CHAPTER 6

RESULTS AND DISCUSSION

6.1 COMBUSTION, PERFORMANCE AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELLED WITH NEAT NEO AND DIESEL

In order to evaluate the feasibility of *Azadirachta indica* seed oil, the performance and emission characteristics of the diesel engine operating on NeO are investigated and the results are compared with those of diesel.

6.1.1 Characterization of NeO and Diesel

The important fuel properties of diesel and neat NeO are shown in Table 6.1. The properties of oil have a strong relation with the chemical structure of fatty acid composition. The structure of fatty compounds in the fuel can affect many properties such as cetane number, viscosity, lower calorific value, density, bulk modulus and oxidation stability. Increasing the chain length increases the cetane number, calorific value, viscosity and bulk modulus of the fatty compounds. The calorific value, cetane number, viscosity and oxidation stability decreases whereas density and bulk modulus increases as the degree of unsaturation increases (Sara Pinzi et al 2013). The structural formula and fatty acid composition is shown in Table 6.2. The major fatty acid found in the NeO is oleic fatty acid.

The kinematic viscosity of NeO is found to be about 38 cSt, which is 10 times more than that of diesel. The kinematic viscosity, density and surface tension could influence the injection process as well as the process of
mixture formation. It has been reported by Sundarapandian et al (2007) that the surface tensions of diesel and NeO are 0.023 and 0.039 N/m respectively. The specific gravity of NeO is found to be 11.1% higher than that of diesel. This may be attributed to the higher molecular weights of triglyceride molecules present in NeO. The calorific value for NeO is 9.4% lower than that of diesel. This could be due the presence of oxygen molecule in the molecular structure of NeO. The flash point of NeO is found to be 245°C, which is quite higher compared to diesel of 56°C. A similar phenomenon in fire point is also observed. Flash and fire points are important to satisfy the performance and emissions requirements. The cloud and pour points of NeO are higher than those of diesel. The cloud and pour points are important for cold weather conditions.

Table 6.1 Properties of diesel and NeO

<table>
<thead>
<tr>
<th>Property</th>
<th>Diesel</th>
<th>NeO</th>
<th>ASTM code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calorific value, MJ/kg</td>
<td>43.2</td>
<td>39.1</td>
<td>D4809</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.823</td>
<td>0.93</td>
<td>D445</td>
</tr>
<tr>
<td>Kinematic viscosity (at 40°C) cSt</td>
<td>3.9</td>
<td>38</td>
<td>D2217</td>
</tr>
<tr>
<td>Cetane number</td>
<td>49</td>
<td>48</td>
<td>D4737</td>
</tr>
<tr>
<td>Flash point, °C</td>
<td>56</td>
<td>245</td>
<td>D92</td>
</tr>
<tr>
<td>Fire point, °C</td>
<td>64</td>
<td>268</td>
<td>D92</td>
</tr>
<tr>
<td>Cloud point, °C</td>
<td>-8</td>
<td>14.5</td>
<td>D97</td>
</tr>
<tr>
<td>Pour point, °C</td>
<td>-20</td>
<td>5.6</td>
<td>D97</td>
</tr>
<tr>
<td>*Surface tension at 20°C, N/m</td>
<td>0.023</td>
<td>0.039</td>
<td>-</td>
</tr>
<tr>
<td>*Bulk modulus of elasticity, bar</td>
<td>16000</td>
<td>19260</td>
<td>-</td>
</tr>
<tr>
<td>*Latent heat of vaporization, kJ/kg</td>
<td>250</td>
<td>192</td>
<td>-</td>
</tr>
</tbody>
</table>

* From Ref.(Sundarapandian et al 2007)
Table 6.2 Fatty acid composition of NeO*

<table>
<thead>
<tr>
<th>Fatty acid</th>
<th>Chemical structure **</th>
<th>Formula</th>
<th>Fatty acid composition (wt %)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Myristic</td>
<td>14:0</td>
<td>C_{14}H_{28}O_{2}</td>
<td>0.2-0.26</td>
</tr>
<tr>
<td>Palmitic</td>
<td>16:0</td>
<td>C_{16}H_{32}O_{2}</td>
<td>13.6-16.2</td>
</tr>
<tr>
<td>Stearic</td>
<td>18:0</td>
<td>C_{18}H_{36}O_{2}</td>
<td>14.4-24.1</td>
</tr>
<tr>
<td>Arachidic</td>
<td>20:0</td>
<td>C_{20}H_{40}O_{2}</td>
<td>0.8-3.4</td>
</tr>
<tr>
<td>Oleic</td>
<td>18:1</td>
<td>C_{18}H_{34}O_{2}</td>
<td>49.1-61.9</td>
</tr>
<tr>
<td>Linoleic</td>
<td>18:2</td>
<td>C_{18}H_{32}O_{2}</td>
<td>2.3-15.8</td>
</tr>
</tbody>
</table>

* From Ref. (Avinash Kumar Agarwal et al 2007)

** xx:y indicates xx the number of carbons and y the number of double bonds of carbons in the fatty acid chain.

6.1.2 Cylinder Pressure

![Graph of cylinder pressure vs crank angle](image)

Figure 6.1 Variation of cylinder pressure with crank angle for diesel and NeO at BP = 1 kW (low load)
The variation of cylinder pressure with crank angle for diesel and NeO at low load (BP = 1 kW) and full load (BP = 3.5 kW) is shown in Figure 6.1 and Figure 6.2 respectively. It can be observed that at all loads, the combustion starts earlier for NeO than diesel. This may be due to the reduction of ignition delay as well as advance dynamic injection timing for NeO (because of higher bulk modulus of longer fatty acid chains (C18:1) present in the NeO) (Sara Pinzi et al 2013). The peak cylinder pressure is higher for NeO than diesel at low load. This may be due to the increase of ignition delay with the engine load decreasing. Due to longer ignition delay in case of diesel, the combustion starts later for diesel than neat NeO as shown in pressure- crank angle diagram (Figure 6.1 and Figure 6.2). This can be confirmed by the heat release rate diagram (Figure 6.3 and Figure 6.4). Therefore, at low load, the peak cylinder pressure for diesel attains a lower value as it is further away from the TDC in the expansion stroke. As the
engine load increases, the combustion start points come closer for all the
fuels. As a result, at full load, initiation of combustion before TDC takes place
and the pressure rises more quickly for diesel.

### 6.1.3 Rate of Heat Release

The variation of heat release rate with crank angle for diesel and
NeO at low load (BP = 1 kW) and full load (BP = 3.5 kW) is shown in
Figure 6.3 and Figure 6.4 respectively. At the beginning, the negative heat
release is observed due to the vaporization of the fuel accumulated during the
ignition delay, and after the initiation of combustion, the heat release rate
becomes positive. The premixed heat release is always higher for diesel due
to its higher volatility and better mixing of diesel with air. It can also be
observed that the premixed combustion phase is dominant for diesel and NeO
at low load, but at full load, the diffusion burning indicated by the area under
second peak is dominant for neat NeO. This is consistent with the expected
effects of neat NeO viscosity on the fuel spray and reduction of air
entrainment and fuel/air mixing rates (Humke and Barsic 1981). At the time
of ignition, less air-fuel mixture is prepared for combustion with neat NeO.
As a result, more burning occurs in the diffusion burning combustion phase
rather than in the premixed combustion phase. Due to this, more amount of
energy is released in the later part of the combustion process, which lead to
reduction in thermal efficiency (Humke and Barsic 1981). The cumulative
heat release curves for diesel and NeO is shown in Figure A3.1. (APPENDIX
3). The diffusion combustion is high for NeO. Due to this, more amount of
energy is released in the main and later part of the combustion process, which
results in reduction of thermal efficiency and increase in exhaust gas
temperature.
Figure 6.3  Variation of rate of heat release with crank angle for diesel and NeO at BP = 1 kW (low load)

Figure 6.4  Variation of rate of heat release with crank angle for diesel and NeO at BP = 3.5 kW (full load)
6.1.4 Ignition Delay

The variation of ignition delay with BP for diesel and NeO is shown in Figure 6.5. The ignition delay is observed to be lower in case of NeO compared to diesel at all engine loads. The reason may be that a complex and pre-flame reaction takes place at higher temperatures. As a result of higher cylinder temperature existing during fuel injection, NeO may undergo thermal cracking and lighter compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay (Sahoo et al 2009). The increase in fuel viscosity, particularly for the fuels derived from petroleum products, results in poor atomization, slow mixing and reduced cone angle. These result in longer ignition delay. But the vegetable oils are not derived from crude petroleum; the opposite trend (lower ignition delay) is seen in case of neat NeO.

Figure 6.5 Variation of ignition delay with brake power for diesel and NeO
6.1.5 Combustion Duration

The variation of combustion duration with BP for diesel and NeO is shown in Figure 6.6. The combustion duration increases with increase in brake power of the engine with all the fuels due to increase in the quantity of fuel injected. Longer combustion duration is observed with NeO compared to diesel due to the longer diffusion combustion phase. This again confirms the slower combustion characteristics of NeO.

![Figure 6.6 Variation of combustion duration with brake power for diesel and NeO](image)

6.1.6 Equivalence Ratio

Figure 6.7 shows the fuel-air equivalence ratio, i.e., the ratio of actual fuel-air ratio (by mass) to its stoichiometric value, for diesel and NeO. It can be observed that the fuel-air equivalence ratio for NeO curve lies higher.
than the corresponding one for diesel fuel. This shows that engine runs overall at ‘richer’ conditions with NeO for the same engine load and speed conditions.

![Graph showing variation of equivalence ratio with brake power for diesel and NeO.](image)

**Figure 6.7** Variation of equivalence ratio with brake power for diesel and NeO

### 6.1.7 Brake Thermal Efficiency

The variation of BTE with BP for diesel and NeO is shown in Figure 6.8. The thermal efficiency for NeO is observed to be lower compared with diesel at almost all the loads. The reason can be explained as follows. Firstly, it may be due to higher viscosity, specific gravity and lower volatility of neat NeO compared to diesel, which results in poor spray formation and reduced spray angle leading to reduction in air entrainment and fuel-air mixing rates (Humke and Barsic 1981). Secondly, as mentioned earlier in section 6.1.3, more amount of the energy is released in later part of the combustion process resulting in lower thermal efficiency for neat NeO.
Figure 6.8  Variation of brake thermal efficiency with brake power for diesel and NeO

6.1.8 Brake Specific Energy Consumption

The variation of BSEC with BP for diesel and NeO is presented in Figure 6.9. The BSEC is an ideal parameter for comparing the engine performance of fuels having different calorific values and specific gravities. The BSEC is calculated as the product of brake specific fuel consumption and lower calorific value. The BSEC for NeO is observed to be higher compared with diesel at almost all the loads. At full load, the BSEC of NeO is 14% higher than that of diesel. As mentioned earlier, this may be due to the combined effects of lower calorific value, higher viscosity and specific gravity of NeO, so larger amount of NeO is supplied to the engine to maintain the constant brake power.
Figure 6.9  Variation of brake specific energy consumption with brake power for diesel and NeO

6.1.9 Exhaust Gas Temperature

The variation of exhaust temperature with BP for diesel and NeO is shown in Figure 6.10. The exhaust gas temperature for NeO is higher than that of diesel at almost all loads. This may be due to higher viscosity and lower volatility of neat NeO which results in lesser air-fuel mixture formation during the premixed combustion phase and hence higher the diffusion combustion phase, leading to more amount of energy during this phase. This results in slightly higher exhaust gas temperature and lower thermal efficiency when the engine is operating with neat NeO.
6.1.10 Oxides of Nitrogen

The variation of BSNO$_x$ with BP for diesel and NeO is shown in Figure 6.11. The BSNO$_x$ decreases with increase in engine load. However, the NO$_x$ emission expressed in ppm increases with the engine load. This indicates that the NO$_x$ emission strongly depends upon the combustion temperature and availability of oxygen. From Figure 6.11, it can be observed that the BSNO$_x$ emissions from neat NeO are lower than that of diesel. The vegetable oils have higher viscosity, and thus the fuel droplet size is expected to be larger than that of diesel. Larger droplets have longer combustion duration and they demonstrate significant energy release during later part of the combustion process. This suggests that the maximum combustion temperature is possibly lower (less intensity of heat release in the premixed combustion phase), which leads to lower emission of NO$_x$ for neat NeO compared with diesel (Avinash Agarwal and Rajamanoharan 2009b).
Figure 6.11 Variation of BSNO\(_x\) with brake power for diesel and NeO

6.1.11 Hydrocarbon

The variation of BSHC emission with BP for diesel and NeO is shown in Figure 6.12. As shown in Figure 6.12, the BSHC emissions decrease with increase in engine load due to the increase of cylinder temperature. The BSHC emissions from NeO are higher than that of diesel at almost all the loads. The effect of fuel viscosity on fuel spray quality could be expected to produce some HC emission with vegetable oils. Due to high viscosity, the air-fuel mixing process is affected by the difficulty in the atomization and vaporization of NeO. The resulting locally rich mixture causes more incomplete combustion products such as HC and CO (Devan and Mahalakshmi 2009; Avinash Agarwal et al 2009).
6.1.12 Carbon Monoxide

The variation of BSCO emission with BP for diesel and NeO is shown in Figure 6.13. Similar to the behavior of BSHC, the BSCO emissions decrease with increase in engine load due to the increase of cylinder temperature. As shown in Figure 6.13, in case of NeO, the BSCO emissions are 60% higher than that of diesel. The higher viscosity and poor atomization tendency of NeO leads to poor combustion and higher CO emission (Devan and Mahalakshmi 2009). It can also be observed that the engine emits very small CO emissions for NeO and diesel, which are very small values, so that a small change of the CO content will cause a big change in percentages.

Figure 6.12 Variation of BSHC with brake power for diesel and NeO
6.1.13 Smoke Intensity

The variation of smoke intensity with BP for diesel and NeO is shown in Figure 6.14. The smoke emission from NeO is higher than that of diesel. This may due to heavier molecular structure and higher viscosity of NeO; atomization becomes poor and this leads to higher smoke emissions at almost all loads (Devan and Mahalakshmi 2009).

6.1.14 Other Observations

During the operation of engine fuelled with neat NeO, the engine noise has been observed to be higher at lower loads (BP = 0 kW and BP =1 kW) whereas at higher loads (BP = 3 kW and BP = 3.5 kW), the engine noise has been observed to be lower for neat NeO. This could be due to lower cylinder pressure rise during combustion.
Figure 6.14  Variation of smoke intensity with brake power for diesel and NeO

After 100 hours of operation of diesel engine, as shown in Figure 6.15, the carbon deposits on the exterior of injector tip, valve faces, piston head and the cylinder walls have been observed with the use of neat NeO. The deposits have been removed by the method of light scraping.

The high viscosity of neat NeO could be responsible for the above mentioned problems. Therefore, a reduction in viscosity of neat NeO is of prime importance to make it suitable for diesel engines. Blending with diesel/alcohols and transesterification are predominant techniques employed to reduce the viscosity of oil.
Figure 6.15  Carbon deposits formation (a) top of the piston (b) cylinder head (c) after the removal of carbon deposits on the cylinder head and (d) top of the piston

6.2  PERFORMANCE AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELLED BY NEAT NeO AND ITS BLENDS WITH ALCOHOLS

To improve the oil viscosity, four different alcohols are individually blended with neat NeO by manual mixing at room temperature. Based on the experimental results, the performance and emissions of the diesel engine operating on NeO-alcohols blends are investigated and the results are compared with those of neat NeO.

6.2.1  Characterization of NeO and its Blends with Alcohols

The important fuel properties of diesel, neat NeO, ethanol, 1-propanol, 1-butanol and 1-pentanol are shown in Table 6.3. As shown in
Table 6.3, compared to other alcohols, ethanol has high latent heat of vaporization than neat NeO, so that it can extract more heat as it vaporizes, and thus it can lead to cooling effect on the cylinder charge. As a result of cooling effect on the charge, the charge temperature may decrease and therefore emissions of nitrogen oxides would be reduced. The calorific value of neat NeO is approximately 90% of that of diesel. The calorific values of ethanol, 1-propanol, 1-butanol and 1-pentanol are 30.2%, 20%, 13.8% and 9.7% less than the value of neat NeO. Blending of alcohols with neat NeO lowers the calorific values of fuel blends as shown in Figure 6.16. Figure 6.17 shows the oxygen content of neat NeO, N-E, N-P, N-B and N-PT fuel blends. The oxygen content increases with the addition of the alcohols. C/H ratio and stoichiometric A/F ratio of alcohols are lower than those of neat NeO as shown in Table 6.3. Higher oxygen content, lower C/H ratio and stoichiometric A/F ratio are beneficial for improving the combustion and reducing the smoke (Mao Gongping et al 2008). Figure 6.18 shows the flash points of neat NeO, N-P, N-B and N-PT fuel blends. The flash points of N-E, N-P, N-B, N-PT fuel blends are lower than that of NeO because the flash points of alcohols are lower. Flash and fire points are important to satisfy performance and emissions requirements. Figure 6.19 shows the variation of specific gravity of neat NeO, N-E, N-P, N-B and N-PT fuel blends. The specific gravities of N-E, N-P, N-B, N-PT fuel blends are lower than that of neat NeO because the specific gravities of alcohols are lower.

Figure 6.20 shows the variation of kinematic viscosity with the addition of oxygenates. As shown in Figure 6.20, kinematic viscosity decreases with the addition of oxygenates.
### Table 6.3 Comparison of properties of diesel, NeO and alcohols

<table>
<thead>
<tr>
<th>Property</th>
<th>D100</th>
<th>NeO</th>
<th>1-Ethanol</th>
<th>1-Propanol</th>
<th>1-Butanol</th>
<th>1-Pentanol</th>
<th>ASTM Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>C(<em>{14}H</em>{22})</td>
<td>C(<em>{18}H</em>{34}O_{2})</td>
<td>C(<em>{2}H</em>{6}OH)</td>
<td>C(<em>{3}H</em>{8}OH)</td>
<td>C(<em>{4}H</em>{10}OH)</td>
<td>C(<em>{5}H</em>{14}OH)</td>
<td>-</td>
</tr>
<tr>
<td>Calorific value, MJ/kg</td>
<td>43.2</td>
<td>39.1</td>
<td>26.8</td>
<td>30.7</td>
<td>33.1</td>
<td>34.7</td>
<td>D4809</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.823</td>
<td>0.93</td>
<td>0.790</td>
<td>0.804</td>
<td>0.810</td>
<td>0.815</td>
<td>D445</td>
</tr>
<tr>
<td>Cetane number</td>
<td>49</td>
<td>48</td>
<td>8</td>
<td>15</td>
<td>15</td>
<td>-</td>
<td>D4737</td>
</tr>
<tr>
<td>Kinematic viscosity (at 40°C) (cSt)</td>
<td>3.9</td>
<td>38</td>
<td>1.0</td>
<td>2.8</td>
<td>3.6</td>
<td>4.1</td>
<td>D2217</td>
</tr>
<tr>
<td>Latent heat of vaporization, kJ/kg</td>
<td>250</td>
<td>192</td>
<td>923</td>
<td>-</td>
<td>716</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Stoichiometric A/F</td>
<td>14.93</td>
<td>12.35</td>
<td>9.01</td>
<td>10.3</td>
<td>11.12</td>
<td>11.76</td>
<td>-</td>
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<tr>
<td>C (mass %)</td>
<td>87</td>
<td>76.5</td>
<td>52.1</td>
<td>60</td>
<td>64.86</td>
<td>68.18</td>
<td>-</td>
</tr>
<tr>
<td>H (mass %)</td>
<td>13</td>
<td>12.1</td>
<td>13.1</td>
<td>13.3</td>
<td>13.5</td>
<td>13.6</td>
<td>-</td>
</tr>
<tr>
<td>O (mass %)</td>
<td>0</td>
<td>11.3</td>
<td>34.7</td>
<td>26.6</td>
<td>21.6</td>
<td>18.2</td>
<td>-</td>
</tr>
<tr>
<td>C/H</td>
<td>6.71</td>
<td>6.33</td>
<td>3.98</td>
<td>4.5</td>
<td>4.81</td>
<td>5.03</td>
<td>-</td>
</tr>
</tbody>
</table>
Calorific value of diesel = 43.2 MJ/kg
Calorific value of NeO = 39.1 MJ/kg

Figure 6.16 Variations in calorific value with fuel blends

Oxygen content of diesel = 0
Oxygen content of NeO = 11.3

Figure 6.17 Variations in oxygen content with fuel blends
Flash point diesel = 56°C
Flash point NeO = 245°C

Figure 6.18 Variations in flash point with fuel blends

Specific gravity diesel = 0.823
Specific gravity NeO = 0.93

Figure 6.19 Variations in specific gravity with fuel blends
6.2.2 Brake Specific Energy Consumption

Figures 6.21 and 6.22 show the variations in BSEC with blended fuels at BP = 1 kW and BP = 3.5 kW respectively. It is observed that BSEC is lowest for diesel fuel as compared to neat NeO and its blends with different alcohols. It can also be observed that no matter what kinds of alcohol are used, similar results are achieved at all loads. There is a decreasing trend in BSEC with increasing ethanol, 1-propanol, 1-butanol and 1-pentanol content in the blends compared to neat NeO. This is the expected behaviour due to lower calorific values of alcohol blended fuels (Tadashi Murayama et al 1982 and Yi Ren et al 2008).
Figure 6.21 Variation of brake specific energy consumption with fuel blends at BP=1 kW

Figure 6.22 Variation of brake specific energy consumption with fuel blends at BP=3.5 kW
6.2.3 Brake Thermal Efficiency

Figures 6.23 & 6.24 show the variations of BTE with blended fuels at BP = 1 kW and BP = 3.5 kW respectively. It is observed that the BTE is highest for diesel fuel as compared to neat NeO and its blends with different alcohols. It is observed that for all the N-E, N-P, N-B, N-PT blends, the BTE is slightly higher with the increase being higher the higher the percentage of alcohols in the blend. It is to be noted that the BTE is the inverse of the product of the specific fuel consumption and the lower calorific value of the fuel. This relationship then indicates that the increase of BSFC for the NeO-alcohol fuel blends is lower than the corresponding decrease of the lower calorific value of the blends. This behaviour can also be attributed to the higher premixed combustion of the alcohol blended fuels because of the lower cetane number of the alcohols, leading to higher percentage of constant volume combustion (Tadashi Murayama et al 1982).

![Graph showing brake thermal efficiency variation with fuel blends](image)

BTE\textsubscript{diesel} = 14.59% @ 1 kW
BTE\textsubscript{NeO} = 14.44 % @ 1 kW

Figure 6.23 Variation of brake thermal efficiency with fuel blends at BP=1 kW
6.2.4 Carbon Monoxide

Figures 6.25 and 6.26 show the variations of BSCO emissions with blended fuels at \( BP = 1 \) kW and \( BP = 3.5 \) kW respectively. It is observed that the BSCO emission is lowest for diesel fuel when compared to neat NeO and its blends with different alcohols over the entire load. It can be observed that blending all four kinds of alcohols in NeO produces higher BSCO emissions at low load, and lower BSCO emissions at higher loads compared with neat NeO. This is due to the factors such as high latent heat of evaporation causing combustion deterioration (Murat Karabektas and Murat Hosoz 2009).
This leads to a lower combustion temperature at lower loads. At high loads, the combustion is more complete than that of neat NeO because of high oxygen content in the fuels, leading to lower BSCO emissions. At full load, for the NeO-Ethanol blends, from N-E5 to N-E10, the BSCO emissions decrease from 28.5% to 42.3% compared with neat NeO, for the NeO-propanol blends, from N-P5 to N-P20, the BSCO emissions decrease from 14% to 28.7% compared with neat NeO, for the NeO-butanol blends, from N-B5 to N-B20, the BSCO emissions decrease from 14% to 28.5% compared with neat NeO, for the NeO-pentanol blends, from N-PT5 to N-PT20, the BSCO emissions decrease to 14% compared with neat NeO. Moreover, it can be observed that the BSCO emissions for N-E blends are slightly lower than the corresponding N-P, N-B and N-PT blends at higher engine loads.

![Figure 6.25 Variation of BSCO with fuel blends at BP=1 kW](image_url)

*Figure 6.25  Variation of BSCO with fuel blends at BP=1 kW*
6.2.5 Hydrocarbon

**Figure 6.26  Variation of BSCO with fuel blends at BP=3.5 kW**

**Figure 6.27  Variation of BSHC with fuel blends at BP=1kW**
Figures 6.27 and 6.28 show the variations of BSHC emissions with blended fuels at BP = 1 kW and BP = 3.5 kW respectively. It is observed that BSHC emission is low for diesel fuel as compared to neat NeO and its blends with different alcohols at all loads. Similar to the behaviour of CO emission, BSHC emission increases at low loads and decreases at high loads with the addition of alcohols. Higher BSHC emissions at lower loads may be due to the relatively higher latent heat of vaporization of alcohols blended fuels. Since the alcohol blended fuels absorb more heat, the temperature inside the cylinder drops down, which results in lower HC oxidation rate. Low HC emission, at high loads is due to the fact that oxygenated fuels can decrease HC from the locally rich-mixture (Murat Karabektas and Murat Hosoz 2009). For the NeO-Ethanol blends, from N-E5 to N-E10, the HC emissions decrease from 12.5% to 25% compared with neat NeO, for the NeO-propanol blends, from N-P5 to N-P20, the HC emissions decrease from 18.8% to 31.2% compared with neat NeO, for the NeO-butanol blends, from N-B5 to N-B20, the HC emissions decrease from 18.8% to 37.5% compared with neat NeO, for the NeO-pentanol blends, from N-PT5 to N-PT20, the HC emissions
decrease approximately 25% compared with neat NeO. Moreover, it can be observed that the BSHC emissions for N-E blends are slightly lower than the corresponding N-P, N-B and N-PT blends at higher engine loads.

6.2.6 Smoke Intensity

![Graph showing variations in smoke intensity with fuel blends at BP=1 kW](image)

**Figure 6.29 Variations in smoke intensity with fuel blends at BP=1 kW**

Figures 6.29 and 6.30 show the variations of smoke intensity with blended fuels at BP = 1 kW and BP = 3.5 kW respectively. It is found that smoke level is lowest for diesel fuel as compared to neat NeO and its blends with different alcohols. It can be seen that smoke decreases with the addition of alcohols especially with 1-butanol. This may be due to comparatively lower latent heat of evaporation of 1-butanol than ethanol. Since the smoke is produced in the diffusive combustion phase, the addition of oxygenated fuel leads to an improvement in diffusive combustion (Tadashi Murayama et al 1982 and Yi Ren et al 2008). It can also be found that smoke decreases more at higher loads than at lower loads.
Figure 6.30 Variations in smoke intensity with fuel blends at BP=3.5 kW

6.2.7 Oxides of Nitrogen

Figures 6.31 and 6.32 show the variations of BSNO\textsubscript{x} emission with all blended fuels at BP = 1 kW and BP = 3.5 kW respectively. It is seen that NO\textsubscript{x} emission is lower for neat NeO and its blends with different alcohols as compared to diesel fuel at all loads. The NO\textsubscript{x} emission of the NeO-Ethanol is lower than the corresponding neat NeO, with the reduction being higher the percentage of ethanol in the blend. This is due to the fact that the high latent heat of vaporization of ethanol lowers the flame temperature, resulting in lower NO\textsubscript{x} emissions. The NO\textsubscript{x} emissions with the 1-propanol, 1-butanol and 1–pentanol blended neem oil are very slightly lower than those of neat NeO. This is due to the fact that alcohols usually produce lower combustion temperature due to their lower heating values and oxygen content (Murat Karabektas and Murat Hosoz 2009). For the NeO-Ethanol blends, from N-E5 to NE-10, the NO\textsubscript{x} emissions decrease to about 10% compared with neat NeO. Moreover, it can be observed that the BSNO\textsubscript{x} emissions for N-E blends
are lower than the corresponding N-P, N-B and N-PT blends at higher engine loads.

Figure 6.31 Variations in BSNO\textsubscript{x} with fuel blends at BP=1 kW

Figure 6.32 Variations in BSNO\textsubscript{x} with fuel blends at BP=3.5 kW
6.2.8 Other Observations

1. After 100 hours of continuous operation of the engine, very slight carbon deposits on the combustion side of the injector, valve faces and the top of the piston have been observed when fuelled with NeO- alcohols blends. However, the compression and oil scraping rings have been found to be in good condition.

2. Further increase in addition of alcohols will not have a positive effect on the performance and emission characteristics. Moreover, increase in oxygenates causes knocking of the cylinder in the engine.

6.3 COMBUSTION, PERFORMANCE AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELLED BY NOME AND ITS BLENDS WITH DIESEL

With the addition of alcohols into neat NeO, the decrease in viscosity may not be low enough to reach the same level of viscosity as diesel, hence it has been transesterified to produce methyl ester of neem oil (NOME). Based on the experimental results, the performance and emission characteristics of the diesel engine operating on NOME and its blends with diesel are investigated and the results are compared with those of diesel.

6.3.1 Characterization of NOME and its Blends with Diesel

The important properties of NOME and its blends with diesel are shown in Table 6.4. From Table 6.4, it can be observed that after transesterification, the kinematic viscosity of NOME reduces to about 1/7 times neat NeO. But still the viscosity of biodiesel is higher than diesel, which is approximately 25% higher than that of diesel. The standard amount of
viscosity for biodiesel in ASTM is 1.9–6 mm²/s and in EN is 3.5–5 mm²/s. The viscosity of NOME is within the standard range. The specific gravity of NOME is found to be 6.67% higher than that of diesel. The specific gravities are found to increase with the increase of NOME percentage in the blends. In EN the standard amount of density is 860–900 kg/m³ while in ASTM no range is specified. According to the results the density of NOME (867 kg/m³) is within the standard range in EN.

Table 6.4 Properties of NOME and its blends with diesel

<table>
<thead>
<tr>
<th>Property</th>
<th>B10</th>
<th>B20</th>
<th>B30</th>
<th>B50</th>
<th>B100</th>
<th>ASTM code</th>
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</thead>
<tbody>
<tr>
<td>Calorific value (MJ/kg)</td>
<td>43.0</td>
<td>42.74</td>
<td>42.52</td>
<td>42.07</td>
<td>41.0</td>
<td>D4809</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.827</td>
<td>0.832</td>
<td>0.836</td>
<td>0.845</td>
<td>0.867</td>
<td>D445</td>
</tr>
<tr>
<td>Viscosity (at 40⁰C) cSt</td>
<td>4.05</td>
<td>4.2</td>
<td>4.35</td>
<td>4.65</td>
<td>5.4</td>
<td>D2217</td>
</tr>
<tr>
<td>Cetane number</td>
<td>49.2</td>
<td>49.4</td>
<td>49.6</td>
<td>50</td>
<td>51</td>
<td>D4737</td>
</tr>
<tr>
<td>Flash point °C</td>
<td>67</td>
<td>75</td>
<td>89</td>
<td>112</td>
<td>165</td>
<td>D92</td>
</tr>
<tr>
<td>Fire point °C</td>
<td>78</td>
<td>83</td>
<td>97</td>
<td>119</td>
<td>178</td>
<td>D92</td>
</tr>
<tr>
<td>Cloud point °C</td>
<td>3</td>
<td>5</td>
<td>-1</td>
<td>3</td>
<td>3</td>
<td>D97</td>
</tr>
<tr>
<td>Pour point °C</td>
<td>-2</td>
<td>-3</td>
<td>-7</td>
<td>-5</td>
<td>-8</td>
<td>D97</td>
</tr>
<tr>
<td>*Surface tension at 20°C, N/m</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.026</td>
<td>-</td>
</tr>
<tr>
<td>*Bulk modulus of elasticity, bar</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>17150</td>
<td>-</td>
</tr>
<tr>
<td>*Latent heat of vaporization, kJ/kg</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>236</td>
<td>-</td>
</tr>
</tbody>
</table>

*From Ref. (Sundarapandian et al 2007)

The calorific value of NOME is slightly improved and it is 5% lower than that of diesel. This could be due the presence of oxygen molecule.
in the molecular structure of biodiesel. The calorific values of the blends are found to decrease with the increase in NOME concentration in the blends. The viscosity of the blends is found to increase with the increase in NOME concentration in the blends. The flash point of NOME is found to be 165°C respectively, which is quite higher compared to 56°C with diesel. The flash points of all the blends are also higher than that of diesel. A similar phenomenon in fire point is also observed.

The least allowable amount of flash point in ASTM is 130°C and in EN is 120°C. The cloud and pour points of NOME are higher than those of diesel. The cloud and pour points of all the blends are also higher than that of diesel. In EN and ASTM, no standard limits have been specified for cloud and pour points, and they are usually defined according to the climatic conditions of the area. The standard cetane number for biodiesel in ASTM D and EN is 47 and 51 respectively. The surface tension and bulk modulus of NeO after transesterification are improved as shown in Table 6.4. The adiabatic flame temperature for diesel and biodiesel (NOME) is calculated as 2483K and 2683K for an equivalence ratio of 1.0. Compared to diesel, NOME has higher adiabatic flame temperature, which results in higher local combustion temperature, and consequently an increase in NOx emission.

6.3.2 Cylinder Pressure

The variations in cylinder pressure with crank angle for different fuel blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) are shown in Figure 6.33 and Figure 6.34 respectively. It can be observed that combustion starts earlier for NOME blends due to a shorter ignition delay as well as an advanced dynamic injection for NOME and its blends. This is because the biodiesel has lower compressibility (higher bulk modulus of longer fatty acid chains, i.e. C18:1) (Sara Pinzi et al 2013), partially due to the presence of oxygen in the fuel structure (Avinash Agarwal et al 2009a; Sahoo et al 2009).
This can be confirmed in rate of heat release diagram as shown in Figure 6.35 and Figure 6.36. It can be observed that for a specific engine load, the cylinder pressure curves of diesel and its blends with NOME are similar. It can be observed that with increase in engine load, there is a corresponding increase in cylinder pressure and heat release rate. With increasing load, the amount of fuel supplied to the engine increases and due to combustion of larger amount of fuel, the peak cylinder pressure is higher at full load.

![Cylinder Pressure vs Crank Angle](image)

**Figure 6.33** Variation of cylinder pressure with crank angle for diesel and NOME blends at BP = 1 kW (low load)
Compared with diesel, at both low load and full load, the peak cylinder pressure is higher for NOME blends. At low load, the peak cylinder pressure for D100, B10, B20, B30, B50 and B100 is 51.29 bar, 51.55 bar, 51.54 bar, 51.91 bar, 51.98 bar, 52.39 bar respectively and at full load, 59.9 bar, 60.36 bar, 61.36 bar, 61.47 bar, 61.57 bar and 62.2 bar respectively. This may be due to relatively higher amount of injected fuel and early combustion for NOME blends (Shao et al 2009). On the other hand, while running on diesel, the longer ignition delay of diesel leads to later combustion starting which results in lower peak cylinder pressure especially at low load (Avinash Agarwal et al 2009a).

6.3.3 Rate of Heat Release

The variations in heat release with crank angle for different fuel blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) are shown in
Figure 6.35 and Figure 6.36 respectively. It can be observed from Figure 6.35 and Figure 6.36, the peak heat release rates for B10 and B20 blends are evidently higher than that of diesel. The reason may be due to the more percentage of diesel fuel in these blends, which possibly accelerate the combustion process. At both low and full loads, the peak heat release rate for B30 and B50 at the time of premixed combustion stage is comparable with diesel. The peak heat release rate during premixed combustion phase for B100 is lower as compared to diesel. This may be due to the relatively lower volatility and higher viscosity of biodiesel which leads to a reduction in air entrainment and fuel-air mixing rates, leading to lesser amount of fuel being prepared for premixed combustion stage during ignition delay (Sahoo et al 2009). As a result, more fuel burns in the diffusion combustion phase and also more amount of energy is released in the later part of the combustion process, which lead to slight reduction in thermal efficiency. The variation of cumulative heat release with crank angle for NOME blends is shown in Appendix.A3.2

Figure 6.35  Variation of rate of heat release with crank angle for diesel and NOME blends at BP =1 kW (low load)
Figure 6.36  Variation of rate of heat release with crank angle for diesel and NOME blends at BP = 3.5 kW (full load)

6.3.4 Ignition Delay

Figure 6.37  Variation of ignition delay with brake power for diesel and NOME blends
The variation of ignition delay with brake power for diesel and its blends with NOME is shown in Figure 6.37. The ignition delay decreases with the increase of NOME in the blends at higher engine loads. The reason may be that a complex and pre-flame reaction takes place at higher temperatures. As a result of higher cylinder temperature existing during fuel injection, biodiesel may undergo thermal cracking and lighter compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay especially at higher engine loads. The increase in fuel viscosity, particularly for the fuels derived from petroleum products, results in poor atomization, slow mixing and reduced cone angle. These result in longer ignition delay. But the methyl esters are not derived from crude petroleum; the opposite trend is seen in the case of biodiesel blended diesel fuels. (Sahoo et al 2009; Avinash Agarwal 2009).

6.3.5 Combustion Duration

The variation of combustion duration with brake power for diesel and its blends with NOME is shown in Figure 6.38. It can be observed that in Figure 6.38, for each specific engine load, there is an increase in combustion duration with the increase of NOME concentration in the blends which again confirms relatively slower combustion characteristics of biodiesel blended fuels (Edwin et al 2008; Balusamy et al 2010).
6.3.6 Equivalence Ratio

Figure 6.39 shows the fuel-air equivalence ratio for diesel and its blends with NOME. It can be observed that the fuel-air equivalence ratio for NOME blends curve lies little higher than the corresponding one for diesel fuel. This shows that engine runs overall slightly ‘richer’ conditions with biodiesel blended diesel fuels for the same engine load and speed conditions.
Figure 6.39 Variation of equivalence ratio with brake power for diesel and NOME blends

6.3.7 Brake Thermal Efficiency

The variation of brake thermal efficiency with brake power for diesel and its blends with NOME is shown in Figure 6.40. Initially the thermal efficiency of the engine is improved with increasing concentration of NOME in the blend for all loads. The possible reason for this may be more complete combustion and additional lubricity of biodiesel. The molecule of biodiesel has some oxygen, which takes part in the combustion process.
It can be observed that after a certain limit with respect to diesel-biodiesel blend, brake thermal efficiency starts decreasing as a function of the concentration of the NOME in the blend. This could be due to comparatively lower calorific value and higher viscosity, which results in slightly poorer atomization and poorer combustion (Deepak Agarwal et al 2008).

6.3.8 Brake Specific Energy Consumption

The variation of brake specific energy consumption with brake power for diesel and its blends with NOME is shown in Figure 6.41. The BSEC of the blends up to 30% is in well comparable with diesel. Minimum BSEC of 20% blend is 11.784 MJ/kW h against 12.50 MJ/kW h of diesel. The BSEC increases with more increase in percentage of NOME in the blends. This could be due to comparatively lower calorific value and higher viscosity, which results in slightly poorer atomization and poorer combustion (Deepak Agarwal et al 2008).
Figure 6.41 Variation of brake specific energy consumption with brake power for diesel and NOME blends

6.3.9 Exhaust Gas Temperature

The variation of exhaust temperature with brake power for diesel and its blends with NOME is shown in Figure 6.42. The exhaust gas temperature increases with the increase of NOME concentration in the blends at almost all loads, but the difference is more in higher engine loads in comparison with lower engine loads. This may be due to relatively higher viscosity and lower volatility of NOME, which results in more air fuel formation especially during the diffusion combustion phase, leading to more amount of energy in later combustion phase (Devan et al 2009). This results in higher exhaust temperature when the engine is operating with NOME. Another reason may be due to the presence of oxygen content in NOME that enhances the oxidation of soot before it leaves the combustion chamber.
6.3.10 Oxides of Nitrogen

The variation of oxides of nitrogen with brake power for diesel and its blends with NOME is shown in Figure 6.43. The BSNO\textsubscript{x} emissions are sensitive to oxygen content and adiabatic flame temperature. It can be observed that the BSNO\textsubscript{x} emissions increase with the addition of NOME fraction in the blends, with the increase being directly proportional to the percentage of NOME in the blends. The NO\textsubscript{x} formation in biodiesel combustion process is not determined by a change in single fuel property, but this is a result of number of various factors that are dependent on fuel characteristics and fuel injection systems. The increase in NO\textsubscript{x} emission may be due to the increase of bulk modulus of longer fatty acid hydrocarbons, giving an advance injection in this study, where mechanical fuel injection system is used. The higher adiabatic flame temperature is also responsible for the increase in NO\textsubscript{x} emission (Sara Pinzi et al 2013).
6.3.11 Carbon Monoxide and Hydrocarbon

The variations in BSCO and BSHC emissions with brake power are shown in Figure 6.44 and Figure 6.45. The BSCO emission is predominantly formed due to the lack of oxygen. Since biodiesel contains amounts of oxygen, which act as combustion promoter inside the cylinder. Hence, the BSCO emissions decrease with increasing percentage of NOME in the blends at all loads. The HC emissions also show the similar trend observed for BSCO emissions. With increasing NOME percentage in the blends, the HC emissions decrease at almost all loads. The possible reason is the amount of oxygen content in NOME that helps for the complete combustion (Avinash Agarwal et al 2007; Gumus et al 2010).
Figure 6.44 Variation of BSCO with brake power for diesel and NOME blends

Figure 6.45 Variation of BSHC with brake power for diesel and NOME blends
6.3.12 Smoke Intensity

The variation of smoke intensity with brake power is shown in Figure 6.46. In the using of NOME blends, the smoke intensity is lower than the corresponding diesel fuel at almost all loads, with the reduction being higher the higher the percentage of NOME in the blends. Owing to the heterogeneous nature of diesel combustion, there are infinite fuel/air ratios within the cylinder. Smoke formation generally occurs in the rich zone at high temperature, particularly within the core region of fuel spray, and is caused by high temperature decomposition. Since biodiesel contains more amount of oxygen, the locally over rich region decreases. The favorable amount of oxygen in biodiesel helps in restricting the smoke formation (Avinash Agarwal et al 2007; Gumus et al 2010).

![Figure 6.46 Variation of smoke intensity with brake power for diesel and NOME blends](image-url)
6.4 COMBUSTION, PERFORMANCE AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELLED BY NOME AND ITS BLENDS WITH DEE, ETH, DMC AND DGL

The major problem associated with the use of biodiesel, especially that produced from NeO is its relatively higher viscosity, lower volatility and low temperature flow properties than those of diesel. In order to provide significant improvement in combustion, performance and exhaust emissions, experiments are conducted at a constant speed of 1500 rpm under variable load conditions to study the influence of NOME and its blends of 5 vol%, 10 vol%, 15 vol% and 20 vol% with ETH, DEE, DMC and DGL on the combustion, performance and emission characteristics of the diesel engine. Based on the experimental results, the performance and emission characteristics of the diesel engine operating on NOME and its blends with DEE, ETH, DMC and DGL are investigated and the results are compared with those of NOME and diesel.

6.4.1 Characterization of NOME and its Different Blends with Oxygenates

The properties of DEE, ETH, DMC and DGL are shown in Table 6.5. Compared to NOME, the DEE, ETH, DMC and DGL have high latent value of vaporization, so that they can extract more heat as they vaporize, and thus leading to cooling effect over the cylinder charge. As a result of the cooling effect on the charge, the charge temperature may decrease and therefore emissions of nitrogen oxides would be reduced. The calorific value of NOME is approximately 90% of that of diesel. The calorific values of DEE, ETH, DMC and DGL are 17.5%, 34.15%, 67% and 40.24%
less than the value of biodiesel. Blending of DEE, ETH, DMC and DGL with NOME lowers the calorific values of fuel blends as shown in Table 6.6-6.9. The oxygen content increases with the addition of the DEE, ETH, DMC and DGL as shown in Table 6.6-6.9. Compared with NOME, the BD, BE, BC and BG blends have lower C/H ratio and stoichiometric A/F ratio. Higher oxygen content, lower C/H ratio and lower stoichiometric A/F ratio are beneficial for improving the combustion and reducing the smoke. Compared with NOME, the BD, BE, BC and BG blends have lower kinematic viscosity, which will cause an improvement in the shape of fuel spray and atomization. The adiabatic flame temperature at an equivalence ratio of 1.0 for NOME is 2683K and for DEE, ETH, DMC and DGL fuels are shown in Table 6.5. It can be observed that adiabatic flame temperature decreases with the increase in oxygen content of different oxygenated fuels. This may suggest that lower NO\textsubscript{x} emissions can be obtained with the oxygenated fuels having higher oxygen content (Noboru Miyamoto et al 1998; Md. Nurun Nabi 2010).
Table 6.5 Properties of DEE, ETH, DMC and DGL

<table>
<thead>
<tr>
<th>Property</th>
<th>Diesel</th>
<th>NOME</th>
<th>DEE</th>
<th>ETH</th>
<th>DMC</th>
<th>DGL</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>$C_{14}H_{22}$</td>
<td>$C_{18}H_{34}O_2$</td>
<td>$C_{6}H_{10}O$</td>
<td>$C_{2}H_{6}O$</td>
<td>$C_{3}H_{6}O_3$</td>
<td>$C_{6}H_{14}O_3$</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>180</td>
<td>314</td>
<td>74</td>
<td>46</td>
<td>90</td>
<td>134</td>
</tr>
<tr>
<td>Calorific value, MJ/kg</td>
<td>43.2</td>
<td>41.0</td>
<td>33.8</td>
<td>27.0</td>
<td>13.5</td>
<td>24.5</td>
</tr>
<tr>
<td>Latent heat of vaporization, kJ/kg</td>
<td>250</td>
<td>236</td>
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<td>369</td>
<td>322</td>
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<td>Specific gravity</td>
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<td>1.07</td>
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<td>Kinematic viscosity (at 40°C) (cSt)</td>
<td>3.9</td>
<td>5.4</td>
<td>0.23</td>
<td>1.2</td>
<td>0.63</td>
<td>2.0</td>
</tr>
<tr>
<td>Cetane number</td>
<td>49</td>
<td>51</td>
<td>120</td>
<td>8</td>
<td>35</td>
<td>126</td>
</tr>
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<td>Boiling point, °C</td>
<td>180-330</td>
<td>330</td>
<td>34.6</td>
<td>78.4</td>
<td>90.9</td>
<td>161.3</td>
</tr>
<tr>
<td>C (mass %)</td>
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<td>76.5</td>
<td>64.8</td>
<td>52.2</td>
<td>40</td>
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<td>H (mass %)</td>
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<td>12.1</td>
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<td>13</td>
<td>6.7</td>
<td>10.5</td>
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<td>O (mass %)</td>
<td>0</td>
<td>11.3</td>
<td>21.6</td>
<td>34.8</td>
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<tr>
<td>C/H</td>
<td>6.71</td>
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<td>4.83</td>
<td>4.01</td>
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</tr>
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<td>2500</td>
<td>2487</td>
<td>2202</td>
<td>2378</td>
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Table 6.6 Properties of NOME and its blends with DEE

<table>
<thead>
<tr>
<th>Property</th>
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<th>BD10</th>
<th>BD15</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calorific value, MJ/kg</td>
<td>40.6</td>
<td>40.3</td>
<td>40.0</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.859</td>
<td>0.851</td>
<td>0.844</td>
</tr>
<tr>
<td>Kinematic viscosity (@ 40°C) (cSt)</td>
<td>5.14</td>
<td>4.88</td>
<td>4.62</td>
</tr>
<tr>
<td>C (mass %)</td>
<td>75.98</td>
<td>75.46</td>
<td>74.93</td>
</tr>
<tr>
<td>H (mass %)</td>
<td>12.62</td>
<td>12.22</td>
<td>12.28</td>
</tr>
<tr>
<td>O (mass %)</td>
<td>11.44</td>
<td>11.88</td>
<td>12.32</td>
</tr>
<tr>
<td>C/H</td>
<td>6.02</td>
<td>6.175</td>
<td>6.10</td>
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Table 6.7 Properties of NOME and its blends with ETH

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<th>BE10</th>
<th>BE15</th>
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<tr>
<td>Calorific value, MJ/kg</td>
<td>40.4</td>
<td>39.71</td>
<td>39.06</td>
<td>38.4</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.863</td>
<td>0.859</td>
<td>0.855</td>
<td>0.851</td>
</tr>
<tr>
<td>Kinematic viscosity (@ 40°C) (cSt)</td>
<td>5.19</td>
<td>4.98</td>
<td>4.77</td>
<td>4.56</td>
</tr>
<tr>
<td>C (mass %)</td>
<td>75.43</td>
<td>74.34</td>
<td>73.24</td>
<td>72.13</td>
</tr>
<tr>
<td>H (mass %)</td>
<td>12.13</td>
<td>12.17</td>
<td>12.22</td>
<td>12.26</td>
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<tr>
<td>O (mass %)</td>
<td>11.26</td>
<td>13.18</td>
<td>14.29</td>
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<tr>
<td>C/H</td>
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<td>6.10</td>
<td>5.99</td>
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### Table 6.8 Properties of NOME and its blends with DMC

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<th>BC20</th>
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</thead>
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<tr>
<td>Calorific value, MJ/kg</td>
<td>39.625</td>
<td>38.25</td>
<td>36.87</td>
<td>34.51</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.877</td>
<td>0.8870</td>
<td>0.897</td>
<td>0.907</td>
</tr>
<tr>
<td>Kinematic viscosity (@ 40°C) (cSt)</td>
<td>5.161</td>
<td>4.923</td>
<td>4.684</td>
<td>4.446</td>
</tr>
<tr>
<td>C (mass %)</td>
<td>74.89</td>
<td>73.26</td>
<td>71.61</td>
<td>69.94</td>
</tr>
<tr>
<td>H (mass %)</td>
<td>11.86</td>
<td>11.62</td>
<td>11.37</td>
<td>11.12</td>
</tr>
<tr>
<td>O (mass %)</td>
<td>12.735</td>
<td>14.76</td>
<td>16.44</td>
<td>18.07</td>
</tr>
<tr>
<td>C/H</td>
<td>6.31</td>
<td>6.3</td>
<td>6.29</td>
<td>6.28</td>
</tr>
</tbody>
</table>

### Table 6.9 Properties of NOME and its blends with DGL

<table>
<thead>
<tr>
<th>Property</th>
<th>BG5</th>
<th>BG10</th>
<th>BG15</th>
<th>BG20</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calorific value, MJ/kg</td>
<td>40.11</td>
<td>39.22</td>
<td>38.35</td>
<td>37.48</td>
</tr>
<tr>
<td>Specific gravity</td>
<td>0.870655</td>
<td>0.8743</td>
<td>0.8779</td>
<td>0.8816</td>
</tr>
<tr>
<td>Kinematic viscosity (@ 40°C) (cSt)</td>
<td>5.23</td>
<td>5.06</td>
<td>4.89</td>
<td>4.72</td>
</tr>
<tr>
<td>C (mass %)</td>
<td>75.49</td>
<td>74.47</td>
<td>73.44</td>
<td>72.40</td>
</tr>
<tr>
<td>H (mass %)</td>
<td>12.02</td>
<td>11.95</td>
<td>11.88</td>
<td>11.81</td>
</tr>
<tr>
<td>O (mass %)</td>
<td>12.62</td>
<td>13.93</td>
<td>15.17</td>
<td>16.52</td>
</tr>
<tr>
<td>C/H</td>
<td>6.28</td>
<td>6.23</td>
<td>6.18</td>
<td>6.13</td>
</tr>
</tbody>
</table>
6.4.2 Cylinder Pressure and rate of Heat Release

Figure 6.47  Variation of cylinder pressure with crank angle for NOME and BD blends at BP = 1 kW (low load)

Figure 6.48  Variation of cylinder pressure with crank angle for NOME and BD blends at BP = 3.5 kW (full load)
Figure 6.49 Variation of rate of heat release with crank angle for NOME and BD blends at BP = 1 kW (low load)

Figure 6.50 Variation of rate of heat release with crank angle for NOME and BD blends at BP = 3.5 kW (full load)
Figure 6.51  Variation of cylinder pressure with crank angle for NOME and BE blends at BP = 1 kW (low load)

Figure 6.52  Variation of cylinder pressure with crank angle for NOME and BE blends at BP = 3.5 kW (full load)
Figure 6.53  Variation of rate of heat release with crank angle for NOME and BE blends at BP = 1 kW (low load)

Figure 6.54  Variation of rate of heat release with crank angle for NOME and BE blends at BP = 3.5 kW (full load)
Figure 6.55  Variation of cylinder pressure with crank angle for NOME and BC blends at BP = 1 kW (low load)

Figure 6.56  Variation of cylinder pressure with crank angle for NOME and BC blends at BP = 3.5 kW (full load)
Figure 6.57  Variation of rate of heat release with crank angle for NOME and BC blends at BP = 1 kW (low load)

Figure 6.58  Variation of rate of heat release with crank angle for NOME and BC blends at BP = 3.5 kW (full load)
Figure 6.59  Variation of cylinder pressure with crank angle for NOME and BG blends at BP =1 kW (low load)

Figure 6.60  Variation of cylinder pressure with crank angle for NOME and BG blends at BP =3.5 kW (full load)
Figure 6.61  Variation of rate of heat release with crank angle for NOME and BG blends at BP = 1 kW (low load)

Figure 6.62  Variation of rate of heat release with crank angle for NOME and BG blends at BP = 3.5 kW (full load)
Figures 6.47 and 6.48 show the variation of cylinder pressure with crank angle for BD blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively. Figures 6.49 and 6.50 show the variation of rate of heat release with crank angle for BD blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively.

Figures 6.51 and 6.52 show the variation of cylinder pressure with crank angle for BE blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively. Figures 6.53 and 6.54 show the variation of rate of heat release with crank angle for BE blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively.

Figures 6.55 and 6.56 show the variation of cylinder pressure with crank angle for BC blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively. Figures 6.57 and 6.58 show the variation of rate of heat release with crank angle for BC blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively.

Figures 6.59 and 6.60 show the variation of cylinder pressure with crank angle for BG blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively. Figures 6.61 and 6.62 show the variation of rate of heat release with crank angle for BG blends at low load (BP = 1 kW) and full load (BP = 3.5 kW) respectively.

Due to the variation of fuel properties of the blended fuels with the addition of oxygenates, the cylinder pressure and the heat release rate give different behaviors compared to those of NOME.
In the using of blend BD5, the amount of heat release and cylinder pressure decreases at low load, whereas it slightly increases at full load. Due to the lower viscosity and higher volatility of diethyl ether, more air-fuel mixture is formed in ignition delay which results in strong premixed burning phase and gives rise to the peak cylinder pressure and rate of heat release at full load (Masoud Iranmanesh et al 2008; Qi et al 2011). As the engine load decreases, due to high latent heat values of DEE, the temperature inside the cylinder decreases by the vaporization of DEE in the blends, which results in lower rate of heat release and hence the peak cylinder pressure. The BD10 and BD15 blends show a slight variation in cylinder pressures and heat release rates due to the fluctuations in engine speed and power output. This erratic operation may be due to the high volatility of diethyl ether that could lead to vapor lock in fuel line.

For the BE and BC blends, at low load, the amount of heat release during premixed combustion phase increases slightly with the increase of ETH and DMC fraction in the blends as shown in Figure 6.53 (for BE blends) and Figure 6.57 (for BC blends), and at full load, it increases more remarkably as shown in Figure 6.54 (for BE blends) and Figure 6.58 (for BC blends), while the amount of heat release during diffusion combustion phase decreases slightly with the increase of ETH and DMC fraction in the blends as shown in Figures 6.53 and 6.54 (BE blends) and Figures 6.57 and 6.58 (BC blends). The longer ignition delay, which is caused by lower cetane number of ETH and DMC, results in more fuel being formed during the period of ignition delay, subsequently increasing the combustion rate of the premixed combustion phase and resulting in the increase of peak heat release rate in the premixed combustion phase. As a result, less fuel is burned in the diffusion combustion phase, and consequently a reduction of combustion
duration (Zhang et al 2005; Yi Ren et al 2008; Lei Zhu et al 2011). However, as shown in Figures 6.51 and 6.52 and Figures 6.55 and 6.56, for the BE and BC blends, the peak cylinder pressure is slightly lower than that of NOME at both low and full loads. This may be due to the effect of lower combustion temperature environment associated with lower calorific value and higher latent heat of vaporization of ETH and DMC’s, almost counterbalancing the opposing effect of ETH and DMC’s lower cetane number, leading to more fuel combusted in the premixed combustion phase and hence the higher peak cylinder pressure. For BG blends, as shown in Figure 6.59, the cylinder pressure decreases at low load, while there is a slight variation in the cylinder pressure at full load as shown in Figure 6.60. The peak cylinder pressure is slightly lower for BG5 and BG10 than that of NOME at both low and full loads. For the BG15 and BG20 blends, the cylinder pressure is slightly higher than that of NOME. This may be due to the higher cetane number of DGL that causes the decrease of ignition delay, leading to an early combustion of DGL in the blends, which results in higher peak cylinder pressure. However, as shown in Figures 6.61 and 6.62, the fraction of heat release during premixed combustion phase decreases, indicating the milder stage combustion phase, which in turn reduces the combustion temperature, and consequently a reduction of NOx emission. This may be due to the decrease in ignition delay and so the amount of ‘prepared fuel’ for combustion after the start of combustion (Noboru Miyamoto et al 1998; Yi Ren et al 2008). The variation of cumulative heat release for with crank angle for BD, BE, BC and BG blends is given in Appendix.A3 (Figure A3.3-A3.6).

6.4.3 Ignition Delay

Figures 6.63-6.66 show the variation of ignition delay with BP for BD, BE, BC and BG blends respectively. For BD blends, as shown in
Figure 6.63, at higher engine loads, the ignition delay decreases with the increase of DEE fraction in the blends due to the increase of cetane number. However, at lower engine loads, the ignition delay increases with the increase of DEE fraction in the blends. As the engine load decreases, the wall temperature and residual gas temperature decrease, leading to the reduction in charge temperature which results in longer ignition delay. In addition, the vaporization of DEE in the blends may cause the injected fuel spray into a lower temperature environment due to its high latent heat value (Qi et al 2011).

For the BE, BC blends, as shown in Figures 6.64 and 6.65, for a specific engine load, the ignition delay slightly increases with the increase of ETH and DMC fraction in the blends. The increase in ignition delay with BE and BC blends could be explained by the vaporization of ETH and DMC in the blends, which cause the injected fuel spray into a relatively lower in cylinder temperature due to their high latent heat values (Huang et al 2003; Lei Zhu et al 2011). In addition, cetane number of DMC and ETH is low, which can further explain the longer ignition delay. The increase in ignition delay is more in the case of BE blends compared with BC blends. The reason may be due to the fact that ETH has high value of latent heat of vaporization and low value of cetane number compared to DMC.

For the BG blends for a specific engine load, as shown in Figure 6.66, the ignition delay slightly decreases with the increase of DGL fraction in the blends due to the increase of cetane number (Yi Ren et al 2008).
Figure 6.63  Variation of ignition delay with brake power for NOME and BD blends

Figure 6.64  Variation of ignition delay with brake power for NOME and BE blends
Figure 6.65  Variation of ignition delay with brake power for NOME and BC blends

Figure 6.66  Variation of ignition delay with brake power for NOME and BG blends
6.4.4 Combustion Duration

Figures 6.67-6.70 show the variation of combustion duration with BP for BD, BE, BC and BG blends respectively. Compared with NOME, the BC, BG and BE blends show shorter combustion duration especially at higher engine loads. Yi Ren et al (2008) showed that the enrichment of oxygen (from oxygenated fuel) will be beneficial to the improvement of diffusion combustion phase, subsequently decreasing the duration of diffusion combustion phase as well as decreasing the total combustion duration with the increase of ETH, DMC and DGL fraction in the blends. However, the BD blends show a slight variation in the combustion duration. This may be due to the erratic operation observed in the engine speed with the use of BD blends.

![Graph showing the variation of combustion duration with brake power for NOME and BD blends](image_url)

**Figure 6.67** Variation of combustion duration with brake power for NOME and BD blends
Figure 6.68  Variation of combustion duration with brake power for NOME and BE blends

Figure 6.69  Variation of combustion duration with brake power for NOME and BC blends
6.4.5 Brake Thermal Efficiency

Figures 6.71-6.74 show the variation of BTE with BP for BD, BE, BC and BG blends. At full load, compared with NOME, the BD5 (shown in Figure 6.71), BE5, BE10 (shown in Figure 6.72), BC5, BC10 (shown in Figure 6.73), BG5, BG10 and BG15 (shown in Figure 6.74) blends give slightly higher BTE, while for the BE, BC and BG blends containing higher percentage of ETH, DMC and DGL, the cooling effect could be the dominant factor for the lower BTE, but the BTE is almost same as that of NOME. The reason for the increase in BTE can be explained as follows. Firstly, adding DEE, ETH, DMC and DGL to biodiesel decreases the viscosity of blends and leads to improvement in the shape of fuel spray and atomization. These finer fuel droplets tend to mix thoroughly with air and hence improve the combustion. Secondly, the combustion is improved on account of increased
oxygen content in the blends, especially during the diffusion combustion phase. Moreover, based on the rate of heat release analysis, faster combustion process could result in an increase of BTE (Noboru Miyamoto et al 1998; Yi Ren et al 2008; Fujia Wu et al 2008; Lei zhu et al 2011).

However, for BD blends greater than 10%, as shown in Figure 6.71, the BTE decreases due to the fluctuations in the engine speed and power output. The reason for the fluctuations in engine output may be due to the high volatility of diethyl ether that leads to vapor lock in the fuel line. At full load, the increase in BTE for BD5 is 1.15%; for BE5-BE10, the increase in BTE is 0.92% to 1.68%; for BC5-BC10, the increase in BTE is from 0.85% to 1.4%; and for BG5-BG15, the increase in BTE is from 0.6% to 1.83%.

Figure 6.71 Variation of brake thermal efficiency with brake power for NOME and BD blends
Figure 6.72  Variation of brake thermal efficiency with brake power for NOME and BE blends

Figure 6.73  Variation of brake thermal efficiency with brake power for NOME and BC blends
Figure 6.74  Variation of brake thermal efficiency with brake power for NOME and BG blends

6.4.6  Brake Specific Energy Consumption

Figure 6.75  Variation of brake specific energy consumption with brake power for NOME and BD blends
Figure 6.76  Variation of brake specific energy consumption with brake power for NOME and BE blends

Figure 6.77  Variation of brake specific energy consumption with brake power for NOME and BC blends
Figures 6.75-6.78 show the variation of BSEC with BP for BD, BE, BC and BG blends respectively. At full load, compared with NOME, the BD5, BE5, BE10, BC5, BC10, BG5, BG10 and BG15 blends give slightly lower BSEC, while the other blends show almost the same BSEC as that of NOME. The BSEC is simply the product of BSFC and lower calorific value. The increase of BSFC of the blends is lower than the corresponding decrease of calorific values of the blends. For the oxygenated fuels, the BSFC increases slightly with the increase of oxygenate fraction in the blends. Two factors are considered to explain this. One is the decrease in the calorific value of the blends with the use of DEE, ETH, DMC and DGL, and requires more fuel to be injected into the cylinder to maintain the same BP, leading to the increase in BSFC. Another is the improvement of combustion on account of the increased oxygen content, and this will decrease the BSFC. As a result of these conflicting factors, there is an increase of BSFC with the use of oxygenates (Noboru Miyamoto et al 1998; Yi Ren et al 2008; Fujia Wu et al
For the blends BE15, BE20, BC15, BC20 and BG20, the BSFC increases quickly which in turn increases the BSEC of these blends.

6.4.7 Exhaust Gas Temperature

Figures 6.79-6.82 show the variation of exhaust gas temperature with BP for BD, BE, BC and BG blends respectively. For the BD, BE, BC and BG blends, the exhaust gas temperatures are lower than that of NOME. The exhaust gas temperatures of the BC blends are lower than the corresponding values of BD, BE and BG blended fuels as a consequence of the difference in lower calorific values of DGL, DEE and ETH. Oxygenated fuels usually cause lower combustion temperatures because of lower energy content and higher oxygen content (Murat et al 2009).

![Graph showing variation of exhaust gas temperature with brake power for NOME and BD blends]

**Figure 6.79** Variation of exhaust gas temperature with brake power for NOME and BD blends
Figure 6.80 Variation of exhaust gas temperature with brake power for NOME and BE blends

Figure 6.81 Variation of exhaust gas temperature with brake power for NOME and BC blends
6.4.8 Carbon Monoxide

Figures 6.83-6.86 show the variation of BSCO emissions with BP for BD, BE, BC and BG blends respectively. For BD5 (as shown in Figure 6.83), BE5, BE10 (as shown in Figure 6.84), BC5 and BC10 (as shown in Figure 6.85), the BSCO emissions are lower than that of NOME at higher engine loads, while for the other BD, BE and BC blends, the BSCO emissions are higher than that of NOME at almost all loads. Two factors are considered to explain this. One is that the BD, BE and BC blends have higher oxygen content, which could improve the combustion and lead to lower BSCO emissions. The other is that the BD, BE and BC blends could reduce the cylinder temperature, which could lead to increase in CO emissions. In this study, as a result of these conflicting factors, for the BD10, BD15, BE15, BE20, BC15 and BC20 blends, the cooling effects of the DEE, ETH and...
DMC could be the dominating factor for the higher BSCO emissions at all loads.

For the BD5, BE5, BE10, BC5 and BC10 blends, BSCO emissions decrease especially at higher engine loads compared with NOME. The possible reason is that the small amount of DEE, ETH and DMC could increase the oxygen content and reduce the viscosity of the blended fuel, leading to improved spray and atomization, better combustion and hence lower BSCO emissions. Moreover, when the percentage of DEE, ETH and DMC is 5-10%, the cooling effect might be small, while the improvement in combustion leads to reduction in BSCO. However, at lower engine loads the cylinder temperature is inherently low. The lower calorific value and higher latent of vaporization of DEE, ETH, and DMC could further reduce the combustion temperature, which results in lower CO oxidation rate. In addition, some of the mixture would occupy the crevice volumes where flame will not be able to penetrate. When these gases mix with hot combustion gases during the latter part of the power stroke and also in exhaust manifold, the oxidation reactions occur, but do not have time to undergo combustion. Thus at lower engine loads, the increase in BSCO emissions is due to the increased amount of mixture present in the quench and crevice zones (Heywood 1998). Moreover, it can be observed that the BSCO emissions for BC blends are lower than the corresponding BE blends at almost all loads as a consequence of the combined effects of latent heat value and lower calorific values in the blends.

As shown in Figure 6.86, for BG blends, DGL has much higher cetane number and lower auto-ignition temperature compared with NOME, leading to a shorter ignition delay. Thus, the total combustion duration is shortened. As a result, the BSCO emissions decrease at all loads with the addition of DGL. Xiangang Wang et al (2011) reported that cetane number is
the most important variable associated with HC and CO emissions. In addition, the higher local oxygen concentration promotes CO oxidation in the case of DGL addition. In this study, higher cetane number could be the dominating factor for the lower BSCO emissions for BG blends. As shown in Figure 6.86, the higher the proportions of DGL in the blends, the lower the BSCO emissions are (Noboru Miyamoto et al 1998; Xiangang Wang et al 2011; Yage et al 2010). Moreover, it can be observed that the BSCO emissions for BG blends are lower than the corresponding BC and BE blends at almost all loads.

Figure 6.83 Variation of BSCO with brake power for NOME and BD blends
Figure 6.84 Variation of BSCO with brake power for NOME and BE blends

Figure 6.85 Variation of BSCO with brake power for NOME and BC blends
Figure 6.86  Variation of BSCO with brake power for NOME and BG blends

6.4.9  Hydrocarbon

Figures 6.87-6.90 show the variation of BSHC emissions with BP for BD, BE, BC and BG blends respectively. Compared with NOME, for BD5 (as shown in Figure 6.87), BE5 (as shown in Figure 6.88) and BC5 blends (as shown in Figure 6.89), the BSHC emissions are lower at higher engine loads, but it increases at lower engine loads. As known, the formation of unburned BSHC originates from various sources and varies widely with operating conditions. For other BC, BD and BE blends, BSHC emissions are higher compared with NOME, but lower in comparison with diesel. This can be explained as follows. Firstly, the increase of HC may be due to the higher latent heat of evaporation of DEE, ETH and DMC causing lower combustion temperature, especially the temperature near the cylinder walls during the mixture formation. In this case, more HC will be produced from the cylinder boundary. Secondly, it may explain the late escape into the cylinder of the
fuel left in the nozzle sac volume, because with the addition of DEE, ETH and DMC this is easier to evaporate and ‘slipped’ into the cylinder (at low velocity late in the expansion stroke) (Heywood 1998; Qi et al 2011).

As shown in Figure 6.90, for BG blends, similar to the behavior of BSCO emissions, the BSHC emissions slightly decrease with the increase of DGL fraction in the blends (Noboru Miyamoto et al 1998; Xiangang Wang et al 2011; Yage et al 2010). Moreover, it can be observed that the BSHC emissions for BG blends are lower than the corresponding BC and BE blends at almost all loads.

![Figure 6.87 Variation of BSHC with brake power for NOME and BD blends](image)
Figure 6.88 Variation of BSHC with brake power for NOME and BE blends

Figure 6.89 Variation of BSHC with brake power for NOME and BC blends
Figure 6.90 Variation of BSHC with brake power for NOME and BG blends

6.4.10 Oxides of Nitrogen

Figures 6.91-6.94 show the variation of BSNO$_x$ with BP for BD, BE, BC and BG blends respectively. The NO$_x$ not only depends on the amount of the oxygenated fuel, but also on the type of oxygenated fuel, because oxygenated fuels, such as ester, DEE, ETH, DMC and DGL which have different properties, such as cetane number, latent heat of evaporation, adiabatic flame temperature and so on. These factors play an important role in NO$_x$ formation in the combustion process. It is recognized that local rich combustion, combustion temperature and residence time in the high temperature environment are important to NO$_x$ formation (Noboru Miyamoto et al 1998; Xiangang Wang et al 2011; Yage et al 2010).

Regarding the effect of BD blends, as shown in Figure 6.91, the NO$_x$ emissions varies slightly for BD blends. It can be observed that the NO$_x$
emission increases with the use of BD5. This may be due to the more complete combustion which results in higher combustion temperature, leading to higher NO$_x$ emissions. The NO$_x$ emission is lower with BD10 blend as compared to BD5. This may be due to reduction in ignition delay as shown in Figure 6.91. However in the case of BD15 blend, the NO$_x$ emission increases. This may be due to higher peak cylinder pressure and heat release rate, which is produced under some erratic conditions, leading to increased NO$_x$ emissions because of some instability and fluctuations observed in engine speed and power output.

For the BC and BE blends except BE5 and BE10, the BSNO$_x$ emissions decrease at almost all engine loads as shown in Figure 6.92 and Figure 6.93. ETH and DMC affects NO$_x$ formation in two aspects. First, the effect of later combustion in the lower temperature environment associated with relatively their higher latent heat of vaporization and lower calorific value could reduce the combustion temperature and hence suppress the NO$_x$ emissions. Second, the ETH and DMC could reduce the cetane number of the blended fuel, leading to longer ignition delay and higher fraction of the premixed combustion phase, and hence higher NO$_x$ emissions. These two factors act against each other. The temperature lowering effect of ETH and DMC could be the dominant factor, leading to the reduction of NO$_x$ emissions and the noise level of the engine noticeably. (Lei Zhu et al 2011; Huang et al 2003; Zhang et al 2005).

For BG blends, the BSNO$_x$ emissions decrease with the increase of DGL fraction in the blends at almost all engine loads as shown in Figure 6.94. The higher cetane number of DGL leads to a shorter ignition delay and a lesser amount of the fuel burned in the premixed combustion phase, and hence a lower NO$_x$ emissions for BG blends. Meanwhile, the lower calorific
value of DGL could reduce the fraction of heat release in the premixed combustion phase, resulting in milder stage premixed combustion phase, which leads to the reduction of NO\textsubscript{x} emission. It is reported that lowering aromatic content in fuel could reduce the NO\textsubscript{x} emission (Noboru Miyamoto et al 1998; Xiangang Wang et al 2011; Yage et al 2010). The lower adiabatic flame temperature of oxygenates could also reduce the NO\textsubscript{x} emissions. (Noboru Miyamoto et al 1998; Md. Nurun Nabi 2010). At full load, from BE15-BE20, the reductions are 5% to 9%; BC5-BC20, the reductions are 7% to 19%; and BG5-BG20, the reductions are 10% to 28%.

![Figure 6.91 Variation of BSNO\textsubscript{x} with brake power for NOME and BD blends](image)

**Figure 6.91** Variation of BSNO\textsubscript{x} with brake power for NOME and BD blends
Figure 6.92  Variation of BSNO\textsubscript{x} with brake power for NOME and BE blends

Figure 6.93  Variation of BSNO\textsubscript{x} with brake power for NOME and BC blends
Figure 6.94  Variation of BSNO\textsubscript{x} with brake power for NOME and BG blends

6.4.11 Smoke Intensity

Figures 6.95-6.98 show the variation of smoke intensity with BP for BD, BE, BC and BG blends respectively. Regarding the effect of BD blends, the smoke intensity varies with BD10 and BD15. At full load, the smoke intensity decreases by 10% for BD5 compared to biodiesel as shown in Figure 6.95. However, in case of BD10 and BD15 blends, the smoke intensity seems to increase but it is still lower than that of NOME and diesel. This may be due to phase separation of the blend, which results in incomplete combustion of the fuel (Mohanan et al 2003).

It can be observed that for BE (shown in Figure 6.96), BC (shown in Figure 6.97) and BG blends (shown in Figure 6.98), the smoke intensity
decreases with the increase of ETH, DMC and DGL fraction in the blends especially at almost all engine loads, in comparison with NOME. It is reported that the reduction of smoke for oxygenated fuels is due to the following. First, dilution of fuel polycyclic aromatic hydrocarbons and sulfur content which are the main contributors for smoke; second, increase of oxygen content which is favorable to the reduction of soot precursors and decreases the engine smoke; Third, different fuel properties (such as cetane number, viscosity and so on) of blends influence the fuel injection, atomization, combustion process, and consequently the emissions. Meanwhile, as smoke is mainly produced during the diffusive combustion phase, the addition of DEE, ETH, DMC and DGL will provide the additional oxygen in combustion and decreases the rich-mixture regions and improves the diffusive combustion phase, leading to the decrease of exhaust smoke (Noboru Miyamoto et al 1998; Zhang et al 2005; Yi Ren et al 2008; Xiangang Wang et al 2011; Yage et al 2010; Qi et al 2011).

Daniel et al. also concluded that DMC and ETH could reduce the soot precursors (and therefore soot) due to the production of OH radicals by the DMC and ETH. At full load, from BE15-BE20, the reductions are 7 % to 30%; from BC5-BC20, the reductions are 7% to 45%; and from BG5-BG20, the reductions are 13% to 58%.
Figure 6.95  Variation of smoke intensity with brake power for NOME and BD blends

Figure 6.96  Variation of smoke intensity with brake power for NOME and BE blends
Figure 6.97  Variation of smoke intensity with brake power for NOME and BC blends

Figure 6.98  Variation of smoke intensity with brake power for NOME and BG blends