CHAPTER 4

RESEARCH METHODOLOGY

4.1 MODELLING

4.1.1 General Procedure

The change in cylinder volume and the rate of combustion have considerable influence on the cylinder pressure changes during the operation of engine cycle. Therefore, the analysis of pressure data can give the combustion rate information. The cylinder pressure versus crank angle data over the compression and expansion strokes of the engine cycle is obtained from the experiment.

Combustion in an actual CI engine is not instantaneous. The combustion event happens after the injected fuel is evaporated and locally well mixed with surrounding air. Then the combustion is considered to be kinetically controlled. The small amount of fuel that is burned immediately after ignition is premixed and mainly kinetically controlled. In order to achieve, or to approximate, constant pressure during combustion process, fuel must be injected or burned over a finite span of crankshaft rotation. Hence, the diesel combustion is considered progressive. The essential idea underlying the analysis of progressive combustion is expressed in the equation.

\[ \Delta p = \Delta p_p + \Delta p_c \]  

(4.1)
\[ \Delta p = -p k \frac{\Delta V}{V} + (p_3 - p_2) \frac{V_{\text{dc}}}{V} \Delta n \] (4.2)

where \( p \) is the pressure at the beginning of compression stroke
\( k \) is ratio of specific heats (\( k = C_p/C_v \)),
\( \Delta n \) is the mass fraction of fuel burned,
\( \Delta V \) is the volume change due to piston movement,
\( p_2 \) is the pressure at the end of compression stroke (at TDC),
\( p_3 \) is the pressure that would be reached if all the fuel is burned instantaneously at TDC (i.e. at constant volume).

The first term of equation (4.2) represents the pressure change due to piston movement and the second term is due to combustion. The end of combustion is fixed by the amount of fuel injected, which can be varied over a wide range of limits. The upper limit is set by the amount of smoke that can be tolerated in the engine exhaust. The appearance of smoke suggests the presence of unburned carbon, indicating that the engine is operating at the limit of maximum burning rate or at the limit of available oxygen, or both (Ganesan 1996).

The governing equations are the non-flow energy equation together with the semi-perfect gas state equation i.e. Non-flow energy equation:
\[
\sum_{i=1}^{n_i} C_{vi} Td\alpha_i + \sum_{i=1}^{n_i} R_0 \left( \Delta n_i \right) + \sum_{i=1}^{n_i} R_0 \Delta T \Delta n_i + \Delta Q_L + p\Delta V - dm_f CALVAL \tag{4.3}
\]

The first two terms in the above equation represent the change in internal energy of the cylinder charge, \(dQ_L\) the incremental heat loss from the cylinder, \(p\Delta V\) the piston work and \(dm_f CALVAL\) the fuel heat release.

Equation of state in differential form:

\[
pdv + Vdp = \sum_{i=1}^{n_i} R_0 \left( \Delta n_i \right) + \sum_{i=1}^{n_i} R_0 \Delta T \tag{4.4}
\]

These equation may be solved, using appropriate numerical procedures, for the temperature and pressure increments \(dT\) and \(dp\), for any crank angle step \(d\theta\), i.e. time step, \(d\theta = \Delta\theta/6N_E\)

4.1.2 Compression and Expansion Processes

For compression process, assuming trapped conditions at inlet closure, \(p_i, T_i\), together with an assumed degree of charge purity determining the composition of the trapped charge, equations (4.3) and (4.4) are solved step-by-step, in suitably simplified form. In equation (4.3), the heat release term \(dm_f\) CALVAL is omitted while in both equations (4.3) and (4.4) the terms representing change of composition, i.e. \(\sum_{i=1}^{n_i} C_{vi} Td\alpha_i\) and \(\sum_{i=1}^{n_i} R_0 T \Delta n_i\) may be omitted.

For expansion process the heat release from fuel and heat transfer to walls are iterated up to the exhaust valve-opening period.

4.1.3 Combustion Process

4.1.3.1 Adiabatic Flame Temperature

Assuming complete combustion, we can decide the composition of the combustion products merely by inspecting the reactant mixture. For a
reactant mixture that contains C, H, O, N atoms we will formulate the rules as follows: (Ganesan V, 1996)

i. All hydrogen is first oxidized to H₂O

ii. All carbon is initially oxidized to CO

iii. If any oxygen remains, part of the CO is oxidized to CO₂

iv. If there is sufficient oxygen to oxidize all the carbon to CO₂, the excess appears as O₂

v. All nitrogen appears as N₂, i.e. nitrogen is assumed to be inert

Let, \( N_{mC} = \) number of moles of carbon

\( N_{mH2} = \) number of moles of hydrogen

\( N_{mO2} = \) number of moles of oxygen

Let, \( Y_{cc} \) denotes the chemically correct amount of oxygen per mole of fuel required for combustion.

\[
Y_{cc} = N_{mC} + 0.25 N_{mH2} - 0.5 N_{mO2} \quad (4.5)
\]

Let \( Y_{min} \) demote the minimum amount of oxygen required in the reactants per mole of fuel, i.e., the minimum amount of oxygen required to convert H₂ to H₂O and C to CO

\[
Y_{min} = 0.5(N_{mC} - N_{mO2}) + 0.25 N_{mH2}
\]

\[
= Y_{cc} - 0.5 N_{mC} \quad (4.6)
\]

Hence, for a reactant mixture containing

1 mole of fuel + \( YO_2 \) + \( 3.76N_2 \) \quad (4.7)
The product mole number can be written depending on the value of Y to be considered as shown below:

\[
\begin{align*}
Y_{\text{min}} & \leq Y \leq Y_{cc} & Y & \geq Y_{cc} \\
\text{CO} & : \ N_1 = 2(Y_{cc} - Y) & N_1 & = 0 \\
\text{CO}_2 & : \ N_2 = 2(Y - Y_{\text{min}}) & N_2 & = N_{mC} \\
\text{H}_2\text{O} & : \ N_3 = 0.5 \ N_{mH_2} & N_3 & = 0.5 \ N_{mH_2} \\
\text{N}_2 & : \ N_4 = 3.76 \ Y & N_4 & = 3.76 \ Y \\
\text{O}_2 & : \ N_5 = 0 & N_5 & = Y - Y_{cc}
\end{align*}
\]

Adiabatic flame temperature is the maximum temperature attained when one k mole of fuel is completely burned in chemically correct amount of oxygen.

\[
Q_p = \sum N_i \Delta h_i
\]
\[
\Delta h_i = h_i(T) - h_i(298) \quad (4.8)
\]

Substituting the value of enthalpy in the above equation, we get

\[
\sum [N_i \ h_i(T) - N_i \ h_i(298)] - Q_p = 0 \quad (4.9)
\]

From the above equation, adiabatic temperature is calculated by using Newton-Raphson iterative scheme.

The algebraic expression for enthalpy \( h_i \ (T) \) is given as follows:

For \( 400 \leq T \leq 1600 \)

\[ h(T) = AL + BL \ast T + CL \ln(T) \quad (4.10) \]

For \( 1600 \leq T \leq 6000 \)

\[ h(T) = AH + BH \ast T + CH \ln(T) \quad (4.11) \]
The heat capacity \( C_p \) (T) is given as follows:

For \( 400 \leq T \leq 1600 \)
\[
C_p(T) = BL + CL / T \tag{4.12}
\]

For \( 1600 \leq T \leq 6000 \)
\[
C_p(T) = BH + CH / T \tag{4.13}
\]

In order to calculate the flame temperature, the Newton-Raphson iterative scheme could be conveniently employed. For this, the method is to start with a hypothetical value of the flame temperature. Then, the new flame temperature is obtained from the following equation.

\[
T_{\text{new}} = T - Hp(T) - Hp(298) - Qp / C_p(T) \tag{4.14}
\]

If the difference between \( T \) and \( T_{\text{new}} \) is within, say 5 degrees, the iteration can be stopped. Otherwise, the new temperature value \( T_{\text{new}} \) is used in next iteration

The values of the coefficients \( AL, BL, AH, BH, H_{rp} \) and \( U_{rp} \) values at 298 K for various gases are listed in Appendix 1.

4.1.3.2 Combustion Modelling

The simplest single-zone combustion model specifies the heat release pattern in advance so that the cycle calculation merely involves adding energy to the cylinder contents in accordance with this pattern at the appropriate points in the calculations. This approach considers the cylinder contents to be a homogeneous mixture of air and gases, which are always in thermodynamic equilibrium and dealt as ideal gases. Together with this assumption, in the present work, dissociation is neglected and only four gases
are considered; oxygen, nitrogen, water vapor and carbon dioxide (Ganesan V, 1996) as complete combustion is assumed. However, disassociation losses are compensated in computing the values of specific heats, where temperature is accounted.

For the purposes of analysis as used in cycle simulation, equations (4.3) and (4.4) have to be used in their full form. Before proceeding to solve them for the pressure and temperature increments dp and dT, it is necessary to specify the fuel burning rate (FBR) from which can be derived the burnt fuel increment dm fb for any calculation step involving a crank angle increment dθ during the combustion period. It is convenient to express the fraction of fuel burnt, FB, up to any crank angle θ from start of combustion (SOC) in terms of a dimensionless crank angle which is defined as

\[ y = \frac{\text{crank angle } \theta \text{ from start of combustion}}{\text{total crank angle occupied by combustion } \theta_{\text{comb}}} \]

### 4.1.3.3 Wiebe’s Combustion Function

Wiebe specifies a function to represent the combustion curve, for the cumulative fuel burnt, as a fraction of the total fuel injected

\[ x = 1 - \exp(-a.y^{m+1}) \]  

or in differential form (rate of fuel burning)

\[ \frac{dx}{dy} = a.(m+1).y^m \exp(-a.y^{m+1}) \]

The variable ‘x’ represents the fraction of the mass of fuel burnt relative to the fuel injected, the variable ‘y’ represents the time relative to the combustion. The parameter ‘a’ and ‘m’ characterize the shape of combustion
curve. Therefore, the heat release pattern is defined by the total amount of fuel injected, the start of combustion, the combustion duration and the shape parameters of Wiebe’s function. The start of combustion may be defined by the start of fuel delivery, the injection delay and the ignition delay. The start of fuel delivery and the injection delay is known from the geometric data of injection system. However, the ignition delay is related to the cylinder gas temperature, pressure and fuel properties, so it is considered for calculation according to a semi empirical expression. The combustion duration is an arbitrary period in which combustion must be completed. The actual point at which combustion ceases has little real significance, since the combustion rate decays exponentially to almost zero long before combustion truly stops.

Parameter ‘a’ in the Wiebe’s function can be considered as a combustion efficiency term, since the fraction of the cumulative fuel burnt to the total fuel injected is only dependent upon the parameter ‘a’. Therefore, the value ‘a’ is chosen such that all the fuel is burnt at the end of combustion (for example, $x = 0.999$ at $y = 1$ if $a = 6.9$). Parameter ‘m’ in the function characterizes the rate of combustion. The small values for ‘m’ means a high rate at the beginning of combustion, while large values of ‘m’ mean a high rate by the end of combustion.

**Table 4.1 Calculated values of ‘m’ for diesel and blends**

<table>
<thead>
<tr>
<th>Blending level</th>
<th>3.0%</th>
<th>6.0%</th>
<th>9.0%</th>
</tr>
</thead>
<tbody>
<tr>
<td>2EE</td>
<td>0.506</td>
<td>0.493</td>
<td>0.479</td>
</tr>
<tr>
<td>DEC</td>
<td>0.508</td>
<td>0.495</td>
<td>0.482</td>
</tr>
<tr>
<td>DEE</td>
<td>0.521</td>
<td>0.511</td>
<td>0.501</td>
</tr>
</tbody>
</table>
The value of ‘m’ is chosen for oxygenated diesel based on cetane number and oxygen content. The calculated values of ‘m’ for diesel and blends are shown in Table 4.1.

4.1.3.4 Ignition Delay

Ignition delay time can be calculated as the difference between the time at which combustion starts, $t_{\text{ign}}$, and the time at which injection starts, $t_{\text{inj}}$. In general, ignition delay is a complicated function of mixture temperature, pressure, equivalence ratio, and fuel properties. For the purposes of diesel combustion simulations, ignition delay is correlated by

$$\tau_{\text{id}} = A p^{-n} \exp \left( \frac{E_A}{RT} \right)$$

(4.17)

Arrhenius equation (Heywood 2000), is used, where $p$ is gas pressure, $T$ is gas temperature, $E_A$ is the apparent activation energy for the fuel auto ignition process, $R$ is the universal gas constant, and ‘$A$’ and ‘$n$’ are constants dependent on the fuel properties.

According to Wolfer, the constants are $n = 1.19$, $A = 0.00044$, $E_A/R=4650$. This expression is adopted here. In an engine, pressure and temperature change considerably during the ignition delay period due to the compression resulting from piston motion. To account for these changing conditions, $t_{\text{ign}}$ can be obtained by integrating the reciprocals of instantaneous estimates of the ignition delay, based on above equation, until the following relationship is satisfied.

$$t_{\text{id}} \int_{tsi}^{t_{\text{id}}} \frac{1}{\tau} \, dt = 1$$

(4.18)

The flow chart for the program is shown in the Figure 4.1. The computer program for simulation is furnished in Appendix 3.
The basic idea behind developing this model is to test the oxygenates for higher blending levels. Higher blending levels of oxygenates has many limitations such as reduction of heating value, loss of lubricity, corrosion of engine components etc. Despite such limitations oxygenates affect highly the smoke and particulate emissions. They also affect HC, CO and NO\textsubscript{X} emissions beside some improvement in brake thermal efficiency. Before actually testing the oxygenates in an engine, simulation model shall be used for evaluating the oxygenates. The above model is a zero dimension and single zone model and uses Wiebe’s equation, which assumes complete combustion (Tree 2001). For accurate results two zone (burned and unburned
zones) with three dimensional modeling shall be tried (Xianghui Meng et al 2004). This calls for CFD analysis and extensive research, which shall be taken up by research scholars in future.

4.2 EXPERIMENTAL INVESTIGATION

4.2.1 General

The CI engine inducts a constant amount of air on the intake stroke. At part load a small amount of fuel is injected into the engine cylinder, will not need all the air in the cylinder. As the load is increased, greater amount of fuel is injected and more and more amount of air is required for combustion. At some stage, a part of the injected fuel is not fully oxidized and leads to production of smoke. At this condition, part of the air in the engine may not react because of the failure of the injected fuel to find the air. Thus with the oxygenated fuel it is easier for the injected fuel to burn completely with the oxygen readily available within the fuel (SAE 1972).

The aim of this research is to carry out a simulation study and experimental investigation of the technical facts of the conclusions of literature survey, to make a comparative analysis of the effects of the three selected oxygenates, 2EE, DEC and DEE in combustion, performance and emission and select the suitable oxygenate for blending with diesel.

4.2.2 Preparatory Work

The experimental investigation is carried out in different stages. Firstly, an extensive literature survey of the diesel additives is carried out. Based on the physical and chemical properties suitable oxygenates are selected. The selected oxygenate, (from oxygenates 2EE, DEC and DEE), is then blended with diesel in different proportions. The properties of these blends are evaluated and experimented for effective implementation and
usage in diesel engines. Simulation model is developed based on the properties of oxygenated diesel and Wiebe’s combustion equation for mass fraction of fuel burned. The simulated and experimental pressure values are analysed for validation of the model and compared with standard values obtained from the literature. Experiments are then conducted to study and evaluate the effects of these oxygenates on combustion, performance and emissions. Lastly, the evaluated results are compared and analysed with the values obtained for diesel from standard technical papers, to ensure whether the values are satisfactory in the acceptable level.

4.2.3 Details of Investigation

It is observed from the literature survey that, 1 to 8 percent of oxygen, by weight, is required to be blended with diesel to have maximum emission reduction. Based on the above fact and the properties of diesel and oxygenates, the three oxygenates 2EE, DEC and DEE are blended with diesel fuel in the ratios of 3.0, 6.0 and 9.0 percent by weight to have oxygen content varying between 0.65% and 4.14%. The percentage of oxygen in the blends by weight for these blending levels is shown in Table 3.2. The corresponding volumes of the oxygenates to be blended with diesel were calculated and blended to have the above mentioned ratio of blending and oxygen content as shown in Table 3.2.

4.2.4 Experimental Set up

Figure 4.2 shows a schematic layout sketch of the test set up. Figures 4.3 to 4.4 show the measurement sensors and their locations. A single cylinder, four stroke, direct injection, water cooled diesel engine was used for the experiments. The details of the engine are furnished in Table 4.2.
Figure 4.2 Schematic Lay out of the Experimental Set up
1. Engine        2. Flywheel
3. Dynamometer   4. Data acquisition system
5. Control unit  6. Fuel flow meter

Figure 4.3 Photographic view of the experimental set up
1. Pressure sensor
2. Water inlet temperature sensor
3. Magnetic filter
4. Dynamometer water inlet piping
5. Exhaust temperature sensor
6. Speed sensor
7. Dynamometer water outlet piping
8. Water outlet temperature sensor

Figure 4.4 Photographic view of the measurement sensors
Table 4.2 Details of the diesel engine

<table>
<thead>
<tr>
<th>Make and Model</th>
<th>Kirloskar DM 10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore and stroke</td>
<td>102 × 116 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.5:1</td>
</tr>
<tr>
<td>Cubic capacity</td>
<td>0.948 Lits</td>
</tr>
<tr>
<td>Rated speed</td>
<td>1500 rpm</td>
</tr>
<tr>
<td>Rated power</td>
<td>7.5 kW @ 1500 rpm</td>
</tr>
<tr>
<td>Injection Timing</td>
<td>26° BTDC</td>
</tr>
<tr>
<td>Injector opening pressure</td>
<td>200-205 bar</td>
</tr>
<tr>
<td>Valve timing</td>
<td></td>
</tr>
<tr>
<td>Inlet valve opens BTDC</td>
<td>4.5 Deg.</td>
</tr>
<tr>
<td>Inlet valve closes ABDC</td>
<td>35.5 Deg.</td>
</tr>
<tr>
<td>Exhaust valve opens BBDC</td>
<td>35.5 Deg.</td>
</tr>
<tr>
<td>Exhaust valve closes ATDC</td>
<td>4.5 Deg.</td>
</tr>
<tr>
<td>Valve clearance (In cold condition)</td>
<td></td>
</tr>
<tr>
<td>Inlet</td>
<td>0.18 mm</td>
</tr>
<tr>
<td>Exhaust</td>
<td>0.2 mm</td>
</tr>
</tbody>
</table>

4.2.5 Test Instruments for Measurements

The instruments used in the test set up were, electronic flow meter for fuel consumption, air flow measurement system, eddy current dynamometer to apply the needed brake torque, pressure transducer (‘Kistler’, Model No. 701A) for pressure measurement and Diesel tune Smoke meter (Bosch principle) to measure the smoke emission.
The emissions such as NO\textsubscript{x}, smoke, HC and PM are directly obtained from Diesel tune Smoke meter DX.230. Software version GB 1.0X is used for the required results.

4.2.5.1 Instruments for Performance Assessment

The ever-increasing output of diesel engines together with rapid advances in engine design and construction has demanded corresponding improvements in performance testing and quality control techniques. The dynamometer design is based on the philosophy of determining a design factor, and applying this constant to such parameters as power, torque, speed, moment of inertia and basic dimensions, yielding a standardized system of machine sizing to be used.

The eddy current dynamometer is a dry gap rotor machine, which operates in air gap and is inherently capable of bi-directional operation. The construction comprises of a thin spoke steel rotor rotating in a casing, which houses the field coil and the cooling system.

The excitation current generated in the coil produces a magnetic flux field. An equal and opposite magnetic field is induced in the walls cooling system facing the rotor and this field is a pulsating form corresponding to the frequency of the passing rotor spokes. This changing field produces eddy currents in the cooling system walls, which oppose the excitation field, thus causing retardation action to the rotor rotation. Water passing through the dynamometer cooling chambers of the cooling system jackets is for heat dissipation only. Hence, load control is entirely a function of the level of excitation.
The fluctuations and pressure of water flow have no effect whatsoever on the stability of load absorption. Funnel type arrangement is provided to collect the water coming out of dynamometer outlet pipe before it is sent back to water sump outside the test cell. This is to avoid any effect on the indication of load/torque due to the backpressure of cooling water (generated because of the blockage of water flow in piping or any reduction of flow of water coming out of dynamometer due to blockage of dynamometer cooling water passage).

Dynamometer controller is the basic instrument for testing the engine with the help of an eddy current dynamometer. The dynamometer controller supplies the direct current necessary to excite the dynamometer. By loading the dynamometer electrically, the control achieved is very fine and reliable at different load and speed of the engine.

4.2.5.1.1 Measurement of Fuel Flow

The fuel consumed during the testing of the engine is measured by an electronic fuel flow meter. Using the selector switch on the control panel of the dynamometer, it can be set to record 100, 200, or 300 cc of the fuel consumed according to the requirements.

By turning the toggle switch to ‘Auto’ position, the timer is set to the auto mode. The timer starts and stops when the selected quantity of fuel was consumed, once the reset push button is pressed. The time taken for consuming the required quantity of fuel is then noted. The fuel flow meter is shown in the Figure 4.5.
4.2.5.1.2 Measurement of Airflow

The actual quantity of air intake was measured by using a 20 mm diameter orifice meter with differential ‘U’ tube water manometer. The difference in the water column of the manometer is noted (Figure 4.6). The equivalent air column is then calculated. The volume of air $V_a$, through the orifice of diameter, 0.02 m, is then given by

$$V_a = C_d \times A \times \sqrt{2gH_a}$$  \hspace{1cm} (4.19)

where $V_a$ is volume of air in m$^3$

$C_d$ is coefficient of discharge = 0.62

$A$ is the cross sectional area of the orifice

$H_a$ is the column of air in m$^3$
4.2.5.1.3 Measurement of Coolant and Exhaust Temperature

The temperature sensors located at the respective places as shown in Figure 4.3 give the inlet and outlet temperatures of the coolant and the exhaust.

4.2.5.2 Instrument for Pressure Measurements

A pressure transducer (Model No.701A) was used to measure the cylinder pressure. It is a very precise and robust pressure sensor with an inline charge amplifier for combustion pressure measurement application. The charge amplifier accepts a power supply between 7 to 32V DC and has a range of 0 to 100 bar (40mV/bar) and works with a time constant of 5s. PC based combustion analysis hardware and software were used to acquire and analyse the pressure data. The pressure pick up was mounted on the cylinder head with an adopter. It is a very accurate and precise quartz pressure sensor,
whose signal cable is connected to the charge amplifier with an integrated high temperature Viton cable for combustion pressure measurement.

Table 4.3 Technical data of the pressure sensor

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure range</td>
<td>0 – 250 bar</td>
</tr>
<tr>
<td>Calibration at 200 °C</td>
<td>0 – 100 bar</td>
</tr>
<tr>
<td>Sensitivity (±0.5%)</td>
<td>40 mV/bar</td>
</tr>
<tr>
<td>Frequency range (-3dB)</td>
<td>0.016 – 20000 Hz</td>
</tr>
<tr>
<td>Time constant</td>
<td>5 s</td>
</tr>
<tr>
<td>Signal output (at 1mA load)</td>
<td>4.4–5V max</td>
</tr>
<tr>
<td></td>
<td>&gt; 0V min</td>
</tr>
<tr>
<td>Supply current</td>
<td>6 mA</td>
</tr>
</tbody>
</table>

The quartz pressure sensor 701A is used for measuring dynamic pressures up to 250 bars. The measured pressure acts through the diaphragm on the quartz crystal measuring element, which transforms the pressure (bar) into an electrostatic charge (Pico Coulomb). The stainless steel diaphragm is welded flush and sealed hermetically to the stainless steel sensor body. The quartz elements are mounted in a highly sensitive arrangement in the quartz chamber, which is welded hermetically to the body.

The charge amplifier is a rugged one channel (converts only one channel of input signal), charge amplifier with two ranges and an integrated peak memory. It serves to convert the electrical charge yielded by the sensor into a proportional voltage. The circuitry incorporates the latest hybrid technique, making it reliable and vibration proof. The charge amplifier has a very stable temperature input stage, which together with the differential input provides excellent condition for operation. For offset error adjustment,
provision is made in the amplifier. At every 1°, crank angle interval, the PC based engine-indicating system, measures the cylinder pressure for all the four strokes. Thus, 720 angle and pressure values are collected.

The hardware for pressure measurements consists of a high-speed A/D data acquisition system and a dedicated digital signal processor. The inbuilt software performs the thermodynamic analysis of the pressure data in real time. Accurate cylinder pressure vs. crank angle data is obtained with these systems. The parameters described above ensure that the correct reference pressure is used to convert the measured pressure signals to absolute pressure. The pressure vs. crank angle phasing is made accurate to within the crank interval of about 0.2° and the transducer temperature variations (which can change the transducer calibration factor) due to variation in wall heat flux during the engine cycle are held to a minimum.

4.2.5.3 Emission Measuring Instruments and Set up

4.2.5.3.1 Smoke and Particulate Measurement

For measurement of smoke, Diesel-tune DX.230 smoke meter is used. It is a partial flow opacimeter suitable for Free Acceleration and Full Load testing of all types of diesel engines. The smoke meter works on the principle of shining a light beam through the exhaust smoke to determine the density or opacity.

It consists of a sampling head and an interface unit. The sampling head operates at low voltage (30V AC) and at temperatures to -15°C. It consists of a green LED and a photodiode mounted at opposite ends of a measuring tube, such that the effective light path length through the smoke is 250 mm. A schematic diagram of smoke sampling head is shown in Figure 4.7.
The measuring tube and the LED/ Photodiode assemblies are heated such that their minimum temperature is 70°C, thereby preventing condensation of the oil and water content of the smoke. The temperature of the smoke is measured at the inlet to the measuring tube so that, if necessary, an automatic correction can be applied to the opacity readings for smoke temperatures other than 100°C.

The smoke sampling head contains a fan to draw the smoke sample into the measuring tube via a bypass system, so that the pressure inside the system tube cannot normally exceed ambient by more than 7.5 mbar. This provides for rapid response and fast clearance of the smoke sample after testing. An air curtain reduces sooting and protects the lenses. In addition, the
sampling head has an auto range feature that can provide a step change in LED intensity to compensate for dirtiness.

The measuring tube contains rings to eliminate reflections, and the LED is pulsed so that the deduction system can ignore any ambient light level. Mid point calibration is provided by automatically interrupting this pulse train to give correct calibrated light level. For example, three pulses out of eight give a calibrated light level of 3/8 or 37.5%.

The interface unit is connected to the sampling head by a single cable of length up to 20m. The cable carries power to the head and data in both directions. All information are combined in the interface unit and sent to the computer through a link. This link also carries the control commands from the computer to the smoke meter system.

The smoke meter gives smoke readings on a percentage scale, 0% denotes clean air (i.e. no smoke at all), and 100% denotes total obscuration of light beam (i.e. totally black smoke). Nevertheless, this percentage scale will be different for every opacimeter having a different light path length. Therefore, it is expressed in terms of factor ‘K’ coefficient of light absorption by the equation.

\[ K = \frac{-1}{L} \log_e [1 - N/100] \]  \hspace{1cm} (4.20)

where \( N \) = the percentage of light obscuration
\( L \) = light path length in metres

The two readings, 0% and 100% are set automatically, the first by having a light beam in clean air, and the second by switching the beam out, so that the sensor sees no light at all. As an extra precaution, the light level is checked at the mid point of 37.5% (equivalent to coefficient of light
absorption, \( K = 1.88/\text{m} \). Any soot, which may land on the lenses during use affects it.

The smoke meter automatically checks its calibration at the start of each test or at any time that the set zero is pressed. The tolerance on the midpoint calibration check is 34.4% to 40% (equivalent to \( K = 1.88 \pm 0.2/\text{m} \)). If it falls outside this limit, a warning is displayed.

The particulate levels are obtained from the conversion chart of the smoke meter manual (ULX900 HSU manual) and on switching to particulate mode the readings are displayed in \( \text{mg/m}^3 \). The conversion chart scales include compensation for the temperature of the smoke.

Schematic diagram of smoke sampling head is shown in Figure 4.8.

![Schematic diagram of smoke sampling head](image)

**Figure 4.8 Schematic diagram of smoke sampling head**

### 4.2.5.3.2 Measurement of HC, CO, CO\(_2\), and NO\(_x\)

The emissions such as NO\(_x\) and HC were obtained from 5 - Gas analyser 290 series. Software version GB 1.0X is used for the required results.
The emissions HC, CO, CO₂ and NOₓ are directly obtained from Crypton emission analyser with computer interfaced arrangement. Emission analyser is also self-calibrating type. If the calibration is within the required time span, the analyser skips to the leak test to ensure that there are no air leaks in the unit. It will not pass to the next stage until the leak test has passed. It will automatically perform an auto zero test whilst warming up, and at intervals during use to ensure that the measurement accuracy is maintained. HC (Non dispersive infra red analyser) and NOₓ values (chemical sensor and catalytic reaction) are measured in ppm by volume. CO and CO₂ readings (Non dispersive infra red analyser) are given as percentage by volume.

Features

- 5-gas CO/HC/CO₂/O₂/NOₓ readings
- RPM display with 4 stroke or 2 stroke-DIS option
- Calculated Lambda with leaded/unleaded/LPG/CNG settings
- On-screen operator prompts, menus and readings
- Automatic calibration, zeroing and O₂ spanning
- Automatic leak detection and HC residue tests
- Oil temperature probe
- Name/address header facility
- Optional trolley and A4 printer
- Software for computerization as per SIAM/regional RTO’s available
- Flow rate 5 litres / min – nominal
- Operating pressure 750 – 1100 mbar
- Response time – 11 sec – 95% of final reading with 6m sample hose
- Power requirements – 100-250V, AC, 50-60hz
- Warm up time - < 10 minutes (self controlled) at 20° C
- NOₓ sensor – TRAD 103
Figure 4.9 Schematic diagram of gas analyser

1. RS232 Port - CMT Communications  11. Water Output
2. RS485/RS232 Communications  12. Sample Probe Input
5. Calibration Gas Input  15. Parallel Port
6. Water Trap and Primary Filter  16. Viewing Slot for status LEDs
7. Gas Filter  17. Tachometer Probe Input
8. Mains On/Off  18. Parallel port - printer
9. Mains Input  19. Sample Exhaust – NOx
10. Mains Auxiliary Output  20. NOx Sensor
### Range and accuracy

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO</td>
<td>0-10% volume</td>
<td>0.01%</td>
</tr>
<tr>
<td>HC</td>
<td>0-2000 ppm</td>
<td>1 ppm</td>
</tr>
<tr>
<td>CO₂</td>
<td>0-20% volume</td>
<td>0.1%</td>
</tr>
<tr>
<td>O₂</td>
<td>0-25% volume</td>
<td>0.01%</td>
</tr>
<tr>
<td>NOₓ</td>
<td>0-1000 ppm</td>
<td>1 ppm</td>
</tr>
<tr>
<td>RPM</td>
<td>0-10000</td>
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<tr>
<td>AFR</td>
<td>5-50</td>
<td></td>
</tr>
<tr>
<td>Oil temperature</td>
<td>0-120°C</td>
<td></td>
</tr>
<tr>
<td>Lambda</td>
<td>0.2-5°C</td>
<td></td>
</tr>
</tbody>
</table>

#### 4.2.6 Experimental Test Standards, Specifications and Error Analysis

For comparing the performances of engines, a number of standards are available. The power output of the engine at full load is directly related to atmospheric conditions. If the engine is operated in a region of low pressure, there will be a considerable reduction in the output. Similarly, if the temperature of the air entering the engine is high, the output will be considerably reduced. Therefore, there is a need to correct the engine performance to the same standard environment. For the corrected BP, it is assumed that the FP is the same at test and at standard conditions.

Experimental investigations are carried out to determine the performance of the engine based on BIS: 10003 – 1981 “Specification for performance requirements for CI engines for automotive purposes”. The test were carried out at ambient conditions and the power, torque and specific fuel consumption determined during the tests were corrected to the standard
reference conditions, IS: 10000 (PART II) – 1980. Smoke emission levels were measured according to IS: 10000 (part X) – 1980 and Federal automotive exhaust emission test procedures.

Since deviations occur in experimentally obtained observations, the normal error distribution, (Guassian distribution), is applied for corrections. When the data abide by Gaussian or normal distribution rules, plus and minus errors are likely to be equal and small errors are more probable than large errors. It is a fact, that many experimentally determined sets of data with random errors fit the Gaussian distribution better than any other. Since the actual value by which a result is in error is never known, the mean deviation (probable error) and standard deviations are widely used for expressing error estimate. In taking the readings with the instruments in the test set up, this has been followed and it is observed that the errors are negligible. For normal distribution, the probability of obtaining a deviation greater than $2\sigma$ is about 5 percent; greater than $3\sigma$, about 0.33 percent; and greater than $4\sigma$, approximately 0.006 percent. Normally one criterion for discarding suspected data is the $3\sigma$; any data having a deviation of more than $3\sigma$ are arbitrarily thrown out.

The above statistical methods apply only to random errors and they do not account for systematic errors. Therefore, it is more a treatment of precision than it is of error. Systematic error is a much more important factor than random error and must be minimized through careful attention to procedures, calibration and system design, selection and application. This has been followed very carefully through out.

In the experimental procedure, in order to offset the inaccuracies in measurements and to study the effect of cycle-to-cycle variation, while taking the pressure readings 50 cycles average readings are taken and at each loading
the experiment is repeated three times to minimize the errors in noting the various readings, so as to have the deviations of at least $3\sigma$. This reduces the errors in measurements to below 0.5 percent. As far as possible, the above methodology is followed in all the readings taken with the instruments for accuracy and utmost care has been taken in following the test procedures to avoid the errors in experimental investigations.

4.2.7 Test Procedure for Performance and Emissions

The engine is started and allowed to warm-up for about 15 minutes. The readings from dynamometer, time taken for 100cc of fuel consumption, pressure reading from the computer based data acquisition system and smoke values from Diesel tune smoke meter are recorded. The readings are taken three times for each case to minimize the errors in the analysis.

The engine is equipped with sensors (pressure and temperature). Load is applied by an eddy current dynamometer. The time taken for fuel consumption is recorded by an optoelectronic liquid level sensor. The pressure readings are taken from the signal obtained from piezoelectric transducer (Kistler) through the data acquisition system. The computer plot of pressure and crank angle is obtained to a minimum scale reading of $1^\circ$ for crank angle and 1 bar on pressure. Smoke readings are obtained in Hatridge Smoke Units from Diesel Tune Smoke Meter interfaced with computer. The smoke meter is self calibration type and can show smoke readings in Fuel Smoke Number (Bosch Units) to a scale of 1 to 10 with an accuracy of 0.01 units. The details are shown in Appendix 5. The smoke meter is capable of measuring particulates derived from smoke readings in mg/m$^3$. HC, CO, CO$_2$ and NO$_X$ readings are obtained from the gas analyzer interfaced with computer. HC and NO$_X$ values are measured in ppm by volume. CO and CO$_2$ readings are given as percentage by volume. The analyser is computer
interfaced and gives CO and CO₂ readings in volume percentage only. The gas analyzer is equipped with self calibrating facility. To ensure accuracy, five readings were taken at each stage and average was calculated. This procedure was repeated by varying from no load to full load. The emission readings obtained were converted into mass emission standards as given by Blair (1988).

The eddy current dynamometer is set to ‘n = const’ mode, so that once the engine speed is fixed by the ‘SET DEMAND’ dial the engine load/torque is varied by increasing or decreasing the throttle. An input signal from the Load –Cell, a transducer, is given to the torque indication circuitry. The indicator display is calibrated with the help of standard weights to indicate torque in the desired unit, Nm or kgm.

When the excitation current is passed through the dynamometer coil, the engine is braked and as a result the equal and opposite reaction is exerted on the dynamometer stator assembly. This reaction torque is seen on the torque indication assembly. By using the torque value and the corresponding speed value, the power developed by the engine is then calculated.

This procedure is repeated for different brake mean effective pressure (BMEP), by changing the load from no load to full load. All the measurements are recorded as above for the three blending levels of 3.0, 6.0 and 9.0 percent by weight of the three oxygenates.

The rate of heat release based on the data of the recorded cylinder peak pressure is then evaluated. The various performance parameters such as friction power (FP) (calculated by Willan’s line method), brake power (BP), brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC) (calculated from BSFC), torque and the emissions are then
calculated. These results are compared with values of the diesel fuel combustion and performance to evaluate the effect of oxygenates addition.

4.2.8 Cylinder Pressure and Heat Release Analysis

In an engine cycle, the period from inlet valve closing to the exhaust valve opening is very significant as power is developed during this period. For analysis, in a direct injection diesel engine, the cylinder contents could be taken as a single open system during the whole process for one cycle and as a closed system at uniform temperature at each instant in time during the combustion process of the cycle (Obert 1973).

The cylinder pressure changes because of cylinder volume change, combustion rate, heat transfer to the cylinder walls, flow in and out of crevice region and leakage of gases past the piston and valves. The first two of these effects are the largest. The effect of volume change on the pressure can be readily accounted for but the effect of combustion on the pressure is difficult to determine. Under normal operating conditions, 70 to 95 percent of the injected fuel is in the vapour phase at the start of combustion. Evaporation is more than 90 percent complete after 1 ms. However, only 10 to 35 percent of this vapourised fuel is mixed with in flammability limits. Initially the combustion is largely mixing limited and rapid. There after it is stabilized. Under these circumstances, the effect of combustion on pressure cannot be precisely calculated.

The pressure rise due to combustion is proportional to the amount fuel chemical energy released. Although the polytrophic index is not constant during combustion, by using a heat release approach based on the first law of thermodynamics. The effects of heat transfer, crevices and leakage of the gases can directly relate the pressure changes to the amount of fuel chemical energy released by combustion. This can be incorporated into the cylinder
pressure data analysis by selecting a suitable mathematical model (Sorension et al 1998).

Thus, the first law of thermodynamics for such a system is:

\[
\delta Q_{ch} = dU_s + \delta Q_{ht} + \delta W + \sum h_i dm_i
\]  
(4.21)

where, \( \delta Q_{ch} \) represents the chemical energy released by combustion, \( \delta Q_{ht} \) is the heat transfer to the chamber walls, \( \delta W \) is the piston work and is equal to \( \rho dV \) and \( \sum h_i m_i \) is the mass flux term that represents the flow across the system boundary. In the absence of fuel injection, \( \sum h_i m_i \) represents the flow into and out of the crevice regions only, and this is very small compared to the rest of the terms in the equation and is difficult to quantify precisely. Hence, this is neglected in our calculations.

The accuracy with which this energy balance can be made depends on how adequately each term in the above equation can be quantified. Assuming that \( U_s \) is given by \( mU(T) \), where \( T \) is the mean gas temperature and ‘m’ is the mass with in the system boundary, then \( dU_s = mC_vdT + \rho dm \). The leakage to the crankcase here can be neglected being very small. Therefore the equation (2.3) on substitution for \( dU_s \) and \( dm_i \) (= \( dm_{cr} = - \rho dm \)), becomes,

\[
\delta Q_{ch} = mC_vdT + (h' - \rho)dm_{cr} + \rho dV + \delta Q_{ht}
\]  
(4.22)

where, \( dm_{cr} > 0 \) when flow is out of the cylinder into the crevice, and \( < 0 \) when the flow is from the crevice to the cylinder \( h' \) is the specific enthalpy.

Using the ideal gas law (neglecting the change in the gas constant \( R \)), the equation (4.4) then becomes,
The heat transfer rate, 

\[ \delta Q_{ch} = \frac{C_v}{R} V dp + \frac{C_v}{R + 1} pdV - (h' - u + C_v T)dm_{cr} + \delta Q_{ht} \]  

(4.23)

The heat transfer rate,

\[ \frac{dQ_{hv}}{d\theta} = A h_c (T_g - T_w) \]  

(4.24)

where A is the chamber surface area (m²), \( T_g \) is the mean gas temperature (K), calculated by using the ideal gas equation \( pV = mRT \), \( h_c \) is the heat transfer coefficient. \( T_w \) is the mean wall temperature calculated by averaging the gas and coolant temperatures, where the later is assumed to have a constant value of 353K. (Marcello Canova et al 2005)

Many formulae for calculating the instantaneous heat transfer coefficient, \( h_c \), have been proposed. The basis of these correlations is the assumption that the Nusselt, Reynolds and Prandtl number relationship follows that found for turbulent flow in cylindrical enclosures, i.e.

\[ Nu = a (Re)^m (Pr)^n \]  

(4.25)

where \( Nu = \) Nusselt number
\( Re = \) Reynolds number
\( Pr = \) Prandtl number
\( a, m, n \) are proportionality constants.

In this research work \( h_c \) is modeled considering the Woschni correlation \( h_c = 3.26B^{-0.2} p^{0.8} \times v^{0.8} \) and this correlation is found to be more suitable for the given engine dimensions, pressure and mean piston speed (Richard Stone 1999, Heywood 2000, Marcello Canova et al 2005).

The crevice effects, the effect of fuel vaporization and heat up are small in comparison to the gross heat release rate, \( dQ_{ch}/d\theta \) and heat transfer
rate, \( \frac{dQ_{\text{in}}}{d\theta} \). The enthalpy of vaporisation of diesel fuel is generally less than 1% of its heating value; the energy change associated with heating up fuel vapor from injection to typical compression air temperature is about 3% of the fuel heating value. Hence, they were neglected. Then the equation (4.5) can be written as,

\[
\frac{dQ_{\text{sh}}}{d\theta} = (1 + \frac{C_p}{R}) p \frac{dV}{d\theta} + \frac{C_v}{R} \frac{dp}{d\theta} + \frac{dQ_{\text{ht}}}{d\theta} = (\frac{\gamma}{\gamma - 1}) p \frac{dV}{d\theta} + (\frac{1}{\gamma - 1}) \frac{dp}{d\theta} + \frac{dQ_{\text{ht}}}{d\theta} \tag{4.26}
\]

where \( \gamma \) is the ratio of specific heats, \( C_p/C_v \).

In a diesel engine, both the compression of the unburned mixture prior to combustion and the expansion of the burned gases following the end of combustion are only close to adiabatic isentropic processes, \( pV^n = \text{constant} \) \( \gamma = C_p/C_v \). The effects of thermal cycling are significant. Therefore the compression and expansion processes are well fitted by a polytrophic relation, \( pV^n = \text{Constant} \), where \( n \) is the polytrophic index. Also in a polytrophic process, both heat and work transfer take place unlike in an adiabatic process. Hence, \( n \) is taken for the analysis in equation (4.8) instead of \( \gamma \).

Because the diesel engine always operates with lean fuel/air ratios and with very lean fuel/air ratios at part load, the effective value of \( \gamma \) over the expansion process is higher. This is because with leaner mixtures, the burned gas temperatures after combustion decrease. This decreases the burned gas specific heats and thereby increases the effective value of \( \gamma \) over the expansion stroke. More specifically, for diesel engine heat release analysis, we would expect the ratio of specific heat value to have values (for air and the
trapped burned gases), appropriate to air at the end of compression stroke temperatures prior to combustion (which is \( \approx 1.35 \)) and to burned gases at the overall equivalence ratio following combustion (which is \( \approx 1.26 - 1.3 \)). However, the appropriate values for ratio of specific heats during combustion that will give most accurate heat release information is not well defined (Cheng and Gentry 1986, Gatowski et al 1984).

The value of ‘n’ for compression and expansion processes is 1.3 (\( \pm 0.05 \)) for conventional fuels. It is comparable to the average value of ratio of specific heats for the unburned gas mixture over the compression process, but is larger than ratio of specific heats for burned gas mixture during the expansion process due to heat loss to the combustion chamber walls. Taking into consideration the above facts the value of ‘n’ for the oxygenated blends is taken as 1.32, since the appropriate range for ratio of specific heats for diesel heat release analysis is 1.3 to 1.35 (Heywood 2000). The value of ‘n’ can also be obtained by plotting the values log of ignition delay verses \( 1/T_g \). Therefore,

\[
\frac{dQ_{ch}}{d\theta} = \left( \frac{n}{n-1} \right) p \frac{dV}{d\theta} + \left( \frac{1}{n-1} \right) V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \quad (4.27)
\]

The difference between the gross heat release rate, \( Q_{ch} \), and the heat transfer to the walls, \( Q_{ht} \), is the net heat release, \( Q_n \), which is equal to the work is done on the piston plus the change of internal energy of the cylinder contents. This is given by the first two terms of equation (4.9).

The volume at any particular crank angle can be (for air and the trapped burned gases), calculated by using the formula,

\[
V(\theta) = V_{disp} \left[ \frac{r}{r-1} - \frac{1}{2} \cos \theta + \frac{L}{S} \frac{1}{2} \sqrt{\left(2L/S\right)^2 - \sin^2 \theta} \right] \quad (4.28)
\]
where $S$ is the stroke, $L$ is the connecting rod length and $r$ is the compression ratio of the given engine (Heywood 2000).

A computer program developed to calculate all the above values is given in Appendix 3.

To summarize the data collected from the simulation results and experimental results as well as the data using the empirical relations discussed in this chapter, are analyzed in detail in the next chapter for evaluating the performance of the selected oxygenates.