Comparison and analysis of the combustion process of three combustion modes of an ethanol - diesel direct injection engine

Yu Liang¹, Liying Zhou¹,²,*

¹School of Mechanical Engineering, Guiyang University, 103 Jianlong Road, Guiyang, 550005, China
²School of Mechanical Engineering, Guizhou Institute of Technology, 1 Caiguan Road, Guiyang, 550003, China

*Corresponding author e-mail: zhoulijing0330@163.com

Abstract. A diesel engine was modified into an ethanol - diesel direct injection engine. Diesel was injected through the engine’s original pump injection system, whereas ethanol was injected through an installed electronic controlled injection system. 100% of ethanol injection at 240°CA (referred to as B100), 50% of ethanol injection at 240°CA and 50% injected subsequently with diesel at 344°CA (referred to as B50L50) and 100% of ethanol injection with diesel at 344°CA (referred to as L100) were the three combustion modes. Based on test data, a 3D combustion chamber model was established for the analysis of the combustion modes. The result showed that B50L50 had the earliest combustion starting point, the highest combustion pressure and maximum average temperature in the cylinder, the largest maximum cumulative heat release and sufficient combustion. In addition, NO generation was the highest but the soot generation during the initial combustion period was not the highest even though it was high. The soot generation of B100 was the highest. For L100, the combustion starting point was the latest, after burning was severe, cylinder pressure and maximum temperature were lower, NO and soot generation were the lowest, but CO generation was the highest, comparatively.

1. Introduction

The Statistical Review of World Energy conducted by BP in June 2018 showed that China is still the largest energy consumer in the world, accounting for 23.2% of global energy consumption.[1] China’s petroleum output reduced by 4.1 % in 2017, but its petroleum consumption increased by 6.0 %, hence increasing China’s dependence on foreign petroleum to 70%, which is the apex so far.

China’s renewable energy consumption has an annual growth of 31 %. In 2017, China’s renewable energy consumption was 36 % globally, surpassing that of the USA to become the world’s largest renewable energy consumer. But the total output of ethanol and biodiesel was 205.3 million tons of oil equivalent, only accounting for 2.5% of the world’s total output. China has made it clear that it will vigorously develop renewable energy, including ethanol. China’s Medium and Long-term Development Plan for Renewable Energy involves increasing the annual production capacity of ethanol to 10 million tons by 2020.[2] In China, ethanol from non-food crops such as cassava, sweet...
Ethanol is volatile, and therefore easily forms combustible mixture that facilitates fast and full combustion. Also as an oxygen-bearing substance, ethanol enhances sufficient combustion that represses the generation of carbon smoke. In addition, ethanol has high latent heat of vaporization which helps in reducing high combustion temperature and the emission of NOx.[4] The main modes of ethanol application in diesel engines include either mixing at the inlet or directly injecting the ethanol into the cylinder.

From research investigations, [5-9] engines fueled with diesel-ethanol mixture have almost the same power performance with extended ignition delay period, improved in-cylinder combustion process, slightly improved thermal efficiency and significantly reduced smoke intensity, as compared to those fueled with diesel only. However, to obtain a comprehensive economics and emission performance, the optimal ethanol blend ratio for different engine working conditions has to be realized.

The biggest advantage of using diesel-ethanol mixture in diesel engines is that the fuel supply system does not need any modification to handle the fuel. However, with low cetane number, ethanol’s affinity to easily catch fire is low and therefore this presents a disadvantage of its usage. Also, as it is difficult for ethanol to be mutually soluble with diesel, the mixed fuel has poor stability at low temperature [10-15] and poor safety due to low flash point.

Ethanol’s injection in the inlet pipe accompanied by premixing in the inlet can be adjusted with regards to the working conditions of the engine to alter the substitution rate. In addition, the effective thermal efficiency can also experience improvement as the premixed combustion of the mixed fuel is fast.[16-18] However, problems such as ethanol deposition in the inlet due to poor atomization, ethanol emission (small amount) during valve overlapping and scavenging, easy detonation in the case of increased substitution rate, increased combustion circulation changes and reduced engine charging efficiency[19,20] are series of issues that can be experienced.

With direct injection, the problems such as additional cost and preparation process associated with the forming of a stable diesel-ethanol mixture can be avoided. Since the amount of ethanol injected into the cylinder can be precisely controlled, the working mode of fixed proportion blending fuel can be changed. Also fuel deposition in the inlet owing to poor atomization and the emission of a small amount of ethanol due to valve overlapping and scavenging can equally be avoided. Hence this increased the performance and thermal efficiency of diesel engines. [21-23]

In order to go further in the knowledge of combustion process of ethanol-diesel direct injection engine, this article is structured into four additional sections, the contents of which will be presented now. The test equipment and experimental methods were summarized in section “Test equipment and methods”. The combustion model established for the analysis of three combustion modes was described in section “Combustion model”. In section “Combustion simulation results and analysis”, the combustion simulation results were presented and discussed. Finally, the relevant conclusions of the study were presented in the “Conclusions” section.

2. Test equipment and methods
Two diesel generators were transformed and used for the test. The first transformation was based on DG7500SE diesel generator, and the second one was based on KS7500SE diesel generator. The main technical parameters of the two engines are shown in Table 1.
Table 1. Summary of engine specifications

| Parameters                                      | No.1 DG7500SE                                      | No.2 KS7500SE                                      |
|------------------------------------------------|----------------------------------------------------|----------------------------------------------------|
| Type                                           | Vertical, air cooled, natural aspirating, direct injection, four-stroke | Vertical, air cooled, natural aspirating, direct injection, four-stroke |
| Cylinder bore × stroke / mm×mm                  | 86×72                                              | 88×75                                              |
| Length of the connecting rod / mm               | 115                                                | 115                                                |
| Compression ratio                               | 19                                                 | 20.8                                               |
| Rated rotation speed / r·min⁻¹                  | 3000                                               | 3000                                               |
| Diesel supply advance angle (°CA in front of TDC) | 20                                                 | 21                                                 |
| Inlet valve closing (°CA behind BDC)            | 54                                                 | 54                                                 |
| Exhaust valve open (°CA in front of BDC)        | 54                                                 | 54                                                 |
| Number of nozzles of the diesel injector        | 4                                                  | 4                                                  |
| Diameter of nozzles of the diesel injector / mm | 0.32                                               | 0.32                                               |
| Number of nozzles of the ethanol injector       | -                                                 | 6                                                  |
| Diameter of nozzles of the ethanol injector / mm| -                                                 | 0.183                                              |

Table 1 shows that the main technical parameters of the KS7500SE diesel generator were similar to the main technical parameters of the DG7500SE diesel generator. On the first diesel generator (DG7500SE), ethanol-diesel mixture test experiment was conducted and the results were used to verify the developed 3D combustion model. Details are presented in reference.[24] Since the KS7500SE single-cylinder diesel engine had larger cylinder bore, its cylinder head was equipped with the electronic control gasoline injection system for ethanol injection. Under the calibrated working condition of the original diesel engine, pure diesel experiment was conducted on the KS7500SE diesel generator, where the fuel consumption required to analyze the total calorific value was obtained. The advance angle of the diesel injection, diesel and ethanol fuel injection law were measured on an injection test bench.[25]

The ethanol-diesel direct injection test was carried out on the KS7500SE diesel generator. A diagrammatic representation of the test bench is shown in Figure 1. The diesel fuel (0# diesel) was purchased from a gas station (PetroChina), and the ethanol (content≥99.5%) was produced by Chongqing Chuandong Chemical (Group) Co. Ltd.
Figure 1. Schematic diagram of the test bench

1. engine, 2. diesel injection pump, 3. exhaust muffler, 4. dynamometer, 5. generator, 6. opacimeter, 7. exhaust gas analyzer, 8. diesel conveying pump, 9. diesel filter, 10. diesel tank, 11. diesel consumption display 12. diesel consumption tester, 13. diesel injector, 14. ethanol high pressure oil rail, 15. ethanol injector, 16. ethanol consumption tester, 17. ethanol consumption display, 18. ethanol tank, 19. ethanol filter, 20. motor, 21. ethanol high pressure pump, 22. air intake filter, 23. cylinder pressure sensor 24. crank shaft angle sensor, 25. combustion test analyzer, 26. Computer

For the test experiment, the generator was first started with diesel and preheated for 5 min under no-load. Afterwards, it was run subsequently for 10 min, 10 min and 5 min under 1.5 kW, 3.0 kW and 4.5 kW loads respectively. After running for 5 min under 4.5 kW load, ethanol was then injected at this engine load. As the generator continued to run at 4.5 kW, the quantity of ethanol (diesel fuel replacement rate) was gradually increased and the diesel injection quantity was gradually reduced as the engine electronic governor operated to keep the engine running constantly at 3000 rpm. The fuel consumption per minute was then determined. That is when the diesel consumption was 29–33 mL per minute, ethanol consumption was about 28–34 mL per minute, and therefore this made the rate of consumption of diesel and ethanol almost 1:1. Cylinder pressure of 100 cycles was continuously collected to analyze fuel consumption and emissions. The tested power refers to the power consumed by the electric load, which was lower than the actual power of the engine.

3. Combustion model
The prototype of the combustion model was the KS7500SE diesel generator. The initial boundary conditions required for the simulation were measured from the experiment described above. For the simulation, the geometric model of the combustion chamber at TDC (top dead center) was developed and meshed with Pro Engineering (ProE) and Hypermesh software, respectively. Afterwards, the meshed model was imported into the CFD work environment (AVL FIRE software) for the final adjustment and refinement and also the setting of the boundary conditions, sub-models and computation parameters. After the simulation, the post-processing analysis was performed with the same CFD software.

The meshed combustion chamber is shown in Figure 2, where it can be seen that the area close to the nozzle boundary was densely meshed with 110,496 cells at TDC and 348,096 cell at BDC (bottom dead center). Relative to the piston center, the combustion chamber center was 3 mm offset.
The diesel injector nozzle had 4 uniformly distributed holes while the ethanol injector nozzle had 6 non-uniform distributed holes. The simulation settings for the injector positions and the nozzle holes are shown in Table 2 and Table 3 respectively.

| Nozzle | X-coordinate (m) | Y-coordinate (m) | Z-coordinate (m) | X-direction | Y-direction | Z-direction |
|--------|------------------|------------------|------------------|-------------|-------------|-------------|
| Ethanol | -0.0016          | -0.001           | -0.0016          | -0.0016     | 0.00226274  | -0.0016     |
| Diesel | 0.003            | -0.003           | 0                | 0.007       | 0.0397      | 0           |

**Table 2. Nozzle position**

| Nozzle number | Nozzle diameter at hole center positions (m) | Number of nozzle holes | Spray angle delta 1 (deg) | Spray angle delta 2 (deg) | Circumferential hole distribution (deg) |
|---------------|---------------------------------------------|------------------------|--------------------------|----------------------------|----------------------------------------|
| Ethanol 1     | 0.001                                       | 1                      | 120                      | -175                       | 0                                      |
| Ethanol 2     | 0.0004                                      | 1                      | 140                      | 140                        | 0                                      |
| Ethanol 3     | 0.001                                       | 1                      | 120                      | 95                         | 0                                      |
| Ethanol 4     | 0.001                                       | 1                      | 90                       | 35                         | 0                                      |
| Ethanol 5     | 0.0008                                      | 1                      | 60                       | -40                        | 0                                      |
| Ethanol 6     | 0.001                                       | 1                      | 90                       | -115                       | 0                                      |
| Diesel 1      | 0.003                                       | 1                      | 160                      | 135                        | 0                                      |
| Diesel 2      | 0.003                                       | 1                      | 160                      | -135                       | 0                                      |
| Diesel 3      | 0.003                                       | 1                      | 140                      | -45                        | 0                                      |
| Diesel 4      | 0.003                                       | 1                      | 140                      | 45                         | 0                                      |

**Table 3. Nozzle hole data**

The chemical reaction kinetics were achieved with the standard ethanol and diesel transportation sub-model in AVL FIRE. The main sub-models selected for the computation are shown in Table 4.

Figure 3 shows the spray cloud from the two injectors. In order to describe the spray cloud image (i.e. post-processing of simulation results), two sections were defined (section A is the section through the center of the diesel injector and section B is the section through the center of the ethanol injector as shown in the figure).
Three cases were simulated with the established combustion model and in all the three cases, the diesel injection starting point was 344°CA and the injection quality of diesel and ethanol was the same. However, the differences in the cases were the ethanol injection points. In the first case, all the ethanol (100 %) was injected from the injection starting point of 240°CA (B100) while in the second case, the injection point of ethanol was at two stages (B50L50). That is 50% of the ethanol was injected at the first stage (at the injection starting point of 240°CA) and the remaining 50% was then injected at the second stage (at the injection point of 344°CA). In the third case, all the ethanol (100 %) was injected from the injection starting point of 344°CA (L100). Some important parameters for spray and combustion simulation are shown in Table 5.

### Table 5. Parameters for spray and combustion simulation

| Parameter                        | Value                  |
|----------------------------------|------------------------|
| Injected diesel mass             | 17.5 mg/cycle          |
| Injected ethanol mass            | 16.4 mg/cycle          |
| Initial in-cylinder temperature  | 373K                   |
| Initial in-cylinder pressure     | 118KPa                 |
| Diesel injection duration        | 14.1°CA                |
| Ethanol injection duration       | 24.6°CA                |
4. Combustion simulation results and analysis

4.1. Cylinder pressure and combustion heat release rate

![Figure 4](image)

**Figure 4.** Cylinder pressure and combustion heat release rate

Figure 4 shows cylinder pressure and combustion heat release rates for all the combustion modes. The cylinder pressure curve shows that at the compression stage, the cylinder pressure of B50L50 was slightly lower than that of the B100. The reduction in the cylinder’s temperature and pressure was attributed to the vaporization and endothermic characteristics of the ethanol. After ignition, the cylinder pressure (in the case of B50L50) had the fastest increment and therefore attained the highest maximum combustion pressure, comparatively. Even though the B100 followed similar trend as the B50L50, the B100’s cylinder pressure increment and the maximum combustion were slightly lower than those of the B50L50. The cylinder pressure (in the case of L100) had the slowest increment. And also the crank angle of the maximum combustion pressure was significantly behind those recorded by B100 and B50L50. Hence after TDC, the maximum combustion pressure attained under L100 was relatively lower.

The heat release rate curve shows that the combustion starting point under L100 was significantly behind the combustion starting points attained under B100 and B50L50. The heat release rate curves obtained under B100 and B50L50 followed the same trend, however, under B50L50 the heat release rate curve had a steeper gradient and a slightly higher maximum value. The heat release rate attained under L100 (see Figure 4) had double-peak characteristics with a longer duration and a more obvious diffusion combustion characteristic.

4.2. Average temperature in the cylinder and cumulative heat release

![Figure 5](image)

**Figure 5.** Average temperature in the cylinder and accumulated heat release
Figure 5 shows the in-cylinder average temperatures and cumulative heat release rates attained under the three combustion modes. From the figure, it can be seen that the higher the initial quantity of ethanol injected, the lower the in-cylinder temperature. For B100 and B50L50, the temperature rise was fast at the initial combustion period, but the in-cylinder temperatures under B100 maximized earlier while those under B50L50 were still on the rise. Because of the late ignition, the fast temperature rise period noted under L100 occurred after TDC, however, the temperature maximized with a lower value comparatively due to the increasing volume of the cylinder. From the cumulative heat release rate curves, it was realized that though the cumulative heat release under B100 occurred earlier than the cumulative heat release under L100, they were almost the same. The cumulative heat release rate attained under B50L50 was the highest, comparatively. This means more sufficient combustion heat release and the highest fuel utilization rate under B50L50.

4.3. NO, soot, and CO fraction
In Figure 6 and Figure 7, the in-cylinder average mass concentration of NO and soot is respectively shown, while in Figure 8, the in-cylinder average CO mole fraction is shown.

The three figures show that the NO and soot generation from L100 were the lowest comparatively, however, the CO generation from L100 was significantly higher than those from B100 and B50L50. The NO generation from B50L50 was the highest while the highest soot generator was from B100.

The trend of the curves shows that under all the cases, the NO generation gradually increased till it stabilized. Also under all the cases, the CO generation sharply increased initially and then decreased due to the gradual oxidation. However, for the soot generation, the three cases had different trends. The curves for B100 and L100 had the same trend, while the curve for B50L50 had different trend. Under B50L50, the maximum soot generation was higher than those attained under the other two cases because the soot oxidation occurred in the post-combustion stage. However, due to the soot oxidation, the soot generation from B50L50 was lower than that from B100. Also because of the soot oxidation, the heat release and cumulative heat release rate obtained under B50L50 increased.

![Figure 6. NO mass concentration in the cylinder](image-url)
4.4. Discussion and analysis

The ethanol injection began soon after the inlet valve was turned off. At this stage the piston was at a lower position, the cylinder pressure and temperature were low and the ethanol injection backpressure was also low. The ethanol’s spray injection speed was high and cone angle was small, since it only encountered small air resistance. In addition, the ethanol spray had weak swirl and little disturbance due to low airflow in the cylinder. Furthermore, as a result of low in-cylinder air temperature and slow gasification of the ethanol, the spray penetration speed of the ethanol was high and the ethanol spray had easy interaction and adhesion with the wall surface. [25-28]

Figure 9 shows the distribution of the ethanol components in the cylinder at 270°CA (30°CA after ethanol injection) (sections A and B in the figure are defined in Figure 3). As shown in Figure 9, due to the installation position of the ethanol injector and the structural characteristics of the injection hole, the ethanol spray was mainly injected at the cylinder wall and at the top of the piston ring surface. With the upward movement of the piston and air in the cylinder, the ethanol gradually diffused and mixed with the air in the cylinder.[29] Nonetheless, the cylinder wall surface and the top surface of the piston ring were still the places with the highest concentration of ethanol.
Figure 9. Ethanol mass fraction at 270°CA

Figure 10 shows the distribution of the ethanol components in the cylinder at 344°CA (section C is perpendicular to the axis of the cylinder and is located at the top of the piston bowl). It can be seen that a lot of ethanol did not mix in the combustion chamber. Under the B100 case, more ethanol was concentrated at the top surface of the piston ring than under the B50L50 case. Due to less air, the formation of quick and complete combustion of the ethanol mixture was not possible. Under B100, the large concentration of ethanol at the top surface of the piston ring close to the cylinder wall surface resulted in lower heat release rate, lower cylinder pressure rise rate and lower maximum combustion pressure and temperature as compared to those obtained under B50L50 during the initial combustion period.
Figure 10. Ethanol mass fraction at 344°CA

Figure 11 shows the distribution of the in-cylinder temperature, diesel mass fraction and ethanol mass fraction at the combustion starting point under the three combustion modes. Due to the high quantity of ethanol injected at the early period, the B100 ethanol’s vaporization and endothermic characteristics were high, thus causing lower cylinder pressure and temperature, and delayed ignition starting point as compared to those of B50L50. For both B100 and B50L50, the diesel quickly ignited the premixed ethanol after catching fire because the diesel particles vaporized and mixed with air easily after injection. [30,31] For L100, however, since the diesel and ethanol were injected at the same time, the large amount of ethanol concentrated in the combustion chamber absorbed the heat and atomized, thus affecting the atomization and evaporation of the diesel spray. Yao Chunde, et al of Tianjin University [32] also confirmed that the occurrence of ethanol mixture extended the ignition delay period and lift-off length of diesel because of the strong impediment on diesel ignition. This resulted in late ignition starting point.

Although the earliest ignition positions were different under the three combustion modes, the ignitions occurred within the outer layer of the diesel spray. Since the diesel stayed longer in the cylinder, it had sufficient contact with air to form premixed gas of adequate ignition concentration. In addition, the evaporating diesel absorbed the gas heat from the areas involved, cooling down the fuel spray center. However, since the gas in the outer layer of the fuel spray had higher temperature, the outer layer of the diesel spray reached burning conditions earlier. [33]

Under B50L50 and B100, the adequate concentration of ethanol mixture was present in the cylinder, hence as soon as ignition occurred, the mixture burnt quickly. For B50L50, high temperature and strong airflow disturbance due to burning accelerated the mixing of the diesel spray with ethanol. This led to quick combustion and heat release, and therefore resulted in higher heat release rate and shorter interval of heat release. With the downward movement of the piston, the thick ethanol on the top surface of the piston ring gradually participated in the combustion due to air disturbance. Nevertheless, since B100 had more concentrated ethanol than B50L50, its combustion was not as sufficient as that of B50L50.

Under B50L50, the combustion temperature was the highest, and therefore NO generation was the highest, comparatively. Under L100, however, the late ignition gave rise to noticeable combustion characteristics of the ethanol such as long combustion duration, severe post-combustion, lower cylinder pressure and maximum temperature, and finally, concentrated fuel in the combustion chamber, which led to insufficient oxygen content. NO generates in a high-temperature oxygen-rich environment while soot generates in a high temperature oxygen deprived environment. Therefore, under L100, the low temperature and oxygen deprived local areas ensured that NO and soot generation was the lowest, however, due to insufficient combustion, CO generation was significantly higher as compared to the other cases.

Soot experiences two stages of generation and oxidation. The high temperature and oxygen deprived areas around the diesel spray during the initial combustion period led to soot generation [34,35] with a higher component fraction in the case of B50L50 than that in the case of B100. For
B50L50, by the end of the combustion period, high temperature and rich oxygen conditions together with a high temperature flame were formed due to high temperature and strong airflow disturbance that occurred. Hence soot oxidation was accelerated in this case.[36]

![Temperature in the cylinder](image1)

![Temperature in the cylinder](image2)

![Temperature in the cylinder](image3)

**Figure 11.** Temperature distribution and diesel and ethanol mass concentration at the ignition starting point

5. **Conclusion**

A set of electronic controlled gasoline injection system was installed on a single-cylinder diesel engine. Diesel injection into the cylinder was through the original pump injection system, while ethanol injection was by the electronic control injection system. Three different combustion modes were realized from different fuel injection strategies. In the first mode, all the ethanol was injected at the beginning of the compression stroke to form a pre-mixed combustion ignited by diesel (B100). In the second mode, a quantity of ethanol was injected at the beginning of the compression stroke and then another quantity was injected together with diesel, to form partial premixing combustion-partial diffusion combustion (B50L50). In the third mode, all the ethanol was injected together with diesel to form diffusion combustion (L100). A 3D combustion model was established and used to analyze the three combustion modes. Based on the installation position of the injectors, the following conclusions were drawn after the analysis.

1) Under the B50L50 combustion mode, earlier combustion starting point resulted in higher maximum combustion pressure and average temperature in the cylinder, higher cumulative heat release, higher fuel utilization rate and adequate combustion, as compared to the B100 and L100 combustion modes. In addition, NO generation was the highest. Soot generation was higher during the initial combustion period, however, the soot was gradually oxidized during the later period of
combustion. The two-stage ethanol injection (to form premixed ignition combustion -partial diffusion combustion) ensured the complete use of the fuel to obtain higher thermal efficiency.

2) Under the B100 combustion mode, the high concentrated ethanol at the top surface of the piston ring close to the cylinder wall surface resulted in the reduction of ethanol in the combustion chamber than in the case of B50L50. This resulted in lower in-cylinder pressure, lower heat release rate and lower average temperature in the cylinder as compared to the B50L50 combustion mode. Due to lack of high temperature and oxygen rich conditions for soot oxidation in the later period, soot generation was the highest under the B100 combustion mode.

3) Under the L100 combustion mode, the late ignition period resulted in late combustion starting point, long combustion duration, severe post-combustion, lower in-cylinder pressure and lower maximum temperature. Furthermore, lower NO and soot were generated as compared to B100 and B50L50 combustion modes, but the generated CO was the highest.

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