Numerical Assessment of the Effect of Injection Pressure and Load on Performance of a Diesel Engine Running on Different Biodiesel-Diesel Blends

Adam Adham, El Mostafa Mabsate
Mechanical and Energy Engineering Research Team: Modeling and Experimentation [ERG2(ME)]
Mohammadia School of Engineers, Mohamed V University
Rabat, Morocco
aadham.adam@gmail.com, mabsate@gmail.com

Abstract— The impact of fuel injection pressure and engine load on combustion characteristics and exhaust emissions of a compression ignition engine powered with different biodiesel-diesel blends is conducted in the paper. The study was carried out by means of Computational fluid dynamics. Combustion of 10%, 20%, 50% and 100% Soybean biodiesel-diesel blends was conducted first with 4 injection pressures (60 to 120 MPa) at constant load and then with 4 loads (from 5 to 12 bar indicated mean effective pressure) maintaining a constant injection pressure. For fuels chemistry, two reduced chemical reaction mechanisms were used to account for diesel and biodiesel blends combustion. Performance parameters reported are peak cylinder pressure and its location, exhaust gas temperature and ignition delay, while emissions of carbon monoxide, carbon dioxide and Oxides of nitrogen were discussed. It is observed that fuel injection pressure increased maximum cylinder pressure for all blends, while its location was moved towards top dead center for blends higher than 20%. Exhaust gas temperatures also increase with increasing injection pressure and biodiesel percentage. Ignition delay was lowered for growing injection pressure but with increasing injection pressure and biodiesel percentage. Injection pressure also increases NOx and reduces CO emissions. It was reported that CO2 is not sensitive to injection pressure variations. The raise in load also increases cylinder pressure, exhaust gas temperatures and CO2 emissions while it reduces ignition delay, CO and NOx emissions.

Index Term— biodiesel, injection pressure, load, diesel engine, CFD, combustion, emissions.

I. INTRODUCTION

The growing interest in biodiesel fuels comes, among others, from their availability, environment friendliness and specially their capability to yield comparable performances to diesel fuel. In fact, biodiesel can be obtained from different sources and feedstocks.[1] It is also among oxygenated fuels, which is very interesting for the combustion process by providing more oxygen to the reaction. However, this oxygen content is also responsible for enhancing combustion temperature to the point of increasing NOx emissions.[2] Besides, it is known that combustion stoichiometry and efficiency are improved, which has a lowering effect on carbon monoxide and soot emission. Carbon dioxide is then expected to increase. However, some of the CO2 emissions are absorbed at the croplands[3]. It is worth mentioning that, depending on the source, biodiesel fueled engines can have different sorts of pollutants emissions trends compared to conventional diesel.

The increasing interest is observed in the number of research papers that report some positive outputs for the use of biodiesel-diesel blends, with biodiesel derived from animal fats as well as waste cooking oils and various plants. Ghazali et al.[1] presented a large review on the varying effects biodiesel fuels can yield. They concluded that, biodiesel can lead to similar performance than diesel, which makes it a promising alternative, as long as the blend ratios do not necessitate substantial changes of the engine itself.

Since performance and emission characteristics of biodiesel-fueled engines vary with the feedstock, production, blending ratio and operating conditions, it becomes mandatory to study each parameter thoroughly. Another challenge biodiesel fuels face is the engine setup. In fact, Biodiesel fuels, despite having similar properties to diesel, are not fully taken into consideration during the design of the engine. They have higher viscosity and density, which invariably changes the injection parameters. The viscosity raise in mechanical injection systems leads to fast pressure evolution and advanced injection timing[2]. To overcome this difference from diesel, the fuel injection system must be reshaped so it accounts for the higher density and viscosity. While electronically controlled injection systems can be programmed to compute different fuel properties, this is not the case for traditionally mechanically controlled fuel injection systems.

The pressure at which fuel is injected can be a determining step in the injection process. Actually, increasing injection pressure can enhance the atomization and spray penetration inside the combustion chamber[4]. The decrease in droplet size improves the mixing and stoichiometry is reached faster, leading to faster heat release and improved combustion efficiency.

Agarwal et al[5] used different blends of Karanja biodiesel with diesel to assess the rate of fuel injection, performance and pollutant emissions of a diesel engine, at different injection pressures, ranging from 30 to 100 MPa. They observed that injection rate is influenced by injection pressure. Duration of injection substantially decreases with injection pressure. Brake thermal efficiency (BTE) also improves with increasing injection pressure. NOx emissions were found higher than in...
diesel in the case of 10% and 20% Karanja volume percentage while at 50% Karanja, NOx were almost identical. They concluded that using up to 10% of Karanja biodiesel could help improve efficiency and lower emissions.

Anbarasu et al. [6] Studied experimentally the effect of using an emulsion of Canola at different pressures of injection on the performance exhaust pollutant of a single cylinder compression ignition engine. Injection pressure was set from 20 to 24 MPa. They found that BTE increased to reach 28.48% at 24 MPa. Specific fuel consumption (SFC) was the lowest also at 24 MPa. NOx, carbon monoxide (CO) unburnt hydrocarbon (UHC) were found lower at 20 MPa. They concluded that Performance parameters are improved at high injection pressure, while emissions are found to be lowest at 20 MPa.

Gumus et al.[7] discussed the exhaust emissions and SFC of a diesel engine fueled with biodiesel-diesel blends (ranging from B0 for pure diesel to B100 for pure biodiesel) at four injection pressures, from 18MPa to 24MPa and from low to full load. They concluded that CO2, and NOx emissions increase while CO and UHC decrease with the raise in injection pressure. Increasing injection pressure showed that SFC was lowered for high biodiesel percentage blends, while it had opposite behaviors on low percentage biodiesel fuels.

The different sources of biodiesel fuels and the different operating conditions imply that for every engine configuration and every feedstock, a proper study has to be carried out to fully grasp the extent of biodiesel usage. This stems from the complexity in biodiesel composition. It is comprised of fatty acid methyl esters (FAME). Depending on the structure of the acids, i.e. number of carbons of the fatty acid, the carbon-carbon double bonds and the saturated or unsaturated species[8], biodiesel may have different behavior inside an engine. The need for more efficient tools is inevitable to understand how the composition of biodiesel fuels affects their chemistry in combustion engines. Computational fluid dynamics (CFD) coupled with chemical kinetics can allow studying the effect of biodiesel composition on combustion engines.

The objective of this study is to numerically evaluate the effect of injection pressure and load on the performance and emissions characteristics of a single cylinder, direct injection diesel engine, when running with five biodiesel-diesel blends. We use Soy biodiesel as the base methyl ester. Simulations on five blends are conducted: B10, B20, B50, B10O and B0, where BX implies that X % of biodiesel in volume is mixed with diesel and B0 is pure diesel. Injection pressures selected are 60 (original), 80, 100 and 120 MPa. Loads considered are 5 bar, 7 bar, 10 bar and 12 bar. Performance parameters are peak cylinder pressure and its location, exhaust gas temperature and ignition delay, while emissions of CO, CO2 and NOx are reported. The study is implemented using the finite volume software Ansys Fluent. Biodiesel blends and diesel fuel are represented by two chemical reactions mechanisms.

II. NUMERICAL MODEL

This section establishes the numerical model for the simulations. The simulations were conducted with finite volume Ansys Fluent software. The steps of the simulation along with the corresponding modeling are:

A. Geometry and mesh settings

The geometry of the combustion chamber was created using the Design Modeler tool. The fluid domain of the combustion chamber at TDC (top dead center) is modeled. The engine details are shown in table 1. In order to reduce computation time, only a 60° sector (corresponding to one hole of the injector) was considered and a hexahedral mesh was created. To analyze mesh grid dependency, three mesh resolutions were considered: 8063, 16095 and 24173 cells at TDC. Before launching proper calculations, simulations were conducted with these three different resolutions to check the grid independency. Table 2 resumes the comparisons of maximum pressure and the three resolutions. It is concluded that the second mesh resolution (12063) is the best for the simulations. The accuracy achieved was good and the computation time is 40% less than the third resolution. Fig. 1 shows the chosen mesh at TDC.

<table>
<thead>
<tr>
<th>Type</th>
<th>Diesel DI single cylinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement</td>
<td>300 cm³</td>
</tr>
<tr>
<td>Bore</td>
<td>70 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>78 mm</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>139 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>19.5</td>
</tr>
<tr>
<td>Valves/cylinder</td>
<td>4</td>
</tr>
<tr>
<td>Swirl ratio</td>
<td>2.5</td>
</tr>
<tr>
<td>Intake valve opening</td>
<td>13 CAD after TDC</td>
</tr>
<tr>
<td>Intake valve closing</td>
<td>20 CAD After BDC (bottom dead center)</td>
</tr>
<tr>
<td>Exhaust valve Opening</td>
<td>33 CAD before BDC</td>
</tr>
<tr>
<td>Exhaust valve closing</td>
<td>18 CAD after TDC</td>
</tr>
<tr>
<td>maximum valve opening</td>
<td>7.30/7.67 mm (intake/exhaust)</td>
</tr>
<tr>
<td>Injection timing</td>
<td>10° before TDC</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>600 bar</td>
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</table>

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Corse</th>
<th>Intermediate</th>
<th>Fine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cells</td>
<td>8063</td>
<td>16095</td>
<td>24173</td>
</tr>
<tr>
<td>Computation time</td>
<td>55min</td>
<td>2h40min</td>
<td>4h05min</td>
</tr>
<tr>
<td>Peak pressure (bar)</td>
<td>69.82</td>
<td>70.78</td>
<td>71.31</td>
</tr>
<tr>
<td>Difference from experimental peak pressure</td>
<td>1.45%</td>
<td>0.2%</td>
<td>0.65%</td>
</tr>
</tbody>
</table>
B. Fluent models

The key step in the combusting modeling is the setup of fluent models and boundary conditions. The models used are:

1) Turbulence model

Flow turbulence was accounted for using the Renormalization Group (RNG) $k - \varepsilon$ [9]. Transport equations representing turbulent kinetic energy $k$ and rate of dissipation $\varepsilon$ are [9]:

$$
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + P_i + \rho \varepsilon \tag{1}
$$

$$
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{\mu} \frac{\varepsilon}{k} P_i + C_{\mu2} \frac{\varepsilon}{k} \tag{2}
$$

With $C_{\mu2} = C_{\mu} + \frac{C_{\mu} \eta’ (1 - \eta)}{1 + \beta \eta}$

Where: $\rho$ is the mixture density, $u$ is the velocity component in the $i$ direction, $\mu$ is the dynamic viscosity, $\mu_t$ is the eddy viscosity, $\rho, \eta, \eta’, C, \sigma_k, \sigma_\varepsilon, C_\mu, C_{\mu2}$ are model constants and $P_i$ is the production of turbulent kinetic energy.

2) Turbulence interaction model–combustion chemistry

The diesel unsteady Flamelet model [10] was adopted for this simulation. This model establishes the chemistry modeling in a single, 1D laminar Flamelet, which allows for a great reduction in the calculation time while maintaining the accuracy[11].

For the fuels chemistry, two reaction mechanisms were taken for the representation of diesel and biodiesel blends. The biodiesel was modeled by a reaction mechanism consisting of 69 species and 192 reactions by Brakora[12]. The main species for biodiesel chemistry are n-heptane, Methyl Decanoate (MD) and Methyl 9-decenoate (MD9D). It also takes in consideration NOx emissions. This mechanism was reported to be efficient and less time costly, provided that appropriate proportions of the saturated and unsaturated species in the actual biodiesel fuel are correctly set. The saturated species are represented by MD and unsaturated ones are represented by MD9D. Table 3 lists the proportions of MD and MD9D used for the blends from B10 to B100[13]. Diesel fuel was also represented by a surrogate mechanism of 29 species and 52 reactions[14]. The main species representing chemistry fuel is n-heptane.

Since the diesel mechanism does not consider NOx emissions, thermal NOx modeling of fluent was enabled. Thermal NO obeys the following transport equation: [11]:

$$
\frac{\partial}{\partial t} (\rho Y_{NO}) + \nabla \cdot (\rho \vec{v} Y_{NO}) = \nabla \cdot (D \nabla Y_{NO}) + S_{NO} \tag{3}
$$

Where: $Y_{NO}$ is the mass fraction of NO in the gas phase, $D$ is the effective diffusion coefficient and $\rho$ is the density of the gas phase. The Governing equation for the formation of NO is given with the extended Zeldovich mechanism [15]:

$$
O + N_2 = NO + N \tag{4}
$$

$$
O_2 + N = NO + O \tag{5}
$$

$$
N + OH = NO + H \tag{6}
$$

3) Spray breakup and droplet properties

The liquid fuel is injected at the end of compression stroke. Droplets injected in the combustion chamber undergo a series of forces and shear stresses to atomize and breakup. The droplets breakup is considered by means of the wave model [16]. The model was used with its recommended constants. Fuel droplets properties are listed in table 4[17].

4) Injection settings and calculations

The solid cone spray was used to set up the injected fuel. The setup of the injection influences injection pressure and load. A single rectangular shape for injection was adopted. It starts at the injection timing and lasts the required time for the fuel quantity to be loaded inside the cylinder. Fuel quantity was calculated to match engine load.

5) Boundary and initial conditions

The simulation covers the time where the valves are closed, i.e., from IVC to EVO. The initial conditions for temperature and pressure were taken as 313K and 1 bar respectively. As for cylinder walls and piston bowl, temperature is 313K[13]. The injection velocities were calculated with the equation:

$$
V = C_\mu \sqrt{\frac{2(P_{rail} - P_{inj})}{\rho_{fuel}}} \tag{7}
$$

Where $P_{rail}$ is the fuel rail pressure and $P_{inj}$ is the cylinder pressure at time of injection and $C_\mu$ is the discharge coefficient. The cylinder pressure (=5MPa) is very small compared to injection pressure. The discharge coefficient was fixed to 0.6[18].

![Fig. 1. Meshed domain at TDC](image-url)
6) Solution methods and simulation convergence conditions

For the calculations to yield accurate results, convergence criteria for the different residuals must be carefully set. Convergence conditions for the residuals are: for continuity: 0.1, x, y and z velocities: 10e-3, energy: 10e-6, k and ɛ: 10e-3. For this study, a dynamic mesh with the layering approach is adopted. The PISO method (Pressure Implicit with Splitting of Operator) for Pressure velocity coupling method is used. Skewness and neighbor correction factors are set to 0 and 1 respectively. Spatial discretization scheme for gradient is least squares cell based. For pressure, density, turbulent kinetic energy, turbulent dissipation rate, energy, mean mixture fraction and mean fraction variance, second order upwind is used.

7) Comparison with experimental data

To ensure the validity of the model, simulations were compared to experimental results [19]. Pressure traces and NOx emissions for B0, B20, B50 and B100 from experimental setup are used. Load and speed were kept constant (2 bar and 1500 RPM respectively). In Figure 2 (a) to (d), pressure traces for the four blends and NOx emissions are compared. Simulated cylinder pressures are in good agreement with the experimental results. The ignition delay is also predicted (with a difference less than 15% degree for the three fuels). The NOx emissions are shown in figure 3. Emissions in the case B0 are 26% higher than experience. This difference was expected, as the NOx modeling in Fluent is very sensitive to temperature. The other cases are in better agreement with the experience since the NOx pollutant is already included in the mechanism. The next simulations are carried out with B10 as well, even if there is no experimental result to compare with. It is assumed that since the modeling for all biodiesel blends produce satisfactory results, B10 would at least follow the same trends.
III. RESULTS AND DISCUSSION

A. Effect of injection pressure on performance and emissions characteristics

1) Maximum cylinder pressure and location

Figure 4 presents the maximum cylinder pressure for each injection pressure and blends. For all the blends, peak pressure is increased with higher injection pressure. Air-fuel mixing is best achieved and atomization is improved with injection velocity. The latter increases turbulence in the combustion chamber near TDC. For all the blends, increase in peak pressure goes from 12.24% (B10), to 14.3% (B20). At injection pressure of 60 MPa, B20 shows a slightly low pressure than B10. For each case of injection pressure, peak pressure also increases with the increase of biodiesel percentage. The biodiesel contains a high oxygen percentage, which yields an improved combustion.

The positions of cylinder peak pressure for the different blends are shown in figure 5. For all the injection pressures, an increase in biodiesel percentage tends to retard the location of peak pressure. The increasing viscosity of the fuel can postpone the spray atomization and breakup. The growing oxygen content and the decreasing lower heating values of the fuels also delay the position. For B20, B50 and B100, peak pressure location tends to move closer to TDC as injection pressure is increased. The mixing and atomization are enhanced for these three fuels. B0 shows very little sensitivity to injection pressure in terms of peak pressure location. In the case of B10, peak pressure goes from 363.8° to 364.9° for 60 MPa and 80MPa respectively, and then it decreases to 364°.

2) Exhaust gas temperatures

Variations of exhaust gas temperatures are shown in figure 6. For all the injection pressures, the increase in biodiesel percentage increases the exhaust temperatures. This is due to the oxygen content in biodiesel fuels that lead to better combustion. Increase in injection pressure also increases exhaust gas temperature. The mixing and atomization are improved and fuel droplets are smaller, hence improving the reaction and increasing temperatures. For every blend, injection pressure increase yields small increment in Exhaust temperature. The maximum increase is observed for B10 (4.94% increase between 60 and 120 MPa).

3) Ignition delay

Figure 7 shows the change in ignition delay as a function of fuel blends and injection pressures. For all the fuels, ignition delay is reduced as the injection pressure increases. Since the same quantity of fuel is injected at a fixed load, the increase in injection pressure reduces the injection duration and the fuel droplets[5]. The atomization is then faster and combustion takes place earlier than with conventional injection pressure. For each case of injection pressure, the increase in biodiesel percentage shows an increase in ignition delay, since the viscosity and latent heat of biodiesel is bigger that diesel’s. If the injection pressure is too high, the droplet sizes can be too
small for a homogenous mixture to be formed which leads to very poor combustion and increases the formation of soot particles[20].

4) CO emissions
CO rate at the exhaust is an indicating parameter of the combustion quality in a diesel engine. Figure 8 shows the emissions of CO for the different fuels at respective injection pressures. Biodiesel effect is visible since the increase in biodiesel percentage decreases CO emissions. CO emissions has a 58% decrease from B0 to B100 in the case of 60MPa. Injection pressure also reduces the CO emissions. As for injection pressure, CO emissions in the case of B0 are reduced by 52.4% while this reduction tends to decrease with the biodiesel percentage. Higher injection pressures better combustion efficiency, air fuel mixing as well as evaporation and thus lower CO emissions[20,21]. The growing volume fraction of biodiesel is also responsible of the lower CO emissions.

5) CO2 emissions
CO2 emissions are shown in figure 9. The effect of biodiesel percentage is an increase in CO2 emissions for all injection pressures. This also indicates that the combustion efficiency is improved as biodiesel percentage increases. Biodiesel is known for producing less CO2 emissions than diesel[1]. Simulations showed that increase in injection pressure has no substantial effect on the CO2 emissions. Different biodiesel sources yield different CO2 emissions. Kannan [23] reported an increase in CO2 emissions while using a blend of waste cooking palm oil, diesel and ethanol, with 30%, 60% and 10% respectively for each. The fast vaporization with improved combustion increases CO2 at the exhaust. In contrast, when Nanthagopal et al.[24] used Calophyllum inophyllum biodiesel at higher injection pressure, CO2 emissions fell from 7.2% to 6.9% when raising injection pressure from 20 to 40 MPa. This insensitivity of the results to CO2 can be attributed to the high injection pressure compared to all experiences found or to the production mechanism of CO2. However, the trend is well captured and in agreement with experiments.

6) NOx emissions
Figure 10 shows the variations of NOx emissions for the different injection pressures and blends. It is observed that for B50 and B100, NOx emissions increase with the injection pressure. This is explained by the higher temperatures reached in the combustion chamber due to higher heat release. Sayin et al.[25] reported similar founding when increasing injection pressure from 20 to 24 MPa on a C.I engine fueled with biodiesel-diesel blends. The same comment can be made about the other fuels for injection pressures below 120MPa. At 120 MPa, it is observed that NOx emissions slightly decrease for B0, B100 and B20. The high velocity of the injected fuel enhances the fast vaporization, which justifies the decreasing tendency. NOx at the exhaust are found higher for increasing biodiesel fraction, for all injection pressures. This is because the oxygen content in biodiesel leads to higher temperature and hence greater NOx emissions.

Fig. 7. Ignition delay with respect to fuels at different injection pressures
Fig. 8. CO mass fractions with respect to fuels at different injection pressures
Fig. 9. CO2 mass fractions with respect to fuels at different injection pressures
Fig. 10. NOx emissions with respect to fuels at different injection pressures
B. Effect of engine load on performance and emissions characteristics

1) Maximum cylinder pressure and its location

Figure 11 shows the variations of peak cylinder pressure for each fuel at different loads. The raise in engine load leads to an increase of cylinder pressure. As load increases, injected mass into the cylinder also increases, providing more fuel for combustion. Cylinder pressure also increases with biodiesel percentage. For B0, peak pressure goes from 84.5 bar to 90.58 bar at 5 bar and 12 bar IMEP, while in the case of B100, maximum pressure is 92.2 bar and 99.63 bar.

Figure 12 shows the variations of peak pressure location for all the fuels at different loads. It can be seen that, like in figure 5, the increase in biodiesel percentage moves the peak pressure position away from TDC. For every fuel, the increase in load also pushes it away from TDC. This behavior is justified by the fact that higher oxygen content in biodiesel enhances the combustion efficiency, which yield a higher pressure. Gopal [26] also reported similar trends when using Pongamia biodiesel.

2) Exhaust gas temperatures

Figure 13 shows the variations of exhaust gas temperatures for each fuel at different loads. It is visible that the raise in load produces higher temperatures at the exhaust. It can be said that the more mass is injected in the combustion chamber, the longer it is combusting. Temperature at the exhaust also increases with biodiesel volume proportion. This stems from the content of oxygen in biodiesel that enhances the combustion process. Higher latent heat of vaporization makes the combustion last longer, thus increasing the temperature at the opening of the exhaust valve[27].

3) Ignition delay

Figure 14 shows the variations of ignition delay for all the fuels at different loads. As a common result, the increase in engine load leads to a common decrease in ignition delay for all the fuels. The increase in engine load lads to more injected fuel mass and provides more oxygen so a homogenous mixture is obtained at the spray envelope, leading to a rapid combustion, which lowers the ignition delay. However, B100 exhibits higher ignition delay at 5 bar IMEP, but the trends vis-à-vis the engine load is still maintained. Ignition delay for B0 varies from 6.8° to 7.4° while biodiesel blends give ignition delays from 6.5° (for B50 at 12 bar IMEP) to 8.5° (for B100 at 5 bar IMEP).

4) CO emissions

Figure 15 shows the effect of load on the CO emissions for all the blends. It can be seen that CO emissions are decreased as the biodiesel percentage increases. Biodiesel has a significant proportion of oxygen, which is suitable for better combustion, and resulting in more CO oxidation. This result is in agreement with the experimental results in the literature [2,3]. As for engine load, the CO mass fraction decreases for all the fuels. It can be said that the increase in engine exhaust temperature leads to the oxidation of CO into CO₂. In the case of B100, the simulation of B100 showed a very high value for CO emissions at 5 bar IMEP (0.0012). Although the model was rechecked, this discrepancy remained in the final solution. However, the trend for the increasing load is captured.
5) CO₂ emissions

Figure 16 shows the variations of CO₂ emissions for the different cases. It is found that the raise in load boosts the CO₂ at the exhaust. The increase in Exhaust gas temperature can be the reason behind this increase since high temperatures allow for more complete CO oxidation into CO₂. For every case of load, the increase in load also yields higher CO₂ emissions. Along with the high exhaust temperature, the oxygen content of biodiesel makes the combustion more complete; hence, more CO₂ is produced. These trends are in agreement with the literature results and analysis [1].

6) NOx emissions

Figure 17 depicts the effect of engine load on the NOx emissions for all the fuels. As expected, NOx emissions are higher in biodiesel blends than diesel. Due to the high temperatures reached, the oxygen content reacts more with nitrogen. However, the engine load does not have the same effect on all fuels. NOx emissions were found to be higher in the case of 7 bar IMEP for all the fuels. Beyond 7 bar, NOx emissions decreases for all fuels. NOx emissions vary from 850 ppm to 1046 ppm for B0, while for blended biodiesels it goes from 771 ppm (B10, 5bar) to 1354 ppm (B100, 7 bar). This decrease can be explained by the reduction of the time of reaction inside the cylinder, leading to a shorter time of residence of the gases in the combustion chamber [1].

IV. CONCLUSION

Throughout this study, the effect of injection pressure and load were evaluated using CFD and chemical kinetics mechanisms. A 60° sector of a single cylinder diesel engine fueled with 10%, 20%, 50% and 100% biodiesel-diesel blends was simulated. Chemical reaction mechanisms were used as surrogates for diesel and biodiesel fuel blends. Injection pressures ranged from 60 to 120 MPa while load ranged from 5 to 12 bar, maintaining constant engine speed in all cases. Results of the simulations showed:

- For all blends, increase of injection pressure caused a raise in maximum cylinder pressure. The high injection velocity improved air-fuel mixing and atomization increases turbulence in the combustion chamber near TDC. The Raise in cylinder pressure was between 12.24% for B10 and 14.3% for B20. The location of peak pressure was moved to TDC as injection increases for B20, B50 and B100 while there was a difference of one degree between the cases 60 and 80 MPa B10. B0 showed very little difference in peak pressure location. Ignition delay also is lowered with injection pressure for all fuels and increases with biodiesel percentage. Exhaust gas temperatures also are increased with biodiesel percentage increase due to the more oxygen content. However, for each blend, less than 5% increase was observed for different injection pressures. The increase was more observed with engine load increase. CO emissions are reduced with injection pressures and biodiesel percentage. The combustion seemed to be more complete as injection pressure improved the fuel-air mixing. However, CO₂ was barely affected by the injection pressure variation, but increased with biodiesel percentage. NOx emissions at the exhaust were higher with both injection pressure and biodiesel percentage.

- Increase in load and biodiesel percentage led to increased peak cylinder pressure and pushed its position away from TDC. The ignition delay is lowered for increasing load but showed no unique trend for biodiesel percentage at each IMEP. The increase in load and biodiesel percentage reduced CO emissions. CO emissions of B100 at 5 bar was very high compared to other values. In contrast, CO₂ emissions are augmented as load and biodiesel percentage increased. While NOx emissions increased with biodiesel percentage, the trend caused by engine load is not established for each blend.
Injection pressure can have a beneficial effect on combustion performance and some pollutant emissions (CO), while it increases NOx. It is to be noted that the study is conducted with fixed engine speed, boundary and ambient parameters for all the cases, which is not the case in a long time running engine. However, it is a fair estimation of engine performance since physicochemical properties of biodiesel fuels can vary greatly.

REFERENCES


