CHAPTER 6

NUMERICAL SIMULATION OF IN-CYLINDER FLOW

6.1 WORKING FLUID MOTION WITHIN THE CYLINDER

In-cylinder basic motions are concentrated in terms of swirl squish, tumble and turbulence.

6.1.1 Swirl

Swirl is defined as flow that is spinning concentrically with the axis of the cylinder. It is steady charge flow in a diesel engine, based on a stationary geometry, resulting in a simple and stable flow. The main axis of motion is aligned with the cylinder axis and is constant in time as shown in Figure 6.1. The spatial resolution of the single timestep of computational domain is high with a combination of different irregular shaped cells on an adaptive resolution grid.

Figure 6.1 Swirl Motion about Cylinder Axis
6.1.2 Squish

The piston approaches TDC at the end of compression stroke, the volume around the outlet edges of the combustion chamber is suddenly reduced to a very small volume. The gas mixture occupying the volume at the outer radius of the cylinder is forced radially inward. This radially inward motion of the gas mixture is called squish. It adds to other mass motions within the cylinder to mix the air and fuel and to quick spread of the flame front.

6.1.3 Tumble

The piston- approaching near TDC squish motion generates a secondary rotational flow called ‘tumble’. This rotation occurs about a circumferential axis near the outer edge of the piston bowl as shown in Figure 6.2. Tumble is defined as in-cylinder flow that is rotating at right angles to the cylinder axis. As the piston moves down, the cylinder volume increases by an order of magnitude and the fuel-air mixture entering the cylinder is drawn into a gradually developing tumble pattern.

Figure 6.2 Tumble Motion about Circumferential Axis
6.1.4 Turbulence

Turbulence consists by randomly dispersed vortices of different sizes which become superimposed into the air or air and fuel mixture flow stream. Turbulence is the three-dimensional unsteady random motion observed in fluids at moderate to high Reynolds numbers. As technical flows are typically based on fluids of low viscosity, almost all technical flows are turbulent.

6.2 COMPUTATIONAL APPROACH

6.2.1 Computational Domain

The CFD code FLUENT for finite volume method has been utilised to solve the discretised continuity and Navier-Stokes equations. Fully combined meshes of combustion chamber are utilized here due to the requirements of moving mesh. The piston motion was carried out by cell activation and deactivation and supported by vertex motion routines. The code is competent of handling the complex geometry and enabling the computational domain to include the combustion chamber with moving piston. The numerical methodology and computation of this ICE case is based on the pressure-correction method and the PISO algorithm. The second upwind differencing scheme as the spatial discretisation is used for the momentum, energy and turbulence equations. The temporal discretisation is the implicit method, with variable time step depending on the stage of the engine cycle. The time step is set up at the 0.25 CAD per iteration by reason of the highly computational cost, since there are no high local velocities and the effect of the squish and tumble during expansion stroke does not have so much effect on the calculation.
The numerical running calculation starts when inlet valve opens. A time step calculation was continued from the suction stroke to the exhaust stroke. The initial values for pressure and temperature for engine operating speed of 1500 rpm were assumed from atmospheric conditions. The initial turbulence intensity was set at 3% of the mean flow, which is quite sufficient for fully turbulent fluid flow, whereas the integral length scale was specified proportionally at 0.4% based on the Prandtl’s work as a result of the distance to the nearest solid wall. The constant temperature boundary conditions were allocated independently for the cylinder head, the cylinder wall, and the piston crown that outline the walls of the combustion chamber. The temperature on each of these walls will be calculated numerically in the form of iteration for every time step automatically. A standard two-equation k-ε turbulence model was used in this simulation and the model constants (Fluent 2006) are tabulated in Table 6.1.

### Table 6.1 Model Constants

<table>
<thead>
<tr>
<th>Constant</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_\mu$</td>
<td>0.09</td>
</tr>
<tr>
<td>$C_1$</td>
<td>1.44</td>
</tr>
<tr>
<td>$C_2$</td>
<td>1.93</td>
</tr>
<tr>
<td>$k$</td>
<td>1.00</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>1.19</td>
</tr>
<tr>
<td>$K$</td>
<td>0.4187</td>
</tr>
</tbody>
</table>

#### 6.2.2 Computational Methods

The equations employed to describe mass, momentum, energy and k-ε turbulence model for high Reynolds number in the vector notation without
source terms from spray and chemical reactions due to under motoring condition are expressed as follows:

\[
\frac{\partial \rho}{\partial t} + \nabla (\rho \, \bar{u}) = 0 \quad (6.1)
\]

\[
\frac{\partial (\rho \, \bar{u})}{\partial t} + \nabla (\rho \, \bar{u} \, \bar{u}) = \nabla \cdot \left( \frac{2}{3} \rho k \right) + \nabla \cdot \sigma + \rho g \quad (6.2)
\]

\[
\frac{\partial (\rho \, U \, \bar{u})}{\partial t} + \nabla (\rho \, U \, \bar{u}) = -P \nabla \bar{u} - \nabla \left( \frac{2}{3} \rho k \right) + \nabla \cdot \bar{j} + \rho \varepsilon \quad (6.3)
\]

\[
\frac{\partial \rho \, k}{\partial t} + \nabla \left( \rho \, k \, \bar{u} \right) = -\frac{2}{3} \rho k \nabla \cdot \bar{u} + \sigma \nabla \cdot \bar{u} + \nabla \left[ \left( \frac{\mu_t}{\nu_{k}} \right) \nabla k \right] - D k - \rho \varepsilon \quad (6.4)
\]

\[
\frac{\partial \rho \, \varepsilon}{\partial t} + \nabla \left( \rho \, \varepsilon \, \bar{u} \right) = - \left( \frac{2}{3} C_{t1} - C_{t3} \right) \rho \, \varepsilon \, \nabla \cdot \bar{u} + \nabla \left[ \left( \frac{\mu_t}{\nu_{k}} \right) \nabla \varepsilon \right] + \frac{\varepsilon}{k} \left[ C_{t1} \sigma \, \nabla \bar{u} \cdot C_{t2} \sigma \varepsilon \right] \quad (6.5)
\]

The origins of the above equation are based on the mass, momentum and energy balance method.

### 6.2.3 Turbulence Models

Turbulence modelling is important in internal combustion engines. Since turbulence directly affects spray, mixing and combustion in an engine, adequate prediction of turbulence behaviour is necessary for better understanding of these phenomena in order to improve engine performance and emissions. In this section, the k-ε turbulence model with high Reynolds number was used for CFD calculation. The typical turbulence parameter consists of turbulence viscosity, turbulence integral length scale and turbulence intensity. The equations and formula are referred from Fluent user manual. The turbulence kinematic viscosity term is described as below:

\[
\mu_t = C_{\mu} \rho \left( \frac{k^2}{\varepsilon} \right) \quad (6.6)
\]
The computed turbulence integral length scale is obtained from the k-ε results as:

$$L_v = C_p (3/4) \frac{k(3/2)}{\kappa \varepsilon}$$  \hspace{1cm} (6.7)

where $\kappa$ is the Von Karman constant. The accompanying result of the k-ε model is the turbulence intensity which is assumed as:

$$u = \sqrt{\frac{2k}{3}}$$  \hspace{1cm} (6.8)

### 6.2.4 Heat Transfer Models

The internal combustion engine heat transfer is among the most complicated engineering heat transfer problems, involving rapid changes in gas temperature, pressure and velocity, together with local complexities of temperature and velocity distributions. In principle, the physical phenomena involved in engine heat transfer are functions of diffusion, convection, pressure work and heat generation by combustion and radiation. With the exception of radiation, these are governed by the well-known energy equation. But the complexities of coupling these terms as well as the insufficiency of information to solve the energy equation have raised many difficulties in modelling engine heat transfer. Here the heat transfer models deal with overall empirical heat transfer coefficients generally assumed for all the heat transfer surfaces in the cylinder. As the basis of modelling a quasi-steady assumption is employed, this may be described by an expression as follows:

$$q'' = h(T' - T_w)$$  \hspace{1cm} (6.9)

where $q''$ is heat flux, $h$ is heat transfer coefficient, $T'$ is mass-averaged of bulk mean gas temperature obtained from an equation of state with given
pressure and $T_w$ is wall surface temperature. With regard to the CFD code utilised, the governing equations for heat transfer are implemented through the following general form of the enthalpy conservation equation for a fluid mixture.

$$
\frac{1}{g} \frac{\partial}{\partial t} (gh) + \frac{\partial}{\partial x_j} (\rho \bar{u} h - F_{h,j}) = \frac{1}{g} \frac{\partial}{\partial t} (gp) + \frac{\partial}{\partial x_j} (\bar{u} j p) - p \frac{\partial u_j}{\partial x_j} + \tau_{ij} \frac{\partial u_j}{\partial x_j} sh \quad (6.10)
$$

Here, $h$ is the static enthalpy and it is defined by:

$$
h = c_p T - c_p^0 T_0 + \sum m_m H_m = h_t + \sum m_m H_m \quad (6.11)
$$

It should be noted that the static enthalpy $h$ is defined as the sum of the thermal and chemical components, the latter being included to cater for chemically reacting flows. For a constant-density approximation to an ideal gas like air at standard temperature and pressure, the enthalpy is transported with all pressure dependent terms. For solids and constant density fluids, such as liquids, the used CFD code solves the transport equation for the specific internal energy, $e$, where:

$$
e = \hat{c} T - c^0 T_0 + \sum m_m H_m \quad (6.12)
$$

it does not contain the pressure-related terms.

### 6.2.5 Inlet

Pressure boundary conditions are used to define the fluid pressure at the flow inlet. Pressure inlet boundary conditions are used when the inlet pressure is known but the flow rate or velocity is not known. The inlet temperature and inlet pressure are considered as 300K and 1.03 bar respectively.
6.2.6 Outlet

The outlet pressure boundary conditions require the specification of static pressure at the outlet boundary.

6.2.7 Wall

In any flow, the Reynolds number of the flow becomes very low and turbulent fluctuations are damped considerably near the walls. The laminar viscosity starts to play a significant role. In the present case, walls are assumed to be adiabatic with no slip condition.

6.2.8 Modelling Non-Premixed Combustion

In non-premixed combustion, the fuel and oxidizer enter the reaction zone in distinct streams. This is in contrast to premixed systems, in which reactants are mixed at the molecular level before burning. Under certain conditions, the thermo chemistry can be reduced to a single parameter: the mixture fraction. The mixture fraction denoted by the mass fraction that originated from the fuel stream. In other words, it is the local mass fraction of burnt and unburnt fuel stream elements in all the species. The approach is elegant because atomic elements are conserved in chemical reactions. In turn, the mixture fraction is a conserved scalar quantity, and therefore its governing transport equation does not have a source term. The combustion is simplified to a mixing problem, and the difficulties associated with closing non-linear mean reaction rates are avoided. Once mixed, the chemistry can be modelled as being in chemical equilibrium with the Equilibrium model, being near chemical equilibrium with the Steady Laminar Flamelet model, or
significantly departing from chemical equilibrium with the Unsteady Laminar Flamelet model.

### 6.2.9 Spray Break up Model

FLUENT offers two spray break up models, the TAB and the wave model. In the present work TAB model is used. The TAB model is based on the analogy between an oscillating and distorting droplet and a spring mass system.

### 6.2.10 Droplet Collision Model

Droplet collision model includes tracking of droplets for estimating the number of droplet collisions and their outcomes in a computationally efficient manner. The model is based on O’Rourke’s method, which assumes stochastic approximation of collisions. When two parcels of droplets collide then algorithm further establish the type of collision. Only coalescence and bouncing outcomes are measured. The probability of each outcome is calculated from the collision Weber number and fit to experimental observations.

### 6.2.11 Combustion Model

The combustion model was combined with species transport and finite rate chemistry with simplified chemical reactions to simulate the overall combustion process in a diesel engine. This approach is based on the solution of transport equations for species mass fractions.

### 6.3 ESTABLISHING THE BOUNDARY CONDITIONS

The prediction of engine flow using a CFD code strongly depends on the initial and boundary conditions. The simulation generally starts from
the intake valve opening for computational efficiency. The calculations are evaluated for intake valve opening with the piston and the intake valve assigned their appropriate position. The initial fluid temperature, mean density and pressure are assumed to be uniform inside the cylinder as well as inside the intake valve. Initial turbulence kinetic energy and its dissipation rate are scaled to the mean piston speed. Since a finite flow domain is specified, physical conditions are required on the boundaries of the flow domain. The simulation generally starts from an initial solution and uses an iterative method to reach a final flow field solution. The valve timings and temperature of the wall are given as initial and boundary conditions as shown in Table 6.2 and the species selection is tabulated in Table 6.3.

<table>
<thead>
<tr>
<th>Boundary</th>
<th>Momentum boundary condition</th>
<th>Thermal boundary condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake port surface</td>
<td>Wall</td>
<td>350K</td>
</tr>
<tr>
<td>Inlet</td>
<td>Pressure=100kPa, TKE=1m²/s², Turbulence length scale=0.02m</td>
<td>315K</td>
</tr>
<tr>
<td>Cylinder wall</td>
<td>Wall</td>
<td>400K</td>
</tr>
<tr>
<td>Cylinder head</td>
<td>Wall</td>
<td>400K</td>
</tr>
<tr>
<td>Piston top</td>
<td>Wall with boundary velocity</td>
<td>450K</td>
</tr>
<tr>
<td>Valve surface</td>
<td>Wall with boundary velocity</td>
<td>450K</td>
</tr>
</tbody>
</table>

Table 6.2 Boundary Conditions used for Suction Simulation
### Table 6.3 Selection of Species Transport

<table>
<thead>
<tr>
<th>Species</th>
<th>Formula</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel liquid</td>
<td>C_{10}H_{22}</td>
</tr>
<tr>
<td>Air(oxygen)</td>
<td>O_{2}</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N_{2}</td>
</tr>
<tr>
<td>Water vapour</td>
<td>H_{2}O</td>
</tr>
</tbody>
</table>

#### 6.4 MESH GENERATION AND COMPUTATION

There are two basic formulations of the finite volume method: node-centred and cell-centred. In the node-centred finite volume formulation, the finite volumes, used to satisfy the integral form of the equations, are elements of the computational mesh, while for the cell-centred finite volume approach, the finite volumes are the mesh elements themselves. The past works have proven the use of cell-centred and higher order element (Payri et al., 2004). A mesher program has been exploited to generate the grid to create the hexahedral cells for the engine model. The computational domain for the CFD calculation covers the piston bowl as shown in Figure 6.3. The number of cells varies from 70,000 cells in TDC and around 2,30,000 cells in BDC. The tetrahedral cells have been adopted for the mesh generation because they provide a better accuracy and stability compared to the cubical cells. The important motivations about the use of tetrahedral cells are the requirements of moving meshes and boundaries to accomplish the CFD calculation. Because of the complexity of the engine model, the computational mesh is divided into four areas with different topologies, where each area has been meshed separately. The simulation required a moving mesh and boundary algorithm embedded into the FLUENT programme.
The moving mesh and boundary algorithm for this engine model has been developed inside FLUENT by declaring the events for each time step we define and then activating the grid in order to move the mesh. The concept of moving mesh is that the cell is squeezed to zero volume over one time step, with all its contents such as pressure, temperature, mass, momentum and enthalpy being expelled into the neighbouring cells. Hence, conservation is satisfied exactly even with removal of any cell layer. On the other hand, when the cell layers are added, they grow from zero size to their full volume, absorbing the conserved variables through their faces. As the total number of computational cells is around 70,000-2,30,000 the typical CPU time taken for the simulation of a complete cycle with fluctuating time steps is more than a day.

Figure 6.3 The computational domain of all piston model of engine


6.5 RESULTS AND DISCUSSION

The two parameters required to characterize the turbulent flow characteristics are the turbulence kinetic energy and its dissipation rate. The turbulence kinetic energy is related to the turbulence intensity, which is a measure of the characteristic speed of the turbulent flow over a distance characteristic. On the other side, the turbulence dissipation rate is associated to the turbulence length scale, which is a quantitative measure of the distance characteristics of the flow structure. An important small scale mixing parameter which is a combination of turbulence kinetic energy and its dissipation rate is referred to as the turbulence kinematic viscosity.

In-cylinder turbulence can be beneficial for gaining the optimum air-fuel mixing preparation before fuel injection and increasing combustion rates. The turbulence kinetic energy during compression stroke can be found out from the XZ cutting plane to verify the generation of air structure inside cylinder, shown in Figures 6.4, 6.5, 6.6 and 6.7 for the suction stroke of the engine. The turbulence intensity field is presented in the form of isoline contours for better appearance. It appears clearly distributed uniformly for ellipsoid piston and elliptical piston where the large values occurred in the area near the top region of the cylinder and close to the cylinder wall where there is a jet impingement was shown in Figures 6.4, 6.5, 6.6 and 6.6. The other important thing to be mentioned is that the level of velocity is relatively low in the bottom part of the cylinder and varies in the piston bowl. As can be seen, the isoline contours from four pistons are slightly different, where Ellipsoid piston has the highest value of turbulence field and other has the lowest one. It is important to note here that the behaviour of air flow before fuel injection among four pistons shows the dissimilar homogeneity. This means that the combustion chamber in the engine does play a major role in generating in-cylinder turbulence during the compression stroke. The
maximum value was observed on the exhaust side of the Ellipsoid and Elliptical chambers. On the other hand, the maximum value for Hemisphere and Double concave almost disappeared and also occur at the bottom side of chambers. From this numerical prediction of air structure, it can be concluded that the vortex homogeneity of air structure needed before the fuel injection process tends to occur in Rankine type pistons due to the nature of profile.

During the early stages of the compression stroke, the jet flow interactions are the most important mechanism for the production of turbulence intensity. Nearer to TDC, the turbulence intensity increases rapidly and reaches its maximum value in the end of the compression stroke. This behaviour continues throughout the compression stroke. As the piston moves downward during the expansion stroke, the production of the turbulence kinetic energy is investigated along the rim of the piston bowl side of the cylinder and along the cylinder top. The highest turbulence kinetic energy is produced by Ellipsoid and Elliptical where their condition is more homogeneous along cylinder head that can be utilized importantly to prepare the flow field structure before ignition timing for the combustion process later. The actual condition for other piston is that they have the capability to produce the higher turbulence kinetic energy on the region around the fuel injector location, but its value is not as high as that of Ellipsoid and Elliptical. The production and destruction of turbulence kinetic energy are always closely associated with the turbulence dissipation rate. The dissipation rate $\varepsilon$ is large where production of $k$ is large. The appearance of turbulence dissipation is quite obliging in locating the shearing rates occurs within cylinder which is a disadvantage for engine performance. The turbulent kinetic energy decreases but the turbulence dissipation rate increases at the end of compression stroke. Both turbulence intensity and length scales decrease to certain value near TDC during the closed period. This occurs due to the compression effect of the moving piston. Lastly, the other in-cylinder
feature characteristic to examine the air flow inside cylinder and perceive the effect of combustion chamber shape is the unsymmetrical in structure profile.

It is interesting to notice in the computation that the investigation of air motion such as swirls and tumbles in the combustion chamber can be carried out using the CFD code. The higher heat transfer to the combustion chamber walls within engine cylinder will lower the average combustion gas temperature and pressure as well as reducing the work per cycle transferred to the piston. Hence, the specific power and efficiency are affected by the magnitude of engine heat transfer. Heat transfer between the unburned charge and the chamber walls in compression ignition engines influences the knock, which cause the effect to the engine power and efficiency. Within the engine cylinder, the temperature of the charge relative to the wall temperature and the flow field varies extremely throughout the cycle. Both of these variables have an important influence on heat transfer. During intake stroke, the incoming charge is usually cooler than the walls and the flow velocities are high. During compression stroke the charge temperature rises above the wall temperature and flow velocities decrease due to the closed period. In this circumstance, the heat transfer occurred from cylinder gases to chamber walls. When the air is compressed to the upper position until TDC, the piston shape has little influence to cause the increment of heat transfer occurred within cylinder. Due to favourable squish and swirl conditions combustion was better than Double concave and Hemisphere piston engines. The velocity profiles of the Rankine half body like Ellipsoid and Elliptical were compared with other piston engines and shown in Figures 6.5 and 6.6. The Ellipsoid and Elliptical pistons have deep shallow bowl. It was used for sucking the squish motion of the air and also utilizing the swirl motion of the working medium. Because of this air whirl movement fuel droplet received at the end of TDC was mixed thoroughly and almost fuel is ready for burning. So that fuel was
burned within the available time for combustion. The efficiency of the Rankine type profile piston engine was better than that of other piston engine.

Figure 6.4  The velocity profile for Double concave piston profile
Figure 6.5 The velocity profile for Ellipsoid piston profile
Figure 6.6  The velocity profile for Elliptical piston profile
Figure 6.7  The velocity profile for Hemisphere piston profile
CHAPTER 7

EXPERIMENTAL VIBRATION ANALYSIS

The cylinder displacements were analyzed in Diesel engine using experiment. The Indicated Mean Effective Pressure (IMEP) was calculated using indicated power and speed. Theoretical calculation is done using thick cylinder relation for all the four piston profiles. The theoretical and experimental Cylinder displacement were compared for all four pistons separately at different speed.

7.1 EXPERIMENTAL PROCEDURE

The experiment was carried out in a single cylinder 4-stroke Texvel Engine. The IMEP is the pressure value applied constantly on the pistons during the only expansion phase; it yields the work produced by a complete engine cycle. Hence IMEP was used for the theoretical evaluations. The experimental setup is shown in Figure 7.1; the force for driving the reciprocating and rotating part is in-cylinder combustion pressure, the piston moves along the cylinder with a relatively constant speed mainly maintained by the flywheel. Figure 7.1 also shows a schematic representation for the measurement of the cylinder displacement. In order to directly reach the cylinder wall and pick up the displacement signals, an accelerometer is applied and connected to a computer. The signals were collected from engine block only and its represents the radial vibration, the vertical one is excluded.
from this analysis. Displacement of the cylinder with the four piston profiles was taken at different speeds with varying load conditions. Sequentially, the signals are retrieved for analysis.

Figure 7.1 Cylinder Displacement Measurement Arrangements
7.2 TIME - FREQUENCY ANALYSIS USING FOURIER TRANSFORM OF AUTOCORRELATION

The most commonly used device for vibration measurement is the piezoelectric accelerometer, which gives an electric signal proportional to the vibration acceleration. This signal can readily be amplified, analysed, displayed and recorded. The engine was treated as elastic system with effective damping. The engine was supported by an elastic system of stiffness \( k \) and effective viscous damping of coefficient \( c \). The amplitude \( u \) is given as (Beards, 1995)

\[
u = A' \left( \frac{\omega}{\omega_0} \right)^2 \sqrt{ \left[ 1 - \left( \frac{\omega}{\omega_0} \right)^2 \right]^2 + \left[ 2 \zeta \left( \frac{\omega}{\omega_0} \right)^2 \right] } \tag{7.1}
\]

The spectral density \( S(\omega) \) of a stationary random process is the Fourier transform of the autocorrelation function \( R(\tau) \). It is given by

\[
S(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} R(\tau) e^{-j\omega \tau} d\tau \tag{7.2}
\]

The inverse, which also holds true, and yields FTAF

\[
R(\tau) = \int_{-\infty}^{\infty} S(\omega) e^{-j\omega \tau} d\omega \tag{7.3}
\]

If \( \tau=0 \); gives the mean square value of random processes spectral density.

\[
R(0) = \int_{-\infty}^{\infty} S(\omega) d\omega = \mathbb{E}[x^2] \tag{7.4}
\]

The engine in-cylinder pressure correlation with the displacement of the engine block in the radial direction was taken from the following theoretical Lame’s equations using the in-cylinder pressure/indicated mean
effective pressure $P$, inner radius $R_1$, outer radius $R_2$ and at any radius $r$ (Timoshenko 1940).

\[
A = P(R_1^2)/(R_2^2 - R_1^2) \tag{7.5}
\]

\[
B = P(R_1^2 R_2^2)/(R_2^2 - R_1^2) \tag{7.6}
\]

\[
\sigma_c = (B/r^2) + A \tag{7.7}
\]

\[
\sigma_r = (B/r^2) - A \tag{7.8}
\]

\[
\sigma_\alpha = P(R_1^2)/(R_2^2 - R_1^2) \tag{7.9}
\]

\[
\bar{A} = (\pi D^2/4) \tag{7.10}
\]

\[
\varepsilon = (1/E) \left( \sigma_\alpha - \mu(\sigma_c - \sigma_r) \right) = (\delta r/r) \tag{7.11}
\]

Theoretical displacement $\delta r = \varepsilon r$, but the experimental value measured is given as function of many parameters.

\[
u = f(\varepsilon r, F_r, F_{kc}, F_v) \tag{7.13}
\]

The measured displacement ($u$) and the calculated theoretical displacement $\delta r$ are different. The displacement in radial direction was calculated for all four category piston, and the results are shown in Figures 7.4 to 7.7.
7.3 ENGINE TEST PROGRAMME AND EXPERIMENTAL DATA ACQUISITION

A single cylinder compression ignition engine was used to generate the test data. This direct-injection 4-stroke diesel engine was fitted with fixed valve timing and speed control. The engine was coupled with a mechanical dynamometer. Knock sensor was used to measure engine block vibrations. The engine specifications are tabulated in Table 7.1.

Table 7.1 Specification of the engine

<table>
<thead>
<tr>
<th>Engine Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of Cylinder (n)</td>
<td>One</td>
</tr>
<tr>
<td>Bore D (mm)</td>
<td>85</td>
</tr>
<tr>
<td>Stroke L (mm)</td>
<td>110</td>
</tr>
<tr>
<td>Maximum Speed N (rpm)</td>
<td>1500</td>
</tr>
<tr>
<td>Connecting rod length (mm)</td>
<td>235</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18:1</td>
</tr>
<tr>
<td>Intake pressure (bar)</td>
<td>1</td>
</tr>
<tr>
<td>Intake temperature (K)</td>
<td>300</td>
</tr>
<tr>
<td>Inner radius of engine block R₁ (mm)</td>
<td>44</td>
</tr>
<tr>
<td>Outer radius of engine block R₂ (mm)</td>
<td>112</td>
</tr>
</tbody>
</table>

7.3.1 Collection of data

Two different sampling methods were available, namely: crank-angle-based sampling and time-based sampling. Crank-based sampling is generally the most suitable for sampling engine test data. Most combustion
events occur at different angular positions of the crank shaft and therefore can easily be located with crank-based sampling. However control over the sampling frequency is not possible, this is determined by the engine-crank speed and by the pulse per revolution of the encoder. Crank-based data sampling is therefore not readily suitable for identifying fixed-step discrete-time mathematical models. Temporal sampling by contrast, involves fixed-time-step sampling controlled as usual by an internal clock, such that the sampling rate is independent of the crank speed. The temporal sampling data is readily suitable for identifying discrete-time mathematical models. For vibration analysis involving combustion events, high sampling-rates are needed for capturing acceleration data. This is because activities such as valve-impact and injector-pulse forcing are high-frequency events. For these reasons, constant time-sampling was used instead of crank-based-sampling. Time sampling was used here for signal data acquisition. Every vibration acceleration pickups, the device store 300 wave forms and data. The experimental programme was divided into four categories. For the entire category the engine was loaded using dynamometer. The load starting from 0-15 kg in steps of 5 kg, and for each load the speed of the engine has varied from 1100 rpm to 1500 rpm in steps of 100 rpm. First the Double Concave profile piston was fitted in the engine and it was run in normal mode. Second the Hemisphere profile piston was fitted, and the load and speed were varied, then the engine was run in typical way.

Similarly in the third category, the engine was operated with Elliptical profile piston and finally the Ellipsoid piston was fitted with the engine and operated for the normal test conditions. For the above set of categories using Sendig acceleration was picked up through accelerometer and stored in data collector as shown Figure 7.2. MCME software was used for analysing the collected data for various correlations. All four pistons are shown in Figure 7.3. For entire categories the performance parameters were
recorded and Equations (7.5) to (7.13) were used for calculating the displacements of engine block in radial direction.

Figure 7.2 Engine experimental setup with Sendig
7.4 RESULTS AND DISCUSSION

Experimental values for all four categories were taken, and analyzed for the pressure contribution, Fourier Transform Autocorrelation Function (FTAF) and the combined contribution of moving masses to displacement. The results of the FTAF analysis and displacement correlation are explained in the sub sections.
7.4.1 Double Concave Profile Piston Fitted Engine

The contribution of in-cylinder pressure induced displacement and experimental displacement is shown in Figure 7.4, and the experimental values follow the same trend as the theoretical values. Similarly the comparison between theoretical and experimental displacement values for Hemisphere profile piston fitted engine, Elliptical profile piston fitted engine and Ellipsoid profile piston fitted engine were shown in Figures 7.5, 7.6 and 7.7 respectively, and explained in the subsequent sections. The inlet valve, exhaust valve and fuel injection are common for all the four type of piston fitted engines. Theoretical and experimental displacement of Double Concave piston engine was tabulated in Table 7.2 and 7.3. Deviation of experimental values from the theoretical values is due to the unbalanced reciprocating forces in the engine ($F_r$), thrust force exerted by all connecting kinematic chains in the engine ($F_{kc}$) and valves impact ($F_v$). At low speed the fuel consumption was high and part of the power produced was exerting forces on the side walls of the cylinder. Autocorrelation is the cross-correlation of a signal with itself. Informally, it is the similarity between observations as a function of the time lag between them. It is a mathematical tool for finding repeating patterns, such as the presence of a periodic signal obscured by noise, or identifying the missing fundamental frequency in a signal implied by its harmonic frequencies. It is often used in signal processing for analyzing functions or series of values, such as time domain signals. The auto correlation function is used to detect a weak recurring signal which may be buried in a truly random noise. However, the auto correlation of this signal, which is a periodic function, predominates since a truly random noise has its auto correlation equal to zero. In Figure 7.8, auto correlation values varying from the mean represent instability in combustion and consequently in the vibration. The main goal is to recover the information related to the combustion phenomena from the engine block vibration. Normally the
damping takes too much time compare than that of the combustion cycle. Every two revolution one combustion cycle occurs. Within the combustion cycle time the damping is not happened. So the next cycle effect comes and accumulated with available signals. Because of the cycle-to-cycle variation, the combustion was dominating in the particular piston engine. At low load the cycle-to-cycle phenomena cause periodic behaviour in combustion timing; together with cylinder deviations this is found responsible for decreasing the operating regime. The combustion from the other cylinders noise sources are identified from Figure 7.9 (a), the peak amplitude marks the combustion at the top dead centre. The other cylinder noise is in between the valley and peak amplitude over the whole engine mapping for the double concave combustion chamber piston engine. The signal shown in Figure 7.9 is the one received from engine with respect to time. This signal wave forms same as those of the signal generated theoretically and was shown in Figures 8.2 to 8.5; this is giving an idea about the crank based signal changes.

Table 7.2 Theoretical Displacement of Double concave profile piston engine

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.550204</td>
<td>0.634617</td>
<td>0.616632</td>
<td>0.601216</td>
<td>0.585184</td>
</tr>
<tr>
<td>5</td>
<td>0.814827</td>
<td>0.899241</td>
<td>0.881255</td>
<td>0.86584</td>
<td>0.849807</td>
</tr>
<tr>
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<td>1.079451</td>
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<td>1.145879</td>
<td>1.130463</td>
<td>1.11443</td>
</tr>
<tr>
<td>15</td>
<td>1.344074</td>
<td>1.428487</td>
<td>1.410502</td>
<td>1.395086</td>
<td>1.379054</td>
</tr>
</tbody>
</table>
Table 7.3 Experimental Displacement of Double concave profile piston engine

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1.62</td>
<td>1.439</td>
<td>1.34</td>
<td>1.587</td>
<td>1.52</td>
</tr>
<tr>
<td>5</td>
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<td>1.579</td>
<td>1.455</td>
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<td>1.859</td>
<td>1.685</td>
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</tbody>
</table>

7.4.2 Hemisphere Profile Piston Fitted Engine

Hemisphere profile pistons are normally used in most internal combustion engines and theoretical displacement at 1200 and 1300 rpm was slightly diverging, as represented in Figure 7.5. Fourier transformed autocorrelation analysis was shown in Figure 7.8(b). The displacement amplitude recorded was 0.74μm, and is one of the highest amplitude received from the engine through vibration pick up. The combustion phenomenon in the chamber is adverse to produce a constructive pressure increase in the cylinder. But it is used to increase the unbalanced excess forces on the kinematic chain, consequently it produces the vibration and then the amplitude is increased. An off-line time-frequency analysis of the cylinder pressure and the resulting knock signal determines the frequency range where the information about the combustion can be extracted was shown in Figures 7.9 and 7.10. One can observe in the first angular-frequency represents the pressure signal within the same combustion cycle where the combustion process occurs.
Figure 7.4 Displacement of Double concave profile piston engine.

The low frequency component may be related to the auto ignition and appears a few crank angle degrees before the top dead centre for every combustion cycle. The spectrum has the lower amplitude and is related to the slow increase of the cylinder pressure after the auto ignition. From Figure 7.5 the combustion chamber pressure induced displacement amplitude is depicted; it increases rapidly to the experimental value. These effects are due to the nature of combustion pressure generated in-cylinder. As per the piston profile, the flow field varies as shown in Figures. 6.3 to 6.4. The better flow
producing good air-fuel mixing and consequently the combustion too become a best one. It results the pressure rise in the cylinder and the related displacement.

**Table 7.4 Theoretical Displacement of Hemisphere profile piston engine**

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
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<td>0.582998</td>
<td>0.668018</td>
<td>0.665963</td>
<td>0.629846</td>
<td>0.614577</td>
</tr>
<tr>
<td>5</td>
<td>0.847621</td>
<td>0.932641</td>
<td>0.930586</td>
<td>0.894469</td>
<td>0.8792</td>
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<tr>
<td>10</td>
<td>1.112244</td>
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<tr>
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<td>1.408446</td>
</tr>
</tbody>
</table>

**Table 7.5 Experimental Displacement of Hemisphere profile piston engine**

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
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<tr>
<td>10</td>
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<td>1.526</td>
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<td>1.564</td>
<td>1.75</td>
<td>1.641</td>
<td>1.687</td>
<td>1.605</td>
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</tbody>
</table>
7.4.3 Elliptical Profile Piston Fitted Engine

The Elliptical profile piston engine exhibits different vibration behaviour, as shown in Figure 7.6. Theoretical value of displacement at 1400 and 1500 rpm are nearly drawn-out at the same point, the vibration amplitude due to combustion in cylinder pressure are stabilised in this speed. From Figure 7.8(a), the vibration velocity vector was in opposite direction and it will increase the rubbing action between the cylinder liner and compression rings. The amplitude of the vibration was shown in Figure 7.9(b) and is higher.
than that of any other profile. Time-domain value in Figure 7.10(b) gave the
displacement amplitude as cyclic variation with time. The energy released by
the combustion is the estimated source of vibration. This corresponds to the
squared sum of Fourier coefficients. Auto correlation the other source of noise
was nearly equivalent to the noise source in the Double Concave piston
engines, but its pressure generation inside the cylinder is better than the
Double Concave and hemisphere profile piston engine. The theoretical
displacement variations for speed and load were very close to each other,
indicating that the energy release and combustion of the engine was uniform
and follows quasi-adiabatic heat release. This pressure becomes the source of
vibration and noise.

Table 7.6 Theoretical Displacement of Elliptical profile piston

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm µm</th>
<th>Displacement at 1200 rpm µm</th>
<th>Displacement at 1300 rpm µm</th>
<th>Displacement at 1400 rpm µm</th>
<th>Displacement at 1500 rpm µm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
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<td>0.68</td>
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</tr>
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<td>1.46</td>
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<td>1.52</td>
</tr>
</tbody>
</table>

Table 7.7 Experimental Displacement of Elliptical profile piston engine

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm µm</th>
<th>Displacement at 1200 rpm µm</th>
<th>Displacement at 1300 rpm µm</th>
<th>Displacement at 1400 rpm µm</th>
<th>Displacement at 1500 rpm µm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1.46</td>
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<td>1.65</td>
<td>1.55</td>
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<td>1.98</td>
<td>2.04</td>
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<td>2.08</td>
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</table>
7.4.4 Ellipsoid Profile Piston Fitted Engine

In-cylinder pressure contribution to the displacement for ellipsoid profile piston engine was shown in Figure 7.7, at low speed amplitude was minimum value and it was increased with speed as well as load. Energy release and combustion phenomena are constantly increasing for the load as well as speed. The ellipsoid profile piston engine exhibits the habitual changes for entire range of operating conditions. Auto correlation of this

Figure 7.6 Displacement of Elliptical profile piston engine
profile gave smooth variation, and the fluctuation was very small compared to the other three piston profile fitted engine depicted in Figure 7.8(c). The noise production in the cylinder due to stable combustion is less. The amplitude was smaller as depicted in Figures 7.9 and 7.10 because of the stable combustion, the unbalance forces acting on the reciprocating elements, kinematic chain and rotary elements are reduced. Oscillation of the Ellipsoid piston engine has stable motion over long combustion cycle as shown in Figure 7.8, but the rest of piston engines, oscillation decay for long period and growth for long period shows uncertainty of combustion cycle.

Table 7.8 Theoretical Displacement of Ellipsoid profile piston engine

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
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<td>0.701419</td>
<td>0.678295</td>
<td>0.652749</td>
<td>0.614577</td>
</tr>
<tr>
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</tr>
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<td>1.408446</td>
</tr>
</tbody>
</table>

Table 7.9 Experimental Displacement of Ellipsoid profile piston engine

<table>
<thead>
<tr>
<th>Load kg</th>
<th>Displacement at 1100 rpm μm</th>
<th>Displacement at 1200 rpm μm</th>
<th>Displacement at 1300 rpm μm</th>
<th>Displacement at 1400 rpm μm</th>
<th>Displacement at 1500 rpm μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1.582</td>
<td>1.549</td>
<td>1.683</td>
<td>1.32</td>
<td>1.448</td>
</tr>
<tr>
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<td>1.667</td>
<td>1.624</td>
<td>1.758</td>
<td>1.425</td>
<td>1.538</td>
</tr>
<tr>
<td>10</td>
<td>1.752</td>
<td>1.699</td>
<td>1.833</td>
<td>1.53</td>
<td>1.628</td>
</tr>
<tr>
<td>15</td>
<td>1.837</td>
<td>1.774</td>
<td>1.908</td>
<td>1.635</td>
<td>1.718</td>
</tr>
</tbody>
</table>
In the time-domain analysis the peak value is lower than the other four major peak values are occurred, that is equivalent to full combustion cycle. Like other engines the cycle-to-cycle variation was not affecting this type engine.

Figure 7.7 Displacement of Ellipsoid profile piston engine
Figure 7.8  Comparison of auto-correlation analysis of engine Double Concave piston (a) Elliptical (b) Hemisphere (c) Ellipsoid

Figure 7.9  Amplitude spectrum of engine with varies pistons (a) Double Concave (b) Elliptical  (c) Hemisphere (d) Ellipsoid
Figure 7.10 Time-domain wave form of engine with varies pistons
(a) Double Concave (b) Elliptical (c) Hemisphere (d) Ellipsoid

Figure 7.11 Comparison of wave analysis of engine Double Concave piston (a) Elliptical (b) Hemisphere (c) Ellipsoid
In the wave analysis the peak mode represents the high vibration point and the combustion at top dead centre as shown in Figure 7.11. The wave form analysis for all four pistons is windowed for a particular period and same was shown in Figure 7.11. It is a fraction of the full signal received from the engine. If consider the double concave piston engine, the peak is once in two combustion cycle, because of the accumulation of the cycle-to-cycle pressure variation in the cylinder and auto-ignition or the entrapped gases from the earlier cycle. The narrow band frequencies exist in the ellipsoid piston engine, and sudden localised lower-intensity broad band responses caused by motorised cylinder pressure. The Ellipsoid piston engines were superior in amplitude modulation; it has minimum cycle-to-cycle in-cylinder variation and the vibration amplitude and noise. The inlet valve, exhaust valve and fuel injection impacts are common for all the combustion chamber profile engines researched at this juncture.