant circuit diagrams and the waveforms are shown in Fig.4.7.

4.6.10 EXHAUST EMISSION MEASUREMENT SYSTEM

In order to ensure a complete mixing of the exhaust so that a truly representative average exhaust gas was available for emission measurements, a mixing chamber with a volume of 10 times that of the cylinder displacement was introduced in the exhaust system ahead of the muffler and the exhaust sample probe. Exhaust gas temperature was measured by means of calibrated chromel-alumel thermocouple introduced 2 cms downstream of the cylinder head exhaust flange.

4.6.11 EMISSION MEASUREMENTS

For the determination of the concentration of gaseous pollutant species, representative exhaust sample was extracted continually from the downstream of the mixing chamber. Heated line Thermoelectron chemiluminescent analyser
FIG. 4.5. CRANK DEGREE MARKER - SIGNAL PROCESSOR AND WAVE FORMS AT DIFFERENT STAGES
FIG. 4.6. PRESSURE DIFFERENTIATOR

FIG. 4.7. TRIGGER AMPLIFIER

WAVE FORMS
model-10A was used for the measurement of NO/NOx emissions. In order to prevent any water condensation which may probably affect the performance of the analyser, the entire sampling line was kept at a temperature of 140 ± 5°C. For the measurement of HC and CO, an unheated portable HORIBA model MEXA 441-NDIR-analyser was used in the investigation and the unburned hydrocarbon was measured as equivalent to n-hexane. The significant features of the NO/NOx and HC and CO analysers are given in Appendix 3.

4.6.12 TEST FUEL DESCRIPTION

Commercial diesel fuel with cetane number ranging from 48 to 52 has been used in all the test runs. Ethanol fuel of different proofs were prepared by adding the required quantity of water with 190 proof ethanol which was received directly from the distillery. The fuel properties shown in Table 4.1 were made use of in computation.

TABLE 4.1 FUEL PROPERTIES

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Fuel</th>
<th>Density g/cm³</th>
<th>Latent heat kJ/kg</th>
<th>Lower heating value MJ/kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Diesel</td>
<td>0.835</td>
<td>-</td>
<td>44.380</td>
</tr>
<tr>
<td>2</td>
<td>190 proof ethanol</td>
<td>0.805</td>
<td>921</td>
<td>25.162</td>
</tr>
<tr>
<td>3</td>
<td>&quot;</td>
<td>0.830</td>
<td>1130</td>
<td>21.625</td>
</tr>
<tr>
<td>4</td>
<td>&quot;</td>
<td>0.854</td>
<td>1298</td>
<td>18.254</td>
</tr>
</tbody>
</table>
After the completion of the wear studies, the engine was reassembled after cleaning all deposits from the cylinder head, piston and valves. A number of trial runs were made to check, correct and calibrate all the measuring devices including the electronic circuits intended for superimposing the crank degree marks on the cylinder pressure trace and for determining the point of dynamic injection and duration of injection.

The static balancing of the dynamometer and the calibration of the spring balance were carried out at regular intervals. The cylinder pressure measuring system consisting of the quartz crystal pick up, charge amplifier, electronic circuit for superimposing the crank degree marks on the pressure trace and the oscilloscope were checked for accuracy by the application of the required intensity of pressure to the transducer with the aid of a pressure gauge calibrator and observing the output signal on the oscilloscope screen. Throughout the experimental investigation, it was observed that the accuracy of the transducer was within ±1.0 percent.

In order to provide a data base for the purpose of comparing dual fuel runs, a thorough investigation on the performance of the engine was completed on a base line diesel. During the entire investigation, the temperature of
the lubricating oil and the mean temperature of engine cooling water were held constant so as to eliminate their influence from biasing the observed results. Before allowing the ethanol fuel the engine was stabilized with injected fuel only as confirmed by the attainment of a lubricating oil temperature of 65°C ±5°C and mean cooling water temperature of 70°C ±5°C.

For the investigations of dual fuel operation at any combination of speed and load, the desired load was set on the dynamometer by running the engine with injected fuel only. Then the desired flow rate of ethanol was introduced by controlling the area of opening of the main jet. The increase in speed as a result of ethanol induction was controlled automatically by the action of the centrifugal governor. Any further adjustment necessary to maintain the speed within ±5 RPM of the originally set value was done manually by adjusting the fuel pump rack position.

After steady state conditions were reached, the following parameters were recorded.

(1) Consumption of inducted and injected fuels
(2) Exhaust temperature
(3) Peak combustion pressure
(4) Point of dynamic injection
(5) Ignition point
(6) Photographs of pr. and dp/dθ or pr. and duration of diesel injection
(7) NO/NOx emissions
(8) Temperature of air-ethanol mixture at salient points
(9) Inlet manifold and exhaust pressures.

For all the dual fuel runs, admission of ethanol was limited to the occurrence of misfiring or violent knocking as inferred from the visual observation of the pressure trace on the oscilloscope screen.

Experimental investigation carried out include:

I Hot motoring tests and
II Dual fuel tests under various conditions.

4.7 HOT MOTORING TESTS

The aim of this series of tests was to understand and establish the effect of aspirating aqueous ethanol along with the intake air on the development of compression temperature and pressure, both of which are considered to play a vital role in controlling the physical and chemical phenomena leading to the start of ignition and subsequent combustion. For hot motoring runs the engine was started on straight diesel and allowed to run for some time for warming up at any specified speed. The operation of the engine was changed to dual fuel mode and the flow of ethanol was adjusted to any desired value. Then quickly cutting off the diesel supply, the engine was motored at the same specified speed. The cylinder pressure trace on the oscilloscope was then...
photographed. This was repeated for various flow rates and proofs of ethanol and speeds.

4.8 DUAL FUEL TESTS

The following experimental runs were carried out with three different proofs of ethanol.

(i) Constant speed and variable load tests.

(ii) Tests with 5° crank angle advanced and 5° crank angle retarded injections of the injected fuel.

(iii) Tests with constant inlet temperatures and increased temperature of air-ethanol mixture.

(iv) Tests with varying the injection pressure.

The analysis of the data collected in these experimental investigations are presented in the next Chapter.