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Comparison of performance and emission characteristics of diesel and diesel-water blend under varying injection timings

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Abstract

The present work is focused on comparison of diesel and diesel-water blend on emissions and performance parameters of the diesel engine. The emission parameters such as NO, CO and soot are investigated numerically by using commercially available CFD software AVL FIRE. In addition to that, performance parameters in terms of thermal efficiency, indicated power, torque and brake specific fuel consumption is also analyzed numerically. For better understanding of the software, two validations are provided in the present study. In first validation, the commercial code is validated by experimental data available in the literature; and it is found to be satisfactory in terms of pressure and temperature. Whereas, second validation is taken out with model-TV1 of Kirloskar single cylinder diesel engine and results of pressure and heat release rate are acceptable. Numerical simulation is conducted on a four-cylinder, 1.9L GM diesel engine with piston having a rectangular cavity. The blend is prepared by using 95% diesel and 5% water and compared with diesel fuel. As a part of the above study, injection timing is varied as 6°, 10° and 14° BTDC for diesel as well as diesel-water blend. The result shows that, diesel-water blend reduces emission without affecting the engine performance parameters, when engine fueled with diesel alone. Nevertheless, at an injection angle of 6° BTDC, diesel-water blend with 5% by mass is seems to be an effective substitute in diesel engines in terms of engine emissions and operation parameters.

Keywords: Diesel engine, combustion modeling, diesel-water blends, NO, CO and soot emission, efficiency and power

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1. Introduction

Emission control is of paramount concern to most of the researchers and scientists around the world with ever increasing threat of global warming. It is an established fact that the majority of the emissions which affect the surroundings are mainly led by automobiles and manufacturing industries. Vehicular emission is expected to grow rapidly in future owing to the development being on top of the agenda of most of the developing nations around the world. The pollutants emitted from vehicles, particularly from diesel engines are recognized as a major source of air pollution, which affects human life and the environment adversely. It is nearly impossible to manipulate the number of vehicles; thus, control of emission may be the only plausible answer.

Significant progress has been made during the last decade in the area of emission control. Many researchers focused in compression ignition engines owing to higher rate of emissions from diesel engines and the fact that they are used as the major propulsion power source for land and marine transport applications. Researchers are working in many ways to reduce emissions from the CI engines. However, methods such as blending of diesel with water are quite popular. Diesel - water blends exhibited greater effectiveness to reduce emissions as compared to other methods of reducing emission. The relative merits and demerits of blend are well documented in the literature (Park et al., 2004; Alam-Fahd et al., 2013; Alahmer, 2013).

There are four different ways to introduce water in the diesel engines; direct injection of water in the engines through separate injectors, fumigation of water into the intake air, an unstable mixture (mixing of water and fuel before injection), preparation of mixing using stable blends. These four ways are widely used to reduce emissions by diesel - water blending technique. Numerous analyses are available on diesel engines using diesel blended with water in different proportions. Liang et al. (2004) reported that the performance of engine is better with a certain quantity of water (20%) in fuel having low oxygen content. Subramanian (2011) worked on two different methods of introducing diesel water blends in diesel engine and reported that the diesel - water blend method has greater potential to reduce smoke and NOx emissions at different load conditions as compared to injection method. Sahin et at. (2014) experimented with the use of a carburetor to inject water with varied amount of intake air. It reported that reduction of NO emission and smoke formation without affecting engine power and fuel consumption, however, peak pressures and indicated power remained invariant with the addition of water into the intake air. Zhang et al. (2013) carried out experiments on a 4 cylinder diesel engine with varying diesel - water blend ratio from 0% to 20% and oxygen concentration from 21% to 23% at a constant speed and load condition of 2000 rpm and 180 Nm respectively.

Result showed considerable reduction in NO and smoke emissions. Lif et al. (2006) reported reduction of NO_x emissions by 30% and particulate matter (pm) by 60% using diesel - water blend with water contents from 5% to 45%. Khan et al. (2014) investigated the effect of various surfactants with several blends of emulsified fuels on the combustion characteristics, emission formation and engine behavior. It was found that the stability of diesel - water blend was dependent on the suitable emulsification technique and speed of the agitator. Experiments were also carried out with 44% and 86% load conditions. Zaid (2004) and Badrana et al. (2011) reported an increase in the engine torque, power and brake thermal efficiency with an increasing diesel - water blend ratio while conducting experiments on a DI single cylinder diesel engine. Further, Zaid [10] found that the addition of 2% surfactant to the blends had no effect on engine performance. Murotani et al. (2007) carried out experiments in which both fuel and water were injected through the fuel injector. Experimental measurements and prediction of CFD analysis reported reduction in local combustion temperature and combustion speed as well as reduction in NO_x emission and soot formation when the water injection approximately timed. Bedford et al. (2007) performed CFD analysis to describe the practical NO_x reduction technology for the DI diesel engine.

Results showed that latent heat was responsible for further penetration of spray of fully mixed diesel - water blends than a spray of the same volume of diesel alone. The role of CFD in engineering is not new and it is improving day by day. Benny et al. (2010) used an earlier version of CFD software GAMBIT with STAR CD to mesh helical, spiral and helical - spiral inlet manifold to study the effect of turbulence in air movement. Now a day's researcher is more focused on advanced I.C engine software like KIVA 3V, AVL FIRE etc., those have inbuilt mesh operation. Wagner et al. (2008) performed numerical simulation using a 3D CFD code KIVA 3V in a DI diesel engine with diesel - water blend and predicted an improvement in the NO emissions. The numerical simulation can be done by using mathematical formulation of different models. Sharma et al. (2013) performed theoretical investigation to analyze the effect of air cavity in diesel engine valves in terms of radial thermal stress, heat flux and temperature. The present study is also based on numerical simulation by using commercially available CFD software AVL FIRE.

The above review of literature clearly indicates that there is ample experimental work has been carried out using water mixed with the base fuel. The performance of the blend is found to be passably safe for several operating conditions of speed, load, and blend mass fractions. Diesel - water blend tends to behave differently under different operating conditions; yet, no general comment can be prepared with regard to effectiveness to control emission. The major reason behind the use of diesel water blend is to reduce emissions up to possible minimum level and proper injection timing can help to reduce emission level as well. So, it seems an interesting point to analyze the combined effect of blend and injection timings on emissions. There has not been any study which has undertaken effect of injection timing on diesel - water blends. Therefore, there is a need to find their performances. Although the task of making a meaningful comparison between diesel and diesel - water mixtures with varying injection timings on a single experimental setup under similar operating conditions may prove to be painstakingly difficult, such a comparison can be carried out using numerical simulation with ease. In the present investigation a comprehensive comparative analysis of the base fuel and its blend with water by employing commercially available AVL FIRE software has carried out on a four cylinder diesel engine with a piston having rectangular bowl geometry. In this work, simulations are performed using diesel and diesel – water blend through a single injector with variable injection timings to predict NO, CO and soot emissions along with engine performance parameters.

2. Description of CFD model

The modeled engine sample is a 4.1 L, 4 cylinders, DI diesel engine, which is used for whole simulation process. The specifications of the engine and initial conditions are listed in Table 1 and Table 2 respectively. In order to perform the combustion simulation, a hexahedral mesh of rectangular bowl geometry with 35976 cells is created by using AVL FIRE ESE diesel module shown in Fig 1. The grid independent test is conducted for three different sets of cells, i.e. 35976, 58452 and 78332 of the same geometry and the cylinder pressure of the three meshes is depicted in Fig 2; the variation in pressure is found to be ±1%. Thus, the simulation is carried out with 35976 cells to reduce the process time.

Table 1. Engine specifications.

Bore	0.105 m	Compression ratio	16.00
Stroke	0.120 m	Swirl ratio	1.6
No. of cylinders	4	Engine speed	1800 rpm
Displacement volume	4.1 L	Fluid mass	1.7286e-05 Kg
Connecting rod length	0.200 m		

Table 2. Initial/boundary conditions

Initial pressure	0.256 bar	Liner temperature	470° K	Spray angle	160°
Initial temperature	386° K	Head temperature	570° K	Fuel injection type	Single injection
Piston temperature	570° K	Fuel injection timing	Varied		

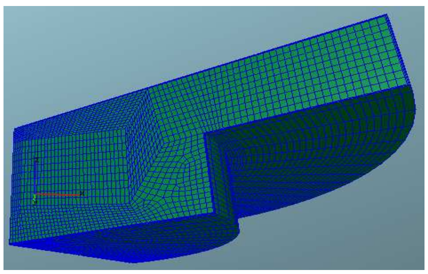


Fig 1. Computational grid at TDC.

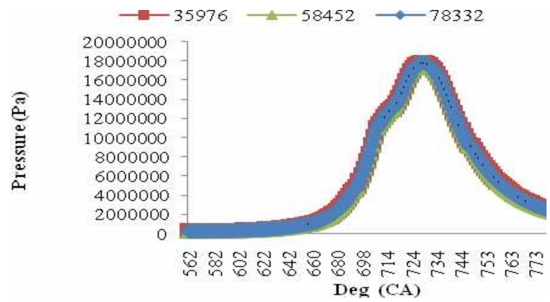


Fig 2. Grid independency test.

3. Applied models

In the present work, commercial CFD software AVL FIRE is used, which specially developed for IC engine applications. This software is based on a finite volume approach by applying the Cartesian coordinate system. This software is validated by Tatschl et al. (2007). Simulation performed by using several improved physical and chemistry models, listed in Table 5 .The k-zeta -f turbulence model is used in the present study was developed by Hanjalic et al. (2004). This model is used to improve numerical stability by solving transport equation. The wave breakup model is used for spray modeling, which depends upon the physical and dynamic parameters of the injected fuel and the domain fuel; however, it depends mainly on the wavelength of the speed of the droplets. This model is used for diesel fuel spray simulation (Reitz 1987). Spray wall interaction model is used for accounting the effect of non - atomized or non- evaporated fuel particles striking the wall of the combustion chamber.

A spray wall interaction model known as 'walljet1' is used (Uludogan, 1996). In the case of walljet1, the droplets get a rebound or slide over the wall by the formation of vapor cushion under the droplets. The extended zeldovich mechanism is used to calculate the NO emissions and this mechanism considers the effect of hydrocarbon radicals, nitrogen and oxygen on NO formation. NO formation of nitrogen, oxygen and hydrocarbon radicals very much depends on combustion temperature (Zeldovich et al. 1947). This model can be coupled with ECFM-3Z combustion model based on equilibrium approach. The kinetic soot model is used in the present simulation as it can be used for different fuel classes to describe the behavior of soot formation and oxidation (Appel et al. 2000). The kinetic soot model can solve the 1850 gas phase reaction, 186 species and 100 heterogeneous reactions with the participation of micro-heterogeneous particles of different types (AVL FIRE user manual V2013.1). In this study, both Dukowicz model (Liu et al. 1993) and multi-component evaporation model are used subject to type of fuel involved in the combustion process.

Further, ECFM-3Z shown in Fig 3, a type of combustion model based on laminar flame let concept is applied in the present work.

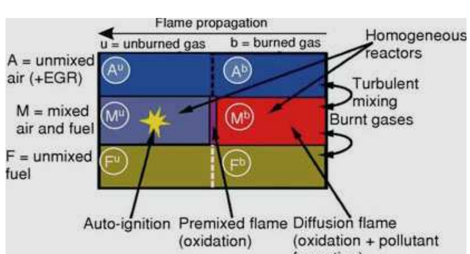


Fig 3. ECFM-3Z model computational cell.

Such models are useful for the treatment of chemistry and turbulence. The flame let models assume that the reaction takes place in a relatively thin layer separating unburned gases from the burned gases (AVL FIRE user manual V2013.1. - Colin et al., 2004). ECFM-3Z model is applicable for auto ignition case as shown in Fig 3. This model has capability to handle both ignition procedure i.e. auto ignition and spark ignition. The simulation of compression and power strokes, the volume of gases has been considered enclosed between three wall boundaries namely piston, for cylinder head and liner. In order to close the system of equations, boundary conditions needs to be prescribed at these boundaries; the temperatures of cylinder head and piston are 570.15 K and that for the liner is 470.15 K as listed in Table 2.

3. Validation of the CFD software

Validation is carried out on a GM diesel engine model and a kirloskar single cylinder diesel engine (Model TV-1) by using parameters listed in Table 3 and Table 4 respectively. The predictions of numerical simulation of GM diesel engine model compared with experimental results (Aggarwal, 2011). Fig 4 shows good agreement between experiment and simulation results in terms of in cylinder pressure and temperatures at 1500 rpm. Furthermore, piston of kirloskar single cylinder diesel engine (Model TV-1) is validated in terms of in-cylinder pressure and heat release rate (HRR) against the experimental data as shown in Fig 5; although, qualitative agreement between predicted and experimented results is found to be good, however, some discrepancies were observed particularly at TDC. This may attributed to the uncertainty of input parameter such as injection duration.

Table 3. Engine parameters of first validation (GM diesel engine)

Make	GM diesel engine	Stroke	90.4 mm	Fuel	C_7H_{16}
Number of cylinders	4	Connecting rod length	145.4 mm	Spray cone angle	9°
Bore	82 mm	Engine speed	1500 rpm	Mass of fuel	1.2495e-06 kg
Stroke	90.4 mm	Compression ratio	16.8	Injection duration	8.5°

Table 4. Engine parameters of second validation (Kirloskar engine, TV-1)

Make	Kirloskar engine, TV-1	No. of nozzle holes	3
Number of cylinders	1	Connecting rod length	234 mm
Bore X Stroke	87.5 mm X 110 mm	Compression ratio	17.5
Swept volume	661 cc	Rated speed	1500 rpm

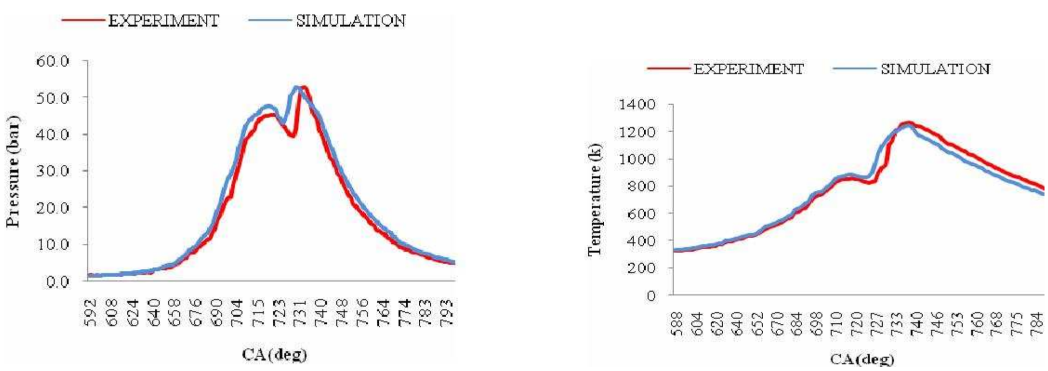


Fig 4. Comparison of cylinder pressure (bar) and temperature (K).

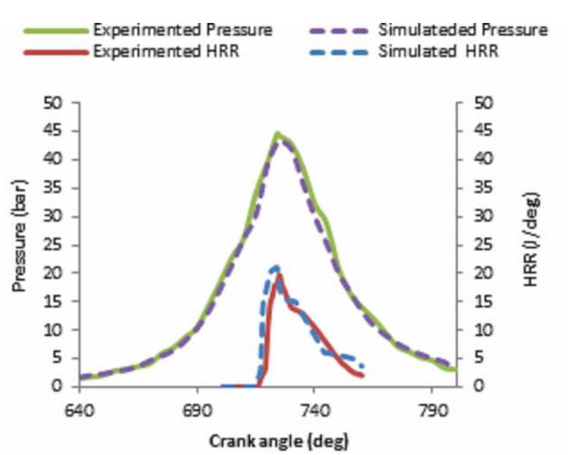


Fig 5. Comparison of cylinder pressure (bar) and heat release rate (J/deg).

4. Case study

Since the main focus of the present investigation is to compare the performance of diesel and diesel - water blend, all the computations are carried out under identical conditions of injection timing using the same piston bowl geometry. For the present investigation, two different cases are considered. Engine is simulated with base fuel i.e. pure diesel and blend of diesel - water with water fraction of 5%. Further, both cases are then simulated for injection timings of 6,10◦ and 14◦ BTDC.

5. Combustion process

The combustion of water diesel blend depends on the difference of boiling temperature of the two constituents. Their difference in boiling temperature influences the evaporation rate of molecules. Thus, water molecules tend to attain superheated stage faster than the diesel molecules (Watanabe et al. 2010, Saito et al. 2010, Watanabe et al. 2010, Biao et al. 2002, Lin et al. 2006, and Hagos et al. 2011). The two phenomena occurs at this stage; first is called the micro explosion, in which liquid is distributed in tiny droplets, and the second is called puffing, in which water leaves in term of droplets in very fine mist. Then, secondary atomization occurs due to micro explosion which causes fast evaporation and improving air fuel mixing. The process of mixture formation and combustion is homogeneous inside the combustion chamber therefore it reduces mass fraction. Moreover, some amount of heat is utilized for evaporation process (evaporation enthalpy) which causes sinking of peak temperature in the combustion chamber. Accordingly, NO formation decreases because the NO formation increases exponentially with the combustion chamber temperature. Murozumi and Saito (2010) concluded that the micro explosion depends on three major factors namely volatility of base fuel, water content and type of emulsion used. They also stated that an increase in emulsifier quantity will increase micro explosion temperature and waiting time.

6. Effect of injection timings

NOx formation depends on temperature as well as residence time. The burnt gas arising from the part of the combustion, which occurs before peak pressure, is compressed due to the rising pressure in the combustion chamber. This means it remains at high temperatures for a long time compared with the burnt gas from the later stages of combustion. This allows more time for NOx to form. Delayed injection leads to lower pressure and temperature throughout most of the combustion, thus NOx formation reduced. Delayed injection increases fuel consumption due to later burning, as less of the combustion energy release is subject to the full expansion process and gas temperatures remain high later into the expansion stroke, resulting in more heat losses to the walls. Smoke also increases due to reduced combustion temperatures and thus less oxidation of the soot produced earlier in the combustion (Colin et al. 2004).

It is well known that fuel injection timing has a strong bearing on the start of combustion and then the attainment of in-cylinder pressures and temperatures. If injection timing is advance from 6° to 14^0 BTDC, ignition delay period will higher and result shows high in-cylinder pressures and temperatures would develop resulting in favorable conditions for the formation of NOx. It can be seen that any given blend, the NOx levels are lower for retarded injection timing. Thus it is established from the present investigation that lower NOx would be obtained with the retarded injection timings.

7. Result and discussion

In order to make a meaningful comparison of the performance of diesel with diesel - water blend, three injection timings are used. Simulations are carried out with injection timings varying from 6◦ to 14◦ BTDC. The predictions in terms of the emissions of CO, NO and soot mass fractions respectively are plotted against crank angle. Further, predictions are also made for the performance parameter such as indicated thermal efficiency, power, torque and brake specific fuel consumption for various injection timing.

7.1. Effect of diesel fuel on emission and performance parameters

CO is not only considered as undesirable emission but also represents loss of energy. Poor mixing, local rich regions and incomplete combustion are sources of CO emission. As the injection timing advanced from 6° BTDC to 10° BTDC, mixing of air-fuel tends to improve. At more advanced injection timing from 10° BTDC, combustion becomes uncontrolled due to improper mixing, high temperature and pressure inside combustion chamber resulting in an increase in CO mass fraction at 14° BTDC.

NO is mostly created by the presence of nitrogen air. At low temperature, nitrogen exists as a very stable form of diatomic molecule (N_2). The diatomic molecule is not responsible for formation of NO but small amount of NO is there because of it. However at high temperatures of combustion chambers, diatomic nitrogen breaks in to mono atomic nitrogen (N), which is highly reactive. Formation of NO also depends on pressure and air-fuel ratio. Rich concentration of fuel causes high temperature inside the cylinder which results to high amount of NO formation in that area. Combustion duration is also a reason for NO formation. If combustion process is fast then formation of NO is reduced. As the NO formation depends on temperature so it's continuously increasing from 6° BTDC to 14°BTDC because of high temperature.

Solid carbon soot particles are generated within fuel rich zones inside combustion chamber. Soot is also known as exhaust smoke. Most of the soot formation occurs when engine is loaded and maximum amount of fuel is injected for greater power production. Soot is the combination of carbon spheres and traces of other components. Carbon spheres are generated within the combustion chamber at fuel rich zone where oxygen concentration is low. If mixing of fuel and air is good then 90% to 95% of carbon particles are converted in to CO_2 during combustion. Additionally, lubricating oils are producing 25% of carbon in soot which vaporize and react during combustion. A single soot particle may contain about 5000 carbon spheres.

Gases inside the cylinder reach to low temperature during the power stroke because of high compression ratio. This leads to condensation of remaining high boiling point components (found in fuel and lubrication oils) on the surface of the carbon soot particles. These soot particles are known as soluble organic fraction (SOF). It can be seen that at low temperature, SOF is as high as 50% of the total soot mass fraction. At high temperature it reduces to 3% of total soot mass fraction. If combustion time is extended, then soot formation can be reduced because of carbon particles get enough time to mix with oxygen and converted to CO_2. As the temperature increases from 6° BTDC to 14°BTDC, soot mass fractions reduce continuously.

But longer combustion time is responsible for high temperature and more NO formation. EGR techniques used to reduce NO level but they tend to increase soot emissions. Finer droplet size of fuel can reduce soot formation by increasing injection pressure but it also increases in cylinder pressure and NO emissions. Engine design must be focused to reduce CO, NO and soot formation by controlling the injection timing, injection pressure and valve timing but there is need of some sort of compromise. The effect of diesel fuel on performance parameters like power, thermal efficiency, torque and brake specific fuel consumption are reported by many researchers and their findings are listed in the literature.

7.2. Effect of diesel-water blend on emission and performance parameters

In this study, water is mixed with diesel fuel in 5% of total mass. The results in terms of NO, CO and soot emissions and performance parameters of the engine are plotted against crank angle. The combined study of CO mass fraction shows that, graphical trend is decreasing continuously from 6° BTDC to 14° BTDC. The water particles get vaporize inside the cylinder by absorbing heat of combustion and convert to finer droplets which lead to better mixing of fuel with air; subsequently, CO mass fraction seems to be decreasing. However, soot mass fraction appears to be increasing due to low temperature inside the combustion chamber and reduction of temperature is attributed to absorption of a large fraction of heat of combustion in the evaporation process, which leads to incomplete combustion. NO mass fraction increases at different injection timing because of higher temperatures.

Addition of water has positive effect on the combustion efficiency of the engine. The improvement in thermal efficiency is observed due to better mixture formation and enhanced premixed combustion phase particularly at 6° BTDC; whereas at other injection timings, thermal efficiency of diesel fuel is almost equal to the blend. The output torque increases with water content over the entire rpm range. Water is turned into steam, While, the charge is fired inside the cylinder under very high temperature and pressure. The another possible reason for the improved combustion efficiency is that, the presence of the oil - water interface with very low interfacial tension, leads to finer atomization of fuel during injection. Finer dispersion of fuel droplets facilitates higher contact with air and thus increases the burning process, which is advantageous for the combustion. It has been postulated that water in fuel improves the combustion process owing to simultaneous rupture of drops, to elevate evaporation surface of drops and facilitates the better mixing of fuel burning in air.

The power output of diesel-water blend is higher than diesel fuel at 6° BTDC because of prolonged ignition delay. Long ignition delay absorbs less compression work at the end of compression stroke and it pushes pressure curve to expansion side which results to more power at the expansion stroke than diesel fuel. This effect is less observed in case of 10° and 14° BTDC due to short ignition delay. The torque produced by diesel-water blend is higher than diesel fuel at 6° BTDC due to better mixing of fuel with air; while, poor air fuel mixing is observed due to short ignition delay at other injection timings. It produces same level of torque for diesel and diesel-water blend at 10° and 14° BTDC as shown in Fig 23(a). Fig 23(b) shows break specific fuel consumption of diesel and diesel-water blend at different injection timings. It can be seen from the Figure that, high amount of fuel is consumed by the engine because of long ignition delay, to produce same amount of power at 6° BTDC. Whereas, ignition timing of 10° and 14° BTDC is consuming less quantity of fuel with diesel and blend due to shorter ignition delay; however, blend consumed more fuel than diesel to produce same power.

7.3. Effect of injection timing on emissions and performance parameters

7.3.1. 6°BTDC

Fig 5 & 6 show reduction in pressure and temperature which may be due to proper mixing and micro explosion of water droplets by taking heat of evaporation from combustion chamber. As shown in Fig 7, CO formation gets reduced because it depends on air fuel mixing, local rich regions and quality of combustion. The addition of water leads to enhance combustion process and proper mixing inside chamber. Soot and NO formation depend on temperature. The temperature of diesel-water blend is lower than diesel alone, due to which Soot increases but NO decreases in blend as shown in Fig 8 & 9. The improved combustion process results to high efficiency and power output by diesel-water blend at 6°BTDC in Fig 10. However, Fig 23 (a & b) shows that, torque and brake specific fuel consumption is high for diesel-water blend than diesel fuel at 6°BTDC.

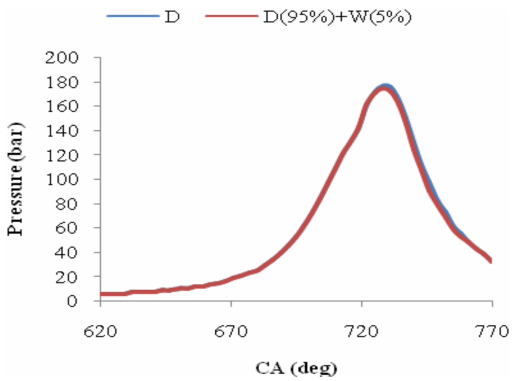


Fig 5. Pressure.

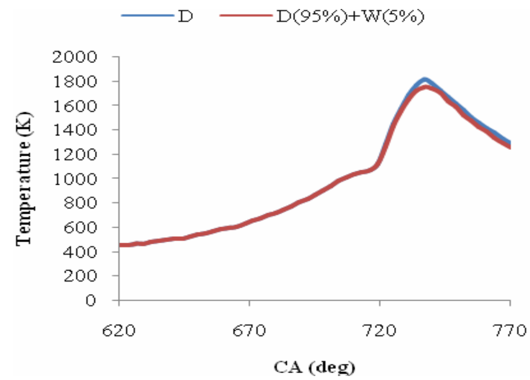


Fig 6. Temperature

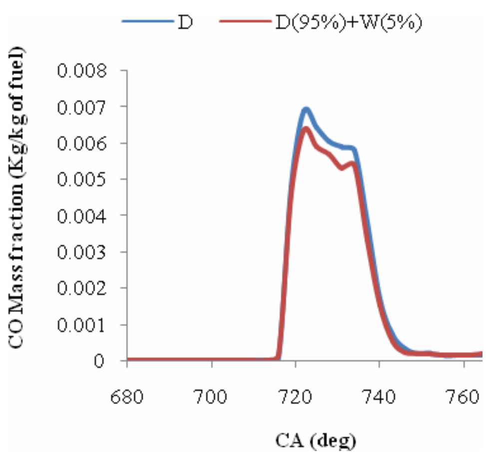


Fig 7. CO Mass fraction.

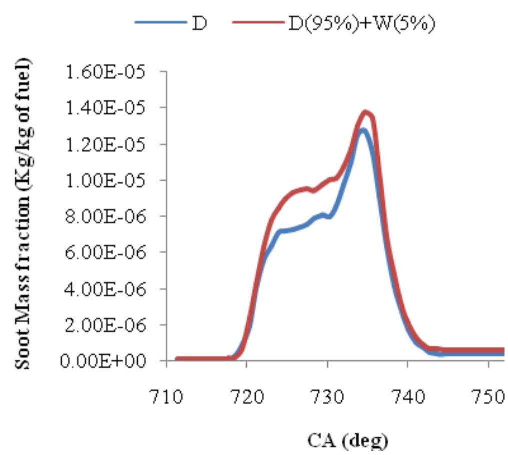


Fig 8. SOOT Mass fraction.

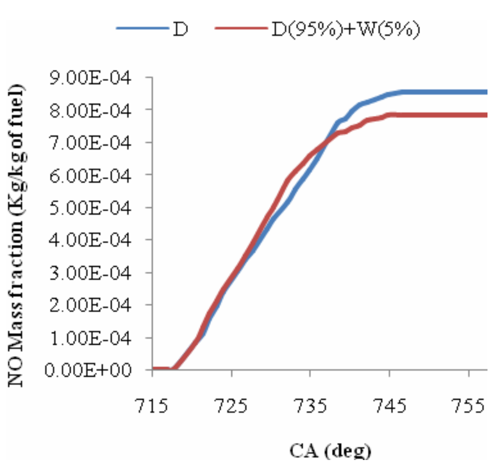


Fig 9. NO Mass fractions

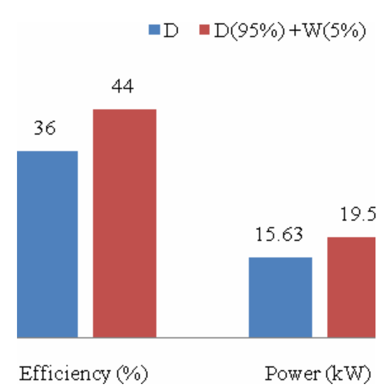


Fig 10. Efficiency and Power at 6°BTDC.

7.3.2. 10°BTDC

The shorter delay period is obtained, when injection timing is advanced from 6°BTDC to 10°BTDC. Fig 11 & 12 shows that, shorter delay period results to higher pressure and temperature of diesel and diesel-water blend at 10°BTDC. It is shown in Fig 13 that, CO mass fraction is improved due to proper mixing of blend with air. The temperature of diesel-water blend is low comparison to diesel fuel which tends to high soot and low NO formation as presented in Fig 14 & 15. The low temperature is reported due to micro explosion phenomena in diesel-water blend. Fig 16 revealed that, efficiency and power output is approximately same, while fuel is injecting with 10°BTDC injection timing. Fig 23 reports that, torque is almost same for both fuels whereas brake specific fuel consumption is high for diesel- water blend than diesel fuel at 10°BTDC.

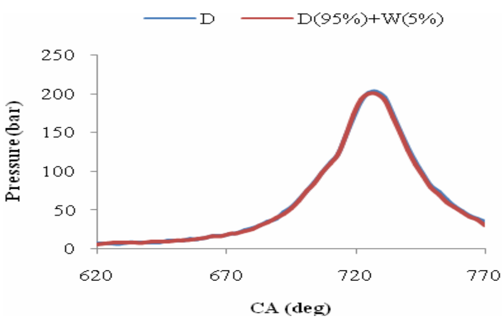


Fig 11. Pressure.

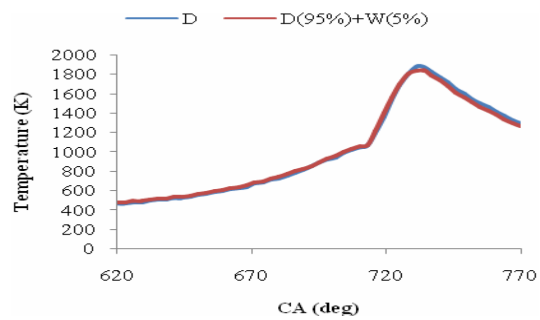


Fig 12. Temperature.

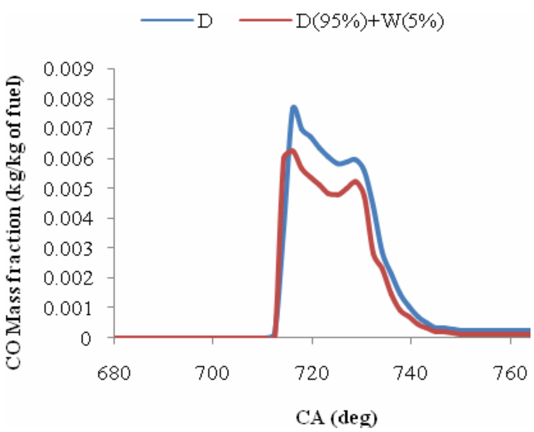


Fig 13. CO Mass fraction.

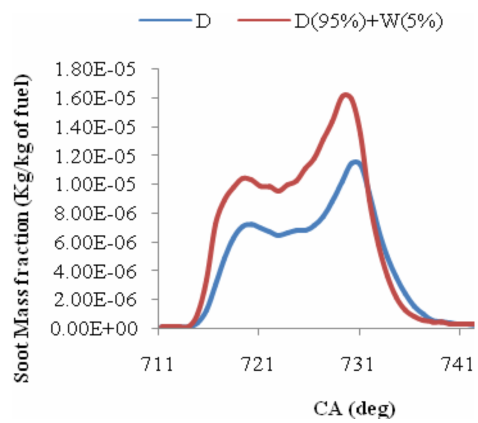


Fig 14. SOOT Mass fraction

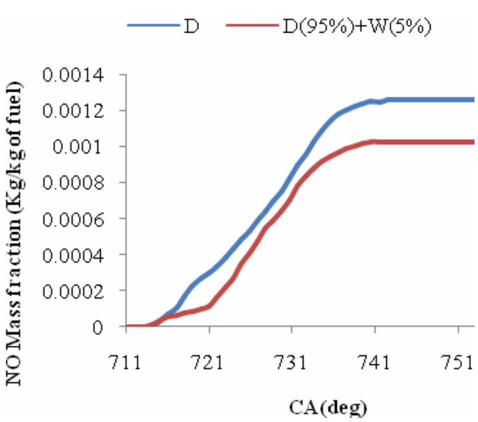


Fig 15. NO Mass fractions.

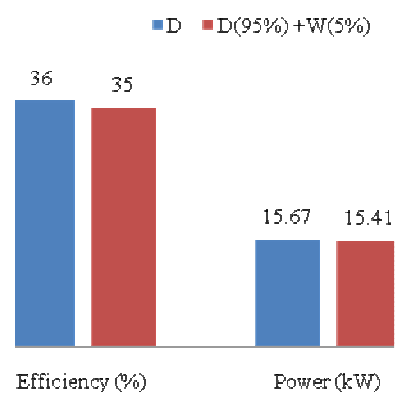


Fig 16. Efficiency and Power at 10°BTDC.

7.3.3. 14°BTDC

The advancement of injection timing from 10°BTDC to 14°BTDC is not showing any major change. It can be seen in Fig 17 that, pressure is higher at 14°BTDC than other injection timings; however it is almost same for both fuels. Furthermore, the graphical trend of temperature, CO mass fraction and NO mass fraction is low by using blend than diesel alone (Fig 18, 19 & 21). Soot formation is high due to low in cylinder temperature with blends as shown in Fig 20. Fig 22 shows that, efficiency and power output is approximately same at 14°BTDC injection timing. However, Fig 23 revealed that, torque and brake specific fuel consumption is high for diesel- water blend than diesel fuel at 6°BTDC. The trend of torque and brake specific fuel consumption is same as observed for 10°BTDC.

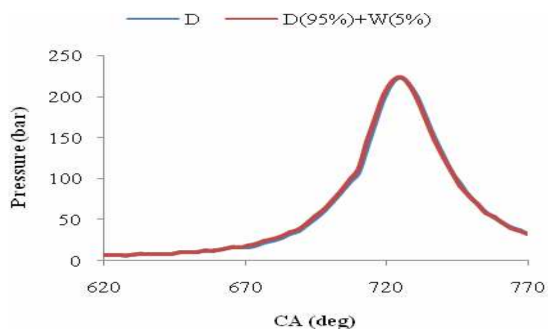


Fig 17. Pressure.

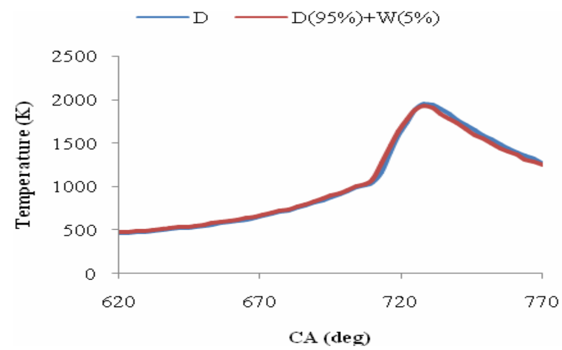


Fig 18. Temperature.

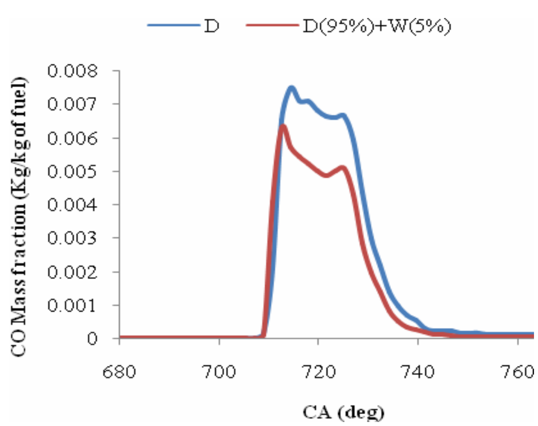


Fig 19. CO Mass fraction.

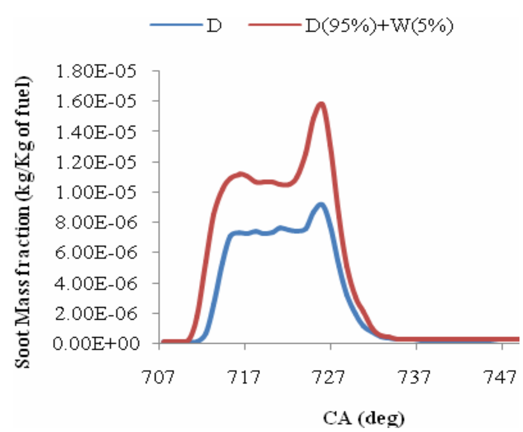


Fig 20. SOOT Mass fraction

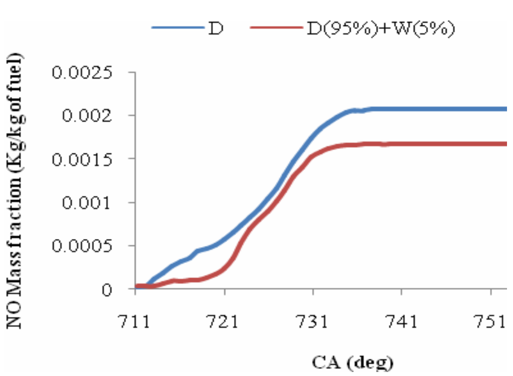


Fig 21. NO Mass fractions.

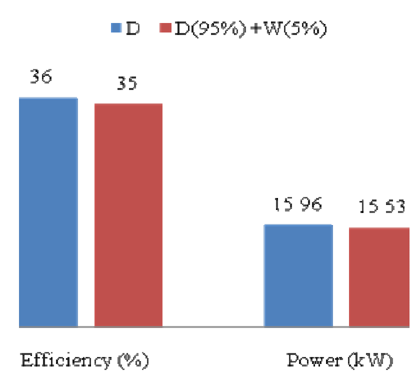
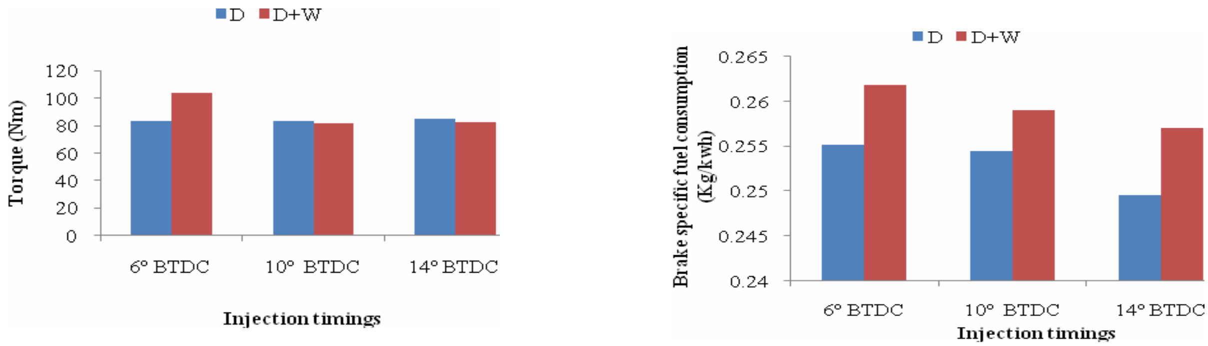


Fig 22. Efficiency and Power at 14°BTDC.



(a) (b)

Fig 23. (a) Torque and (b) brake specific fuel consumptions at different injection timings

8. Conclusion

It is clear from above discussion that diesel - water blend exhibits the best performance in terms of emission levels as well as engine power and efficiencies are concerned. The need of emission reduction forces researchers to find a way to introduce diesel - water blend to reduce emission as well as cost of power production. The diesel and diesel - water blend is simulated by AVL FIRE software. CO, NO and Soot formation and performance parameters are examined to find proper way to reduce emissions. Following results can be concluded from analysis.

a. The mass fractions of CO and NO decreases when engine fueled with diesel and diesel-water blend at 6°BTDC injection timing. The CO mass fraction and NO mass fraction are decreased from 0.0070 to 0.0065 and from 0.00086 to 0.00080 respectively, but soot formation increases simultaneously, from $1.25e^{-5}$ to $1.4e^{-5}$ due temperature dependent nature.

b. At 10°BTDC and 14°BTDC injection timing, same reduction pattern is observed for CO and NO mass fraction. It can be seen from Fig 13 that, CO mass fraction is reduced from 0.0068 to 0.0060; whereas, NO mass fraction is reduced from 0.0012 to 0.0010 and soot formation increases from 1.8e-5 to $6.5e^{-5}$ for 10°BTDC. In case of 14°BTDC injection timing, CO mass fraction is decreased from 0.0075 to 0.006 and NO mass fraction is decreased from 0.0021 to 0.0016 but soot formation is increased from $9e^{-6}$ to $1.6e^{-5}$.

c. The comparison of performance parameter in terms of power and efficiency shows that diesel-water blend has much more efficiency (44%) and power (19.5kW) than diesel fuel while conducting simulation on 6°BTDC injection timing. Furthermore, diesel-water blend is also dominating in terms of torque over diesel fuel and it is high at 6°BTDC than other injection timings, whereas lowest BSFC is achieved for both fuels at 14°BTDC injection timing.

So, it is concluded that, diesel-water blend with 6°BTDC injection timing is better to use in place of pure diesel for rectangular bowl piston geometry. Amount of reduced emissions are good in 6°BTDC injection timing than other injection timings. Above simulation analysis of rectangular bowl geometry can be useful in future to reduce emissions from engines.

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