CHAPTER 5

RESULTS AND DISCUSSION

Simultaneous reduction of NO\textsubscript{x} and Particulate Matter (PM) in diesel engines is a challenging task for researchers. In the present work, reduction of NO\textsubscript{x} and PM with Partially Premixed Charge Compression Ignition (PPCCI) combustion was investigated. For PPCCI mode of operation, the premixed ratio (Rp) is defined as the ratio of energy of premixed fuel $Q_p$ to total energy $Q_e$.

In the present work, diesel was used as a premixed fuel along with conventional injection of either diesel or fish oil methyl ester (FOME) blend or jatropha oil methyl ester (JOME) blend with premixed ratios of 0.25, 0.50 and 0.75. Diesel fuel was injected into the intake manifold using a solenoid operated injector controlled by Electronic Control Unit (ECU) to form premixed charge.

The pre-mixed charge was burnt in the cylinder along with the fuel directly injected into the cylinder by a conventional injection system. To control the start of combustion and NO\textsubscript{x} emissions, EGR was adopted and the exhaust gas was varied from 10\% to 30\% in steps of 10\%.

For the conventional compression ignition direct injection (CIDI) mode, the optimum percentage blend of JOME and FOME with diesel was determined and the same was used for PPCCI mode of operation.
Experiments were conducted to study the performance, emission and combustion characteristics as follows:

Part I : Determination of optimum blends of JOME and FOME with conventional diesel fuel for use in diesel engines in CIDI combustion mode and optimum percentage of EGR to reduce NOx emission for the same.

Part II : Investigation of PPCCI combustion mode with diesel as premixed fuel and diesel / optimised blend of JOME / FOME with diesel as main fuel with and without EGR.

The following notations are used in this chapter:

- JOME and 100% JOME - Jatropha oil methyl ester
- FOME and 100% FOME - Fish oil methyl ester
- xx % JOME/FOME – xx refers to the percentage of JOME/FOME in the blend of JOME/FOME and Diesel

5.1 EFFECTS OF JOME AND FOME BLENDS IN CIDI COMBUSTION MODE

Experiments were conducted with 20%, 40%, 60%, 80% and 100% JOME and FOME blends with diesel. The results show that 40%, 60% and 80% blends follow the same trend as 20% blend and the measured values of various parameters are observed to lie between those of 20% and 100% methyl esters. Hence, results are presented only for diesel, 20% and 100 % methyl esters.
5.1.1 Specific Energy Consumption and Brake Thermal Efficiency

Figures 5.1 and 5.2 show the variation of specific energy consumption (SEC) and brake thermal efficiency respectively with power output for diesel, 20% JOME, 20% FOME, JOME and FOME. The SEC and brake thermal efficiency are inversely proportional to each other. SEC is higher and hence brake thermal efficiency is lower for both the methyl esters (JOME and FOME) and their blends at all power outputs compared to diesel. At rated power output (4.4 kW) the brake thermal efficiency of various fuels tested varies from 28.4% to 24.8%.

![Graph showing variation of specific energy consumption with brake power](image)

**Figure 5.1 Variation of Specific Energy Consumption with Brake Power**

Higher kinematic viscosities of JOME blends (3.7 to 4.5 cSt) and FOME blends (3.9 to 5.2 cSt) compared to that of diesel (3.5 cSt) result in poor atomisation, vapourisation and dispersion of fuel in the combustion chamber. Lower calorific value of JOME blends (39,640 to 42,730 kJ/kg) and
FOME blends (38,650 to 42,530 kJ/kg) compared to diesel (43,500 kJ/kg) result in higher fuel consumption of blends compared to diesel at all the power outputs resulting in higher SEC and lower brake thermal efficiency. The late burning of long chain fatty acid fractions in the expansion stroke increases the temperature of exhaust gases (Nagaraju et al 2008) and hence the heat carried away by the exhaust gases is higher for methyl esters at all the power outputs compared to diesel.

![Graph showing variation of brake thermal efficiency with brake power](image)

**Figure 5.2 Variation of Brake Thermal Efficiency with Brake Power**

FOME blends have higher viscosities and lower calorific values compared to JOME blends. In addition, long carbon chain fatty acid content in the range of C20-C22 in different types of fish oil methyl esters varies from 33.83 to 39.52 % (Behcet, 2011). FOME supplied to our investigation contains 36.7 % of long carbon chain fatty acid content in the range of C20-C22 (Table 3.2), whereas it is almost nil in JOME (Table 3.2). These long chain fatty acid fractions burn late in the expansion stroke and increase the heat carried away by exhaust gases (Nagaraju et al 2008). It will also increase
the fuel consumption for a given power output and decrease the brake thermal efficiency. A good agreement is obtained between the results obtained for brake thermal efficiency and specific energy consumption in the present investigation and experimental results of Godiganur et al (2010).

5.1.2 Exhaust Gas Temperature

The variation of exhaust gas temperature with power output for the fuels tested is shown in Figure 5.3. It is observed that the exhaust gas temperature increases with power output as more fuel is burnt at higher power output to meet the power requirement.

![Exhaust Gas Temperature with Brake Power](image)

**Figure 5.3 Variation of Exhaust Gas Temperature with Brake Power**

Figures 5.8 and 5.9 reveal higher cylinder pressures and Figures 5.12 and 5.13 reveal higher heat release in the expansion stroke for both methyl esters due to late burning of heavier fractions of methyl esters. This increases the heat carried away by exhaust gases and also the temperature of exhaust
gases. It will also increase the fuel consumption for a given power output and decrease the brake thermal efficiency. It is also observed that the exhaust gas temperature increases with increase in percentage of methyl esters in the fuel at all the power outputs compared to diesel. Better combustion of both the methyl esters due to their intrinsic oxygen content may result in higher exhaust gas temperatures at all the power outputs. Increase in the quantity of fuel injected in the case of methyl esters at a given power output increases the in-cylinder pressure and hence in-cylinder temperature resulting in higher exhaust gas temperatures.

FOME and its blends show higher exhaust gas temperature at various power outputs compared to JOME due to their higher kinematic viscosity and presence of long chain fractions in the range of C20-C22 of FOME compared to JOME blends.

At rated power output the exhaust gas temperatures of methyl esters of Jatropha are in the range from 460°C to 475°C while those of methyl esters of Fish oil are in the range from 470°C to 490°C compared to 445°C of diesel. As the exhaust temperature is higher with increased methyl esters content, the possibility of exhaust gas heat recovery is higher when running the engine on biodiesel.

### 5.1.3 Unburnt Hydrocarbon Emissions

The emissions of UBHC, CO and NO\textsubscript{x} are measured using AVL five gas analyser and converted to g/kWh. For example emission of UBHC in g/kWh is calculated using the following equation:

\[
\text{UBHC in g/kWh} = \frac{\text{(UBHC in g/mass of dry exhaust)} \times \text{(mass of dry exhaust/kg of fuel)}}{\text{SFC (kg/kWh)}}
\]  

(5.1)
Where

\[
\text{mass of dry exhaust/kg of fuel} = \frac{\text{Carbon per kg of fuel}}{\text{Carbon per kg of dry exhaust}} \quad (5.2)
\]

- Carbon per kg of fuel is available from the mass percentage of constituents of the fuel.
- Carbon per kg of dry exhaust is calculated from the volumetric analysis of products of combustion.
- SFC is from calculated from total fuel consumption per unit time and brake power.

Same procedure is used for other constituents like NOx and CO emissions. The variation of Unburnt Hydrocarbons (UBHC) with power output for 20% JOME, 20% FOME, JOME and FOME compared with that of diesel is shown in Figure 5.4. It is observed that UBHC emissions decrease as the power output increases for all the fuels tested.

![Graph showing variation of Unburnt Hydrocarbon Emission with Brake Power](image)

**Figure 5.4** Variation of Unburnt Hydrocarbon Emission with Brake Power
For diesel, the UBHC emissions vary from 1.5 g/kWh at 1.1 kW (25% of rated power output) to 0.7 g/kWh at rated power output while for 20% JOME and JOME the variation is from 1.2 to 0.7 g/kWh and 0.9 to 0.5 g/kWh respectively. The UBHC emissions for 20% FOME vary from 1.3 to 0.7 g/kWh while for FOME they vary from 1.1 to 0.6 g/kWh.

A significant reduction in UBHC emissions is noticed for both the methyl esters and their blends at all the power outputs compared to diesel. This may be due to their intrinsic oxygen content (JOME - 10.8 % by wt. and FOME - 8.1% by wt.) whereas, diesel does not have oxygen. The addition of methyl ester to diesel increases the oxygen content enhancing the combustion reactions resulting in high combustion temperature which lowers UBHC emissions (Nabi et al 2009).

Even though the oxygen content in methyl esters is low compared to overall oxygen present inside the combustion chamber it plays a significant role during combustion. Intrinsic oxygen present in the fuel (about 10.8 % by wt. in JOME and 8.1 % by wt. in FOME) will help the combustion process better because diesel combustion is basically heterogeneous in nature. The oxygen in biodiesel finds it easy to participate in the combustion reaction. The addition of biodiesel to diesel provides more oxygen to the combustion reaction and promotes complete combustion especially for those areas at the core of the fuel spray (Lapuerta et al, 2008). The oxygen surrounding the fuel droplets would take relatively larger time to assist the combustion of fuel droplet. The reduction of UBHC in CI engines is also due to lower content of carbon to hydrogen ratio than the normal diesel due to the presence of oxygen in its molecular structure.
It can be observed that FOME and its blends show marginally higher UBHC emissions compared to JOME. It may be due to the lower intrinsic oxygen of FOME (8.1% by wt., Table 3.3) compared to JOME (10.8% by wt., Table 3.3).

5.1.4 Carbonmonoxide Emissions

Figure 5.5 shows the variation of Carbonmonoxide (CO) emission with brake power. Formation of CO is due to insufficient oxygen and time in the combustion chamber during the combustion process. In general, the CO emissions are low for diesel engines as they are operated with lean mixtures.

![Graph showing variation of Carbonmonoxide Emission with Brake Power](image)

**Figure 5.5 Variation of Carbonmonoxide Emission with Brake Power**

It is observed that both JOME and FOME blends show lower CO emissions compared to diesel at all the power outputs. The CO emission varies from 23.0 g/kWh at 25% of rated power output to 16.8 g/kWh at rated
power output for diesel fuel while for 20% JOME the CO emission varies from 16.5 g/kWh to 10.3 g/kWh. JOME shows lower CO emission which varies from 11.8 g/kWh at 25% of rated power to 7.5 g/kWh at rated power. The CO emission for 20% FOME varies from 19.2 to 12.5 g/kWh while for FOME it varies from 14.6 to 9.3 g/kWh.

Due to intrinsic oxygen content in JOME and FOME, the oxygen available for oxidation of CO is more compared to diesel which results in reduced CO emission. Thus CO emission which is already low in diesel engines is further reduced by the use of methyl ester and its blends.

As the percentage of methyl ester in the blend increases it is observed that the CO emission decreases. The CO emission also depends upon the carbon to hydrogen ratio and in-cylinder temperature. In the case of JOME and FOME blends, carbon to hydrogen ratio is comparatively lower and the in-cylinder temperature is higher compared to diesel. These factors reduce the formation of carbon monoxide (Nabi et al 2009). It can be observed that CO decreases with increase in power output except at rated power for all the fuels tested. At rated power, the quantity of fuel injected is more while the quantity of air inhaled is constant and this may result in an increase in CO emission for all the fuels tested.

It can be further observed that FOME and its blends show marginally higher CO emission compared to JOME and its blend. CO emission of FOME is higher due to their lower oxygen content (8.1% by weight) compared to that of JOME (10.8 % by weight).

5.1.5 Oxides of Nitrogen Emissions

Oxides of Nitrogen (NO\textsubscript{x}) consist of Nitric oxide (NO) and Nitrogen dioxide (NO\textsubscript{2}). Nitric oxide is predominant in the oxides of nitrogen
produced inside the engine cylinder. NO\textsubscript{2}/NO in a diesel engine is approximately in the ratio of 0.1 to 0.3 (Heywood 1988). NO is produced in the engines through the following routes: thermal, prompt and fuel.

The thermal route revealed by Zel'dovich in 1946 is predominant in fuels without nitrogen. This mechanism exhibits an exponential dependence on temperature and becomes active at temperatures over approximately 1800 K. The three core reactions are:

\[ O + N_2 \leftrightarrow NO + N \]  \hspace{1cm} (5.3)

\[ N + O_2 \leftrightarrow NO + O \]  \hspace{1cm} (5.4)

\[ N + OH \leftrightarrow NO + H \]  \hspace{1cm} (5.5)

Other than temperature, residence time also has an influence on the production of NO at equivalence ratios above 0.5. Lower equivalence ratios have negligible effect on the production of NO.

The prompt mechanism revealed by Fenimore describes the NO formation early in the flame when the thermal mechanism has not had time to fully establish itself. It is due to the hydrocarbon radicals reacting with molecular nitrogen forming hydrogen cyanide (HCN) through the reactions given below:

\[ CH + N_2 \leftrightarrow HCN + N \]  \hspace{1cm} (5.6)

\[ CH_2 + N_2 \leftrightarrow HCN + NH \]  \hspace{1cm} (5.7)

\[ CH_2 + N_2 \leftrightarrow H_2CN + N \]  \hspace{1cm} (5.8)

\[ C + N_2 \leftrightarrow CN + N \]  \hspace{1cm} (5.9)
HCN then reacts to create NO through reactions (5.10) to (5.13). The main reactions which govern prompt NO formation are as follows:

\[
\text{HCN} + \text{O} \leftrightarrow \text{NCO} + \text{H} \tag{5.10}
\]

\[
\text{NCO} + \text{H} \leftrightarrow \text{NH} + \text{CO} \tag{5.11}
\]

\[
\text{NH} + \text{H} \leftrightarrow \text{N} + \text{H}_2 \tag{5.12}
\]

\[
\text{N} + \text{OH} \leftrightarrow \text{NO} + \text{H} \tag{5.13}
\]

NO formation also occurs through the nitrogen present in the fuel. The fossil fuels do not contain nitrogen or very negligible amount of nitrogen. Hence, the amount of NO formed from the fuel is very low and it is observed that NO increases slowly with increasing temperature and increased NO levels can occur at increased flame temperatures.

As NO\textsubscript{2} formation occurs especially near the flame zone, it is a source of NO\textsubscript{x} emissions in diesel engines. The governing reactions for this mechanism are:

\[
\text{NO} + \text{HO}_2 \leftrightarrow \text{NO}_2 + \text{OH} \tag{5.14}
\]

\[
\text{NO}_2 + \text{H} \leftrightarrow \text{NO} + \text{OH} \tag{5.15}
\]

\[
\text{NO}_2 + \text{O} \leftrightarrow \text{NO} + \text{O}_2 \tag{5.16}
\]

The variation of NO\textsubscript{x} emission with power output for various fuels tested is shown in Figure 5.6. It can be observed that NO\textsubscript{x} decreases with increase in power output for all the fuels. A gradual increase in the emission of NO\textsubscript{x} is observed at all the power outputs for the both methyl esters and
their blends. As methyl esters are oxygenated fuels their use results in higher combustion temperatures and higher NO\textsubscript{x} emissions at all the power outputs.

![Graph](image)

**Figure 5.6** Variation of Oxides of Nitrogen Emissions with Brake Power

Hence, Oxides of Nitrogen emissions (NO\textsubscript{x}) emissions are higher for both the methyl esters and its blend compared to diesel. For diesel, the NO\textsubscript{x} emissions vary from 15.6 g/kWh at 25% of rated power output to 7.3 g/kWh at rated power output. For 20% JOME and JOME, the variations are from 15.1 to 7.4 g/kWh and 19.3 to 11.0 g/kWh respectively. For 20% FOME and FOME, the variations are from 16.3 to 7.6 g/kWh and 19.9 to 11.2 g/kWh respectively at rated power output.

FOME and its blends show marginally higher values of NO\textsubscript{x} emissions compared to JOME and its blends. The exhaust gas temperatures and cylinder peak pressures are higher for FOME at all the power outputs.
compared to JOME and diesel. This indicates that peak combustion temperatures may be higher for FOME which may result in higher NOx emissions compared to JOME even though the oxygen content (8.1% by wt.) of FOME is lower compared to that of JOME (10.8 % by wt.).

5.1.6 Soot Emissions

Diesel particulate matter consists of highly agglomerated solid carbonaceous material (soot), ash, volatile organic and Sulphur compounds resulting from incomplete combustion of hydrocarbons present in fuel; it is also partly contributed by the burning of lubricating oil. Soot is formed in locally rich regions during combustion. Most of the soot formed is subsequently oxidised. A small fraction of the evaporated fuel and lubricating oil does not undergo oxidation and appear as soluble organic fractions (SOF) in the exhaust (Stratakis & Stamateslos 2003). The increase in soot emission for diesel fuel is also due to higher boiling point and thermal stability of aromatic hydrocarbons present in it. In particular, the presence of branched and ring (multi-ring or poly) structure of diesel fuel can increase the soot levels (Rao 2011). Higher boiling point paraffins or aromatics adsorbed from particulates emitted by the engine are reported to increase volatile organic fractions (Stratakis & Stamateslos 2003).

AVL 415 Variable Sampling Smoke meter which gives smoke value in FSN and soot concentration in mg/m^3 is used to measure soot in the exhaust. Figure 5.7 shows the variation of soot emission with brake power for diesel, 20% JOME, 20% FOME, JOME and FOME.
Figure 5.7  Variation of Soot Emission with Brake Power

It can be observed that soot emission increases with increase in power output for all the fuels due to lower air fuel ratio at higher power outputs as shown in Table 5.1.

Table 5.1 Air Fuel Ratio

<table>
<thead>
<tr>
<th></th>
<th>100% Diesel</th>
<th>20% JOME</th>
<th>100% JOME</th>
<th>20% FOME</th>
<th>100% FOME</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air/Fuel Ratio at 25 % of Rated Power Output (measured)</td>
<td>54.7</td>
<td>51.7</td>
<td>41.5</td>
<td>52.6</td>
<td>45.3</td>
</tr>
<tr>
<td>Air/Fuel Ratio at Rated Power Output (measured)</td>
<td>24.7</td>
<td>23.5</td>
<td>19.1</td>
<td>23.9</td>
<td>20.5</td>
</tr>
</tbody>
</table>

The soot emissions at rated power output for diesel, 20% JOME, 20% FOME, JOME and FOME are 166 mg/m³, 120 mg/m³, 125 mg/m³, 88 mg/m³ and 102 mg/m³ respectively. Soot emissions for JOME and FOME are significantly reduced compared to diesel due to the following reasons:
• Their higher oxygen content (Sahoo et al 2009).

• They have no aromatic compounds which are considered as soot precursors typically available in petroleum diesel fuel (Wang et al 2000, Lapuerta et al 2002, Rao 2011).

• In addition, oxygen atoms in fatty acid methyl esters bonds to carbon atoms, and therefore prevents carbon atoms from participating in soot growth reactions. The oxidation of fatty acid methyl esters and other oxygenated intermediates also forms OH radicals, which readily attack unsaturated hydrocarbons and prevent their participation in soot growth reactions. Reduced delay period of methyl esters and their blends advances the start of combustion and increases the residence time of soot particles at elevated temperature and promotes their oxidation (Cardone et al 2002).

It can be further observed that FOME and its blend show marginally higher values of soot emission due to their lower intrinsic oxygen content compared to JOME and its blend. Higher kinematic viscosity of FOME (5.2 cSt) compared to JOME (4.5 cSt) results in poor atomisation and deterioration in combustion leading to formation of soot. In addition, long carbon chain fatty acid content in the range of C20-C22 in FOME is 36.7 % which is also responsible for higher soot emission.

5.1.7 Pressure - Crank Angle Diagram

Figures 5.8 and 5.9 show the variation of cylinder pressure with crank angle at rated power output for diesel, 20% and 100% methyl esters of jatropha oil and fish oil respectively. In-cylinder peak pressure is higher for 20% JOME and JOME compared to diesel.
Figure 5.8  Variation of In-cylinder Pressure with Crank Angle at Rated Power Output

Figure 5.9  Variation of In-cylinder Pressure with Crank Angle at Rated Power Output
The cylinder pressures at all the crank angles are higher for both the methyl esters and their blend compared to diesel due to their higher cetane number and shorter ignition delay compared to diesel. Hence, combustion starts early in the case of methyl esters and their blend and results in higher cylinder pressure. Methyl esters are oxygenated fuels and their higher oxygen content results in better combustion and higher cylinder pressure. At any power output, the specific energy consumption is also high for both the methyl esters and its blends compared to diesel as observed from Figure 5.1 which also results in higher cylinder pressure.

It is observed from Figures 5.8 and 5.9 that both the methyl esters and their blends show marginally higher in-cylinder pressure during the expansion stroke due to late burning of higher fatty acid components present in them (Nagaraju et al 2008).

In the case of 20% FOME and FOME, the pressures at various crank angles are marginally higher compared to JOME and its blend. From Table 3.2 it can be observed that FOME contains 36.76% long chain fatty acids in the range of C:20 – C:22 which burn late in the expansion stroke resulting in higher in-cylinder pressure compared to JOME which does not contain any fatty acid component beyond C:18.

5.1.8 Ignition Delay

Figures 5.10 (a) and (b) show the variation of ignition delay with brake power in degrees crank angle (CAD) and milliseconds (ms) respectively for all the fuels tested.
Figure 5.10 Variation of Ignition Delay with Brake Power
The ignition delay is usually defined as the period from the start of injection to the start of combustion. Start of injection is given by the engine manufacturer as 23.4°bTDC (Static injection timing). The start of combustion is the point at which the heat release rate curve starts rising from zero determined from cylinder pressure data. Start of combustion is determined from the AVL indimodule.

At rated power output, the ignition delay of diesel, 20% JOME and JOME is determined to be 15.8, 14.9 and 13.9 °CA respectively while for 20% FOME and FOME, it is 15.4 and 14.6 °CA respectively. The marginally lower ignition delays for both the methyl esters (EL-Kasaby & Nemit-allah, 2013) and their blends compared to diesel are due to their higher cetane index. The calculated cetane index values for diesel, 20% JOME and JOME are 49, 50 and 54 respectively while for 20% FOME and FOME the values are 50 and 53 respectively. The ignition delay decreases with increase in percentage of methyl ester in the blend and this is attributable to their higher cetane index compared to diesel. As the amount of biodiesel increases in the blend, the cetane index of the blend increases and this results in a decrease in ignition delay. The ignition delay decreases as the percentage of methyl esters in the blend increases due to the intrinsic oxygen (JOME – 10%, FOME – 8.1%) present in them.

Biodiesel usually includes a small percentage of diglycerides having higher boiling point compared to diesel. However chemical reactions during the injection of biodiesel at high temperature result in the breakdown of high molecular weight esters leading to the formation of gases of low molecular weight. Rapid gasification of these lighter fractions in the fringe of spray spreads out the jet and thus volatile combustion compounds get ignited earlier and reduce the delay period (Buyukkaya 2010). Large amounts of Oleic fatty acid (40.6 %) and Linoleic fatty acid (33.4 %) methyl esters
present in JOME split into smaller compounds when they enter the combustion chamber resulting in higher spray angles. Hence ignition takes place earlier (Bari et al 2004).

Another contributing factor that decreases the ignition delay of methyl esters and their blends is higher density of blends compared to diesel fuel. When methyl esters with high density (high bulk modulus) are injected, the pressure wave travels faster through the high pressure line from pump to nozzle compared to diesel. This causes early lift of needle in the nozzle when using biodiesel, resulting in advanced injection time leading to an apparent decrease in ignition delay (Rao 2011). Hence, combustion starts earlier in the case of 20% blends and as the percentage of methyl ester in the blend increases it is observed that combustion is advanced further.

Higher ignition delay observed in the case of FOME blends at various power outputs is due to their lower cetane index, lower oxygen content, higher density and higher kinematic viscosity compared to JOME blends. It is observed that ignition delay decreases as the power output increases. At higher power outputs, exhaust gas dilution is reduced due to high in-cylinder temperature and lower density. Both these effects contribute to early ignition.

5.1.9 Peak Pressure

It is observed that for both the methyl esters and their blends the peak pressure increases and occurs later compared to diesel as the percentage of methyl esters in the blend increases. Figure 5.11 shows the variation of peak pressure with brake power for diesel, 20% JOME, JOME, 20% FOME and FOME. The peak pressure is observed to increase with increase in percentage of methyl ester (Scholl & Sorenson 1993, Agarwal & Atul Dhar 2010) and decrease is also observed (Suryawanshi & Deshpande 2005,
Kinoshita et al (2006). The results of the present study show that the peak pressure increases with increase in power output for both the methyl esters and their blends as the oxygen content of methyl esters is higher compared to diesel and the amount of fuel burned increases with increase in power output.

The peak pressure at various power outputs is higher for FOME and its blends compared to JOME and its blends. The peak pressure at rated power output is 72 bar and 74 bar for 20% JOME and JOME respectively while the peak pressure is 73 bar and 76 bar for 20% FOME and FOME respectively.

![Graph showing variation of peak pressure with brake power](image)

**Figure 5.11** Variation of Peak Pressure with Brake Power

### 5.1.10 Heat Release Rate

The heat release rate is used to identify the start of combustion, the fraction of fuel burned in the premixed mode and differences in combustion rates of fuel (Buyukkaya 2010). Figures 5.12 and 5.13 show the variation of heat release rate with crank angle at rated power for diesel, 20% and 100% methyl esters of jatropha oil and fish oil. It can be observed that prior to the
start of combustion the heat release rate is negative for all the fuels tested as the fuel injected absorbs heat for vapourisation.

It is also observed that the peak heat release rates for JOME and its blends are lower than that of diesel. The peak heat release rates at rated power output for diesel, 20% JOME and JOME are 77.5, 66.5 and 56.9 J/°CA respectively. The peak heat release rates for 20% FOME and FOME are 70.6 and 62.6 J/°CA respectively.

![Graph showing heat release rate vs crank angle](image)

**Figure 5.12  Variation of Heat Release Rate with Crank Angle at Rated Power Output**

The lower peak heat release rates for both the methyl esters and their blends are due to their shorter ignition delay which result in less intense premixed combustion phase. Accumulation of fuel during the ignition delay period is less. In the case of diesel, the increased accumulation of fuel during the relatively longer delay period results in higher heat release rate. Between 20% and 100% methyl esters the peak heat release rate is higher for 20% methyl esters due to their longer delay period. It may also be due to lower
viscosity of 20% blend, which results in better spray formation and atomisation. The higher oxygen content of biodiesel (JOME and FOME) results in reduction of carbon to hydrogen ratio and causes lower heat of combustion of biodiesel compared to conventional diesel fuel. This is one of the reasons for lower peak heat release rate compared to diesel.

There is an increase in heat release rate in the diffusion combustion phase for methyl ester and their blends compared to diesel after TDC. Increase in methyl ester percentage in the blend results in significant diffusion combustion phase due to higher viscosity and larger percentage of longer chain fatty acid compounds compared to diesel.

![Graph showing heat release rate variation with crank angle](image)

**Figure 5.13 Variation of Heat Release Rate with Crank Angle at Rated Power Output**

The peak heat release rate of FOME and its blends is lower than that of diesel and higher than those of JOME and its blends. This is due to their longer ignition delay compared to that of JOME and its blends. The effect is more pronounced in the case of FOME compared to JOME.
5.1.11 Summary

Figure 5.14 (a) shows the comparison of brake thermal efficiency, exhaust gas temperature, ignition delay and peak pressure at rated power output for 20%, 40%, 60% 80% and 100% JOME with diesel. Figure 5.14 (b) shows the comparison of UBHC, CO, NOx and soot emissions for various blends of JOME with diesel. Figures 5.15 (a) and (b) show the comparison of the above characteristics for FOME and its blends. The following are the summary of the investigations carried out to study the effect of JOME and FOME blends in CIDI combustion mode:

- Both JOME and FOME and their blends can be used as fuels for CI engines without any modification in the engine.

- Brake thermal efficiency decreases marginally as the methyl ester in the blend increases.

- At rated power output the ignition delay decreases as the percentage of methyl ester in the blend increases compared to diesel.

- Peak pressure is marginally higher for methyl esters at rated power output.

- UBHC, CO and soot emissions decrease as the percentage of methyl esters in the blend increases.

- NOx emissions are significantly higher for methyl esters at rated power output and increasing the percentage of methyl ester in the blend increases NOx emissions.

- Both the methyl esters have higher viscosity compared to diesel. Hence, better lubricity than diesel fuel which will reduce the need for lubricity additives being added to diesel fuel
Figure 5.14 Comparison of JOME and its Blends with Diesel at Rated Power Output (a) Percentage Variation of Performance and Combustion Parameters (b) Percentage Variation of Emissions

* Positive variation denotes increase in values compared to CIDI mode and Negative variation denotes decrease in values compared to CIDI mode
Figure 5.15  Comparison of FOME and its Blends with Diesel at Rated Power Output (a) Percentage Variation of Performance and Combustion Parameters (b) Percentage Variation of Emissions

* Positive variation denotes increase in values compared to CIDI mode and Negative variation denotes decrease in values compared to CIDI mode
An optimum percentage of methyl esters in the blend is necessary as simultaneous reduction of soot and NO\textsubscript{x} is desirable. The variation of soot and NO\textsubscript{x} values normalised with respect to baseline diesel operation at rated power output for various percentages of JOME and FOME in the blends is shown in Figures 5.16 and 5.17 respectively.

![Graph showing tradeoff between Oxides of Nitrogen and Soot Emissions for Various Percentage of JOME in Blend](image)

**Figure 5.16** Tradeoff between Oxides of Nitrogen and Soot Emissions for Various Percentage of JOME in Blend

19% JOME and 22% FOME are observed to be optimum considering both NO\textsubscript{x} and soot emissions and this is approximated to be 20% for both JOME and FOME.

The cost of operation per hour at rated power output, various emissions and brake thermal efficiency at rated power output are given in Table 5.2.
Figure 5.17  Tradeoff between Oxides of Nitrogen and Soot Emissions for Various Percentage of FOME in Blend

Table 5.2  Comparison of 20% JOME, 20% FOME and Diesel Fuel at Rated Power

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Diesel</th>
<th>20% JOME</th>
<th>20% FOME</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbonmonoxide (CO)</td>
<td>g/kW h</td>
<td>16.75</td>
<td>10.33</td>
<td>12.60</td>
</tr>
<tr>
<td>Unburnt Hydrocarbon (UBHC)</td>
<td>g/kW h</td>
<td>0.74</td>
<td>0.67</td>
<td>0.72</td>
</tr>
<tr>
<td>Oxides of Nitrogen (NOₙ)</td>
<td>g/kW h</td>
<td>7.25</td>
<td>7.46</td>
<td>7.60</td>
</tr>
<tr>
<td>Soot</td>
<td>mg/m³</td>
<td>166</td>
<td>120</td>
<td>125</td>
</tr>
<tr>
<td>Brake Thermal Efficiency</td>
<td>%</td>
<td>28.4</td>
<td>28.1</td>
<td>27.7</td>
</tr>
<tr>
<td>Cost Per Litre</td>
<td>INR ($)</td>
<td>54 (0.87)</td>
<td>61(0.98)</td>
<td>55 (0.88)</td>
</tr>
<tr>
<td>Cost Per Hour at Rated Power Output</td>
<td>INR ($)</td>
<td>83 (1.33)</td>
<td>96 (1.54)</td>
<td>88 (1.41)</td>
</tr>
</tbody>
</table>

(1 US dollar ($) = INR 62 as of Nov 2013)
The brake thermal efficiency of 20% JOME and 20% FOME is marginally lower than that of diesel. It can be observed that CO, HC and soot emissions are lower while NO\textsubscript{x} emissions are higher for 20% methyl ester blends compared to diesel. The JOME and FOME are procured in the open market at Rs.90/-($1.5) per litre and Rs. 60/-($1.0) per litre respectively when compared to Diesel at Rs.54/- ($0.9) per litre. Cost of operation per hour is highest for 20 % JOME as the basic cost of JOME per litre is high.

The cost of methyl esters are likely to come down in the near future with mass production where as the cost of diesel fuel will increase continuously. From the present experimental results, it can be concluded that 20% JOME and 20% FOME can be successfully used in existing diesel engines without any modifications.

5.2 EFFECT OF 20% JOME AND 20% FOME BLENDS IN CIDI COMBUSTION MODE WITH EXHAUST GAS RECIRCULATION

20% JOME and 20% FOME are observed to be optimum for CI engines from the results of investigations carried out but they exhibit higher NO\textsubscript{x} emissions compared to diesel fuel. Simultaneous reduction of soot and NO\textsubscript{x} is a prime requirement for modern day diesel engines in order to meet the increasingly stringent emission standards. Three important factors leading to the formation of NO\textsubscript{x} in diesel engines are high temperature, availability of oxygen and residence time for the reaction to complete. As observed from the literature, exhaust gas recirculation (EGR) is one of the most effective techniques for reducing NO\textsubscript{x} emissions in diesel engines (Labecki & Ganippa 2012), as it reduces the in-cylinder temperatures and availability of oxygen. However, EGR leads to higher fuel consumption and increase in HC, CO, and Soot emissions.
The recirculation of part of exhaust gases into the engine intake air increases the specific heat capacity and reduces the oxygen concentration of the intake mixture. It also lowers the peak combustion temperature and reduces the NOx emission. In the present work, part of the exhaust gases from the engine exhaust was cooled down to 30°C and admitted along with the intake air in the inlet manifold. Cooled EGR was used throughout the experimental investigation. A control valve was provided in the intake manifold to regulate the quantity of exhaust gases mixed with the intake air. The temperature of the exhaust gas-air mixture was measured just before the entry into the combustion chamber using a thermocouple. At high percentages of EGR, high levels of UBHC, CO and soot emissions were observed in the exhaust. The result shows that diesel engines must be typically operated significantly below their maximum EGR potential, thus penalising NOx emissions. EGR rate was calculated from the measurement of CO2 in the intake charge and exhaust gas and the ratio was limited to 0.3. The results are compared with those of diesel without EGR.

Experiments were conducted with 10%, 20% and 30% EGR for 20% blends of JOME and FOME. The optimum EGR rate was decided considering the variation of NOx and soot emissions at various EGR rates for both the methyl ester blends. Figures 5.18 and 5.19 shows the tradeoff between NOx and soot emissions with various percentages of EGR for 20% JOME and 20% FOME at rated power output. Since the units of NOx and smoke density are different, normalised values of NOx and smoke are indicated. It can be observed that, approximately 20% EGR gives the optimum NOx and soot emission for both 20% JOME and 20% FOME. Hence, further results are presented only for 20% EGR for both the blends of JOME and FOME.
Figure 5.18  Tradeoff between Oxides of Nitrogen and Soot Emissions for 20% JOME for Various EGR Percentages

Figure 5.19  Tradeoff between Oxides of Nitrogen and Soot Emissions for 20% FOME for Various EGR Percentages
5.2.1 Specific Energy Consumption and Brake Thermal Efficiency

The variation of SEC and brake thermal efficiency with brake power for 20% JOME and 20% FOME respectively with and without 20% EGR is shown in Figure 5.20 and 5.21 respectively in comparison with that of diesel.

![Graph showing Specific Energy Consumption vs Brake Power](image)

**Figure 5.20 Variation of Specific Energy Consumption with Brake Power**

The recirculation of exhaust gas increases SEC at all the power outputs for 20% JOME and 20% FOME without EGR. Experimental investigations show that when EGR is increased, SEC is increased and the brake thermal efficiency is decreased. The decrease in thermal efficiency is due to the dilution of fuel–air mixture with exhaust gases and consequent deficiency in oxygen concentration in the combustion chamber. Hence, combustion deteriorates when EGR is introduced. This reduces the peak pressure and temperature inside the combustion chamber resulting in a
decrease in brake thermal efficiency at all the power outputs. The effects of 30% EGR are investigated and the results are shown in Figure 5.33 (a) from which it can be observed that the effect is more pronounced at higher percentage of EGR.

![Graph showing variation of brake thermal efficiency with brake power]

**Figure 5.21  Variation of Brake Thermal Efficiency with Brake Power**

At rated power output the brake thermal efficiency decreases from 28.0% to 26.0% for 20% JOME when EGR flow rate is varied from 0 to 30% while the variation is from 27.7% to 25.6% for 20% FOME compared to 28.4% for diesel. The results of brake thermal efficiency with EGR in the present work are in a good agreement with the experimental data obtained by Nitin Shrivastava et al (2012).

The brake thermal efficiency of 20% FOME with EGR is still lower at all the power outputs compared to 20% JOME due to lower intrinsic oxygen present in 20% FOME (2.16 % by wt.) compared to that of 20% JOME (1.62 % by wt.).
5.2.2 Exhaust Gas Temperature

The variation of exhaust gas temperature with brake power for diesel and for 20% methyl esters with and without EGR is shown in Figure 5.22.

![Exhaust Gas Temperature Graph](image)

**Figure 5.22 Variation of Exhaust Gas Temperature with Brake Power**

At rated power output, the exhaust gas temperature without EGR is 460°C for 20% JOME and 470°C for 20% FOME. With 20% EGR the exhaust gas temperature is 445°C for 20% JOME and 450°C for 20% FOME whereas in the case of diesel, exhaust gas temperature is 445°C.

It is observed that EGR decreases the exhaust gas temperature for both the methyl esters tested. This may be due to deterioration in combustion and reduction in peak combustion temperature with EGR. From the experimental results obtained the effect is observed to be more pronounced at higher percentages of EGR. Higher exhaust gas temperature observed in the case of 20% FOME at all the power outputs is due to late burning of higher
proportion of long chain fractions in the methyl esters of fish oil (36.7\%) which are not present in JOME.

5.2.3 Unburnt Hydrocarbon Emission

Figure 5.23 shows the variation of UBHC emissions with brake power for diesel, 20\% JOME and 20\% FOME with 20\% EGR and without EGR.

![Diagram showing variation of Unburnt Hydrocarbon Emission with Brake Power](image)

**Figure 5.23 Variation of Unburnt Hydrocarbon Emission with Brake Power**

UBHC emissions decrease with power output for all the fuels tested. It can be observed that UBHC emissions at low power outputs are higher than that at rated power since at lighter loads turbulence is less significant and the mixture is too lean (Saleh 2009). EGR increases the UBHC emissions for both the methyl esters at all the power outputs and UBHC emissions are higher for higher percentage of EGR. With EGR the oxygen available for combustion is reduced and the air fuel mixture does not combust properly resulting in higher UBHC emissions.
At rated power output the UBHC emission varies from 0.7 to 0.8 g/kWh for 20% JOME when EGR is varied from 0 to 30% while the variation is from 0.7 to 0.9 g/kWh for 20% FOME compared to 0.7 g/kWh for diesel. 20% FOME with EGR shows higher UBHC emissions at all the power outputs compared to 20% JOME with EGR due to its lower percentage of intrinsic oxygen and higher percentage of longer chain fatty acid components present in it.

5.2.4 Carbonmonoxide Emission

The variation of CO emission with brake power for all the fuels tested is shown in Figure 5.24.

![Figure 5.24](image)

Figure 5.24 Variation of Carbonmonoxide Emission with Brake Power

For both the methyl ester blends, EGR increases CO emissions and higher EGR results in higher CO emissions due to incomplete combustion caused by the diluted mixture. When EGR percentage is increased beyond
20%, UBHC and CO emissions are higher than those of diesel showing that there is a limit for EGR. CO emissions at rated power output for various EGR flow rates used vary from 10.3 to 23.0 g/kWh for 20% JOME while they vary from 13.2 to 25.5 g/kWh for 20 % FOME compared to 16.8 g/kWh for diesel. Higher CO emissions in the case of FOME may be due to lower intrinsic oxygen available for combustion compared to JOME.

5.2.5 Oxides of Nitrogen Emissions

Figure 5.25 shows the NO\textsubscript{x} emissions at various power outputs for the fuels tested with and without EGR compared to diesel.

![Graph showing variation of oxides of nitrogen emissions with brake power](image)

**Figure 5.25 Variation of Oxides of Nitrogen Emissions with Brake Power**

NO\textsubscript{x} emissions are reduced for both the methyl ester blends as the percentage of EGR is increased. This is due to the fact with EGR the availability of oxygen reduces; in addition to this, higher specific heat capacity of the recirculated exhaust gases lowers the global temperature in the
combustion chamber and the flame temperature, which subsequently reduces the reaction rates leading to the formation of NOx emissions.

At rated power output, the NOx emissions without EGR and with 10, 20 and 30% EGR vary from 7.5 to 6.1 g/kWh for JOME while they vary from 7.6 to 6.4 g/kWh for 20% FOME compared to 7.3 g/kWh for diesel. At rated power output, the concentration of CO2 and H2O is considerably higher and specific heat capacity increases. Both CO2 and H2O have higher specific heat capacities than air (Jacobs et al 2003). The specific heat at constant pressure (Cp) of combustion products at different temperatures is shown in Table 5.3 (Luke Franklin, 2010). At higher combustion temperatures, H2O has higher specific heat capacity than CO2. Higher heat capacity of the mixture requires more energy to pre-heat the incoming mixture, and lowers the flame temperature which reduces the NOx emissions at rated power output.

**Table 5.3 Specific Heat of Combustion Products and Air**

<table>
<thead>
<tr>
<th>Products</th>
<th>Specific heat (in kJ/kmol K) at temperatures of</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>727 °C</td>
</tr>
<tr>
<td>CO2</td>
<td>55</td>
</tr>
<tr>
<td>H2O</td>
<td>41</td>
</tr>
<tr>
<td>Air</td>
<td>33</td>
</tr>
</tbody>
</table>

5.2.6 Soot Emissions

Figure 5.26 shows the variation of soot emissions at various power outputs for diesel, 20% JOME and 20% FOME with 20% EGR and without EGR. It is observed that as EGR increases, soot emissions increase but soot emissions are lower than those of diesel at rated power output. The exhaust gases in the intake air decrease the oxygen content and result in lower combustion temperature. The formation of soot and its oxidation are mainly
influenced by the engine operating conditions. The combined effect of fuel properties, lower combustion temperature and lower oxygen content reduces the soot oxidation process which eventually results in higher soot emissions.

![Graph showing the variation of soot emission with brake power](image)

**Figure 5.26  Variation of Soot Emission with Brake Power**

With FOME, soot emissions are marginally higher due to lower intrinsic oxygen content compared to JOME. At rated power output, the soot emissions with no EGR, 10, 20 and 30% EGR vary from 120 to 150 mg/m³ for 20% JOME while the soot emissions vary from 125 to 156 mg/m³ for 20% FOME compared to 166 mg/m³ for diesel.

### 5.2.7 Pressure - Crank Angle Diagram

Figure 5.27 and 5.28 shows the variation of in-cylinder pressure with crank angle at rated power output for 20% methyl esters respectively with 20% EGR and without EGR.
Figure 5.27  Variation of In-cylinder Pressure with Crank Angle at Rated Power Output

Figure 5.28  Variation of In-cylinder Pressure with Crank Angle at Rated Power Output
With EGR, the oxygen available for combustion is reduced and start of combustion is delayed. It is observed that with EGR, the in-cylinder pressure is marginally low before the occurrence of peak pressure compared to that without EGR for both the methyl esters. The recirculated exhaust gas increases the heat capacity of the cylinder charge and reduces the combustion pressure and temperature. It is also observed that in-cylinder pressure with EGR is higher than that of diesel over the entire range of operation.

### 5.2.8 Ignition Delay

Figure 5.29 shows the variation of ignition delay with brake power for diesel, 20% JOME and 20% FOME with and without EGR.

![Graph showing variation of Ignition Delay with Brake Power](image)

**Figure 5.29** Variation of Ignition Delay with Brake Power

The ignition delay at rated power output for 20% JOME without EGR and with 20% EGR is 14.9 °CA and 15.5 °CA respectively compared to 15.8 °CA for diesel fuel. With cooled EGR, the delay period increases
marginally due to reduction in oxygen content. In the case of 20% FOME, ignition delay period without EGR and with 20% EGR is 15.4 and 15.7 °CA respectively at rated power output.

5.2.9 Peak Pressure

The variation of cylinder peak pressure with brake power for diesel, 20% JOME and 20% FOME with and without EGR is shown in Figure 5.30. It can be observed that the cylinder peak pressure decreases with increase in percentage of EGR at all the power outputs due to decrease in oxygen availability.

![Variation of Peak Pressure with Brake Power](image_url)

**Figure 5.30  Variation of Peak Pressure with Brake Power**

The cylinder peak pressure at rated power output for 20% JOME without EGR and with 20% EGR is 72.0 and 72.5 bar respectively compared to 71.7 bar for diesel fuel. In the case of 20% FOME, peak pressure without
EGR and with 20% EGR is 73.1 and 72.5 bar respectively at rated power output.

5.2.10 Heat Release Rate

Figure 5.31 and 5.32 shows the variation of heat release rate with crank angle at rated power for 20% methyl esters with and without EGR.

![Diagram showing heat release rate with crank angle](image)

**Figure 5.31 Variation of Heat Release Rate with Crank Angle at Rated Power Output**

With EGR, the oxygen available for combustion is reduced which retards the start of combustion and decreases the peak heat release rate. This may lower the peak combustion temperature and decrease NOx emission with EGR. Figure 5.31 shows that at rated power output, the peak heat release rate decreases from 66.5 J/°CA for 20% JOME without EGR to 58.7 J/°CA with 20% EGR while it is 77.5 J/°CA in diesel. It is observed that the heat release rate after TDC is higher when exhaust gas is recirculated. In the case of 20%
FOME, the peak heat release rate decreases from 70.5 J/°CA for 20% FOME without EGR to 65.8 J/°CA with 20% EGR at rated power output.

![Graph showing Heat Release Rate vs Crank Angle](image)

**Figure 5.32  Variation of Heat Release Rate with Crank Angle at Rated Power Output**

### 5.2.11 Summary

The variation of performance and combustion characteristics of 20% methyl esters compared to diesel without EGR and with 10, 20 and 30% EGR at rated power output is shown in Figures 5.33(a) and 5.34(a). As EGR is increased the brake thermal efficiency and exhaust gas temperature decrease. The effect is more pronounced at higher EGR rates. It is observed that ignition delay increases and peak pressure decreases with increase in EGR for 20% methyl esters. Figure 5.33(b) and 5.34(b) shows the variation of emissions of 20% methyl esters with and without EGR compared to diesel at rated power output.
Figure 5.33  Comparison of 20% JOME without EGR and with 10, 20 and 30% EGR with Diesel at Rated Power Output
(a) Percentage variation of performance and Combustion Parameters (b) Percentage Variation of Emissions

* Positive variation denotes increase in values compared to CIDI mode and Negative variation denotes decrease in values compared to CIDI mode
Figure 5.34 Comparison of 20% FOME without EGR and with 10, 20 and 30% EGR with Diesel at Rated Power Output
(a) Percentage Variation of Performance and Combustion Parameters (b) Percentage Variation of Emissions

* Positive variation denotes increase in values compared to CIDI mode and Negative variation denotes decrease in values compared to CIDI mode
NOₓ emissions are higher for 20% JOME and 20% FOME blend without EGR. As methyl esters are oxygenated fuels their use results in higher combustion temperature and higher NOₓ emission. Experiments were conducted with 10, 20 and 30 % EGR and the investigations show that with EGR, NOₓ emission is reduced but UBHC, CO and soot emissions are increased.

For both the methyl esters upto 20% EGR, UBHC and CO emissions are lower compared to diesel. Further increase in EGR increases the UBHC and CO emissions significantly. Thus it can be concluded that 20% EGR is optimum considering the emissions from both the methyl esters.

The following are the summary of the investigations carried out to study the effect of EGR with 20% JOME and 20% FOME blends in CIDI combustion mode:

- Brake thermal efficiency decreases marginally with EGR. As the percentage of EGR increases, the brake thermal efficiency decreases further.
- The exhaust gas temperature decreases marginally with EGR. The exhaust gas temperature in the case of 20% JOME with 20% EGR is equal to that of diesel. The percentage variation in exhaust gas temperature is zero hence it is not shown in Figure 5.33 (a).
- At rated power output, the ignition delay increases as the percentage of EGR increases compared to that without EGR in diesel.
- Peak pressure is marginally lower with EGR for both the methyl esters at rated power output compared to without EGR.
- UBHC, CO and soot emissions increase as the percentage of EGR increases.
- With EGR, NO\textsubscript{x} emissions are significantly lowered for both the methyl esters. Increasing the percentage of EGR decreases NO\textsubscript{x} emissions significantly.
- Considering both NO\textsubscript{x} and soot emissions, 20% EGR is observed to be optimum for both 20% JOME and 20% FOME.

The cost per hour of operation at rated power output for Diesel, 20% JOME with 20% EGR and 20% FOME with 20% EGR are Rs.83/- (US $1.30), Rs.99/- (US $1.60) and Rs.92/- (US $1.50). For 20% FOME, it is marginally lower than that of 20% JOME. Use of 20% FOME is recommended for use in coastal areas where it is easily available and 20% JOME is recommended for rural areas where it can be cultivated and procured. It will help to improve rural economy of farmers.

5.3 PPCCI MODE OF COMBUSTION WITH DIESEL AS PREMIXED AND MAIN FUELS (D-D MODE) WITH PREMIXED RATIOS (\(R_p\)) OF 0.25, 0.50 AND 0.75 WITHOUT EGR

Homogeneous charge compression ignition (HCCI) engines are being considered as alternatives to diesel engines (Dale and Oppenheim 1982). The HCCI concept involves premixing fuel and air prior to induction into the cylinder (as in SI engines) then igniting the fuel-air mixture through the compression process (as in CI engines). This is an engine combustion process that is basically a combination of spark ignition (SI) and compression ignition (CI) engine processes (Dale and Oppenheim 1982). PM and NO\textsubscript{x} emissions of HCCI engines are considerably lower (Wontae et al 2007,
Andreas et al 2008) than those of CIDI engines due to their dilute homogeneous charge in addition to their lower peak temperatures. Inlet air fuel mixture can be diluted more than the levels normally used in SI or CIDI engines. In HCCI engines, compression heating by piston motion induces combustion throughout the homogenous charge and results in lower flame temperatures.

In the present investigation, premixed charge was prepared partially in the inlet manifold and the remaining quantity of fuel is injected into the cylinder by conventional mode. This method of operation is named Partially Premixed Charge Compression Ignition (PPCCI). For achieving smooth combustion in PPCCI mode, control over the start of combustion (SOC) is necessary. Indirect methods like variable valve timing (VVT), variable compression ratio (VCR), EGR, varied intake charge temperature, varied equivalence ratio which would help to alter the combustion process are employed. EGR was used to control the start of combustion in the present work. The following notations are used in PPCCI mode:

- **D-D mode**: Diesel (manifold injection) – Diesel (main injection)
- **D-20J mode**: Diesel (manifold injection) – 20% JOME and 80% Diesel blend (main injection)
- **D-20F mode**: Diesel (manifold injection) – 20% FOME and 80% Diesel blend (main injection)

The performance, emission and combustion characteristics of PPCCI combustion mode with diesel as premixed fuel along with diesel or 20% methyl esters of Jatropha oil and fish oil for main injection were investigated with premixed ratios of 0.25, 0.5 and 0.75. Experiments were conducted in PPCCI combustion mode as shown in Table 5.4.
Table 5.4 Experiments Conducted in PPCCI Combustion Mode

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>PPCI mode</th>
<th>Main Injection</th>
<th>Premixed Injection</th>
<th>Performance without EGR for Premixed ratios of 0.25 Rp with EGR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>D-D</td>
<td>Diesel</td>
<td>Diesel</td>
<td>0.25, 0.50 &amp; 0.75, 10%, 20% &amp; 30%</td>
</tr>
<tr>
<td>2</td>
<td>D-20J</td>
<td>20% JOME*</td>
<td>Diesel</td>
<td>0.25, 0.50 &amp; 0.75, 10%, 20% &amp; 30%</td>
</tr>
<tr>
<td>3</td>
<td>D-20F</td>
<td>20% FOME*</td>
<td>Diesel</td>
<td>0.25, 0.50 &amp; 0.75, 10%, 20% &amp; 30%</td>
</tr>
</tbody>
</table>

*20% JOME and 20% FOME were selected based on the performance and emission characteristics as discussed in sections 5.1 and 5.2.

5.3.1 Specific Energy Consumption and Brake Thermal Efficiency

Figure 5.35 and 5.36 shows the variation of SEC and brake thermal efficiency with brake power respectively for PPCCI mode for various values of premixed ratio compared with diesel. While calculating SEC and brake thermal efficiency, premixed fuel energy and main fuel energy both were taken into account.

Increase in SEC and decrease in brake thermal efficiency are observed at all power outputs due to the following reasons:

- Part of the premixed fuel injected into the intake port is accumulated in the premixed chamber in a liquid state at lower temperature and does not enter into the combustion chamber (Dae & Chang 2006).
- Low volatility of diesel fuel results in poor vapourisation.
- External mixture formation also results in wall wetting and over-leaning problems (Kook & Bae 2004).
Figure 5.35  Variation of Specific Energy Consumption with Brake Power

Figure 5.36  Variation of Brake Thermal Efficiency with Brake Power
• The in-cylinder pressure (Shown in Figure 5.42) is higher before TDC due to the advance of start of combustion during the end of the compression stroke compared to base line diesel fuel mode which increases the compression work (Miguel Torres Garcia et al 2009).

• The rapid rates of heat releases that occur with almost instantaneous combustion result in high rates of cylinder pressure rise and excessive peak pressures. This results in higher in-cylinder temperature and increases the heat transfer to the cylinder walls and more fuel is consumed for the same power output.

These effects are more pronounced at higher values of premixed ratio. The SEC varies from 13,238 to 16,165 kJ/kWh while the brake thermal efficiency varies from 27.2 % to 22.3 % for premixed ratios of 0.25, 0.50 and 0.75 compared to 12661 kJ/kWh and 28.4 % for diesel mode at rated power output.

5.3.2 Exhaust Gas Temperature

The variation of exhaust gas temperature with brake power is shown in Figure 5.37.

In PPCCI mode of operation, combustion takes place in the entire cylinder charge which results in low combustion temperatures at part loads. Hence, the exhaust gas temperature is lower for PPCCI mode compared to CIDI mode up to 50% of rated power output. The exhaust gas temperature at 50% of rated power output is 284°C, 277°C, 265°C and 258°C for CIDI mode and premixed ratios of 0.25, 0.5 and 0.75 respectively.
Figure 5.37  Variation of Exhaust Gas Temperature with Brake Power

It can be further observed that the exhaust gas temperature is high at rated power output for all values of premixed ratios compared to base line diesel mode. As the premixed ratio is increased the exhaust gas temperature increases at higher power outputs. The exhaust gas temperature at rated power output is 445°C, 470°C, 491°C and 515°C for CIDI mode and premixed ratios of 0.25, 0.5 and 0.75 respectively. From Figure 5.42 it can be observed that the pressure rise starts earlier for all values of premixed ratios at rated power output which indicates early combustion. In-cylinder peak pressure also increases and occurs closer to TDC. This may result in higher combustion temperature leading to higher exhaust gas temperature.

5.3.3 Unburnt Hydrocarbon Emissions

Figure 5.38 shows the variation of UBHC emissions with brake power for PPCCI mode for various premixed ratios compared with diesel mode. It is observed that when premixed ratio is increased the UBHC
emissions increase significantly at all the power outputs as the premixed fuel injected in the manifold is higher. For a given premixed ratio, as the power output increases the equivalence ratio increases which results in an increase in fuel in the crevice volume. The premixed fuel trapped in crevice volumes may be too cold to get ignited and may not undergo complete combustion in PPCCI combustion process resulting in higher UBHC emissions (Lu Xingcai et al 2006 and Junjun et al 2008).

![Graph showing variation of Unburnt Hydrocarbon Emission with Brake Power](image)

**Figure 5.38 Variation of Unburnt Hydrocarbon Emission with Brake Power**

The thermal boundary layer formed on the cylinder surface of PPCCI engines may also be another source of UBHC emissions. The air fuel mixture in the boundary layer does not undergo complete combustion due to flame quenching by the cooler cylinder wall temperatures. Absorption and desorption of fuel vapour into oil layers on the cylinder wall also gives rise to higher UBHC emission (Junjun et al 2008). The effect is more pronounced at
higher premixed ratios. At rated power output, the UBHC emission in PPCCI mode for premixed ratios of 0.25, 0.5 and 0.75 range from 1.0 to 1.6 g/kWh while it is 0.7 g/kWh for CIDI mode.

5.3.4 Carbonmonoxide Emissions

The variation of CO emission with brake power for PPCCI mode for various premixed ratios compared with diesel mode is depicted in Figure 5.39.

![Graph showing CO emission variation with brake power](image)

**Figure 5.39 Variation of Carbonmonoxide Emission with Brake Power**

It can be observed that CO emission is higher at all power outputs for PPCCI mode compared to base line diesel mode. Higher values of premixed ratios result in higher CO emission as the premixed fuel injected in the manifold is higher. The fuel released from the crevice volume is not fully
oxidised due to low in-cylinder temperature and gives rise to higher CO emissions.

The OH radical concentration gets reduced due to incomplete combustion in intermediate temperature regions (Ghazikhani et al 2009). OH radical concentration is one of the important factors for conversion of CO to CO\(_2\) (Heywood 1988). Hence, CO emission increases in PPCCI combustion mode. Poor vapourisation of premixed fuel injected in the manifold may also increase the CO emissions.

At rated power output, the CO emission in PPCCI mode for premixed ratios of 0.25, 0.5 and 0.75 range from 18.7 to 34.7 g/kWh while it is 16.7 g/kWh for CIDI mode.

5.3.5 Oxides of Nitrogen Emissions

In a CI engine, NO\(_x\) is formed in very hot zones closer to stoichiometric conditions and soot is formed in the fuel rich regions. The in-cylinder average air-fuel ratio is globally always lean but locally it varies from lean to rich values. This means that there is a large potential to reduce emissions of NO\(_x\) and PM by preparing homogeneous mixture of fuel and air before combustion.

PPCCI allows a significant reduction of NO\(_x\) and particulates as homogenous mixture is formed which auto-ignites due to compression heat. This differs from conventional CI engines in the sense that ignition does not occur at a specific place in the spray but simultaneously across the combustion chamber. Hence, high temperature flame front does not appear and results in practically negligible NO\(_x\) emissions. Due to homogenously lean mixture, fuel rich zones are absent and therefore soot formation is also reduced. However, the operating range of engines using HCCI mode is
observed to be too narrow and good performance is obtained only from 25% to 75% of rated power output. Improper combustion or misfire under fuel-lean conditions is observed to limit the minimum load at which the engine can operate.

Figure 5.40 shows the variation of NO$_x$ emissions with brake power. Upto 50% of rated power output, NO$_x$ emissions decrease when the premixed ratio is increased for PPCCI mode of combustion. The NO$_x$ emission for premixed ratios of 0.25, 0.5 and 0.75 at 25% of rated power (1.1 kW) output ranges from 12.6 g/kWh to 10.2 g/kWh compared to 15.6 g/kWh for diesel. At 50% of rated power (2.2 kW) NO$_x$ ranges from 10.8 g/kWh to 9.0 g/kWh compared to 12.2 g/kWh for diesel.

Figure 5.40 Variation of Oxides of Nitrogen Emissions with Brake Power
Near rated power output, HCCI engines are often extremely loud (ringing sound) and strong acoustic oscillations resembling those of a knocking spark-ignited engine are observed to occur (Samveg Saxena & Iván D. Bedoya 2013).

At lower power outputs, PPCCI combustion lowers the flame temperature and increases the homogeneity by giving sufficient time for air and fuel to mix. The increase in pressure and temperature during the compression stroke will give rise to simultaneous auto-ignition in the entire cylinder. As there is no high temperature flame front in PPCCI mode of operation the local temperatures are reduced and NO\textsubscript{x} emissions are significantly decreased. It also results in lower soot formation due to lean and homogenous local air fuel mixture (Suyin Gan et al 2011).

NO\textsubscript{x} emission at rated power output varies from 8.8 to 11.9 g/kWh when premixed ratio is increased from 0.25 to 0.75 for PPCCI mode of operation compared to 7.3 g/kWh for diesel mode.

It is also observed from Figure 5.44 that at rated power output for PPCCI mode of combustion the peak heat release rate is higher and also occurs earlier compared to diesel. The peak heat release rate increases and occurs earlier when premixed ratio is increased. This will result in higher in-cylinder pressure and temperatures. Hence, at rated power output, NO\textsubscript{x} emissions are higher compared to diesel. PPCCI ignition at high loads causes significantly increased SEC and NO\textsubscript{x} levels (Stanglmaier & Roberts 1999). This is one of the factors which limit the operating range of PPCCI mode of combustion.
5.3.6 Soot Emissions

The variation of soot emissions with brake power for PPCCI mode compared with CIDI mode is shown in Figure 5.41. Soot and NO\textsubscript{x} emissions are dependent on equivalence ratio and homogeneity of the charge in the cylinder.

It can be observed that soot emissions are low at all power outputs when compared with CIDI mode with diesel fuel due to the homogeneity of air fuel mixture inside the combustion chamber.

![Graph showing variation of soot emission with brake power]

**Figure 5.41 Variation of Soot Emission with Brake Power**

In PPCCI mode, fuel rich regions inside the combustion chamber which is normally observed in CIDI engines are eliminated which results in lower soot emissions. The absence of diffusion limited combustion and localised fuel rich regions reduce the possibility of formation of soot
emissions (Stanglmaier & Roberts 1999). At rated power, soot emissions
decrease significantly compared to CIDI mode due to high heat release rate
which results in better combustion and consequent reduction of soot. As the
premixed ratio is increased the soot emission decreases significantly at all
power outputs due to the increase in homogeneity of premixed charge.

Soot emission at rated power varies from 32 to 86 mg/m³ when
premixed ratio is decreased from 0.75 to 0.25 for PPCCI mode of operation
compared to 166 mg/m³ for diesel mode.

5.3.7 Pressure - Crank Angle Diagram

Figure 5.42 shows the pressure crank angle diagram at rated power
with diesel as premixed fuel and main fuel for various values of premixed
ratios.

![Pressure vs Crank Angle Diagram](image)

---

**Figure 5.42** Variation of In-cylinder Pressure with Crank Angle at
Rated Power Output
It is observed that prior to combustion, the pressure inside the cylinder is low for PPCCI mode of combustion as the premixed fuel vapourises and absorbs latent heat from the mixture reducing the temperature and pressure.

The combustion starts earlier for PPCCI mode and the angle at which the combustion starts is advanced as premixed ratio is increased compared to CIDI mode. At loads lower than 50%, the peak pressure is lower as well as the angle at which it occurs does not vary much from the base diesel mode. At higher loads (above 50%), the peak pressure is higher and the combustion starts earlier for PPCCI mode and the angle of occurrence is advanced as premixed ratio is increased compared to base diesel mode. The cylinder pressure after combustion is high compared to basic diesel mode over the entire range of operation. This is due to the increase in quantity of fuel at higher power outputs while the quantity of air inducted is nearly constant. This increases the fuel air ratio when the fuel is injected in the manifold and results in higher peak pressures and also advances the start of combustion. The effect is more pronounced at higher power outputs and higher values of premixed ratio.

At rated power, the start of combustion occurs at 9, 13 and 16 °CA bTDC for premixed ratios of 0.25, 0.5 and 0.75 compared to 7 °CA bTDC for CIDI mode.

5.3.8 Peak Pressure

The peak pressure is obtained from p-θ diagram and is shown in Figure 5.43. It can be observed that at higher power output the peak pressure is high. As observed in Figure 5.42, the peak pressure occurs earlier in PPCCI mode at rated power output compared to base diesel mode of operation. The effect is more pronounced at higher values of premixed ratio.
As the power output is increased the quantity of fuel increases while the quantity of air inducted is nearly constant. This increases the fuel air ratio of the premixed fuel which burns almost rapidly resulting in higher peak pressures. At rated power output, the peak pressures are 75.4, 81.0 and 84.4 bar for premixed ratios of 0.25, 0.5 and 0.75 and occur 7, 5 and 4 °CA aTDC respectively for PPCCI combustion mode compared to 71.7 bar for CIDI mode which occurs 8 °CA aTDC.

![Graph showing variation of peak pressure with brake power](image)

**Figure 5.43** Variation of Peak Pressure with Brake Power

### 5.3.9 Heat Release Rate

Figure 5.44 shows the heat release rate at rated power output for various premixed ratios in PPCCI mode compared to CIDI mode. It can be observed that the heat release curves show two peaks - one of smaller magnitude 20 to 30 °CA bTDC and another peak of greater magnitude near TDC. The first stage of heat release is associated with low-temperature
kinetic reactions (cool flames) named Low Temperature Reactions (LTR). The second stage of heat release rate is the main heat release and named as High Temperature Reactions (HTR). The time delay between the LTR and HTR is named as Negative Temperature Coefficient (NTC) Region. The Negative Temperature Coefficient regime is characterised by a decrease in the overall reaction rate even though in-cylinder temperature increases. This leads to a lower reactivity of the system. For diesel (lower octane number fuel) the heat release in the Low Temperature Combustion (LTC) is predominant compared to gasoline (higher octane number fuel).

![Graph](image)

**Figure 5.44 Variation of Heat Release Rate with Crank Angle at Rated Power Output**

Figure 5.45 shows the LTR, HTR and interval between LTR and HTR at rated power output for various premixed ratios in PPCCI mode compared to CIDI mode.
It is observed that the magnitude of LTR and the timing of LTR are dependent primarily on the fuel selection and the temperature time history of the fuel-air mixture (Ryan et al 2004). The increasing pressure and temperature during Low Temperature reactions (LTR), together with heat release rate (HRR) and timing of LTR control the high temperature reactions (HTR) (Xingcai Liu et al 2007). Hence, control of LTR is important for increasing the operating ranges of PPCCI combustion.

HTR peak increases in magnitude as premixed ratio is increased and also occurs earlier. LTR peak also increases and occurs nearly at the same crank angle for all premixed ratios at any given power output. LTR occurs earlier as the power output is increased due to the early start of cool flame reactions at higher power outputs.

![Graph showing LTR, HTR, and interval between LTR and HTR with various premixed ratios at rated power output for D-D Mode](image_url)

**Figure 5.45** LTR, HTR and Interval between LTR and HTR with Various Premixed Ratios at Rated Power Output for D-D Mode
From Figure 5.49 it is observed that the crank angle at which HTR occurs is advanced as the premixed ratio is increased while LTR occurs at nearly the same crank angle for all the premixed ratios. Hence, Negative Temperature Coefficient (NTC) region which is the time interval between the occurrence of LTR and HTR decreases as premixed ratio is increased.

The peak heat release rates during HTR at rated power output are observed to be 102.5, 115.2 and 135.1 J/°CA occurring at 9, 13, 16 °CA bTDC for premixed ratios of 0.25, 0.5 and 0.75 respectively while the peak heat release rates during LTR are 4.8, 7.3 and 11.7 J/°CA occurring at nearly 25 °CA bTDC for all the premixed ratios. The time interval between the LTR and HTR is 16, 12 and 10 °CA for premixed ratios of 0.25, 0.5 and 0.75 respectively.

5.3.10 Summary

Table 5.5 shows the percentage variation of different parameters in D-D mode without EGR at rated power output for premixed ratios of 0.25, 0.5 and 0.75 respectively compared to CIDI mode.

| Table 5.5 Percentage Variation of Various Parameters in D-D mode in PPCCI Operation from CIDI Operation for Various Premixed Ratios |
|---------------------------------|------------------|-----------------|-----------------|
|                                | D-D mode Premixed Ratios of |                |                |
|                                | 0.25  | 0.5   | 0.75   |
| Brake Thermal Efficiency       | -4.2   | -12.7 | -21.5  |
| Exhaust Gas Temperature        | 5.6    | 10.3  | -88.4  |
| Unburnt Hydrocarbon            | 34.9   | 71.5  | 123.9  |
| Carbonmonoxide                 | 11.4   | 54.0  | 107.2  |
| Oxides of Nitrogen             | 20.5   | 43.8  | 63.0   |
| Soot                           | -48.2  | -61.4 | -80.7  |
| In-cylinder Peak Pressure      | 5.9    | 13.9  | 18.5   |
| Peak Heat Release Rate         | 32.3   | 48.6  | 74.3   |
The following is the summary of results:

- Brake thermal efficiency decreases as the premixed ratio increases in D-D Mode. The maximum percentage decrease is 21.5% at premixed ratio of 0.75 and minimum percentage decrease is 4.2% at premixed ratio of 0.25 compared to baseline diesel mode.

- Soot emission decreases as the premixed ratio increases in D-D Mode and the maximum percentage decrease is 80.7% at premixed ratio of 0.75.

- NO\textsubscript{x} emission decreases upto 50% of rated power for all the premixed ratios used. Beyond 50% of rated power output, NO\textsubscript{x} emission increases and at premixed ratio of 0.75, a percentage increase of 63.0% in NO\textsubscript{x} emission is observed compared to CIDI mode. Premixed ratio of 0.25 gives the minimum percentage increase of 20.5%.

- UBHC and CO emissions increase. The maximum percentage increases are 123.9% and 107.1% for UBHC and CO emission respectively at premixed ratio of 0.75. Premixed ratio of 0.25 gives the minimum percentage increases in UBHC (34.9%) and CO (11.4%) emission. UBHC and CO emission can be controlled with oxidation catalytic converters in the exhaust manifold.

- Increase in premixed ratio increases both the in-cylinder peak pressure and heat release rate. Percentage increase in peak pressure increase ranges from 5.9% to 18.5 % while percentage increase in peak heat release rate ranges from 32.3% to 74.3% when premixed ratio is increased from 0.25 to 0.75.
Premixed ratio of 0.25 is observed to be optimum as the decrease in brake thermal efficiency is only marginal compared to CIDI mode and the soot emission is lower than that of diesel. To decrease the NO\textsubscript{x} emission at high power outputs and control the start of combustion, EGR methodology was used.

5.4 PPCCI MODE OF COMBUSTION WITH DIESEL AS PREMIXED AND MAIN FUEL (D-D MODE) WITH PREMIXED RATIO OF 0.25 WITH 10%, 20% AND 30% EGR

When cooled exhaust gas is inducted with premixed diesel-air mixture NO\textsubscript{x} emissions are reduced due to lower combustion temperature and pressure. The exhaust gas acts as a heat absorption sink, primarily due to the heat absorbing capacity of CO\textsubscript{2} (thermal effects) as well as through the dissociation of CO\textsubscript{2} (chemical effect) and results in a reduction in combustion pressure and temperature. The combustion products CO\textsubscript{2} and H\textsubscript{2}O which have higher specific heats compared to air as shown in Table 5.3 reduce the combustion temperature and chemical kinetic reaction rate. Hence, EGR is very crucial in controlling the combustion phase in PPCCI combustion.

EGR can extend the operating range of PPCCI combustion to higher equivalence ratios (Ghazikhani et al 2009). The main constituents of EGR are CO\textsubscript{2}, H\textsubscript{2}O, N\textsubscript{2} and O\textsubscript{2}, partially burned gases such as CO, particulate matter and UBHC. Generally, the chemical constituents present in EGR influence the amount of OH radicals in the cylinder. Since, at higher equivalence ratios more OH radicals are formed, the effect of chemical constituents of EGR in the production of OH radicals is negligible at higher equivalence ratios. Accordingly, the main effect of EGR at high equivalence
ratios is dilution of in-cylinder mixture and reduction of maximum combustion temperature. Therefore, EGR can extend the operating range of HCCI combustion to higher equivalence ratios. At lower equivalence ratios less OH radicals are formed. Therefore, the effect of chemical constituents present in EGR to produce OH radicals is stronger and increases the overall reactivity (Ghazikhani et al 2009). Premixed ratio of 0.25 was selected for further experimental work with EGR as the decrease in brake thermal efficiency was minimum and increase in NOx, UBHC and CO emissions was only marginal.

5.4.1 Specific Energy Consumption and Brake Thermal Efficiency

Figures 5.46 and 5.47 show the variation of SEC and brake thermal efficiency with brake power respectively for CIDI mode and PPCCI mode with 0.25 premixed ratio without EGR and with 10%, 20% and 30% EGR.

![Figure 5.46 Variation of Specific Energy Consumption with Brake Power](image-url)
It is observed that in PPCCI mode, the SEC increases and brake thermal efficiency decreases as the percentage of EGR increases compared to base line diesel mode. The brake thermal efficiency at rated power output without EGR and with 10%, 20%, and 30% EGR for premixed ratio of 0.25 ranges from 27.2 to 25.3 % compared to 28.4 % for CIDI mode. The SEC for the same operating conditions range from 13,238 to 16,165 kJ/kWh compared to 12,667 kJ/kWh in CIDI mode.

![Graph](image)

**Figure 5.47  Variation of Brake Thermal Efficiency with Brake Power**

As premixed charge is formed externally, EGR is useful to control the start of combustion and rapid rate of pressure rise in PPCCI combustion mode. When EGR is increased, the dilution of fuel air mixture with exhaust gases increases and retards combustion. EGR also reduces the oxygen available and heat release in combustion reactions resulting in an increase in the quantity of unburned fuel. Increase in UBHC and CO emissions in PPCCI
mode may also contribute to decrease in brake thermal efficiency (Ganesh & Nagarajan 2010). The effect is more pronounced at higher EGR percentages.

### 5.4.2 Exhaust Gas Temperature

The variation of exhaust gas temperature with brake power for CIDI mode and PPCCI mode with premixed ratio of 0.25 without EGR and with 10%, 20% and 30% EGR is shown in Figure 5.48.

![Variation of Exhaust Gas Temperature with Brake Power](image)

**Figure 5.48** Variation of Exhaust Gas Temperature with Brake Power

The exhaust gas temperature is high at rated power output for D-D Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode. It can also be observed from Figure 5.42 that the pressure rise starts earlier at rated power output which indicates early combustion. In-cylinder peak pressure also increases and occurs closer to TDC. This may result in higher combustion temperature leading to higher exhaust gas temperature for D-D Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode. When EGR percentage is increased, the in-cylinder pressure is
decreased at rated power output as observed in Figure 5.53. As the quantity of EGR increases, the specific heat capacity of the charge increases due to higher specific heat capacity of re-circulated H₂O and CO₂ constituents compared to air. This results in lower peak combustion temperature particularly with high EGR percentage. Hence, the exhaust gas temperature decreases with increase in EGR percentage. The exhaust gas temperature at rated power output without EGR and with 10%, 20%, and 30% EGR for premixed ratio of 0.25 ranges from 470°C to 421°C compared to 445°C for CIDI mode.

5.4.3 Unburnt Hydrocarbon and Carbonmonoxide Emissions

The major disadvantage of PPCCI engine is its higher UBHC and CO emissions. Figures 5.49 and 5.50 show the variation of UBHC and CO with power output for CIDI mode and PPCCI mode with premixed ratio of 0.25 without EGR and with 10%, 20% and 30% EGR.

![Graph showing variation of Unburnt Hydrocarbon Emission with Brake Power](image)

**Figure 5.49** Variation of Unburnt Hydrocarbon Emission with Brake Power
With increase in EGR rate, the combustion reaction rate is reduced due to lack of oxygen and the mean temperature in the cylinder would decrease. CO to CO₂ oxidation reactions in PPCCI are sensitive to combustion temperature and a minimum temperature of 1127 °C is required to oxidise CO (Johnson 2006).

The combustion reactions are incomplete resulting in an increase in UBHC and CO emissions. As discussed in 5.3.3 and 5.3.4, the effects of crevice volume, absorption and desorption of fuel into oil layers on the cylinder wall and flame quenching may also result in high UBHC and CO emissions.

![Figure 5.50 Variation of Carbonmonoxide Emission with Brake Power](image)

The UBHC emissions vary from 1.0 to 1.3 g/kWh at rated power output in PPCCI mode without EGR and with 10%, 20%, and 30% EGR, while it is about 0.7 g/kWh for CIDI mode. The CO emissions vary from
18.6 to 36.4 g/kWh at rated power output in PPCCI mode without EGR and with 10%, 20%, and 30% EGR compared to 16.7 g/kWh in CIDI mode.

5.4.4 Oxides of Nitrogen Emission

The variation of NO\textsubscript{x} emission with brake power for CIDI mode and PPCCI mode with premixed ratio of 0.25 for various percentages of EGR and without EGR is shown in Figure 5.51. With EGR, NO\textsubscript{x} emission is reduced at all power outputs compared to CIDI mode and PPCCI mode with premixed ratio of 0.25 without EGR.

![Figure 5.51 Variation of Oxides of Nitrogen Emissions with Brake Power](image)

The increase in NO\textsubscript{x} emission in D-D mode without EGR compared to CIDI mode at rated power output is due to high heat release rate, temperature and higher equivalence ratio for the given premixed ratio. With EGR, part of the intake air is displaced with exhaust containing CO\textsubscript{2} and H\textsubscript{2}O
which have higher specific heats and reduce the peak combustion temperature. EGR also reduces the available oxygen and heat release in combustion reactions. As EGR percentage is increased, lower pressure rise rates, longer combustion duration and lower peak combustion temperature are observed (Luke Franklin 2010). All these factors limit NO\textsubscript{x} production rates.

At rated power output in PPCCI mode without EGR and with 10%, 20%, and 30% EGR, the NO\textsubscript{x} emissions range from 8.8 to 3.1 g/kWh whereas in CIDI mode it is about 7.3 g/kWh.

5.4.5 Soot Emissions

Figure 5.52 shows the soot emissions of the engine operated in PPCCI mode with and without EGR compared to CIDI mode. Soot emission in PPCCI mode without EGR is significantly low compared to CIDI mode.

![Figure 5.52 Variation of Soot emission with Brake Power](image)
With increase in EGR, in PPCCI mode, soot emission increases at all power outputs due to depletion of oxygen and decrease in heat release rate. The dilution effect delays the timing of low temperature reactions and suppresses the advance of start of combustion (SOC). Consequently, the peak combustion temperature is also reduced and results in an increase in soot emissions. The formation of soot and its oxidation are mainly influenced by the engine operating conditions. With 10% and 20% EGR, soot emissions are lower than that of CIDI mode but higher than PPCCI mode without EGR. Dae & Chang (2006) reported that EGR rates over 30% give rise to sharp increase in soot emissions. In the present work, as the EGR is increased beyond 20% with 0.25 premixed ratio, soot emissions are higher than that of CIDI mode due to decrease in the availability of oxygen and lower combustion temperatures. This limits the maximum EGR in PPCCI mode with premixed ratio of 0.25. In PPCCI mode, soot emissions are 86, 94, 134 and 228 mg/m³ at rated power output without EGR and with 10%, 20%, and 30% EGR respectively compared to 166 mg/m³ in CIDI mode.

5.4.6 Pressure - Crank Angle Diagram

Figure 5.53 shows the pressure crank angle diagram at rated power output with diesel as premixed fuel and main fuel for premixed ratio of 0.25 with 10, 20 and 30% EGR and without EGR.

It can be observed that with EGR, the combustion starts earlier in PPCCI mode compared to CIDI mode. High specific heats of gases with EGR reduce the in-cylinder pressure and temperature. The decrease in oxygen concentration limits the availability of oxygen for combustion. Therefore the start of combustion is retarded as EGR percentage is increased. The peak pressure also decreases as EGR rate is increased.
Figure 5.53 Variation of In-cylinder Pressure with Crank Angle at Rated Power Output

At rated power the start of combustion is between 9 to 5 °CA bTDC for premixed ratio of 0.25 without EGR and with 10%, 20% and 30% EGR compared to 7 °CA bTDC in CIDI mode. In PPCCI engines, EGR serves to control the start of combustion and a reduction in heat release rate inhibiting rapid pressure rise.

5.4.7 Peak pressure

The variation of peak pressure with brake power for CIDI mode and PPCCI mode with premixed ratio of 0.25 without EGR and with various percentages of EGR is shown in Figure 5.54.

As discussed in section 5.3.8, combustion starts earlier and the peak pressure is high at rated power output for D-D Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode.
Figure 5.54 Variation of Peak Pressure with Brake Power

It can be observed that the peak pressure decreases with increase in EGR compared to PPCCI mode without EGR. EGR act as a thermal sink controlling the heat release rate and inhibiting rapid pressure rise.

Lack of oxygen is also responsible for the reduction in peak pressure. The peak pressure varies from 73.7 to 69.3 bar in PPCCI mode with 10% to 30% EGR compared to 75.4 bar without EGR.

5.4.8 Heat Release Rate

Figure 5.55 shows the rate of heat release rate at various crank angles at rated power with diesel as premixed fuel and main fuel for premixed ratio of 0.25 without EGR and with 10, 20 and 30% EGR. Figure 5.60 shows the LTR, HTR and interval between LTR and HTR at rated power output for premixed ratio of 0.25 without EGR and with 10, 20 and 30% EGR in PPCCI mode compared to CIDI mode.
Figure 5.55  Variation of Heat Release Rate with Crank Angle at Rated Power Output

It is observed that the start and peak of LTR are not affected by increase in EGR percentage. Only with higher percentage of EGR, the start of LTR is retarded marginally as shown in Figure 5.56.

But peak heat release rates during HTR decrease and start of HTR is retarded as the percentage of EGR is increased at rated power output as observed from Figures 5.55 and 5.56. It is observed from Figure 5.56 that the Negative Temperature Coefficient (NTC) and hence the combustion duration increases as EGR percentage increases.

The EGR act as a thermal sink absorbing the heat present and reduce the cylinder pressure and temperature. This lowers the heat release rate. The peak heat release rates during HTR at rated power are 102.5, 98.2, 92.2 and 74.0 J/ºCA occurring at 9, 8, 7 and 7 ºCA bTDC for premixed ratios of 0.25 without EGR, with 10%, 20% and 30% respectively while the peak
heat release rates during LTR vary from 4.2 to 4.8 J/°CA occurring at nearly 25 °CA bTDC for all the premixed ratios.

**Figure 5.56**  LTR, HTR and Interval between LTR and HTR with Various Percentages of EGR at Rated Power Output for D-D mode

The time interval between LTR and HTR is 16, 17, 17 and 18 °CA for premixed ratios of 0.25 without EGR, with 10%, 20% and 30% respectively.

**5.4.9 Summary**

Table 5.6 shows the comparison of D-D Mode in PPCCI operation with EGR in CIDI mode.
Table 5.6 Percentage Variation of Various Parameters in D-D mode with EGR from CIDI Operation for Various Premixed Ratios

<table>
<thead>
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<th></th>
<th>Premixed Ratio of 0.25</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>without EGR</td>
</tr>
<tr>
<td>Brake Thermal Efficiency</td>
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</tr>
<tr>
<td>Exhaust Gas Temperature</td>
<td>5.6</td>
</tr>
<tr>
<td>Unburnt Hydrocarbon</td>
<td>34.9</td>
</tr>
<tr>
<td>Carbonmonoxide</td>
<td>11.4</td>
</tr>
<tr>
<td>Oxides of Nitrogen</td>
<td>20.5</td>
</tr>
<tr>
<td>Soot</td>
<td>-48.2</td>
</tr>
<tr>
<td>In-cylinder Peak Pressure</td>
<td>5.9</td>
</tr>
<tr>
<td>Peak Heat Release Rate</td>
<td>32.2</td>
</tr>
</tbody>
</table>

The following is the summary of the results of D-D mode with EGR at rated power output for premixed ratio of 0.25 compared to CIDI mode:

- Percentage decreases in brake thermal efficiency are observed to be 5.5\%, 6.8\% and 10.7\% with 10\%, 20\% and 30 \% EGR respectively compared to CIDI mode.

- Percentage increase in UBHC emissions are found to be 54.0 \%, 65.6\% and 83.0\% and percentage increase in CO emissions are 34.0\%, 63.8\% and 117.3\% with 10\%, 20\% and 30 \% EGR respectively compared to CIDI mode.

- Percentage decreases in NO\textsubscript{x} emission are 18.8\%, 47.7\% and 58.2\% with 10\%, 20\% and 30 \% EGR compared to CIDI mode.

- Soot emission decreases up to 20\% EGR and increases when EGR is increased beyond 20\%. Percentage decreases in soot emission
with 10% and 20% EGR are 43.4% and 19.3% respectively while 30% EGR result in an percentage increase of 37.3% compared to CIDI mode.

- Percentage increase in cylinder peak pressure is found to be 3.5% with 10% EGR while percentage reduction with 20% and 30% are observed to be 0.1% and 2.6% respectively compared to CIDI mode.

- Percentage increases in peak heat release rate are found to be 26.7% and 19% with 10% and 20% EGR while percentage decrease for 30% EGR is found to be 4.5% compared to CIDI mode.

UBHC, CO and Soot emissions increase while, NOx emissions decrease as EGR percentage increases. As the EGR is increased beyond 20% with 0.25 premixed ratio, soot emissions are higher than that of CIDI mode. Hence 20% EGR is found to be optimum that can be used in PPCCI mode with premixed ratio of 0.25. Brake thermal efficiency is marginally decreased compared to CIDI mode.

5.5 PPCCI MODE OF COMBUSTION WITH DIESEL AS PREMIXED FUEL WITH PREMIXED RATIOS (R_p) OF 0.25, 0.50 & 0.75 AND 20%JOME/20%FOME AS MAIN FUEL (D-20J AND D-20F MODE)

5.5.1 Specific Energy Consumption and Brake Thermal Efficiency

Figure 5.57 and 5.58 shows the variation of SEC and brake thermal efficiency with brake power respectively in PPCCI mode for D-20J for various values of premixed ratios. Figures 5.59 and 5.60 show the same for D-20F.
Figure 5.57  Variation of Specific Energy Consumption with Brake Power

Figure 5.58  Variation of Brake Thermal Efficiency with Brake Power
Figure 5.59 Variation of Specific Energy Consumption with Brake Power

Figure 5.60 Variation of Brake Thermal Efficiency with Brake Power
It can be observed that at any given power output the SEC is higher and brake thermal efficiency is lower in PPCCI mode with D-20J and D-20F compared to CIDI mode. With D-20F, the brake thermal efficiency decreases significantly compared to D-20J. As explained earlier in section 5.3.1, external mixture formation results in poor vapourisation due to low volatilility of diesel fuel and wall wetting. Start of combustion advancement and excessive peak pressures due to rapid rates of heat release result in high compression work leading to higher fuel consumption and reduced thermal efficiency.

Higher viscosities and lower calorific values of 20% JOME (3.7 cSt, 39,000 kJ/kg) and 20% FOME blends (3.9 cSt, 38700 kJ/kg) compared to that of diesel (3.5 cSt, 43500 kJ/kg) result in further increase in the quantity of blends of methyl esters injected in the cylinder compared to diesel for the same power output.

When the premixed ratio is increased at any given power output, the brake thermal efficiency of both the blends (D-20J and D-20F) in PPCCI mode decreases. The combustion also advances in both D-20J and D-20F mode of operation due to their higher cetane number leading to higher compression work. All these factors contribute to increase in SEC and decrease in brake thermal efficiency.

From Figure 5.71 and 5.72, it can be observed that D-20F shows comparatively higher pressure than D-20J and diesel which increases the compression work for D-20F. FOME is having a long carbon chain fatty acid content of 36.7 % in the range of C20-C22 where as it is not present in JOME. This long carbon chain fatty acid components burn late in the expansion stroke which increases the temperature of exhaust gases.

SEC varies from 13,420 to 16,636 kJ/kWh for D-20J mode and from 13,652 to 16,823 kJ/kWh for D-20F for premixed ratios of 0.25, 0.50
and 0.75 compared to 12,662 kJ/kWh in CIDI mode at rated power output. At rated power output, the brake thermal efficiency in D-20J mode varies from 26.8% to 21.6% and in D-20F mode from 26.4% to 21.4% when premixed ratio is varied from 0.25 to 0.75 compared to 28.4% in CIDI mode.

5.5.2 Exhaust Gas Temperature

Figure 5.61 and 5.62 shows the variation of exhaust gas temperature with brake power in PPCCI mode for various premixed ratios tested with D-20J and D-20F respectively. In PPCCI mode of operation, combustion takes place over the entire cylinder charge and the exhaust gas temperature is lower compared to CIDI mode up to 50% of rated power output for both D-20J and D-20F.

In PPCCI mode of operation, combustion takes place in the entire cylinder charge which results in low combustion temperatures at part loads for both D-20J and D-20F mode. Hence, the exhaust gas temperature is lower for both D-20J and D-20F mode compared to CIDI mode up to 50% of rated power output. The exhaust gas temperature at 50% of rated power output in D-20J is 284°C, 270°C, 265°C and 269°C in CIDI mode and premixed ratios of 0.25, 0.5 and 0.75 respectively. In D-20F mode, the corresponding values are 284°C, 272°C, 267°C and 270°C.

Exhaust gas temperature is higher at higher power output for all premixed ratios and increases as premixed ratio is increased. At higher power output the equivalence ratio is higher which increases the heat release and results in higher combustion pressure and temperature. The exhaust gas temperature at rated power output in D-20J is 445°C, 485°C, 502°C and 535°C in CIDI mode and premixed ratios of 0.25, 0.5 and 0.75 respectively. In D-20F mode, the corresponding values are 445°C, 490°C, 510°C and 542°C.
Figure 5.61  Variation of Exhaust Gas Temperature with Brake Power

Figure 5.62  Variation of Exhaust Gas Temperature with Brake Power
It can be observed from Figure 5.71 and 5.72 that the pressure rise starts earlier for all premixed ratios at rated power output in D-20J and D-20F mode compared to CIDI mode which indicates early combustion. In-cylinder peak pressure also increases and occurs closer to TDC in D-20J and D-20F mode compared to CIDI mode. This may result in higher combustion temperature leading to higher exhaust gas temperature. With D-20F, the exhaust gas temperature is high at all power outputs compared to D-20J due to its higher molecular fatty acid components which burn relatively in the later part of the expansion stroke.

5.5.3 Unburnt Hydrocarbon Emissions

Figure 5.63 and 5.64 shows the variation of UBHC emissions with brake power in PPCCI mode for various premixed ratios for D-20J and D-20F respectively in comparison with CIDI mode. It is observed that UBHC emissions are higher in both D-20J and D-20F compared to CIDI mode at all the power outputs.

![Figure 5.63 Variation of Unburnt Hydrocarbon Emissions with Brake Power](image-url)
As the premixed ratio in PPCCI mode is increased the fuel introduced in the inlet manifold increases. Due to poor volatility characteristics of premixed fuel, vapourisation is poor and UBHC emissions increase in both D-20J and D-20F mode compared to CIDI mode.

![Graph showing variation of Unburnt Hydrocarbon Emissions with Brake Power](image)

**Figure 5.64** Variation of Unburnt Hydrocarbon Emissions with Brake Power

The crevice volume, flame quenching by the cooler cylinder wall temperatures, absorption and desorption of fuel vapour into oil layers and incomplete combustion of premixed fuel are also the reasons for higher UBHC emissions. Oxygen content of 20% JOME (2.1% by weight) is high compared to 20% FOME (1.6% by weight) and hence UBHC emissions are lower in D-20J compared to D-20F.
The UBHC emissions at rated power output vary from 0.9 to 1.6 g/kWh in D-20J while in D-20F mode the variations are from 1.0 to 1.6 g/kWh when premixed ratio is varied from 0.25 to 0.75 compared to 0.7 g/kWh in CIDI mode.

5.5.4 Carbonmonoxide Emissions

Figure 5.65 and 5.66 shows the variation of CO emission with brake power in PPCCI mode for D-20J and D-20F respectively for various premixed ratios. It can be observed that in both D-20J and D-20F, CO emission is higher at all the power outputs compared to base line diesel mode.

In PPCCI combustion mode, the in-cylinder temperature during combustion is low which will affect the oxidation of fuel released from the crevice volume (Dec & Sjoberg 2003).

![Graph showing variation of Carbonmonoxide Emission with Brake Power](image-url)

**Figure 5.65  Variation of Carbonmonoxide Emission with Brake Power**
The OH radical concentration which is one of the important factors for conversion of CO to CO\textsubscript{2} is reduced due to incomplete combustion in intermediate temperature regions (Ghazikhani et al 2009). Poor vapourisation of premixed fuel injected in the manifold may also increase CO emissions. Hence, CO emission increases in PPCCI combustion mode compared to CIDI mode. In both D-20J and D-20F, CO emission increases as the premixed ratio is increased. D-20F mode gives rise to higher CO emissions as its intrinsic oxygen content is lower compared to that of D-20J mode.

![Figure 5.66 Variation of Carbonmonoxide Emission with Brake Power](image)

At rated power output, the CO emission in D-20J varies from 17.6 to 30.6 g/kWh and in D-20F it varies from 17.9 to 32.6 g/kWh when the premixed ratio is increased from 0.25 to 0.75 compared to 16.7 g/kWh in CIDI mode.
5.5.5 Oxides of Nitrogen Emissions

The variation of NO\textsubscript{x} emission with brake power in PPCCI mode for D-20J and D-20F for various premixed ratios is shown in Figure 5.67 and 5.68 respectively. NO\textsubscript{x} emission decreases in PPCCI mode of operation in both D-20J and D-20F at lower power outputs and is high at high power outputs compared to CIDI mode.

At lower power outputs, PPCCI combustion lowers the in-cylinder temperature and increases the homogeneity by giving sufficient time for air and fuel to mix unlike in CIDI mode. Hence, NO\textsubscript{x} emissions are significantly decreased compared to CIDI mode (Suyin Gan et al 2011). The NO\textsubscript{x} emission for premixed ratios of 0.25, 0.5 and 0.75 in D-20J mode at 25 % of rated power (1.1 kW) output ranges from 13.0 g/kWh to 10.9 g/kWh compared to 15.6 g/kWh in CIDI mode. At 50% of rated power (2.2 kW) NO\textsubscript{x} emission varies from 11.0 g/kWh to 9.3 g/kWh in D-20J compared to 12.2 g/kWh in CIDI mode. In D-20F the variation is from 13.2 to 11.1 g/kWh at 25 % of rated power and from 11.2 to 9.7 g/kWh for 50% of rated power output.

It is also observed from Figure 5.75 and 5.76 that the peak heat release rate at rated power output in D-20J and D-20F mode is higher and also occurs earlier compared to CIDI mode resulting in higher NO\textsubscript{x} emissions.

However, NO\textsubscript{x} emission at rated power output in D-20J mode varies from 9.1 to 12.5 g/kWh when premixed ratio is increased from 0.25 to 0.75 compared to 7.3 g/kWh in diesel mode. In D-20F, the variation is from 9.3 to 12.9 g/kWh at rated power output.
Figure 5.67 Variation of Oxides of Nitrogen Emissions with Brake Power

Figure 5.68 Variation of Oxides of Nitrogen Emissions with Brake Power
It is observed that the in-cylinder peak pressures and heat release rate are higher in the case of D-20F at all power outputs resulting in higher NO\textsubscript{x} emissions compared to D-20J. This indicates that peak combustion temperatures may be higher for 20% FOME which may result in higher NO\textsubscript{x} emissions compared to 20% JOME in spite of the lower oxygen content of 20% FOME (1.6% by weight) compared to that of 20% JOME (2.1 % by weight).

5.5.6 Soot Emissions

Figure 5.69 and 5.70 shows the variation of soot emission with brake power in PPCCI mode for D-20J and D-20F respectively for various premixed ratios.

![Figure 5.69 Variation of Soot Emission with Brake Power](image)
It is observed that the soot emissions are lower at all power outputs in D-20J and D-20F mode when compared to CIDI mode due to homogeneous air fuel mixture inside the combustion chamber. Fuel rich regions inside the combustion chamber are eliminated in PPCCI mode compared to CIDI mode which results in lower soot emissions. Decrease in soot emission is also due to their intrinsic oxygen present in both the methyl ester blends. As the premixed ratio is increased the homogeneity of charge inside the cylinder increases and hence soot emissions are lower for higher values of premixed ratios. In D-20J, the soot emissions are lower at all the power outputs compared to D-20F due to its higher oxygen content, lower viscosity and lower long chain molecular compounds.

![Graph](https://via.placeholder.com/150)

**Figure 5.70  Variation of Soot Emission with Brake Power**

In D-20J mode, the soot emission at rated power output varies from 75 to 22 mg/m$^3$ while in D-20F mode it varies from 80 to 27 mg/m$^3$ when the
premixed ratio is increased from 0.25 to 0.75 in PPCCI mode of operation compared to 166 mg/m³ in diesel mode.

5.5.7 Pressure - Crank Angle Diagram

The pressure crank angle diagrams at rated power in PPCCI mode for D-20J and D-20F for various premixed ratios are shown in Figure 5.71 and 5.72 respectively.

The combustion starts earlier in D-20J and D-20F mode, the peak pressure is higher and occurs earlier as premixed ratio is increased compared to CIDI mode.

![Pressure - Crank Angle Diagram](image)

**Figure 5.71 Variation of In-cylinder Pressure with Crank Angle at Rated Power Output**
Figure 5.72 Variation of In-cylinder Pressure with Crank Angle at Rated Power Output

The cylinder pressure after combustion is high in both D-20J and D-20F modes compared to CIDI mode at all crank angles. This is due to the increase in the quantity of fuel at higher power outputs while the quantity of air inducted is nearly constant. This increases the fuel air ratio of the mixture in the manifold and results in higher peak pressures and also advances the start of combustion.

Both 20% JOME and 20% FOME have higher cetane numbers compared to diesel due to which combustion is advanced further compared to CIDI mode. Methyl esters are oxygenated fuels and their higher oxygen content results in better combustion and higher cylinder pressure. The effect is more pronounced at higher power outputs. It is observed that in D-20F mode, the in-cylinder pressures are marginally higher compared to D-20J in PPCCI mode.
At rated power output, the start of combustion occurs at 11, 15 and 16 °CA bTDC in D-20J for premixed ratios of 0.25, 0.5 and 0.75 respectively and in D-20F combustion starts at 10, 14 and 16 °CA bTDC for premixed ratios of 0.25, 0.5 and 0.75 respectively compared to 7 °CA bTDC in CIDI mode.

5.5.8 Peak Pressure

The peak pressures at various power outputs are shown in Figures 5.73 and 5.74 respectively. The peak pressures at rated power output are higher and occur earlier in the case of D-20J and D-20F compared to CIDI mode. The effect is more pronounced at higher premixed ratios.

![Peak Pressure vs Brake Power Graph](image-url)

**Figure 5.73** Variation of Peak Pressure with Brake Power
Figure 5.74  Variation of Peak Pressure with Brake Power

At rated power output, the peak pressure occurs between 6 and 4 °CA aTDC in D-20J while in D-20F it occurs between 6 and 2 °CA aTDC for premixed ratios of 0.25, 0.5 and 0.75 compared to 8 °CA aTDC in CIDI mode.

5.5.9 Heat Release Rate

The heat release rate at various crank angles at rated brake power for D-20J and D-20F respectively for various premixed ratios in PPCCI mode compared to CIDI mode is shown in Figures 5.75 and 5.76. As discussed in section 5.3.9 the heat release curves show two peaks - one peak of greater magnitude during HTR near TDC and another one of smaller magnitude during LTR nearly 20 to 30 °CA bTDC. The magnitude of LTR and the timing of LTR are dependent primarily on the fuel selection and the temperature time history of the fuel-air mixture.
Figure 5.75  Variation of Heat Release Rate with Crank Angle at Rated Power Output

Figure 5.76  Variation of Heat Release Rate with Crank Angle at Rated Power Output
HTR peak increases in magnitude as premixed ratio is increased and also occurs earlier. LTR peak also increases and occurs nearly at same crank angle for all the premixed ratios at any given power output. LTR occurs earlier as the power output is increased due to early start of cool flame reactions at higher power outputs. From Figure 5.77 and 5.78 it is observed that HTR occurs earlier as the premixed ratio is increased while LTR occurs at nearly the same crank angle for all the premixed ratios in both D-20J and D-20F mode.

Figure 5.77 LTR, HTR and Interval between LTR and HTR with Various Premixed Ratios at Rated Power Output for D-20J Mode

The peak heat release rates during HTR at rated power output for D-20J are 115.8, 124.5 and 152.5 J/°CA occurring at 11, 14 and 18 °CA bTDC for premixed ratios of 0.25, 0.5 and 0.75 respectively while peak heat
release rates during LTR are 5, 7.6 and 9.5 J/°CA occurring at nearly 24 °CA bTDC for all the premixed ratios. The time interval between the LTR and HTR is 14, 11 and 8 °CA for premixed ratios of 0.25, 0.5 and 0.75 respectively.

![Graph showing the relationship between crank angle and interval between LTR and HTR with various premixed ratios at rated power output for D-20F Mode.](image)

**Figure 5.78** LTR, HTR and Interval between LTR and HTR with Various Premixed Ratios at Rated Power Output for D-20F Mode

The peak heat release rates during HTR at rated power output in D-20F are 110.2, 120.1 and 145.2 J/°CA occurring at 10, 14 and 17 °CA bTDC for premixed ratios of 0.25, 0.5 and 0.75 respectively while peak heat release rates during LTR are 5.1, 7.5 and 9.0 J/°CA occurring at nearly 24 °CA bTDC for all the premixed ratios. The time interval between LTR and HTR is 15, 11 and 9 °CA for premixed ratios of 0.25, 0.5 and 0.75 respectively.
5.5.10 Summary

The comparison of D-20J and D-20F modes in PPCCI operation without EGR with CIDI operation for various premixed ratios is shown in Table 5.7.

Table 5.7 Percentage Variation of Various Parameters in D-20J and D-20F modes from CIDI Operation for Various Premixed Ratios

<table>
<thead>
<tr>
<th></th>
<th>D-20J mode</th>
<th></th>
<th>D-20F mode</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Premixed Ratios of</td>
<td></td>
<td>Premixed Ratios of</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.25</td>
<td>0.5</td>
<td>0.75</td>
<td>0.25</td>
</tr>
<tr>
<td>Brake Thermal Efficiency</td>
<td>-5.6</td>
<td>-18.3</td>
<td>-23.9</td>
<td>-7.4</td>
</tr>
<tr>
<td>Exhaust Gas Temperature</td>
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<td>12.8</td>
<td>20.2</td>
<td>10.1</td>
</tr>
<tr>
<td>Unburnt Hydrocarbon</td>
<td>23.0</td>
<td>61.8</td>
<td>112.3</td>
<td>30.0</td>
</tr>
<tr>
<td>Carbonmonoxide</td>
<td>5.1</td>
<td>30.1</td>
<td>82.5</td>
<td>7.0</td>
</tr>
<tr>
<td>Oxides of Nitrogen</td>
<td>24.7</td>
<td>52.1</td>
<td>71.2</td>
<td>27.4</td>
</tr>
<tr>
<td>Soot</td>
<td>-54.8</td>
<td>-65.1</td>
<td>-86.7</td>
<td>-51.8</td>
</tr>
<tr>
<td>In-cylinder Peak Pressure</td>
<td>12.2</td>
<td>17.8</td>
<td>23.3</td>
<td>14.0</td>
</tr>
<tr>
<td>Peak Heat Release Rate</td>
<td>49.4</td>
<td>60.6</td>
<td>96.8</td>
<td>42.2</td>
</tr>
</tbody>
</table>

- It is observed that both 20% methyl esters of JOME and FOME (D-20J and D-20F) can be used in PPCCI mode of operation.

- Brake thermal efficiency decreases marginally in both the modes and decreases with increase in premixed ratio. At rated power output, the percentage decreases in brake thermal efficiency in D-20J mode are found to be 5.6%, 18.3% and 23.9 % and in D-20F mode the percentage decreases are 7.4%, 19.4% and 24.6 % when premixed ratio is varied from 0.25 to 0.75 compared to baseline CIDI mode.
• NO\textsubscript{x} emissions are higher in both the modes. In D-20J the percentage increases are 24.7%, 52.1% and 71.2% while in the case of D-20F the percentage increases are 27.4%, 57.5% and 76.7% respectively for premixed ratios of 0.25, 0.50 and 0.75 compared to baseline CIDI mode.

• UBHC and CO emissions are higher in both the modes. The percentage increases in UBHC emission are 23.0%, 61.8% and 112.3% in D-20J while the percentage increase is 30.0%, 70.9% and 116.1% in D-20F for premixed ratios of 0.25, 0.50 and 0.75 respectively compared to baseline CIDI mode. The percentage increases in CO emission are observed to be 5.1%, 30.1% and 82.5% in D-20J while the percentage increases are 7.1%, 35.7% and 94.9% in D-20F for premixed ratios of 0.25, 0.50 and 0.75 respectively compared to baseline CIDI mode.

• Soot emissions decrease significantly for both the modes. In D-20J the percentage decreases are 54.8%, 65.1% and 86.7% while in D-20F the percentage decreases are 51.8%, 63.8% and 83.7% for premixed ratios of 0.25, 0.50 and 0.75 respectively compared to baseline CIDI mode. Soot emission is lower in D-20J compared to D-20F mode.

• Peak pressure and Heat release rates are higher in both D-20J and D-20F. The percentage increases in peak pressure are 12.2%, 17.8% and 23.3% in D-20J while the percentage increases are 14.0%, 21.3% and 26.1% in D-20F for premixed ratios of 0.25, 0.50 and 0.75 respectively compared to baseline CIDI mode. The percentage peak heat release rate increases by 49.4%, 60.6% and 96.8% in D-20J while the corresponding values are 42.2%, 55.0%
and 87.3% in D-20F for premixed ratios of 0.25, 0.50 and 0.75 respectively compared to baseline CIDI mode.

Premixed ratio of 0.25 is observed to be optimum in both D-20J and D-20F as the decrease in brake thermal efficiency is only marginal compared to CIDI mode and the soot emission is lower than that of CIDI mode. D-20J mode is observed to be better than D-20F mode comparing the performance, combustion and emission characteristics. EGR methodology was adopted to decrease the NO\textsubscript{x} emission at high power outputs and to control the start of combustion.

5.6 PPCCI MODE OF COMBUSTION WITH DIESEL AS PREMIXED FUEL WITH PREMIXED RATIO OF 0.25 AND 20\%JOME/20\%FOME AS MAIN FUEL WITH 10\%, 20\% AND 30\% EGR

5.6.1 Specific Energy Consumption and Brake Thermal Efficiency

When diesel is used as premixed fuel and 20\% methyl ester as main fuel with various premixed ratios it is observed that NO\textsubscript{x} emissions are higher near rated power output. Cooled EGR is one of the most effective technique for reducing NO\textsubscript{x} in PPCCI mode. EGR partially replaces O\textsubscript{2} and N\textsubscript{2} in fresh air with CO\textsubscript{2} and H\textsubscript{2}O from the engine exhaust increasing the specific heat capacity of the exhaust gases. Reduction of oxygen concentration and increase in specific heat capacity result in decrease in NO\textsubscript{x} emissions. EGR also controls the start of combustion and as the EGR percentage increases SOC is retarded in PPCCI combustion mode.

Figure 5.79 and 5.80 shows the variation of SEC and brake thermal efficiency with brake power respectively for CIDI mode and D-20J mode with premixed ratio of 0.25 without EGR and with 10\%, 20\% and 30\% EGR while Figures 5.81 and 5.82 shows the same for D-20F.
Figure 5.79  Variation of Specific Energy Consumption with Brake Power

Figure 5.80  Variation of Brake Thermal Efficiency with Brake Power
Figure 5.81  Variation of Specific Energy Consumption with Brake Power

Figure 5.82  Variation of Brake Thermal Efficiency with Brake Power
In both D-20J and D-20F, as the percentage of EGR increases, the SEC increases and brake thermal efficiency decreases compared to CIDI mode. When EGR is introduced the fuel air mixture is diluted and the decrease in the availability of oxygen retards the combustion. The heat release in combustion reactions is decreased and the quantity of unburned fuel is relatively large. As EGR increases, the brake thermal efficiency decreases. SEC is higher and brake thermal efficiency is lower in D-20F compared to D-20J due to lower intrinsic oxygen concentration of D-20F.

The SEC varies from 13,420 to 14,417 kJ/kWh in D-20J and from 13,652 to 14,746 kJ/kWh in D-20F for premixed ratio of 0.25 without EGR and with 10% 20% and 30% EGR compared to 12,661 kJ/kWh in CIDI mode at rated power output. The brake thermal efficiency varies from 26.8 % to 25.0% in D-20J mode while it varies from 26.4 % to 24.4 % for premixed ratio of 0.25 without EGR and with 10% 20% and 30% EGR compared to 28.4 % in CIDI mode.

5.6.2 Exhaust Gas Temperature

The variation of exhaust gas temperature with brake power for D-20J and D-20F respectively with premixed ratio of 0.25 with 10%, 20% and 30% EGR and without EGR compared with CIDI mode is shown in Figure 5.83 and Figure 5.84.

As discussed in section 5.5.2, combustion starts earlier resulting in higher in-cylinder temperature and pressure. These results in higher exhaust gas temperature at rated power output for both D-20J Mode and D-20F Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode.
Figure 5.83  Variation of Exhaust Gas Temperature with Brake Power

Figure 5.84  Variation of Exhaust Gas Temperature with Brake Power
With EGR, the specific heat capacities of re-circulated H₂O and CO₂ constituents increase resulting in lower peak combustion temperature. The effect is more pronounced at higher EGR percentages.

At rated power output, the exhaust gas temperature varies from 444°C to 431°C with 10%, 20%, and 30% EGR for premixed ratio of 0.25 compared to 485°C without EGR in D-20J mode while the exhaust gas temperature varies from 445°C to 435°C with 10%, 20%, and 30% EGR for premixed ratio of 0.25 compared to 490°C without EGR in D-20F mode.

5.6.3 Unburnt Hydrocarbon and Carbonmonoxide Emissions

Figures 5.85 and 5.86 show the variation of UBHC with brake power for CIDI mode, for PPCCI mode with D-20J and D-20F respectively with 10, 20, 30% EGR and without EGR. The variation of CO emissions is shown in Figures 5.87 and 5.88 for the same operating conditions.

Reduction of oxygen with EGR reduces the combustion reaction rate and temperature inside the cylinder. The burned gas temperature is low which results in increased emissions of UBHC and CO compared to CIDI mode. The peak temperatures are also relatively low to complete the oxidation of CO to CO₂.

Due to lower inlet temperatures, premixed diesel fuel and in-cylinder injection of methyl ester blends are not completely evaporated which also leads to higher UBHC and CO. As already discussed in section 5.3.3 and 5.3.4, the effects of crevice volume and flame quenching may also be responsible for high UBHC and CO emissions.
Figure 5.85 Variation of Unburnt Hydrocarbon Emissions with Brake Power

Figure 5.86 Variation of Unburnt Hydrocarbon Emissions with Brake Power
Figure 5.87  Variation of Carbonmonoxide Emission with Brake Power

Figure 5.88  Variation of Carbonmonoxide Emission with Brake Power
The UBHC emissions vary from 0.9 to 1.2 g/kWh in D-20J and from 1.0 to 1.3 g/kWh in D-20F for premixed ratio of 0.25 without EGR and with 10% 20% and 30% EGR compared to 0.7 g/kWh in CIDI mode at rated power output and the CO emissions vary from 17.6 to 30.9 g/kWh in D-20J mode while it varies from 17.9 to 32.8 g/kWh compared to 16.7 g/kWh in CIDI mode.

5.6.4 Oxides of Nitrogen Emissions

Figures 5.89 and 5.90 show the variation of NO\textsubscript{x} emissions with brake power for CIDI mode and PPCCI mode with D-20J and D-20F respectively with 10, 20, 30% EGR and without EGR.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure589.png}
\caption{Variation of Oxides of Nitrogen Emissions with Brake Power}
\end{figure}

As discussed in section 5.5.5, combustion starts earlier resulting in higher heat release rate, in-cylinder temperature and pressure at rated power output. These results in higher NO\textsubscript{x} formation at rated power output for both
D-20J Mode and D-20F Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode.

Recirculation of exhaust gas reduces the NO\textsubscript{x} emission at all the power outputs compared to CIDI mode as oxygen available for the formation of NO\textsubscript{x} is reduced when using EGR. Peak combustion pressure and temperatures are reduced as EGR percentage increases.

![Oxides of Nitrogen Emissions with Brake Power](image)

**Figure 5.90 Variation of Oxides of Nitrogen Emissions with Brake Power**

At rated power output in D-20J mode with 10%, 20%, and 30% EGR, the NO\textsubscript{x} emissions range from 6.2 to 3.33 g/kWh compared to 9.1 g/kWh without EGR. In D-20F, the values are 6.5 to 3.7 g/kWh with 10%, 20%, and 30% EGR and 9.3 g/kWh without EGR whereas in CIDI mode it is about 7.3 g/kWh.
5.6.5 Soot Emissions

The variation of soot emission with brake power for PPCCI mode of operation with D-20J and D-20F respectively with 10, 20, 30% EGR and without EGR is shown in Figures 5.91 and 5.92. Soot emission in PPCCI mode with EGR is higher compared to that of without EGR in both D-20J and D-20F. The increase in soot emission is due to reduction in oxygen content and decrease in heat release rate with EGR at all power outputs.

![Graph showing variation of soot emission with brake power](image)

**Figure 5.91 Variation of Soot Emission with Brake Power**

It is observed that with 10% and 20% EGR, soot emissions are lower than that of CIDI mode but higher than PPCCI mode without EGR. Soot emissions are higher than that of CIDI mode when EGR is increased beyond 20%. Hence, the quantity of EGR that can be re-circulated in PPCCI mode with premixed ratio of 0.25 is limited to 20% in the present work.
Figure 5.92  Variation of Soot Emission with Brake Power

In PPCCI mode, the soot emissions in D-20J mode at rated power output vary from 82 to 208 mg/m³ with 10%, 20% and 30% EGR while it is 75 mg/m³ without EGR. The corresponding values for D-20F mode are 86 to 220 mg/m³ and 80 mg/m³ without EGR compared to 166 mg/m³ in CIDI mode.

5.6.6  Pressure - Crank Angle Diagram

Figures 5.93 and 5.94 show the pressure crank angle diagram at rated power output for premixed ratio of 0.25 without EGR and with 10, 20 and 30% EGR in D-20J and D-20F mode respectively.

The in-cylinder pressure and temperature are reduced due to higher specific heats of CO₂ and H₂O compared to air, with EGR. The decrease in oxygen concentration limits the availability of oxygen for combustion.
Figure 5.93 Variation of In-cylinder Pressure with Crank Angle at Rated Power Output

Figure 5.94 Variation of In-cylinder Pressure with Crank Angle at Rated Power Output
It can be observed that the start of combustion is retarded with increasing EGR in PPCCI mode. The peak pressure also decreases as the EGR percentage is increased. It is observed that D-20F mode shows higher cylinder pressures compared to D-20J mode.

At rated power output, the start of combustion occurs between 11 and 6 °CA bTDC in D-20J in premixed ratio of 0.25 without EGR and with 10%, 20% and 30% EGR and in D-20F combustion occurs between 10 and 6 °CA bTDC for premixed ratio of 0.25 without EGR and with 10%, 20% and 30% EGR compared to 7 °CA bTDC in CIDI mode.

5.6.7 Peak Pressure

The variation of peak pressure with brake power for D-20J and D-20F mode with premixed ratio of 0.25 without EGR and with various percentages of EGR is shown in Figures 5.95 and 5.96.

As discussed in section 5.5.8, combustion starts earlier and the peak pressure is high at rated power output both D-20J Mode and D-20F Mode with premixed ratio of 0.25 without EGR compared to base line diesel mode.

It can be observed that the peak pressure decreases with increase in EGR compared to PPCCI mode without EGR. EGR act as a thermal sink controlling the heat release rate and inhibiting rapid pressure rise. Decrease in oxygen concentration is also responsible for the reduction in peak pressure. It is observed that D-20F mode shows higher in-cylinder peak pressures compared to D-20J mode.
Figure 5.95  Variation of Peak Pressure with Brake Power

Figure 5.96  Variation of Peak Pressure with Brake Power
The peak pressure varies from 77.6 to 68.9 bar in D-20J mode with 10% to 30% EGR compared to 80.0 bar without EGR. The peak pressure varies from 76.8 to 74.2 bar in D-20F mode with 10% to 30% EGR compared to 81.2 bar without EGR.

5.6.8 Heat Release Rate

Figures 5.97 and 5.98 show the heat release rate at various crank angles at rated power for D-20J and D-20F mode for premixed ratio of 0.25 without EGR and with 10, 20 and 30% EGR.

Figure 5.99 and 5.100 shows the LTR, HTR and interval between LTR and HTR in D-20J and D-20F mode respectively at rated power output for premixed ratio of 0.25 without EGR and with 10, 20 and 30% EGR.

![Heat Release Rate Graph](image)

**Figure 5.97** Variation of Heat Release Rate with Crank Angle at Rated Power Output
Figure 5.98 Variation of Heat Release Rate with Crank Angle at Rated Power Output

It is observed that the peak of LTR is not affected with increase in EGR percentage. But peak of HTR is significantly decreased as the percentage of EGR is increased. Increasing EGR percentage can delay both, the start of LTR and HTR as shown in Figure 5.99 and 5.100.

The EGR act as a thermal sink absorbing the heat present and lowers the heat release rate. The peak heat release rates during HTR at rated power in D-20J are 115.8, 109, 101.2 and 95.2 J/°CA occurring at 11 to 7 °CA bTDC for premixed ratios of 0.25 without EGR and with 10%, 20% and 30% while the peak heat release rates during LTR vary from 4.8 to 5.1 J/°CA occurring at nearly 25°CA bTDC for all the premixed ratios. The time interval between LTR and HTR varies from 14 to 17 °CA for premixed ratios of 0.25 without EGR, with 10%, 20% and 30% EGR respectively.
Figure 5.99 LTR, HTR and Interval between LTR and HTR with Various Percentages of EGR at Rated Power Output for D-20J mode

The peak heat release rates during HTR at rated power output in D-20F are 110.2, 104, 97.2 and 90.0 J/°CA occurring at 10 to 7 °CA bTDC for premixed ratios of 0.25 without EGR, with 10%, 20% and 30% EGR, while the peak heat release rates during LTR vary from 4.5 to 5.1 J/°CA occurring at nearly 25 °CA bTDC for all the premixed ratios. The time interval between LTR and HTR varies from 15 to 17 °CA for premixed ratios of 0.25 without EGR and with 10%, 20% and 30% EGR.
Figure 5.100  LTR, HTR and Interval between LTR and HTR with Various Percentages of EGR at Rated Power Output for D-20F mode

5.6.9 Summary

Table 5.8 shows the percentage variation of various parameters at rated power output for premixed ratio of 0.25 for various EGR percentages in D-20J and D-20F modes compared to CIDI mode.
Table 5.8 Percentage Variation of Various Parameters in D-20J and D-20F modes with EGR from CIDI Operation for Premixed Ratio of 0.25

<table>
<thead>
<tr>
<th></th>
<th>D-20J mode Premixed Ratio of 0.25</th>
<th></th>
<th>D-20F mode Premixed Ratio of 0.25</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>without EGR</td>
<td>with 10% EGR</td>
<td>with 20% EGR</td>
<td>with 30% EGR</td>
</tr>
<tr>
<td>Brake Thermal Efficiency</td>
<td>-5.5</td>
<td>-7.3</td>
<td>-9.2</td>
<td>-12.1</td>
</tr>
<tr>
<td>Exhaust Gas Temperature</td>
<td>9.0</td>
<td>-0.2</td>
<td>-1.3</td>
<td>-3.1</td>
</tr>
<tr>
<td>Unburnt Hydrocarbon</td>
<td>23.0</td>
<td>34.4</td>
<td>54.8</td>
<td>69.7</td>
</tr>
<tr>
<td>Carbonmonoxide</td>
<td>5.1</td>
<td>20.8</td>
<td>50.2</td>
<td>84.2</td>
</tr>
<tr>
<td>Oxides of Nitrogen</td>
<td>24.7</td>
<td>-14.8</td>
<td>-42.8</td>
<td>-54.3</td>
</tr>
<tr>
<td>Soot</td>
<td>-54.8</td>
<td>-50.6</td>
<td>-25.3</td>
<td>25.3</td>
</tr>
<tr>
<td>In-cylinder Peak Pressure</td>
<td>12.2</td>
<td>9.0</td>
<td>6.2</td>
<td>-3.1</td>
</tr>
<tr>
<td>Peak Heat Release Rate</td>
<td>49.4</td>
<td>40.6</td>
<td>30.6</td>
<td>22.8</td>
</tr>
</tbody>
</table>

The results are summarised as follows:

- Brake thermal efficiency decreases in both the modes and decreases with increase in EGR percentages. At rated power output, the percentage variations in brake thermal efficiency in D-20J mode are 7.3%, 9.2% and 12.1% with 10%, 20% and 30% EGR respectively compared to baseline diesel mode. For D-20F mode the corresponding percentage variations are 8.2%, 9.2% and 14.0% with 10%, 20% and 30% EGR respectively compared to baseline diesel mode.

- UBHC and CO emissions are higher in both the modes. Percentage increases in UBHC emission in D-20J mode are 34.4%, 54.8% and
69.8% while the percentage increases in D-20F are 49.7%, 61.5% and 74.0% with EGR 10%, 20% and 30% EGR respectively compared to baseline diesel mode. Percentage increases in CO emission in D-20J are 20.8%, 50.2% and 84.3% while the percentage increases in D-20F mode are 28.2%, 56.1% and 95.9% with 10%, 20% and 30% EGR respectively compared to baseline diesel mode.

- NOx emissions are lower in both the modes. Percentage decreases in NOx emission in D-20J mode are 14.8%, 42.8% and 54.3% while in D-20F the percentage decreases are 10.5%, 40.6% and 49.3% respectively with 10%, 20% and 30% EGR compared to baseline diesel mode.

- Soot emission decreases upto 20% EGR and increases when EGR is increased above 20% compared to CIDI mode. The percentage decreases in soot emission in D-20J mode are 50.6% and 25.3% with 10% and 20% EGR respectively while 30% EGR result in percentage increase of 25.3 % compared to baseline diesel mode. The percentage decreases in soot emission in D-20F are 48.2% and 21.1% with 10% and 20% EGR respectively while 30% EGR results in percentage increase of 32.5% compared to baseline diesel mode. Hence, 20% EGR is found to be optimum that can be used in PPCCI mode with premixed ratio of 0.25 in both D-20J and D-20F modes.

- In D-20J the peak pressure increases upto 20% EGR and decrease when EGR is increased to 30% compared to CIDI mode. The percentage increases in peak pressure in D-20J are 9.0% and 6.2% with 10% and 20% EGR respectively while 30% EGR results in a
percentage decrease of 3.1\% compared to baseline diesel mode. The percentage decreases in peak pressure in D-20F are 7.8\%, 5.9\% and 4.2\% with 10\%, 20\% and 30\% EGR respectively compared to baseline diesel mode.

- The percentage increases in Peak heat release rate in D-20J are 40.6\%, 30.6\% and 22.8\% while the percentage increases in D-20F are 34.2\%, 25.4\% and 16.0\% with 10\%, 20\% and 30\% EGR respectively compared to CIDI mode.

Premixed ratio of 0.25 with 20\% EGR is observed to be optimum in both D-20J and D-20F comparing the performance, combustion and emission characteristics.