CHAPTER 4

MATERIALS AND METHODS

4.1 INTRODUCTION

All experiments performed during the investigation were conducted using a single cylinder DI diesel engine. A test rig was installed in the laboratory with the necessary instruments to measure performance, combustion and emissions characteristics of the engine at different operating conditions. Details of this engine, engine modifications to investigate the combined effects of combustion chamber geometries and injection parameters, description of engine fuel injection system components, the fuel used and its properties, the pollution measuring equipments and the other instruments used for recording various parameters like cylinder pressure, crank angle and TDC position, measurement of fuel consumed, exhaust gas temperature, air flow and cooling water flow etc., are given in this chapter.

4.2 EXPERIMENTAL SET-UP AND INSTRUMENTATION

The various components of experimental set up and engine instrumentation associated with the experimental set up are discussed below. Figure 4.1 shows schematic diagram of experimental set up and Figure 4.2 shows the photographic view of the experimental set up. The important components of the experimental set up were

1. Single cylinder, four stroke, vertical, water cooled, DI diesel engine
a) Fuel injection system

2. Eddy current dynamometer and controller

3. Data acquisition system
   a) Cylinder pressure measurement
   b) charge amplifier
   c) Analog to digital converter
   d) Crank angle encoder

4. Exhaust gas analyser (for HC, CO and CO₂ measurements)

5. NOₓ analyser (for NOₓ measurement)

6. Smoke meter

7. Fuel flow rate measurement

8. Temperature measurement

4.2.1 Test Engine

The engine chosen to carry out experimentation was a single cylinder, compression ignition, direct injection engine. It was a naturally aspirated, four stroke, water cooled, vertical engine. This engine can withstand higher pressures and is used extensively in agriculture and industrial sectors. The detailed specifications of the engine are provided in Appendix-1. The engine operates at a constant speed of 1500 rpm. The engine had a bowl-in-piston combustion chamber, particularly hemispherical shaped open combustion chamber. The engine also had a overhead valve arrangement. The valves are operated by push rods and a camshaft. The water required for the engine cooling was forced by a water pump through the water jacket. A provision for in-cylinder pressure measurement was made on the cylinder head to mount the piezoelectric transducer.
Figure 4.1 Schematic diagram of experimental setup

1. Engine
2. Fuel Injection pump
3. Fuel Injection Nozzle
4. Intake Manifold
5. Intake Air Surge Tank
6. Air Cleaner
7. U-Tube Manometer
8. Fuel Tank
9. Crank Angle Detector
10. Electric Dynamometer
11. Exhaust Manifold
12. Compression Pressure Transducer
13. Injector Needle Lift Sensor
14. Dynamometer Control Panel
15. Exhaust Gas Analyzer
16. Smoke Meter
17. Digital Scope Recorder
18. Exhaust Gas Temp Sensor
19. EGR System
20. Exhaust Pipe

Figure 4.2 Photographic view of the experimental set up
4.2.1.1 Fuel injection system

The engine was fitted with a conventional fuel injection system. The fuel injection pump which pressurized the fuel is shown in Figure 4.3. It was manufactured by MICO and comes with the engine as standard equipment. This was an inline pump used for small low speed diesel engines. The fuel injection pump is operated by the cam shaft and the fuel injection timing can be varied by adding/removing shims placed beneath the pump and pump mounting bracket. The fuel injection system also had an injector which had a three orifice of 0.24 mm separated at 120 degrees, inclined at an angle of 60 degrees to the cylinder axis. The fuel was injected into the centrally positioned open combustion chambers made in the piston crown. The position of injector nozzle and fuel spray penetration within the piston bowl is shown in Figure 4.4. The recommended injection timing by the manufacturer was 23° bTDC and the nozzle opening pressure was 200 bar.

4.2.2 Eddy Current Dynamometer and Controller

It consists of a stator supported on trunnion bearings so that any tendency of the stator to rotate is read on a fixed scale. Electromagnets are fitted to the stator and a rotor is coupled to the output shaft of the engine. When rotor rotates, eddy currents are produced in the stator due to magnetic flux set up by the passage of field current in the electromagnets and the induced flux tends to rotate the stator in the same direction as the shaft. These eddy currents oppose the rotor motion, thus loading the engine. These eddy currents are dissipated in producing heat and so this type of dynamometer needs cooling arrangement. The torque was measured by a moment arm and a load cell type strain gauge used in the dynamometer set up. Loading of the eddy current dynamometer was done by regulating the current in electromagnets. The detailed specifications of the dynamometer and its controller are given in Appendix-2. Figure 4.5 shows the photographic view of the eddy current dynamometer and its controller.
Figure 4.3 Photographic view of the fuel injection pump

Figure 4.4 Schematic diagram of position of injector nozzle and fuel spray within the piston bowl
4.2.3 Data Acquisition System

For acquiring in-cylinder pressure variations with respect to the crank angle, a high speed data acquisition system was employed. This was used for analyzing the measured cylinder pressure data. The components of the data acquisition system include pressure pick up, TDC encoder, charge amplifier, A/D card, combustion analyser and a personal computer. Figure 4.6 shows the photographic view of important components of data acquisition system. The cylinder pressure variation with respect to piston displacement in terms of pressure and crank angle was logged in to computer via data acquisition card and then to an excel work sheet. The large data which were collected during the experimentation have to be systematically processed. The cylinder pressure with respect to crank angle generally had large cycle to cycle variations and hence one such cycle data cannot be used to represent the particular operating condition. Usually an average of 100 cycles of pressure crank angle data were chosen for quantitative analysis. The number of cycles to be averaged depends upon the repeatability of the data. The detailed specifications of the data acquisition system are provided in Appendix-3.

4.2.3.1 Cylinder pressure measurement

In-cylinder pressure was measured with the help of Kistler piezo electric sensor, model 6613CQ09. This sensor was specially developed for the measurement of in-cylinder pressure in diesel engines. Cylinder pressure measurements can be made with high precision because of its very good thermodynamic characteristics. The good linearity and long term stability ensured reliable and repeatable measurements over a long period of time. Piezoelectric transducer is commonly preferred due to its small size, quick response and accuracy. The transducer used in engine testing needs to have very high natural frequency for its mechanical vibrations compared to the frequencies of pressure waves in the engine cylinder to avoid resonance. The
piezoelectric transducer produced a charge output, which was proportional to the in-cylinder pressure. The sensor was connected to the charge amplifier with a high temperature cable. The charge produced by the pressure transducer was converted to analog voltage reading by a charge amplifier. This charge output was then fed to combustion analyser to evaluate and determine cylinder combustion characteristics, and peak pressure balance.

Figure 4.5 Photographic views of the dynamometer and its controller

Figure 4.6 Photographic views of the components of data acquisition system
4.2.3.2  Charge amplifier

Charge amplifiers convert the charge output by a piezoelectric sensor into a proportional voltage, which was used as an input variable for analysis systems, and can be digitized in an analog-to-digital (A/D) converter. A charge amplifier basically consists of an inverting voltage amplifier with a high open-loop gain and capacitive negative feedback. The charge amplifier accepted a power supply between 7-32 V DC and had a range of 0-100 bar (40 mV/bar).

4.2.3.3  Analog to digital converter

Engine cylinder pressure and TDC signals were acquired and stored on a high speed computer based digital data acquisition system. A 12 bit analog to digital (A/D) converter was used to convert analog signals to digital forms. The A to D card had external and internal trigger facility with sixteen ended channels. During experiments, data from 100 consecutive cycles were recorded and signals were then passed to specially developed software to obtain the combustion parameters and also the heat release rate.

4.2.3.4  Crank angle encoder

The crank angle encoder is used wherever engine angle and flow information are needed to calculate the combustion parameters and the heat release rate of an engine. An inductive non-contact type Kistler make engine crank angle encoder, was used. It works based on a proximity switch. Inductive sensors comprise at least a permanent magnet and a coil. If a magnetically conductive mark carrier is moved in front of the sensor a voltage is induced in the coil. The output voltage of inductive sensors, depends very much on the speed. A steel screw and nut was attached to the engine flywheel and the proximity sensor detected its every revolution, thus a squared signal
waveform was emitted. Signals from the sensor were fed into the analog to digital converter and then to data acquisition system along with the pressure signals for recording.

### 4.2.4 Exhaust Gas Analyzer

The Crypton 290 analyser is a fully microprocessor controlled exhaust gas analyser employing Non Dispersive Infra Red (NDIR) techniques. The unit measured CO, CO$_2$ and HC. One more channel was provided employing electrochemical measurement. Exhaust gas analyser is shown in Figure 4.7. NDIR technique was used for the measurement for HC, CO and CO$_2$ emissions while for NO$_x$ and O$_2$ electrochemical measurement method was used in the exhaust analyzer. A three stage filtration system was employed to remove carbon particles and moisture from the exhaust gases. No regular emptying of the water trap was required as water was automatically expelled from the system during normal use. The filtration system comprised a disposable paper pre-filter (fitted in the sampling hose), a nylon mesh primary filter with water trap and a disposable gas filter. When the probe was inserted into the exhaust pipe of the engine, these filters prevent moisture, soot, dust and particles from entering the exhaust gas analyzer. After which the clean and cool sample gas entered the direct sensor measurement through filter arrangement and the readings were displayed on the screen and saved. The emission measurements were carried out on dry basis. The Crypton 290 exhaust gas analyzer specifications are given in Appendix-4.

### 4.2.5 NO$_x$ Analyser

Nitric oxide emission in the exhaust gas was measured with the SIGNAL heated vacuum chemiluminescent NO$_x$ analyser. The instrument works on the principle of reference method technique for monitoring NO$_x$ (combined NO and NO$_2$). This method for detection of NO$_x$ was based on
reaction of NO with ozone to produce nitrogen dioxide and oxygen. The NO$_2$ molecules from their electronically excited state revert to the ground state with the emission of photons as given in Equations (4.1) and (4.2). These photons were directly proportional to the NO$_x$ concentration.

\[
NO + O_2 \rightarrow NO_2 + O_2 \tag{4.1}
\]

\[
NO_2^* \rightarrow NO_2 + h\nu \tag{4.2}
\]

Where $h$ is plank’s constant and $\nu$ is the frequency (Hz).

The intensity of the light emitted was proportional to the mass flow-rate of nitric oxide into the reaction chamber and can be measured with high sensitivity photomultiplier tube. The advantages of this instrument are minimal quenching effects, and a heated reaction chamber allowing processing of hot, wet sample gas. The detection method was continuous with a fast response time. The instrument was fitted with a neon ozoniser lamp. This did not produce NO from ambient air as a by-product, which led to accurate readings. The detailed specifications of the SIGNAL heated vacuum chemiluminescent NO$_x$ analyser are given in Appendix-5. Figure 4.8 shows the photographic view of the NO$_x$ analyser used.

### 4.2.6 Smoke Meter

AVL 437 C smoke meter shown in Figure 4.9 was used to measure the smoke opacity of the test engine (in the measurement chamber of a defined measurement length). The smoke opacity is the extinction of light between light source and receiver. The engine exhaust gas was fed into a chamber with non-reflective inner surfaces. The effective length of the light absorption track was determined depending on the type of the light source and the photocell employed in the instrument. Light that scatters on the photocell
Figure 4.7 Photographic view of exhaust gas analyser

Figure 4.8 Photographic view of NO<sub>x</sub> analyser

Figure 4.9 Photographic view of smoke meter
from reflections or diffused light inside the chamber, was reduced to a minimum by the use of matt black light traps. The light source was an incandescent bulb with a color temperature between 2800°K to 3250 °K. The specifications of the smoke meter are given in Appendix-6.

4.2.7 Fuel Flow Rate Measurement

Fuel flow rate was measured on the volume basis using a burette and stopwatch. The fuel from the tank was sent to the diesel engine through a graduated burette using a two way valve. When the valve was set at 9’O clock position the fuel was sent to the engine directly from the tank and in 6’O clock position the fuel contained in the burette was sent to the engine. For the measurement of the fuel flow rate of the engine, the valve was set at 6’O clock position and the time for a definite quantity of the fuel flow was noted.

4.2.8 Temperature Measurement

Temperature of the exhaust was measured using chromel-alumel (k-Type) thermocouples. A digital indicator with automatic room temperature compensation facility was used.

4.3 TEST FUEL

To carry out the present investigation, six types of fuels were used: PBDF, a neat POME i.e. POME100 and blends of POME viz. POME10 (10% by volume of biodiesel obtained from Pongamia oil + 90% by volume of PBDF), POME20, POME30, POME40. Figure 4.10 shows the photographic view of the different fuels used in the experimental investigation. The fuel source and properties of the various fuels used for the experimentation are presented in this section.
4.3.1 Pongamia Biodiesel

In order to investigate the performance, emission and combustion characteristics of biodiesel fuel, biodiesel derived from Pongamia oil was selected. In India, usage of non-edible oils for the production of biodiesel is found to be most suitable when considering the deficit supply of edible oils and their cost of production. Among the non-edible oils, Tree Borne Oil seeds (TBOs) like Jatropha and Pongamia are important. Tranesterified raw Pongamia vegetable oil is called Pongamia biodiesel or Pongamia Oil Methyl Ester (POME). Transesterification is a chemical process of transforming large, branched, triglyceride molecules of vegetable oils and fats into smaller, straight chain molecules, almost similar to diesel fuel. The process takes place by the reaction of raw Pongamia oil with methyl alcohol in the presence of potassium hydroxide (KOH) catalyst.

4.3.1.1 Source of POME

The Pongamia biodiesel was produced from Pongamia seeds obtained from the tree Pongamia pinnata. Pongamia Pinnata, is one of the promising biofuel crop ideally suitable for growing in the wastelands. It is a
medium sized evergreen tree with a spreading crown and a short bole. Pongamia plant shown in Figure 4.11 is widely distributed in both dry and moist Indian plains, especially along watercourses. Pongamia pinnata belongs to the family of Fabaceae (or Leguminosae). Pongamia seeds contain 30-40% oil. The tree reaches its adult height within 4 to 5 years and yields 9 to 90 kg of seed. The Pongamia seeds are shown in figure 4.12. The yield potential per hectare is 900 to 9000 Kg/Hectare. Traditionally, Pongamia oil has been used to burn household oil lamps. It is one of the few nitrogen-fixing trees and its root, bark, leaves, sap, and flower also have medicinal properties and traditionally used as medicinal plant. It is a preferred species for controlling soil erosion and binding sand dunes because of its dense network of lateral roots. For the experimental investigation the POME was procured from Bannari Amman Sugar company’s biodiesel plant at Sathyamangalam, Tamilnadu.

Figure 4.11 Pongamia tree

Figure 4.12 Pongamia seed
4.3.1.2 Properties of POME

The physico-chemical properties of the raw Pongamia oil and POME were experimentally evaluated. The properties of raw Pongamia oil and POME100 are compared with the PBDF in Table 4.1. The properties of POME and blends of POME viz. POME10, POME20, POME30, POME40 are given in Table 4.2. Most of the properties of POME and its blends such as calorific value, viscosity, kinematic density, cetane number, flash point, cloud point and pour point were comparable with those of PBDF. The fuel properties of raw Pongamia oil indicate that the kinematic viscosity of oil was 37.84 cSt at 38°C, which is very high. The high viscosity of Pongamia oil is because of its high density (912 kg/m³). The flash point of Pongamia oil was very high (above 200°C).

The heating value of raw Pongamia oil was 39.15 MJ/kg when compared to PBDF (about 44.12 MJ/kg). Even though properties of POME were comparable with PBDF, the viscosity of POME100 (5.46 cSt) and POME20 (3.49 cSt) were found to be about 88.27% and 20.34% higher and calorific value were 11.26% and 2.3% lower respectively, when compared to PBDF.

The cetane number of POME and its blends was in the range of 50–57. Cetane number was similar or slightly higher than that of diesel fuel. Long chain saturated, unbranched hydrocarbons are especially suitable for conventional diesel fuel. The long, unbranched hydrocarbon chains in the fatty acids meet this requirement. The fatty acid composition of Pongamia oil biodiesel is given in Table 4.3. POME contains 8-20% saturated acids (palmitic, stearic and lignoceric) and 55-90% unsaturated acids (oleic and linoleic). The above unique properties of vegetable oils helped to replace the diesel fuel.
# Table 4.1 Properties of Raw Pongamia Oil, POME and PBDF

<table>
<thead>
<tr>
<th>Properties</th>
<th>Raw Pongamia Oil</th>
<th>POME 100</th>
<th>PBDF</th>
<th>IS: 15607 specification</th>
<th>Test methods IS1448 / ASTM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>912</td>
<td>898</td>
<td>850</td>
<td>860-890</td>
<td>P16</td>
</tr>
<tr>
<td>Kinematic Viscosity (cSt)</td>
<td>37.84</td>
<td>5.46</td>
<td>2.9</td>
<td>2.5-6.0</td>
<td>P 25/D 445</td>
</tr>
<tr>
<td>Calorific Value (MJ/kg)</td>
<td>34</td>
<td>39.15</td>
<td>44.12</td>
<td></td>
<td>D5865</td>
</tr>
<tr>
<td>Flash Pt (°C)</td>
<td>242</td>
<td>196</td>
<td>76</td>
<td>120</td>
<td>P21/D93</td>
</tr>
<tr>
<td>Cloud Pt (°C)</td>
<td>14.6</td>
<td>10.2</td>
<td>6.5</td>
<td>-</td>
<td>D2500</td>
</tr>
<tr>
<td>Pour Pt (°C)</td>
<td>-</td>
<td>4.2</td>
<td>3.1</td>
<td>-</td>
<td>D2500</td>
</tr>
<tr>
<td>Cetane No</td>
<td>46</td>
<td>57.9</td>
<td>49</td>
<td>51</td>
<td>P9/D613</td>
</tr>
<tr>
<td>Sulphur (mg/kg)</td>
<td>0.007</td>
<td>0.005</td>
<td>29</td>
<td>≤ 50</td>
<td>P83/D5453</td>
</tr>
<tr>
<td>Carbon residue, % mass</td>
<td>1.2</td>
<td>0.0035</td>
<td>0.1</td>
<td>≤ 0.05</td>
<td>ASTM 4530</td>
</tr>
<tr>
<td>Sulfated ash, % mass</td>
<td>0.014</td>
<td>0.002</td>
<td>0.001</td>
<td>≤ 0.02</td>
<td>P4/D874</td>
</tr>
<tr>
<td>Water content, mg/kg</td>
<td>-</td>
<td>340</td>
<td>52</td>
<td>≤ 500</td>
<td>P40/D2709</td>
</tr>
<tr>
<td>Acid value, mg KOH/g</td>
<td>5.06</td>
<td>0.42</td>
<td>0.10</td>
<td>≤ 0.5</td>
<td>P1/D664</td>
</tr>
<tr>
<td>Methanol, % mass</td>
<td>-</td>
<td>0.09</td>
<td>-</td>
<td>≤ 0.20</td>
<td>EN 14110</td>
</tr>
<tr>
<td>Ester content, % mass</td>
<td>-</td>
<td>98</td>
<td>-</td>
<td>≥ 96.5</td>
<td>EN 14103</td>
</tr>
<tr>
<td>Total glycerol, % mass</td>
<td>-</td>
<td>0.19</td>
<td>-</td>
<td>≤ 0.25</td>
<td>ASTM D6584</td>
</tr>
<tr>
<td>Iodine value, g I₂/100 gm.</td>
<td>96</td>
<td>86.5</td>
<td>38.3</td>
<td>≤ 120</td>
<td>EN 14104</td>
</tr>
<tr>
<td>Oxidation stability, at 110 °C, h</td>
<td>-</td>
<td>11.6</td>
<td>-</td>
<td>≥ 6</td>
<td>EN 14112</td>
</tr>
</tbody>
</table>
Table 4.2 Properties of POME and its blends

<table>
<thead>
<tr>
<th>Properties</th>
<th>POME100</th>
<th>POME10</th>
<th>POME20</th>
<th>POME30</th>
<th>POME40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>912</td>
<td>852</td>
<td>862</td>
<td>863</td>
<td>868</td>
</tr>
<tr>
<td>Kinematic Viscosity (cSt)</td>
<td>37.84</td>
<td>3.19</td>
<td>3.49</td>
<td>3.89</td>
<td>4.63</td>
</tr>
<tr>
<td>Calorific Value (MJ/kg)</td>
<td>34</td>
<td>43.623</td>
<td>43.126</td>
<td>42.629</td>
<td>41.864</td>
</tr>
<tr>
<td>Flash Pt (°C)</td>
<td>242</td>
<td>84</td>
<td>91</td>
<td>98</td>
<td>106</td>
</tr>
<tr>
<td>Cloud Pt (°C)</td>
<td>14.6</td>
<td>6.8</td>
<td>7.1</td>
<td>7.8</td>
<td>8.4</td>
</tr>
<tr>
<td>Pour Pt (°C)</td>
<td>-</td>
<td>3.3</td>
<td>3.6</td>
<td>3.7</td>
<td>3.9</td>
</tr>
<tr>
<td>Cetane No</td>
<td>46</td>
<td>50</td>
<td>51</td>
<td>52.4</td>
<td>53.7</td>
</tr>
<tr>
<td>Sulphur, mg/kg</td>
<td>0.007</td>
<td>26</td>
<td>21</td>
<td>19.5</td>
<td>15</td>
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</tbody>
</table>

Table 4.3 Fatty and unsaturated acids in POME

<table>
<thead>
<tr>
<th>Acid</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Palmitic acid</td>
<td>C16:0</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>C18:0</td>
</tr>
<tr>
<td>Lignoceric acid</td>
<td>C24:0</td>
</tr>
<tr>
<td>Oleic acid</td>
<td>C18:1</td>
</tr>
<tr>
<td>Linoleic acid</td>
<td>C18:2</td>
</tr>
</tbody>
</table>

4.3.2 Petroleum Based Diesel Fuel

PBDF was used as test fuel for comparison between POME and its blends with PBDF. The properties of conventional diesel fuel are shown in Table 4.4.
Table 4.4 Properties of PBDF

<table>
<thead>
<tr>
<th>Properties</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cetane number</td>
<td>49</td>
</tr>
<tr>
<td>Net calorific value (MJ/kg)</td>
<td>44.12</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>850</td>
</tr>
<tr>
<td>Kinematic viscosity (cSt)</td>
<td>2.9</td>
</tr>
<tr>
<td>Pour point (°C)</td>
<td>3.1</td>
</tr>
<tr>
<td>C (% mass)</td>
<td>87.3</td>
</tr>
<tr>
<td>H (% mass)</td>
<td>12.5</td>
</tr>
<tr>
<td>O (% mass)</td>
<td>0</td>
</tr>
<tr>
<td>Sulphur (mg/kg)</td>
<td>29</td>
</tr>
<tr>
<td>Water (% mass)</td>
<td>-</td>
</tr>
<tr>
<td>Potassium (mg/kg)</td>
<td>-</td>
</tr>
<tr>
<td>50% Distillation temperature (°C)</td>
<td>278</td>
</tr>
<tr>
<td>Stoichiometric air-fuel ratio</td>
<td>14.2</td>
</tr>
</tbody>
</table>

4.4 ENGINE MODIFICATIONS

To achieve improved performance and further reductions in emissions from biodiesel operated diesel engine, rapid and better air-biodiesel mixing is the most important requirement. To achieve these, investigations on any modifications of engine design, particularly combustion chamber may be required, because mixture formation within the engine cylinder mainly depends upon the shape of the combustion chamber in a DI diesel engine. Development of biodiesel fuelled direct-injection diesel engine also requires modifications to fuel injection system characteristic as the biodiesel fuel properties particularly viscosity, higher bulk modulus of compressibility and density are different from PBDF. Fuel injection system and injection
characteristics strongly influence the combustion of the fuel inside the cylinder, which in turn influences the engine performance and emissions.

4.4.1 Combustion Chamber Modifications

Mixture formation within the engine cylinder mainly depends upon shape of the combustion chamber in a DI diesel engine. In DI diesel engines a single combustion chamber with different piston bowl shapes such as cylindrical, square, hemispherical, shallow depth, toroidal etc. have been used. In the present investigation, to investigate the effects of combustion chamber geometry on performance, combustion and emission characteristics of biodiesel fuelled DI diesel engine the piston bowl geometry was modified to have Shallow depth Combustion Chamber (SCC), Toroidal Combustion Chamber (TCC), Shallow depth Re-entrant Combustion Chamber (SRCC) and Toroidal Re-entrant Combustion Chamber (TRCC) from the baseline Hemispherical Combustion Chamber (HCC). For all the combustion chamber configurations, bowl volume was kept constant so that compression ratio was the same for all the chambers. The photographic view of pistons having the five shapes of combustion chambers employed for this study is shown in Figure 4.13.

In SCC the depth of the cavity provided in the piston is quite small. Since the cavity diameter is large it gives small squish and swirl motion. HCC also gives small squish. However the depth to diameter ratio can be varied to give any desired squish to give better performance. But the TCC provides a powerful squish along with the air movement similar to that of familiar smoke ring (Ganesan 2005). Squish and swirl air motion induced by the combustion chamber enables fast evaporation of highly atomized fuel and fast mixing of fuel vapors with air ensures complete combustion. Numerous test results indicate that, for DI diesel engines, a reentrant combustion chamber shape in which the lip of the combustion chamber protrudes beyond the walls of the
bowl provides a substantial improvement in performance and emissions over the previous open straight sided bowl designs (Asif et al 1996). Researchers at AVL List (Austria) found that a reentrant bowl gave a 20% reduction on PM emissions compared to those measured from a straight sided bowl at the same compression ratio (Michael and Jitendra 1997).

Figure 4.13 Photographic view of combustion chambers employed

Saito et al (1986) investigated the effect of the combustion geometry on combustion with special emphasis focused on the re-entrant combustion chamber using PBDF. They compared the conventional combustion chambers and the reentrant in terms of combustion process, engine performance, NO\textsubscript{x} and smoke emissions. They found that the reentrant chamber reduces ignition lag and provides better fuel economy with delayed injection timing, which is attributed to the effect produced by the hotter surface of the reentrant chamber. Also, combustion is enhanced with reduced smoke emission due to higher velocities induced around TDC. Montajir et al (2000) have studied the effects of combustion chamber geometry, the shape of
the cavity entrance and bottom corner radius to investigate their effects on the spray development in the chamber using a common rail injection system. They found that the reentrant cavity with round lip produces larger spray volumes and wider spray spreading. They also observed that the fuel impingement just on the lip corner produces the maximum spreading area. They concluded that bottom corner radius helps to disperse the fuel accumulated at the bottom corner and the spray volume increases.

4.4.2 Injection System Modifications

The injection timing of the MICO jerk type pump was varied by changing the number of shims under the pump body. The standard engine was fitted with three shims to give standard injection timing of 23° before Top Dead Center (bTDC). By changing the number of shims, the injection timings were varied to 20°, 21°, 22° and 24° bTDC. The photographic view of shims that have been used to vary the injection timing for this study is shown in Figure 4.14. The injector opening pressure was varied by adjusting the spring tension of the injector by screwing or unscrewing the screw provided on the top of the injector. The fuel injection pressure was measured by hand operated fuel injection calibration test bench shown in Figure 4.15. By varying the spring tension in the fuel injector, the nozzle opening pressure could be varied. Then experiments were carried out at different injection opening pressures of 185, 200, 210, 220, and 230 bar.

4.5 Research Methodology

In the present work, the research has been carried out in two phases. The research involves a combined approach of

1. Numerical simulation
2. Experimental investigation
4.5.1 Numerical Simulation

The aim of the numerical simulation was to investigate the effect of bowl geometry on flow characteristics inside the cylinder of a DI diesel engine under transient conditions. Three open combustion chambers namely, HCC, SCC and TCC and two re-entrant bowls namely SRCC and TRCC were

![Photographic view of shims used to vary the injection timing](image1)

Figure 4.14 Photographic view of shims used to vary the injection timing

![Photographic view of fuel injection calibration test bench](image2)

Figure 4.15 Photographic view of fuel injection calibration test bench
considered. The simulation was carried out at non reacting condition using commercially available CFD software. The methodology used was as follows

1. Modeling of flow domain for five different piston cavities using pre-processor tool.
2. Generating hexahedral mesh for the model.
3. Importing the model to CFD software and solving the computational model to obtain the results.
4. Computing various flow parameters such as swirl velocity, swirl ratio, and turbulent kinetic energy for all five models.

Simulation results obtained for these five different combustion chamber geometries are presented and discussed in Chapter 6.

4.5.2 Experimental Investigation

For achieving the objectives of this research work, the following experimental methodology was adopted in the present work.

1. The selected biodiesel i.e. Pongamia Oil Methyl Ester (POME) was procured and its properties like kinematic viscosity, flash and fire points, calorific value, carbon residue etc., were evaluated.

2. In the first phase of the experiment the performance, emission and combustion tests were conducted on baseline engine having HCC using PBDF, POME and its blends as fuels at fuel injection pressure of 200 bar and fuel injection timing of 23° bTDC which were set by the manufacturer. The test results of PBDF were used as baseline values throughout the research for comparison of the experimental results obtained
from POME and its blends. From these tests the best biodiesel blended fuel was shortlisted based on performance, combustion and emission characteristics.

3. In the second phase, tests were carried out on modified engine having SCC, TCC, SRCC and TRCC with the above shortlisted biodiesel fuel as fuel and PBDF, at fuel injection pressure of 200 bar and fuel injection timing of 23° bTDC. From these experiments the best combustion chamber was selected in terms of performance, combustion and emission characteristics.

4. Subsequently the influence of varying the injection timing was studied. Tests were conducted on modified engine having optimized combustion chamber with the shortlisted biodiesel fuel at fuel injection pressure of 200 bar and at different injection timings viz. 20° bTDC, 21° bTDC, 22° bTDC, 24° bTDC. These results were compared and analyzed with standard injection timing of 23° bTDC. From the results the best injection timing was determined for biodiesel operation.

5. Finally the effect of the injection pressure was investigated. Tests were conducted on modified engine having optimized combustion chamber at optimized fuel injection timing. The experiments were carried out at different injection opening pressures of 185, 210, 220, and 230 bar and their results were compared and analyzed with standard injection pressure of 200 bar. From this investigation the optimum injection pressure was decided for biodiesel operation.

6. The engine tests were carried out at 0%, 25%, 50%, 75% and 100% load.
7. In order to have meaningful comparisons of emissions and engine performance, investigations were carried out at the same operating conditions. Engine speed, torque, air-fuel ratio and peak conditions were maintained for all test fuels and respective engine loads. In addition, for the experimental investigations different equipment and instruments were used for measurement of different parameters. These instruments and equipments are made by different manufacturers using different technologies. The accuracy of measurement and their performance may vary depending on the operating conditions and experimental environment. Hence the uncertainty occurs due to fixed or random errors. The uncertainties in the measured parameters were estimated based on analytical methods. The uncertainties computed for the measured quantities are given in Appendix-7.