CHAPTER 3

ANALYSIS OF EMISSION LEVEL AND ENGINE PERFORMANCE FOR DIESEL AND LPG

In this investigation, the application of CFD in the LPG/air and air mixture formation in a dual fuelled engine has been compared and the mixing pattern has been studied. Normally LPG/air is inducted through a gas jet to the inlet manifold of the engine which is placed near the engine, maintaining an angle of inclination to the air-flow direction. The variation of the jet angle changes the performance of the engine. CFD software has been used to solve this problem and is validated with experimental results. The results show that the optimized jet parameters by CFD package show slight improvement in engine performance and good reduction in NOx emission besides the normal performance of induction.

3.1 COMPUTATIONAL FLUID DYNAMICS (CFD) ANALYSIS

An attempt has been adopted on flow analysis of the LPG & air mixture inside the inlet duct of 5.2 kW HP diesel engine. The flow analysis has been carried out in FLUENT 6.3 and ANSYS FLUENT 14.0. Different cases of LPG passing through the duct are taken for analysis. The induction pipe L Bend with cylinder 3D view is shown in Figure 3.1. The effect of turbulence created by the mixture of LPG & air are analyzed as per the procedure. The meshed induction pipe L Bend with cylinder 3D view is shown in Figure 3.2.
Figure 3.1 Induction pipe L bend with cylinder 3D view

Figure 3.2 Meshed Induction pipe L bend with cylinder 3D view
3.2 SIMULATED RESULTS OF DIFFERENT CASES

Different cases ‘a’, ‘b’, ‘c’ and ‘d’ were modeled and these designs were analyzed using FLUENT. Two models were found to be best and these are implemented practically to see the performance of the engine. In the normal case the induction duct of LPG is placed along the direction of flow of air and the LPG is given out at the starting of the inlet duct. Turbulence for normal case is shown in Figure 3.3

The turbulence created was less and an alternative arrangement for better turbulence was sought for.

![Figure 3.3 Turbulence for normal case](image-url)

Case (a)

The induction tube of LPG is placed at the surface level of the inlet L bend. The expected turbulence was not created in this case. Next a curved slot is pasted on the L bend in front of the induction LPG tube, still the expected turbulence was not created.
Case (b)

The induction tube is placed without piercing into the L bend at the center of the curved portion. The turbulence in the walls was less.

Case (c)

In this case slots were placed on the sides on the L bend. The turbulence created was good but this affects the mass flow rate of air which enters the combustion chamber.
Case (d)

In this case the induction duct is pierced into the L bend by 10 mm against the flow of air. Various angle was tried and found that with 30° inlet condition of LPG, the turbulence created was good.
In analyzing all these cases the turbulence produced in case (d) was found to be promising. The turbulence intensity created in these two cases can result in better mixing of LPG and air.

Thus the secondary induction duct for LPG/H$_2$ with 30º, 45º, -180º and 90º angle inclination to the air flow axis was fabricated and these are used for carrying out experiments and the performance of the engine was calculated.

### 3.3 EXPERIMENTAL METHODOLOGY

In this study, a series of experiments is carried out for different fuel ratio of diesel and LPG share as 1:0, 0.8:0.2, 0.6:0.4 and 0.4:0.6 for various ‘d’ values, ‘x’ values and ‘θ’ values for different loading conditions. The experiment is conducted based on four cases as listed below with speed of the engine maintained constant at 1500 rpm. The exhaust gases such as CO, CO2, O2, HC and NOx are examined with the help of separate apparatus called exhaust gas analyzer ‘AVL 437C5’. The experimental setup is shown in Figure 3.8. The nozzle arrangement with air inlet pipe is shown in Figure 3.9. The fabricated secondary fuel duct is shown in Figure 3.10. From the experimental results, better engine performance was observed for the case 60% diesel and 40% LPG share. On the contrary, considerable reduction in emission level of various gases are examined for 40% diesel and 60% LPG share combination for most of the cases which implies that increase in LPG decreases emission level but affects the engine performance for nozzle diameter of 5 mm at x =10 mm subtended angle of 30º in both cases. The experiment involves using two types of fuel for the compression ignition engine, one the common fuel Diesel and the other LPG. The experiment was done for the four models of passing LPG through the inlet duct with the below listed possible combination of conditions and constraints with the speed of the engine maintained constant at 1500 rpm.
Figure 3.8 Schematic of experimental setup

\[ \theta \] – angle between nozzle and vertical axis of air inlet pipe

\[ d \] – inlet diameter of nozzle (Note: outlet diameter of nozzle = \( \frac{1}{5} d \))

\[ x \] – distance between nozzle tip and the intersection of vertical axis of air inlet pipe and elbow.

Figure 3.9 Nozzle arrangement with air induction pipe
The various cases discussed is shown below

**CASE 1: DIESEL**

In this case the engine is run with 100% diesel. These results are mainly used to compare with that of the LPG mixed diesel.

**CASE 2: 80% DIESEL + 20% LPG share**

In this case the engine is tested with the normal condition of the LPG induction jet and 80% DIESEL + 20% LPG share

**CASE 3: 60% DIESEL + 40% LPG share**

In this case the engine is tested with the normal condition of the LPG induction jet and 60% DIESEL + 40% LPG share.
**CASE 4: 40% DIESEL + 60% LPG share**

In this case the engine is tested with the normal condition of the LPG induction jet and 40% DIESEL + 60% LPG share.

The cases are repeated by placing the 30°, 45°,-180° and 90° models of secondary ducts in the L bend of the induction duct. These possible combinations are carried for all the four angle of inclinations of 30°, 45°,-180° and 90° models of secondary ducts were designed and experiments were carried out.

In this case the engine is tested with the normal condition of the LPG induction jet and 80% DIESEL and 20% LPG share.

### 3.3.1 Boundary Conditions

- **Governing equation** – RANS equation (Transport equations)
- **Turbulence Model** – Standard K epsilon.
- **Combustion Model** – Species transport, Eddy Dissipation
- **NO\textsubscript{x} Emission** – Thermal, Prompt & Fuel NO\textsubscript{x}.
- **Injection Model** – Discrete phase, spray model & Droplet.
- **SMD of the droplet** – 0.25 mm
- **Turbulent intensity** – 1 % (Low) & (Slow Incremental)
- **Inlet Condition** – Mass Flow Inlet
- **Flow type** – Steady Flow

The simulated results for 30° inclination secondary fuel duct with 80% Diesel +20% LPG share is shown below in Figure 3.11.
Figure 3.11 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 30°, 80% Diesel + 20% LPG share

The NO\textsubscript{x} emission in the simulated results for 80% diesel + 20% LPG share was found to be 658 ppm. The average temperature produced during combustion was as high as 1053 K which was within the limits of combustion of diesel fuel and LPG fuel. This was achieved when the above mentioned boundary conditions was applied. Contours of mass fraction of LPG shown above validate proper mixing of LPG fuel with air.

The simulated results for 30° inclination secondary fuel duct with 60% Diesel + 40% LPG share is shown below in Figure 3.12.
The NO\textsubscript{x} emission in the simulated results for 60% diesel + 40% LPG share was found to be 529 ppm. The average temperature produced during combustion was as high as 790 K which was within the limits of combustion of diesel fuel and LPG fuel. The figure shows the contours of mass fraction of LPG fuel inside the cylinder which reduced the volumetric efficiency when 40% LPG share was introduced.

The simulated results for 30° inclination secondary fuel duct with 40% Diesel + 60% LPG share is shown below in Figure 3.13
Figure 3.13 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 30°, 40% Diesel + 60% LPG share

The NO\textsubscript{x} emission in the simulated results for 40 % diesel + 60 % LPG share was found to be 702 ppm. The average temperature produced during combustion was as high as 656 K which was within the limits of combustion of diesel fuel and LPG fuel. The mass fraction of LPG fuel increased when 60% LPG share was inducted into the cylinder.

The simulated results for 45° inclination secondary fuel duct with 80% Diesel +20% LPG share is shown below in Figure 3.14.
The NO\textsubscript{x} emission in the simulated results for 80 % diesel + 20 % LPG share was found to be 625 ppm. The average temperature produced during combustion was as high as 1003 K which was within the limits of combustion of diesel fuel and LPG fuel. The contours of mass fraction of LPG fuel show slight density rise when compared with other simulated contours.

The simulated results for 45° inclination secondary fuel duct with 60% Diesel + 40% LPG share is shown below in Figure 3.15.
Figure 3.15 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 45°, 60% Diesel + 40% LPG share

The NO\textsubscript{x} emission in the simulated results for 60% diesel + 40% LPG share was found to be 760 ppm. The average temperature produced during combustion was as high as 853 K which was within the limits of combustion of diesel fuel and LPG fuel. The contours of mass fraction of LPG show an increased density of gaseous fuel when compared with 30° induction simulation.

The simulated results for 45° inclination secondary fuel duct with 40% Diesel + 60% LPG share is shown below in Figure 3.16.
Figure 3.16 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 45°, 40% Diesel + 60% LPG share

The NO\textsubscript{x} emission in the simulated results for 40% diesel + 60% LPG share was found to be 458 ppm. The average temperature produced during combustion was as high as 623 K which was within the limits of combustion of diesel fuel and LPG fuel. The mass fraction contours show increased density of LPG fuel. The volumetric efficiency decreased on par with the fuel concentration.

The simulated results for -180° inclination secondary fuel duct with 80% Diesel +20% LPG share is shown below in Figure 3.17.
Figure 3.17 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for -180°, 80% Diesel + 20% LPG share

The NO\textsubscript{x} emission in the simulated results for 80% diesel + 20% LPG share was found to be 456 ppm. The average temperature produced during combustion was as high as 980 K which was within the limits of combustion of diesel fuel and LPG fuel. The low flame speed of LPG fuel reduces the NO\textsubscript{x} emission. The contours of mass fraction show a proper mixing of gaseous fuel inside the cylinder.

The simulated results for -180° inclination secondary fuel duct with 60% Diesel +40% LPG share is shown below in Figure 3.18.
Figure 3.18 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for -180°, 60% Diesel + 40% LPG share

The NO\textsubscript{x} emission in the simulated results for 60% diesel + 40% LPG share was found to be 359 ppm. The average temperature produced during combustion was as high as 749 K which was within the limits of combustion of diesel fuel and LPG fuel. The gaseous fuel have high diffusivity than diesel fuel, hence the presence of LPG fuel as shown in the contours of mass fraction suggest an uneven density distribution of LPG gas inside the cylinder.
The simulated results for -180° inclination secondary fuel duct with 40% Diesel + 60% LPG share is shown below in Figure 3.19.

![Figure 3.19 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for -180°, 40% Diesel + 60% LPG share](image)

The NO\textsubscript{x} emission in the simulated results for 40% diesel + 60% LPG share was found to be 356 ppm. The average temperature produced during combustion was as high as 686 K which was within the limits of combustion of diesel fuel and LPG fuel. The gaseous fuel have high flame velocity than diesel fuel, hence the presence of LPG fuel as shown in the contours of mass fraction suggest a proper burning of LPG gas inside the cylinder. This led to the decrease of NO\textsubscript{x} emission.
The simulated results for 90° inclination secondary fuel duct with 40% Diesel + 60% LPG share is shown below in Figure 3.20.

![Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 90°, 80% Diesel + 20% LPG share](image)

**Figure 3.20** Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 90°, 80% Diesel + 20% LPG share

The NO\textsubscript{x} emission in the simulated results for 80% diesel + 20% LPG share was found to be 542 ppm. The average temperature produced during combustion was as high as 954 K which was within the limits of combustion of diesel fuel and LPG fuel. The LPG gas has unstable flame at lean gaseous fuel, hence while inducted through 90° induction duct, the
stochiometric ratio is not evenly distributed inside the cylinder which in turn has influenced the emissions.

The simulated results for 90° inclination secondary fuel duct with 60% Diesel + 40% LPG share is shown below in Figure 3.21.

![Simulated results](image)

**Figure 3.21 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 90°, 60% Diesel + 40% LPG share**

The NO$_x$ emission in the simulated results for 80 % diesel + 20 % LPG share was found to be 402 ppm. The average temperature produced
during combustion was as high as 718 K which was within the limits of combustion of diesel fuel and LPG fuel. The mass fraction contour suggest that there is low concentration zone and high concentration zone of LPG fuel inside the cylinder due to the mixing influenced by the angle of induction.

The simulated results for 90° inclination secondary fuel duct with 40% Diesel + 60% LPG share is shown below in Figure 3.22.

Figure 3.22 Simulated results of pollutant, mass fraction, static temperature and velocity magnitude for 90°, 40% Diesel + 60% LPG share
The NO\textsubscript{x} emission in the simulated results for 40 % diesel + 60 % LPG share was found to be 289 ppm. The average temperature produced during combustion was as high as 625 K which was within the limits of combustion of diesel fuel and LPG fuel. When compared with the contours of mass fraction of various induction arrangements, it is evident that 90° induction duct facilitated less mixing of LPG fuel and air. The LPG gas retards the reaction intensity and uneven distribution of this fuel has increased the emission comparatively.

3.4 MATERIALS AND EXPERIMENTAL SET UP

The experimental setup consists of a single cylinder, four strokes, water-cooled, Kirloskar TV1 diesel engine (bore 87.5 mm, stroke 110 mm, rated power 5.2 kW at a constant speed of 1500 rpm) coupled to an eddy current dynamometer. Water is used as a coolant for maintaining the engine temperature and the flow rate of water is measured using rota meter. The inlet air flow of the engine is measured using manometer. The detailed engine specification is given in Table 3.1.

During the experiment, different sensors are employed along with the CI engine to measure the cylinder pressure, engine exhaust temperature, coolant inlet temperature and coolant outlet temperature. The rotary encoder is used to measure the engine speed as well as the crank angle position. In addition, the eddy current dynamometer is coupled with the engine to apply various loads. The flow rates of gaseous fuels are measured using flow meters.
Table 3.1 Engine specifications

<table>
<thead>
<tr>
<th>Engine Make &amp; Model</th>
<th>Kirloskar TV1 Diesel Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>1500 rpm (Constant Speed)</td>
</tr>
<tr>
<td>Power</td>
<td>5.2 kW</td>
</tr>
<tr>
<td>Bore Diameter</td>
<td>87.5 mm</td>
</tr>
<tr>
<td>Stroke Length</td>
<td>110 mm</td>
</tr>
<tr>
<td>Cylinder</td>
<td>Single Cylinder</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>17.5</td>
</tr>
<tr>
<td>Capacity</td>
<td>661cc</td>
</tr>
<tr>
<td>Water cooled adapter</td>
<td>For piezo sensor</td>
</tr>
<tr>
<td>Injection Timing</td>
<td>23° before TDC</td>
</tr>
<tr>
<td>Crank angle sensor</td>
<td>Resolution 1°, Speed 5000 rpm, with TDC marker pulse.</td>
</tr>
<tr>
<td>Engine indicator</td>
<td>For data scanning and interfacing, with speed indicator.</td>
</tr>
<tr>
<td>No. of Stroke</td>
<td>Four stroke</td>
</tr>
<tr>
<td>Dynamometer</td>
<td>Eddy Current Dynamometer</td>
</tr>
<tr>
<td>Piezo Sensor</td>
<td>Range 200 bar(pressure)</td>
</tr>
<tr>
<td>Low noise cable</td>
<td>For piezo sensor</td>
</tr>
<tr>
<td>Temperature indicator</td>
<td>Digital, thermocouple type temperature sensor</td>
</tr>
</tbody>
</table>

The engine is interfaced with the computer through an in-built data acquisition system. The fuel properties of diesel, LPG and hydrogen is given in the Table 3.2.1.
Table 3.2 Properties of diesel, LPG and hydrogen

<table>
<thead>
<tr>
<th>S. No</th>
<th>Properties</th>
<th>Diesel</th>
<th>LPG</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Formula</td>
<td>C14.4H24.9</td>
<td>C3.36H8.72</td>
<td>H₂</td>
</tr>
<tr>
<td>2.</td>
<td>Boiling point (K)</td>
<td>436-672</td>
<td>243</td>
<td>20.27</td>
</tr>
<tr>
<td>3.</td>
<td>Viscosity at 15.5°C, centipoises</td>
<td>2.6-4.1</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>4.</td>
<td>Vapour pressure at 38°C (kPa)</td>
<td>Negligible</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>5.</td>
<td>Diffusivity in air (cm²/s)</td>
<td>-</td>
<td>-</td>
<td>0.63</td>
</tr>
<tr>
<td>6.</td>
<td>Octane number</td>
<td>30</td>
<td>-</td>
<td>130</td>
</tr>
<tr>
<td>7.</td>
<td>Calorific value</td>
<td>42500</td>
<td>46380</td>
<td>-</td>
</tr>
<tr>
<td>8.</td>
<td>Flame velocity (cm/s)</td>
<td>30</td>
<td>-</td>
<td>265-325</td>
</tr>
<tr>
<td>9.</td>
<td>Cetane number</td>
<td>40-55</td>
<td>&lt;3</td>
<td>-</td>
</tr>
<tr>
<td>10.</td>
<td>Auto ignition temperature (K)</td>
<td>530</td>
<td>738</td>
<td>858</td>
</tr>
<tr>
<td>11.</td>
<td>Flammability limits (volume % in air)</td>
<td>0.7-5</td>
<td>2.4-9.5</td>
<td>4-75</td>
</tr>
<tr>
<td>12.</td>
<td>Density at 160 °C and 1.01 bar (kg/m³)</td>
<td>833-881</td>
<td>505</td>
<td>0.0838</td>
</tr>
<tr>
<td>13.</td>
<td>Specific gravity</td>
<td>0.83</td>
<td>0.51</td>
<td>0.091</td>
</tr>
</tbody>
</table>

3.4.1 AVL Gas Analyser

The AVL combustion emission bench II is used to measure gaseous emissions. The bench is composed of six gas specific analyzers. Hot exhaust gases are sampled from the engine’s exhaust pipe by headline filters. NOₓ, CO, CO₂ and HC are measured using the AVL testing bench.
3.4.2 AVL Smoke Meter

The opacity of polluted air, in particular diesel exhaust engines has been measured using AVL 437C Smoke meter. The extinction of light between light source and receiver is known as opacity. Photocell is used as receiver with spectral sensitivity tuned to the sensitivity curve of the human eye. In accordance with ECE-R24 ISO 3173 the absorption coefficient is calculated. When the indicator reaches 90% of the full scale with a completely opaque plate being placed in front of the photocell, the response time produced between 0.9 and 1.1 seconds. To check the measurement process and to store such values as pressure, temperature, opacity and absorption, the equipment uses a microprocessor controlled program.

3.5 UNCERTAINTY ANALYSIS

Uncertainties in computed quantities such as speed, load, temperature, mass flow rates of air and fuel, equivalence ratio, brake power, brake thermal efficiency are estimated. The estimated uncertainty values at a typical operating condition are given below:

\[
\begin{align*}
\text{Brake Power} & : \pm 1.053 \% \\
\text{Brake thermal efficiency} & : \pm 1.138 \% \\
\text{Speed} & : \pm 0.1153 \% \\
\text{Temperature} & : \pm 1^\circ \mathrm{C} \\
\text{Load} & : \pm 1.176 \% \\
\text{Mass flow rate of air} & : \pm 0.1968 \% \\
\text{Mass flow rate of fuel} & : \pm 0.3531 \%
\end{align*}
\]
The uncertainties in cylinder pressure measurement, crank angle, HC, CO and NO\textsubscript{X} emission levels are given below:

\begin{itemize}
  \item Cylinder pressure : $\pm 0.8 \%$
  \item Crank Angle : $0.1^\circ$
  \item HC : $\pm 2 \%$
  \item CO : $\pm 2.99 \%$
  \item CO\textsubscript{2} : $\pm 2 \%$
  \item NO\textsubscript{X} : $\pm 0.59 \%$
\end{itemize}

### 3.6 ENGINE PERFORMANCE ANALYSIS

The brake thermal efficiency, brake specific energy consumption, exhaust temperature and volumetric efficiency were calculated to validate the engine’s performance.

#### 3.6.1 Brake Thermal Efficiency

Brake thermal efficiency is an important key parameter to evaluate the performance of internal combustion engines. Figure 3.23 shows the behavior of brake thermal efficiency with varying flow rate of LPG/diesel fuelled engine. The brake thermal efficiency of 100% diesel fuelled engine at 20\% loading condition (1.61 bar BMEP) is 16.68 \%. At 6.43 bar BMEP (full load condition), the brake thermal efficiency is 31.69\%. The maximum BTE of 32.23\% for diesel is obtained at 80\% loading condition at 5.47 bar BMEP.
The secondary duct induction at 30° for 40% loading with liquefied petroleum gas (LPG) induction at 60% Diesel + 40% LPG share increased the brake thermal efficiency compared with base diesel operation by 0.36. The brake thermal efficiency for 20% load was 16.84%, 17.04% and 16.80% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 40% load was 22.68%, 23.12% and 22.87% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 60% load was 29.50%, 31.56% and 28.67% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 80% load was 32.87%, 32.93% and 32.78% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 100% load was 31.72%, 32.54% and 31.89% at 20%, 40% and 60% LPG share respectively as shown in Figure 3.23. The rise in brake thermal efficiency at higher loads are due to the presence of rich gaseous fuel-air mixture. The higher flame velocity of the gaseous fuel takes less time to reach the cylinder walls and
hence less time is available for heat transfer to the cylinder walls. Therefore heat losses are less, resulting in higher brake thermal efficiency.

![Figure 3.24 BMEP Vs BTE for diesel with LPG induction at 45°](image)

The secondary duct induction at 45° for 80% loading with liquefied petroleum gas (LPG) induction at 60% Diesel + 40% LPG share increased the brake thermal efficiency compared with base diesel operation by 0.73. The brake thermal efficiency for 20% load was 16.71%, 17.06% and 16.63% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 40% load was 22.50%, 22.79% and 22.64% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 60% load was 29.27%, 30.54% and 28.39% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 80% load was 32.61%, 32.96% and 32.46% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 100% load was 31.47%, 32.57% and 31.57% at 20%, 40% and 60% LPG share respectively as shown in Figure 3.24. The brake thermal efficiency when compared with
secondary duct induction at 30° is comparatively less due to the homogeneity of the gaseous fuel-air mixture formation. All other parameters are similar except the secondary duct induction angle.

![Figure 3.25 BMEP Vs BTE for diesel with LPG induction at -180°](image)

Figure 3.25 BMEP Vs BTE for diesel with LPG induction at -180°

The secondary duct induction at -180° for 80% loading with liquefied petroleum gas (LPG) induction at 60% Diesel + 40% LPG share increased the brake thermal efficiency compared with base diesel operation by 0.38. The brake thermal efficiency for 20% load was 16.57%, 16.98% and 16.47% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 40% load was 22.32%, 22.81% and 22.42% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 60% load was 29.03%, 30.57% and 29.43% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 80% load was 32.35%, 32.61% and 32.13% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 100% load was 31.22%, 31.67% and 31.45% at 20%, 40% and 60% LPG share respectively.
share respectively as shown in Figure 3.25. The brake thermal efficiency when compared with secondary duct induction at 45° is comparatively less due to the homogeneity of the gaseous fuel-air mixture formation.

![Figure 3.26 BMEP Vs BTE for diesel with LPG induction at 90°](image)

**Figure 3.26 BMEP Vs BTE for diesel with LPG induction at 90°**

The secondary duct induction at 90° for 80% loading with liquefied petroleum gas(LPG) induction at 60% Diesel + 40% LPG share increased the brake thermal efficiency compared with base diesel operation by 0.43. The brake thermal efficiency for 20% load was 16.44%, 16.98% and 16.31% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 40% load was 22.14%, 22.86% and 22.20% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 60% load was 29.12%, 29.58% and 29.48% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 80% load was 32.09%, 32.66% and 31.82% at 20%, 40% and 60% LPG share respectively. The brake thermal efficiency for 100% load was 30.97%, 31.89% and 31.48% at 20%, 40% and 60% LPG share respectively.
as shown in Figure 3.26. The brake thermal efficiency when compared with secondary duct induction at -180° is comparatively less due to the homogeneity of the gaseous fuel-air mixture formation. All other parameters are similar except the secondary duct induction angle.

3.6.2 Brake Specific Energy Consumption

In this discussion the term brake specific energy consumption is used instead of brake specific fuel consumption as more than one fuel is used in the engine. The brake specific energy consumption for diesel fuelled engine at initial 20% load (1.61 bar BMEP) was 21582.73 kJ/kWh. The specific energy consumption decreased with load to 15915.12 kJ/kWh, 12653.77 kJ/kWh and 11169.71 kJ/kWh for 40%, 60% and 80% load respectively. The energy consumption of diesel steadily increased to 11360.05 kJ/kWh at 100% load (6.44 bar BMEP).

![Figure 3.27 BMEP Vs BSEC for diesel with LPG induction at 30°](image)

Figure 3.27 BMEP Vs BSEC for diesel with LPG induction at 30°
The energy consumption for 20%, 40% and 60% LPG share at 20% load was 21377.67 kJ/kWh, 21126.76 kJ/kWh and 21428.57 kJ/kWh respectively. The energy consumption at 80% load (5.47 bar BMEP) was 10952.23 kJ/kWh, 10932.28 kJ/kWh and 10982.31 kJ/kWh for 20%, 40% and 60% LPG share induction respectively. The energy consumption was 11249.31 kJ/kWh, 11063.31 kJ/kWh and 11288.81 kJ/kWh for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.27. The engine, fuelled with LPG, consumed more energy compared with diesel as the engine was basically designed for diesel. The reason for reduction of brake specific energy consumption was complete combustion occurred in dual fuel operation of LPG induction. The brake specific energy consumption was lowered in 60% diesel +40% LPG energy share induction compared to diesel engine operation due to high calorific value of LPG fuel.

Figure 3.28 BMEP Vs BSEC for diesel with LPG induction at 45°
The energy consumption for 20%, 40% and 60% LPG share at 20% load was 21548.69 kJ/kWh, 21105.63 kJ/kWh and 21642.86 kJ/kWh respectively. The energy consumption at 80% load (5.47 bar BMEP) was 11039.85 kJ/kWh, 10921.35 kJ/kWh and 11092.13 kJ/kWh for 20%, 40% and 60% LPG share induction respectively. The energy consumption was 11440.10 kJ/kWh, 11052.24 kJ/kWh and 11401.69 kJ/kWh for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.28. The reason for reduction of brake specific energy consumption was complete combustion occurred in dual fuel operation of LPG induction and high calorific value of LPG fuel.

![Figure 3.29 BMEP Vs BSEC for diesel with LPG induction at -180°](image)

**Figure 3.29 BMEP Vs BSEC for diesel with LPG induction at -180°**

The energy consumption for 20%, 40% and 60% LPG share at 20% load was 21521.08 kJ/kWh, 21201.41 kJ/kWh and 21859.28 kJ/kWh respectively. The energy consumption at 80% load (5.47 bar BMEP) was
11128.17 kJ/kWh, 11039.56 kJ/kWh and 11203.05 kJ/kWh for 20%, 40% and 60% LPG share induction respectively. The energy consumption was 11531.62 kJ/kWh, 11367.22 kJ/kWh and 11446.74 kJ/kWh for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.29. The reason for reduction of brake specific energy consumption was due to higher intake charge density produced by the LPG mixture.

![Figure 3.30 BMEP Vs BSEC for diesel with LPG induction at 90°](image)

The energy consumption for 20%, 40% and 60% LPG share at 20% load was 21494.85 kJ/kWh, 21201.41 kJ/kWh and 22077.88 kJ/kWh respectively. The energy consumption at 80% load (5.47 bar BMEP) was 11117.20 kJ/kWh, 11039.56 kJ/kWh and 10915.08 kJ/kWh for 20%, 40% and 60% LPG share induction respectively. The energy consumption was 11023.87 kJ/kWh, 10988.81 kJ/kWh and 10446.74 kJ/kWh for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.30. The slight increase in brake specific energy consumption was
due to the decrease in oxygen content when compared with other angles of induction.

### 3.6.3 Exhaust Temperature

Exhaust temperature is produced as a result of combustion in the engine. It depends on the type of fuel used in the engine and also varies with load. The results obtained for exhaust temperature is discussed here. The exhaust temperature for pure diesel fuelled engine is 183°C at no load condition. It steadily increases to 280°C, 384°C, 491°C, 541°C and 634°C for 20%, 40%, 60%, 80% and 100% loads respectively.

![Figure 3.31 BMEP Vs Exhaust temperature for diesel with LPG induction at 30°](image_url)

The exhaust temperature for 20%, 40% and 60% LPG share at 20% load was 177°C, 178°C and 180°C respectively. The exhaust temperature at 80% load (5.47 bar BMEP) was 497°C, 519°C and 544°C for 20%, 40% and
60% LPG share induction respectively. The exhaust temperature was 588°C, 613°C and 632°C for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.31. The exhaust temperature slowly climbed up due to the fast rate of combustion due to proper mixture of air and LPG fuel.

![Figure 3.32 BMEP Vs Exhaust temperature for diesel with LPG induction at 45°](image)

The exhaust temperature for 20%, 40% and 60% LPG share at 20% load was 179°C, 181°C and 182°C respectively. The exhaust temperature at 80% load (5.47 bar BMEP) was 502°C, 529°C and 549°C for 20%, 40% and 60% LPG share induction respectively. The exhaust temperature was 594°C, 626°C and 638°C for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.32. The minor decline in the exhaust temperature when compared with 30° inclination duct is due the turbulence factor and gaseous fuel air entrainment into the pilot diesel spray zone.
Figure 3.33 BMEP Vs Exhaust temperature for diesel with LPG induction at -180°

The exhaust temperature for 20%, 40% and 60% LPG share at 20% load was 181°C, 185°C and 183°C respectively. The exhaust temperature at 80% load (5.47 bar BMEP) was 507°C, 541°C and 551°C for 20%, 40% and 60% LPG share induction respectively. The exhaust temperature was 601°C, 623°C and 640°C for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.33. LPG presence, gave fast and efficient combustion, so when the piston run down in power stroke, all the fuel had been burned, and all burned gases would be cooled in this stroke before exhaust valve opened, emitting lower exhaust gas temperatures.
The exhaust temperature for 20%, 40% and 60% LPG share at 20% load was 181°C, 187°C and 183°C respectively. The exhaust temperature at 80% load (5.47 bar BMEP) was 509°C, 547°C and 552°C for 20%, 40% and 60% LPG share induction respectively. The exhaust temperature was 602°C, 630°C and 642°C for 100% load (6.44 bar BMEP) for 20%, 40% and 60% LPG share respectively as shown in Figure 3.34. The turbulence factor created in the LPG air mixture plays a vital role in the presence of LPG in the fuel burned.

Figure 3.34 BMEP Vs Exhaust temperature for diesel with LPG induction at 90°

3.6.4 Energy Share

The energy sharing is used to determine the amount of energy produced by the fuels. It is calculated to determine the contribution of various fuels during combustion. The fuel sources used are diesel and LPG.
Table 3.3 Energy sharing LPG and diesel

<table>
<thead>
<tr>
<th>S.No</th>
<th>Load (%)</th>
<th>Energy Share</th>
<th>Total Energy Share</th>
<th>Percentage of Energy Share of Diesel</th>
<th>Percentage of Energy Share of LPG</th>
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<td>40</td>
<td>3.4</td>
<td>0.84</td>
<td>4.3</td>
<td>80.3</td>
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<tr>
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<td>60</td>
<td>2.2</td>
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<tr>
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<tr>
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<td>100</td>
<td>0.6</td>
<td>0.84</td>
<td>1.4</td>
<td>41.2</td>
</tr>
</tbody>
</table>

3.6.5 Volumetric Efficiency

This important parameter that decides the performance of a four-stroke engine is volumetric efficiency. The parameter volumetric efficiency defines the breathing ability of an engine. Volumetric efficiency is defined as the ratio of actual volume flow rate of air into the intake system to the rate at which the volume is displaced by the system. It has to be pointed out here that volumetric efficiency is the volumetric rate of air flow and not the volumetric mixture flow. Hence, it can also be described as the ratio of the actual mass of air sucked into the engine during a given period of time to the theoretical mass. It must have been sucked into it during that same period of time, based on the total piston displacement of the engine, and the temperature and pressure of the surrounding atmosphere.
The Figure 3.35 shows that the volumetric efficiency of diesel fuelled engine at 20% loading condition was 77.43%. In 80% load condition volumetric efficiency was 76.05%. The volumetric efficiency was 75.58% for diesel operated engine at full load condition. LPG is inducted into the air intake manifold to improve the engine performance. The volumetric efficiency obtained for 80% loading condition at 20% LPG share was 75.31%. At 40% LPG share the volumetric efficiency reduced to 73.34% at 80% loading condition. Further induction of 60% LPG share reduced the volumetric efficiency to 77.33%, 76.14%, 74.23%, 73.51%, 73.02% and 72.86% at no load, 20%, 40%, 60%, 80% and 100% loading respectively. LPG induction reduced the volumetric efficiency compared with diesel fuelled engine owing to more dense than hydrogen gas and occupied the portion of fresh air as illustrated in Figure 3.35.
LPG is inducted into the air intake manifold to improve the engine performance. The volumetric efficiency obtained for 80% loading condition at 20% LPG share was 75.24%. At 40% LPG share the volumetric efficiency reduced to 74.23% at 80% loading condition. Further induction of 60% LPG share reduced the volumetric efficiency to 77.62%, 76.23%, 74.52%, 74.18%, 73.62% and 72.54% at no load, 20%, 40%, 60%, 80% and 100% loading respectively. LPG induction reduced the volumetric efficiency compared with diesel fuelled engine owing to more dense than hydrogen gas and occupied the portion of fresh air thereby reducing the volumetric efficiency. The significance of angle of induction was not traceable as illustrated in Figure 3.36.

Figure 3.36 BMEP vs volumetric efficiency for diesel with LPG induction at 45°
Figure 3.37 BMEP vs volumetric efficiency for diesel with LPG induction at -180°

LPG is inducted into the air intake manifold to improve the engine performance. The volumetric efficiency obtained for 80% loading condition at 20% LPG share was 75.16%. At 40% LPG share the volumetric efficiency reduced to 73.98% at 80% loading condition. Further induction of 60% LPG share reduced the volumetric efficiency to 77.19%, 76.01%, 74.10%, 74.05%, 73.45% and 72.76% at no load, 20%, 40%, 60%, 80% and 100% loading respectively. LPG induction slightly reduced the volumetric efficiency compared with diesel fuelled engine owing to more dense than hydrogen gas and occupied the portion of fresh air, but less significance was observed with the change in angle of inclination of the induction duct as illustrated in Figure 3.37.
LPG is inducted into the air intake manifold to improve the engine performance. The volumetric efficiency obtained for 80% loading condition at 20% LPG share was 75.16%. At 40% LPG share the volumetric efficiency reduced to 73.98% at 80% loading condition. Further induction of 60% LPG share reduced the volumetric efficiency to 77.19%, 76.01%, 74.10%, 74.05%, 73.45% and 72.76% at no load, 20%, 40%, 60%, 80% and 100% loading respectively. Moderate change in volumetric efficiency was observed for 40% diesel and 60% LPG share at 90° inclination of induction duct. Since the density of LPG gas was roughly 500 times more than air the maximum reduction was obtained only at 40% diesel and 60% LPG share as illustrated in Figure 3.38.
3.7 CYLINDER PRESSURE AND HEAT RELEASE ANALYSIS

3.7.1 Pressure Release Rate

The combustion in CI engine occurs in four different phases as mentioned by (Thipse 2008):

- It is also called the preparatory phase. The first phase is ignition delay period and during which the fuel is injected into the cylinder but not ignited.

- The heat release rate is maximum during this period. The second phase is the uncontrolled combustion period during which the pressure rise is rapid.

- The third phase is the period of controlled combustion during which the temperature and pressure of the air-fuel mixture are high.

- The final phase is after burning period where unburned or partially burned fuel particles remaining in the combustion chamber start burning when they come in contact with fresh oxygen in the air.
Figure 3.39 Crank angle vs cylinder pressure for diesel with LPG induction at 30°

LPG induction decreased the cylinder pressure and the change in crank angle degree of peak pressure was also observed for varying LPG quantity. The peak pressure obtained for LPG induction was slightly higher than that of diesel. The peak pressure of 63.74 bar for diesel was achieved at a crank angle of 369° at 80% load. When LPG was inducted at an angle of 30° inclination to the air flow through secondary fuel duct the peak pressure raised to 64.23 bar, 64.34 bar and 64.88 bar for 20%, 40% and 60% LPG share at a crank angle of 359° as shown in Figure 3.39. The increase in the peak cylinder pressure at light load is mainly controlled by first phase of combustion namely premixed burning of part or whole of the pilot quantity along with small part of LPG fuel entrained in the diesel spray. While at
higher loads it is generally governed by second phase of combustion, for example, auto-ignition of gaseous fuel-air mixture in the close vicinity of the pilot spray and diffusive burning of the remaining diesel fuel.

The LPG induction reduced the cylinder pressure to 63.94 bar at 359° crank angle for 80% load as shown in Figure 3.40. The peak pressure was slightly increased due to unique mixture formation of LPG with air and finally forming combustible mixture with diesel spray. The cylinder pressure of 63.29 bar, 63.35 bar and 63.94 bar was attained for 20%, 40% and 60% LPG share at a crank angle of 359° when the loading was at 80%. An increase of 0.20 bar was achieved when the LPG and air mixture homogeneity was altered through proper mixing of air and LPG fuel. The good mixing of LPG in air increases the combustible mixture available during the ignition delay period. This leads to increase heat released in the premixed combustion phase.
and increases the peak cylinder pressure at combustion period. These gases will run out at lower temperatures after the exhausts are being cooled at power stroke and at exhaust stroke.

Figure 3.41 Crank angle vs cylinder pressure for diesel with LPG induction at -180°

The LPG induction reduced the cylinder pressure to 64.07 bar at 359° crank angle for 80% load as shown in Figure 3.41. The peak pressure was slightly increased due to unique mixture formation of LPG with air and finally forming combustible mixture with diesel spray. The cylinder pressure of 63.42 bar, 63.36 bar and 64.07 bar was attained for 20%, 40% and 60% LPG share at a crank angle of 359° when the loading was at 80%. An increase of 0.33 bar was achieved when the LPG and air mixture homogeneity was altered. Due to the very high flame temperatures of diesel and LPG, the proper homogeneity of LPG and air combustibles has raised the peak pressures.
The LPG induction reduced the cylinder pressure to 63.94 bar at 359° crank angle for 80% load as shown in Figure 3.42. The peak pressure was slightly increased due to unique mixture formation of LPG with air and finally forming combustible mixture with diesel spray. The cylinder pressure of 63.29 bar, 63.35 bar and 63.94 bar was attained for 20%, 40% and 60% LPG share at a crank angle of 359° when the loading was at 80%. An increase of 0.20 bar was achieved when the LPG and air mixture homogeneity was altered. The higher air velocity and gaseous-air entrainment lead to increase in rate of evaporation of the liquid fuel and gives higher rate of heat release resulting in higher peak pressure in the cylinder.
3.7.2 Heat Release Rate

A simple way of representing the combustion process is by heat release rate In the present work, heat release rate diagrams have been obtained from experimental cylinder pressure-crank angle curve. The heat release rate by first phase of combustion at different LPG substitution at 80% load conditions are shown in Figure 3.43. It is observed that the heat release rate by first phase of combustion is less as compared to base diesel at all load conditions.

![Figure 3.43 Crank angle vs heat release rate for diesel with LPG induction at 30°](image)

This results in more entrainment of homogeneous LPG and air mixture in diesel spray zone. By increase in gaseous fuel substitution, the heat release rate in first phase of combustion decreases at all load conditions. At low load condition a major part of the injected diesel fuel along with some
part of the gaseous fuel may burn in the first phase of combustion. As pilot fuel quantity decreases, the heat release rate also decreases. The 60% of LPG share substitution due to higher ignition delay results in more accumulation of diesel fuel which causes higher heat release rate in first phase of combustion compared to base diesel. The 20% share LPG induction for diesel fuelled engine generates the heat release rate of 95.61 J/deg at 357° crank angle for 80% load as shown in Figure 3.43. The heat release rate for 40% and 60% LPG share at 80% loading is 94.72 J/deg and 93.78 J/deg respectively. The heat release rate was decreased as the LPG substitution was increased due to the above said reason.

![Figure 3.44 Crank angle vs heat release rate for diesel with LPG induction at 45°](image)

The 20% share LPG induction for diesel fuelled engine generates the heat release rate of 95.06 J/deg at 357° crank angle for 80% load as shown in Figure 3.44. The heat release rate for 40% and 60% LPG share at
80\% loading is 94.42 J/deg and 93.18 J/deg respectively. As pilot fuel quantity decreases, the heat release rate also decreases. The pilot quantity increase and decrease because of the ignition delay and due to this factor LPG fuel substitution occurs. Hence high heat release is obtained in the first phase of combustion and in second phase less heat release is obtained.

Figure 3.45 Crank angle vs heat release rate for diesel with LPG induction at -180°

The 20\% share LPG induction for diesel fuelled engine generates the heat release rate of 94.96 J/deg at 357°crank angle for 80\% load as shown in Figure 3.45. The heat release rate for 40\% and 60 \% LPG share at 80\% loading is 94.58 J/deg and 93.32 J/deg respectively. The difference in heat release rate is slightly exhibited due to the change in the pilot fuel quantity. As pilot fuel namely diesel fuel quantity decreases, the heat release rate also decreases.
The 20% share LPG induction for diesel fuelled engine generates the heat release rate of 95.26 J/deg at 357° crank angle for 80% load as shown in Figure 3.46. The heat release rate for 40% and 60% LPG share at 80% loading is 94.02 J/deg and 93.03 J/deg respectively. At low load condition a major part of the injected diesel fuel along with some part of the gaseous fuel may burn in the first phase of combustion. Hence for dual fuel combination the heat release rate is low in the first phase of combustion.
3.8 EXHAUST EMISSION CHARACTERISTICS

3.8.1 Nitrous Oxides Emission

The engine exhaust consists of various gases and NOX is such a hazardous gas. The exhaust gas consists of oxides of nitrogen. The most commonly found element is nitrogen oxide. The other nitrogen-oxygen combinations like NO2 are found in very smaller amounts. The combination of all these gases is grouped as NOX. In NOX, the letter ‘X’ denotes some suitable gases. NOX is the major cause for ozone depletion. The control of NOX from engines is carried out as a major step in fixing emission standards across the globe year by year. NOX is created by the mixing of nitrogen in air. Nitrogen is stable at low temperature, during combustion N2 breaks down into nitrogen monatomic molecule due to higher temperature in the range of 2500 to 3000 K. NOX formation is directly proportional to the higher temperature produced during combustion. NOX formation also depends on pressure and air-fuel ratio. Further, the nitrogen oxide emission increases with the rise in cylinder temperature, oxygen concentration and combustion duration. A detailed discussion on NOX observed is included in this chapter.

The engine fuelled with diesel produces NOX of 37 ppm, 140 ppm, 279 ppm, 480 ppm, 575 ppm and 629 ppm for no load, 20%, 40%, 60%, 80% and 100% loading conditions respectively as shown in Figure 3.47.
LPG is inducted into the air intake manifold to improve the emission parameters. The NO\textsubscript{X} emission obtained for 80% loading condition at 20% LPG share was 45 ppm. At 40% LPG share the NO\textsubscript{X} emission increased to 310 ppm at 80% loading condition. Further induction of 60% LPG share increased the NO\textsubscript{X} emission to 32 ppm, 55 ppm, 164 ppm, 247 ppm, 330 ppm and 420 ppm at no load, 20%, 40%, 60%, 80% and 100% loading respectively. LPG induction reduced the NO\textsubscript{X} emission compared with base diesel fuelled engine owing to the comparatively low temperature produced during combustion.

Figure 3.47 BMEP Vs NO\textsubscript{X} for diesel with LPG induction at 30°
LPG is inducted into the air intake manifold to improve the emission parameters. The NO\textsubscript{X} emission obtained for 80% loading condition at 20% LPG share was 292 ppm. At 40% LPG share the NO\textsubscript{X} emission increased to 311 ppm at 80% loading condition. Further induction of 60% LPG share increased the NO\textsubscript{X} emission to 35 ppm, 59 ppm, 169 ppm, 249 ppm, 328 ppm and 428 ppm at no load, 20%, 40%, 60%, 80% and 100% loading respectively as shown in Figure 3.48. LPG inducted at 45° slightly increased the NO\textsubscript{X} emission compared with 30° LPG inducted engine owing to the comparatively low temperature produced during combustion.
LPG is inducted into the air intake manifold to improve the emission parameters. The NO\textsubscript{X} emission obtained for 80% loading condition at 20% LPG share was 296 ppm. At 40% LPG share the NO\textsubscript{X} emission increased to 315 ppm at 80% loading condition. Further induction of 60% LPG share increased the NO\textsubscript{X} emission to 36 ppm, 57 ppm, 168 ppm, 252 ppm, 333 ppm and 431 ppm at no load, 20%, 40%, 60%, 80% and 100% loading respectively as shown in Figure 3.49. Apart from the normal reduction in NO\textsubscript{X} emission compared with base diesel fuelled engine, slight variation in NO\textsubscript{X} emission was visible in correspondence with the change in the angle of LPG induction.
LPG is inducted into the air intake manifold to improve the emission parameters. The NO\textsubscript{X} emission obtained for 80% loading condition at 20% LPG share was 288 ppm. At 40% LPG share the NO\textsubscript{X} emission increased to 312 ppm at 80% loading condition. Further induction of 60% LPG share increased the NO\textsubscript{X} emission to 37 ppm, 60 ppm, 166 ppm, 246 ppm, 329 ppm and 430 ppm at no load, 20%, 40%, 60%, 80% and 100% loading respectively as shown in Figure 3.50. LPG induction reduced the NO\textsubscript{X} emission compared with base diesel fuelled engine owing to the comparatively low temperature produced during combustion. Increasing LPG proportion increases the ignition delay period, the amount of heat release, and the cylinder gas temperatures. This will contribute to increase NO\textsubscript{x} emission. The emitted NO\textsubscript{x} concentration will be the resultant of these two parameters. Hence the NO\textsubscript{X} emission varies with exhaust temperature which is directly proportional to the homogeneity of the LPG and air mixture.
3.8.2 Hydrocarbon Emission

Hydrocarbon is found in the exhaust gas of the engine and their emission reacts with the atmosphere forming photochemical smog. The hydrocarbon is generally formed due to the split up of large fuel molecules into small equilibrium molecules. The exhaust generally consists of 40% unburnt and 60% reacted components. The hydrocarbon formation depends on the type of fuel and it will vary for different fuel blends. The hydrocarbon formation also depends on the geometry of combustion chamber, incomplete combustion, deposits on walls of the combustion chamber, oil on combustion chamber walls and engine operating parameters.

The hydrocarbon emission of engine fuelled with diesel was 25 ppm, 25 ppm, 23 ppm, 29 ppm, 36 ppm and 38 ppm for no load, 20%, 40%, 60%, 80% and 100% loading conditions respectively.

![Figure 3.51 BMEP Vs HC for diesel with LPG Induction at 30°](image)
The hydrocarbon emission of engine when fuelled with 20% LPG share produced 400 ppm at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the hydrocarbon emission produced was 192 ppm, 103 ppm, 66 ppm, 32 ppm and 28 ppm. The HC emission for 40% LPG share was recorded at 619 ppm, 278 ppm, 113 ppm, 68 ppm, 36 ppm and 24 ppm at 20%, 40%, 60%, 80% and 100% loading condition. The HC emission at 60% share increased to 634 ppm, since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at higher loads. The hydrocarbon emission at 20%, 40%, 60%, 80% and 100% loading condition is 270ppm, 113 ppm, 63 ppm, 36 ppm and 18 ppm for 60% LPG share condition as shown in Figure 3.51. The LPG pre-mixture that is pressed into the cooled crevices during compression stroke are difficult to burn. The LPG air mixture shows swirls when inducted through 30° inclined induction duct. Part of this mixture would not take part in combustion and is accounted for the HC emission.

Figure 3.52  BMEP Vs HC for diesel with LPG Induction at 45°
The hydrocarbon emission of engine when fuelled with 20 % LPG share produced 412 ppm at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the hydrocarbon emission produced was 198 ppm, 106 ppm, 68 ppm, 33 ppm and 29 ppm. The HC emission for 40% LPG share was recorded at 632 ppm, 284 ppm, 115 ppm, 69 ppm, 37 ppm and 24 ppm at 20%, 40%, 60%, 80% and 100% loading condition. The HC emission at 60% share increased to 654 ppm, since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at higher loads. The hydrocarbon emission at 20%, 40%, 60%, 80% and 100% loading condition is 278 ppm, 116 ppm, 65 ppm, 37 ppm and 19 ppm for 60% LPG share condition as shown in Figure 3.52. HC emission increases while retarding the fuel injection timings and it decreases with increasing the LPG proportion in the fuel mixture. When the induction angle was altered the HC emissions increased due to homogeneity of LPG and air mixture. Over-lean and over-rich mixtures will increase exhaust the HC emission.

Figure 3.53 BMEP Vs HC for diesel with LPG Induction at -180°
The hydrocarbon emission of engine when fuelled with 20% LPG share produced 425 ppm at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the hydrocarbon emission produced was 204 ppm, 109 ppm, 70 ppm, 34 ppm and 30 ppm. The HC emission for 40% LPG share was recorded at 645 ppm, 289 ppm, 118 ppm, 71 ppm, 37 ppm and 25 ppm at 20%, 40%, 60%, 80% and 100% loading condition. The HC emission at 60% share increased to 674 ppm, since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at higher loads. The hydrocarbon emission at 20%, 40%, 60%, 80% and 100% loading condition is 287 ppm, 120 ppm, 67 ppm, 38 ppm and 19 ppm for 60% LPG share condition as shown in Figure 3.53. The results show that the effect of LPG in the mixtures on the reduction of HC emission is stronger than that from advancing the fuel injection timing.

![Figure 3.54 BMEP Vs HC for diesel with LPG Induction at 90°](image)

Figure 3.54 BMEP Vs HC for diesel with LPG Induction at 90°
The hydrocarbon emission of engine when fuelled with 20% LPG share produced 429 ppm at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the hydrocarbon emission produced was 206 ppm, 111 ppm, 71 ppm, 34 ppm and 30 ppm. The HC emission for 40% LPG share was recorded at 658 ppm, 295 ppm, 120 ppm, 72 ppm, 38 ppm and 25 ppm at 20%, 40%, 60%, 80% and 100% loading condition. The HC emission at 60% share increased to 688 ppm, since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at higher loads. The hydrocarbon emission at 20%, 40%, 60%, 80% and 100% loading condition is 293 ppm, 123 ppm, 68 ppm, 39 ppm and 20 ppm for 60% LPG share condition as shown in Figure 3.54. The HC emission is directly proportional to the flame speed of the fuel used. Flame propagation rate of diesel is 10.5 cm/s and LPG is 83.7 cm/s. The presence of LPG is various pockets of the cylinder and the fresh charge which comes out due to the overlapping of valves increases the HC emissions.

3.8.3 Carbon-monoxide Emission

Carbon-monoxide is found in exhaust gas of the combustion engine. It is generated with the air fuel equivalence ratio. The fuel has carbon content and if the fuel is not burned properly, it ends up as carbon-monoxide. The fuel mixture also plays a vital part in carbon monoxide formation. Carbon-monoxide is also formed due to poor mixing, local rich region and incomplete combustion. The rich mixture generates maximum carbon-monoxide. The results of carbon-monoxide of the experiment are discussed in this section.
The carbon monoxide emission of engine when fuelled with 20% LPG share produced 0.04 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon monoxide emission produced was 0.05 (% Vol), 0.06 (% Vol), 0.09 (% Vol), 0.13 (% Vol) and 0.15 (% Vol). The CO emission for 40% LPG share was recorded at 0.08 (% Vol), 0.08 (% Vol), 0.09 (% Vol), 0.14 (% Vol) and 0.15 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO emission at 60% share increased to 0.15 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at higher loads. The carbon monoxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 0.08 (% Vol), 0.07 (% Vol), 0.07 (% Vol), 0.1 (% Vol) and 0.15 (% Vol) for 60% LPG share condition as shown in Figure 3.55.
The carbon monoxide emission of engine when fuelled with 20% LPG share produced 0.04 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon monoxide emission produced was 0.05 (% Vol), 0.06(%Vol), 0.09(%Vol), 0.14(%Vol) and 0.16 (% Vol). The CO emission for 40% LPG share was recorded at 0.09 (% Vol), 0.09 (% Vol), 0.10 (% Vol), 0.15 (% Vol) and 0.16 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO emission at 60% share increased to 0.10 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon monoxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 0.09(%Vol), 0.07 (% Vol), 0.07 (% Vol), 0.11(%Vol) and 0.16 (% Vol) for 60% LPG share condition as shown in Figure 3.56. The CO emissions show a reduction when compared with other conditions due to better mixing of LPG fuel and air.
The carbon monoxide emission of engine when fuelled with 20% LPG share produced 0.04 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon monoxide emission produced was 0.06 (% Vol), 0.07(%Vol), 0.10(%Vol), 0.14(%Vol)and 0.17(% Vol). The CO emission for 40% LPG share was recorded at 0.09(% Vol), 0.09 (% Vol), 0.11 (% Vol), 0.17 (% Vol) and 0.18 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO emission at 60% share increased to 0.10 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon monoxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 0.09 (% Vol) , 0.08 (% Vol), 0.08 (% Vol), 0.11(% Vol) and 0.18 (% Vol) for 60 % LPG share condition as shown in Figure 3.57. The CO emission did not show a notable change when compared with other angles of induction of LPG fuel. The density of LPG fuel and the homogeneity of charge mixture plays an important role in the CO emission.
The carbon monoxide emission of engine when fuelled with 20% LPG share produced 0.05 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon monoxide emission produced was 0.06 (% Vol), 0.07 (% Vol), 0.10 (% Vol), 0.15 (% Vol) and 0.17 (% Vol). The CO emission for 40% LPG share was recorded at 0.10 (% Vol), 0.10 (% Vol), 0.12 (% Vol), 0.17 (% Vol) and 0.18 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO emission at 60% share increased to 0.11 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon monoxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 0.10 (% Vol), 0.08 (% Vol), 0.08 (% Vol), 0.12 (% Vol) and 0.18 (% Vol) for 60% LPG share condition as shown in figure 3.58. The formation of CO is related to the mixture concentration and fuel composition. The reason for lower emission is the increased burning temperature which created local turbulence and increased flame velocity.

**Figure 3.58 BMEP Vs CO for diesel with LPG induction at 90°**
3.8.4 Carbon-di-oxide Emission

Carbon-di-oxide is found in the exhaust emission of the engine. CO$_2$ is formed during the combustion process. CO$_2$ is mainly formed due to the reaction of air and carbon content of the fuel due to combustion. The air fuel mixture and engine combustion parameters determine the CO$_2$ formation in the exhaust. CO$_2$ observed in the experiments is discussed in this chapter.

![Figure 3.59 BMEP Vs CO$_2$ for diesel with LPG induction at 30°](image)

The carbon dioxide emission of engine when fuelled with 20 % LPG share produced 0.9 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon dioxide emission produced was 1.4 (% Vol), 1.8 (% Vol), 2.2 (% Vol), 2.4 (% Vol) and 3.2 (% Vol). The CO$_2$ emission for 40% LPG share was recorded at 1.3 (% Vol), 1.6 (% Vol), 2.1 (% Vol), 2.5 (% Vol) and 3.1(% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO$_2$ emission at 60% share increased to 0.8 (% Vol),
since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon dioxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 1.2(% Vol), 1.6 (% Vol), 1.9 (% Vol), 2.4(% Vol) and 3.1 (% Vol) for 60 % LPG share condition as shown in Figure 3.59. LPG existence increased hydrogen to carbon percentage in the fuel, which improved the combustion and reduced CO$_2$ emission.

![Figure 3.60 BMEP Vs CO$_2$ for diesel with LPG induction at 45°](image)

The carbon dioxide emission of engine when fuelled with 20 % LPG share produced 0.9(% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon dioxide emission produced was 1.4(% Vol), 1.9(% Vol), 2.2(% Vol), 2.5(% Vol) and 3.1(% Vol). The CO$_2$ emission for 40% LPG share was recorded at 1.2(%Vol), 1.7(%Vol), 2.1(% Vol), 2.4 (% Vol) and 3.2 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO$_2$ emission at 60% share increased to 0.8 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel
increased in the engine at no load. The carbon dioxide emission at 20%, 40%, 60%, 80% and 100% loading condition is 1.2 (% Vol), 1.7 (% Vol), 1.9 (% Vol), 2.5 (% Vol) and 3.2 (% Vol) for 60 % LPG share condition as shown in Figure 3.60. The proper mixing of LPG with air increased the better combustion of fuel which in turn reduced the CO$_2$ emission.

![Figure 3.61 BMEP Vs CO$_2$ for diesel with LPG induction at -180°](image)

The carbon dioxide emission of engine when fuelled with 20 % LPG share produced 0.9 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon dioxide emission produced was 1.4 (% Vol), 1.8 (% Vol), 2.2 (% Vol), 2.4 (% Vol) and 3.2 (% Vol). The CO$_2$ emission for 40% LPG share was recorded at 1.3 (% Vol), 1.6 (% Vol), 2.1 (% Vol), 2.5 (% Vol) and 3.1 (% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO$_2$ emission at 60% share increased to 0.8 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon dioxide emission at 20%, 40%,
60%, 80% and 100% loading condition is 1.2(% Vol), 1.6 (% Vol), 1.9 (% Vol), 2.4(% Vol) and 3.1 (% Vol) for 60 % LPG share condition as shown in Figure 3.61. The molecular structure of LPG is simpler than that of diesel fuel and can be oxidized more easily, as well as increasing the LPG proportion in the blends decreased the carbon fraction in the blends.

![Figure 3.62 BMEP Vs CO₂ for diesel with LPG induction at 90°](image)

The carbon dioxide emission of engine when fuelled with 20 % LPG share produced 1.0 (% Vol) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the carbon dioxide emission produced was 1.6(% Vol), 2.0(% Vol), 2.4(% Vol), 2.7(% Vol) and 3.3(% Vol). The CO₂ emission for 40% LPG share was recorded at 1.4 (%Vol), 1.7(%Vol), 2.3 (% Vol), 2.7(% Vol) and 3.3(% Vol) at 20%, 40%, 60%, 80% and 100% loading condition. The CO₂ emission at 60% share increased to 0.9 (% Vol), since the engine was designed for diesel, the consumption of diesel fuel increased in the engine at no load. The carbon dioxide emission at 20%, 40%,
60%, 80% and 100% loading condition is 1.3 (% Vol), 1.8 (% Vol), 2.1 (% Vol), 2.7 (% Vol) and 3.3 (% Vol) for 60 % LPG share condition as shown in Figure 3.62. The turbulence created with the induction angle played a difference in homogeneity of LPG air mixture. It decreased with increasing the LPG proportion in the fuel mixture.

3.8.5 Smoke Opacity

Smoke is produced as a product of combustion process. It is the combination of gases, liquid particulates and airborne solids emitted during combustion of fuels in engine. The smoke is basically classified as three types such as grey smoke, blue smoke and white smoke. Smoke causes hazards to the environment and humans. Opacity is the term used to measure smoke. The investigations on smoke opacity in the experiment are plotted below.

![Figure 3.63 BMEP Vs Smoke opacity for diesel with LPG induction at 30°](image)

Figure 3.63 BMEP Vs Smoke opacity for diesel with LPG induction at 30°
The smoke opacity of engine when fuelled with 20% LPG share produced 39% at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the smoke opacity produced was 38%, 37%, 36%, 34% and 30%. The smoke opacity for 40% LPG share was recorded at 21%, 21%, 20%, 19% and 19% at 20%, 40%, 60%, 80% and 100% loading condition. The smoke opacity at 60% share at no load is 20%. The smoke opacity at 20%, 40%, 60%, 80% and 100% loading condition is 19%, 19%, 18%, 18% and 14% for 60% LPG share condition as shown in Figure 3.63. Reason for reduction in smoke emission could be because LPG has a lower carbon/hydrogen ratio.

![Figure 3.64 BMEP Vs Smoke opacity for diesel with LPG induction at 45°](image)

The smoke opacity of engine when fuelled with 20% LPG share produced 49% at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the smoke opacity produced was 47%, 46%, 45%, 43% and
38%. The smoke opacity for 40% LPG share was recorded at 23%, 23%, 22%, 21% and 21% at 20%, 40%, 60%, 80% and 100% loading condition. The smoke opacity at 60% share at no load is 20 %. The smoke opacity at 20%, 40%, 60%, 80% and 100% loading condition is 20%, 19 %, 19 %, 18% and 15 % for 60 % LPG share condition as shown in Figure 3.64. When the homogeneity of LPG air mixture was disturbed there was a change in smoke opacity. No other parameter was altered and this change was due to the combustion of the fuel mixture.

![Figure 3.65 BMEP Vs Smoke opacity for diesel with LPG induction at -180°](image)

Figure 3.65 BMEP Vs Smoke opacity for diesel with LPG induction at -180°

The smoke opacity of engine when fuelled with 20 % LPG share produced 52% at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the smoke opacity produced was 50%, 48%, 47%, 45% and 39%. The smoke opacity for 40% LPG share was recorded at 26%, 25%, 25%, 24% and 23% at 20%, 40%, 60%, 80% and 100% loading condition. The smoke opacity at 60% share at no load is 20 %. The smoke opacity at
20%, 40%, 60%, 80% and 100% loading condition is 20%, 20 %, 19 %, 18% and 15 % for 60 % LPG share condition as shown in Figure 3.65. The comparatively less turbulence created in mixing increased the smoke opacity.

![Figure 3.66 BMEP Vs Smoke opacity for diesel with LPG induction at 90°](image)

The smoke opacity of engine when fuelled with 20 % LPG share produced 55(%) at no load conditions. At 20%, 40%, 60%, 80% and 100% loading condition, the smoke opacity produced was 52%, 51%, 49%, 48% and 42%. The smoke opacity for 40% LPG share was recorded at 29%, 28%, 27%, 27% and 26% at 20%, 40%, 60%, 80% and 100% loading condition. The smoke opacity at 60% share at no load is 20 %. The smoke opacity at 20%, 40%, 60%, 80% and 100% loading condition is 21%, 20%, 20%, 19% and 15 % for 60 % LPG share condition as shown in Figure 3.66. The opacity increased at lower loads due to less combustion and varied when LPG concentrations was increased and the hydrogen / carbon content decreased in the fuel mixture.
3.9 RESULTS AND DISCUSSIONS

The readings recorded were computed and the results were obtained. The analysis on engine performance, pressure release rate, heat release rate and emission characteristics were carried out. A detailed discussion of these analyses is explained in this chapter.

From the Figure 3.23 it is found that the brake thermal efficiency for dual fuel mode is increased by 5% when compared to that of diesel. The mechanical efficiency is slightly high when powered with dual fuel mode. 60% Diesel + 40% LPG share condition with normal case was found better when compared to others probabilities.

When comparing the Normal case with the other models for 80% diesel+ 20 % LPG share condition, the 30° degree inclination of secondary fuel duct was found better when compared to other angle inclination.

It is found that NO\textsubscript{x} reduced by 35 % when 30° degree inclination of secondary fuel duct was used when compared with the normal case. The carbon monoxide value decreased with increase in load.

The secondary duct induction at 30° for 40% loading with liquefied petroleum gas (LPG) induction at 60% Diesel + 40% LPG share increased the brake thermal efficiency compared with base diesel operation by 0.36.

The NO\textsubscript{x} emission in the simulated results for 60 % diesel + 40 % LPG share at 30° secondary duct induction was found to be 529 ppm.

The turbulence produced in the 30° secondary fuel duct inclination produced better results compared with other angle of induction of gaseous fuel.
The peak pressure of 63.74 bar for diesel was achieved at a crank angle of 369° at 80% load. When LPG was inducted at an angle of 30° inclination to the air flow through secondary fuel duct the peak pressure raised to 64.88 bar for 40% Diesel + 60% LPG share at a crank angle of 359°.

The increase in the peak cylinder pressure at light load is mainly controlled by first phase of combustion namely premixed burning of part or whole of the pilot quantity along with small part of LPG fuel entrained in the diesel spray.

The 20% share LPG induction for diesel fuelled engine generates the heat release rate of 95.61 J/deg at 357°crank angle for 80% load as shown in Figure 3.43.

The 60% of LPG share substitution due to higher ignition delay results in more accumulation of diesel fuel which causes higher heat release rate in first phase of combustion compared to base diesel.

LPG induction reduced the volumetric efficiency compared with diesel fuelled engine owing to much less dense than hydrogen gas and occupied the portion of fresh air as illustrated in Figure 3.35.

By increase in gaseous fuel substitution, the heat release rate in first phase of combustion decreases at all load conditions. At low load condition a major part of the injected diesel fuel along with some part of the gaseous fuel may burn in the first phase of combustion. As pilot fuel quantity decreases, the heat release rate also decreases.

3.10 CONCLUSIONS

From the analysis carried out the following conclusions is summarized as follows
The thermal efficiency of the engine increases when powered with dual fuel. The percentage increases by approximately 5% for 60% Diesel + 40% LPG share condition, when the secondary fuel duct of 30° inclination was used when compared with normal case.

No considerable change has been found in torque and brake mean effective pressure. NO\textsubscript{x} has been reduced to 35% when compared to that of running at 100% diesel. The NO\textsubscript{x} reduction was mainly due to reduction in peak temperature obtained.

There was an increase in Hydrocarbon emission when going to dual fuel mode. The hydrocarbon emission of engine when fuelled with 20% LPG share produced 412 ppm at no load conditions which is slightly on the higher side when compared to the normal case of dual fuel.

The other performance characteristics of the engine remain more or less same in all the blends given.

The CO emission has been reduced up to 12% in dual fuel mode. The carbon monoxide emission further reduced by 5% with the secondary fuel duct of 30° inclination.

From all the results obtained 60% diesel + 40% LPG, inducted at an angle of 30° to the air flow axis has reduced emission and has better thermal efficiency when compared with the normal case of secondary fuel induction.

On a whole it is found that the thermal efficiency of 60% Diesel + 40% LPG share condition with secondary fuel duct of 30° inclination used, when compared with normal case is high mainly due to the better turbulence created in the flow path. The emissions of CO have been reduced in 80%
Diesel + 20 % LPG share substitution due to better turbulence effect. It reduces by 2% when compared to normal induction of LPG. The NOx emission has been reduced by 10% for the 60% Diesel + 40 % LPG share substitution of fuel with 30 ° inlet conditions of LPG due to better mixing of air and LPG, when compared with the normal case of dual fuel. The results obtained in the performance calculation proved, it was better than the normal case in all parameters.