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Investigation of injection timing and different fuels on diesel engine performance and emissions

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Abstract

Start of fuel injection and fuel type are two important factors affecting engine performance and exhaust emissions in internal combustion engines. In the present study, a one-dimensional computational fluid dynamics solution with GT-Power software is used to simulate a six-cylinder diesel engine to study the performance and exhaust emissions with different injection timing and alternative fuels. Starting the fuel injection was from 10 °CA BTDC to the TDC with an interval between two units and from alternative fuel bases (diesel), including methanol, ethanol, diesel, and ethanol compounds, biodiesel and decane was used. To validate the model, a comparison is made between simulation data and experimental data (including torque and power) showing the validation error is less than 6.12% and indicating the software model validation. Also, the modeling results show that decane fuel has higher brake power and brake torque of more than 6.10 % while fuel is injected at 10 °CA BTDC compared to the base fuel, and illustrates a reduction of 5.75 % in specific fuel consumption due to producing higher power. In addition, with the advance of injection timing compared to baseline, the amount of CO and HC in biodiesel fuel reduces to 83.88% and 64.87%, respectively, and the lowest NOX emission with the retardation of starting injection, to decane fuel is awarded. In general, the results show that decane fuel could be a good alternative to diesel fuel in diesel engines when it starts fuel injection at 10 °CA BTDC.

Nomenclature

BTDC	Before top-dead-center
CA	Crank angle
BSFC	Brake specific fuel consumption
BSEC	Brake specific energy consumption
CI	Compression ignition
PCCI	Premixed charge compression ignition
TDC	Top-dead-center
HC	Hydrocarbons
CO	Carbon monoxide
CO_2	Carbon dioxide
NO_X	Nitrogen dioxide
PM	Particulate matter
H_2	Hydrogen
CNG	Compressed natural gas

1. Introduction

Protecting energy conservation and environmental protection are two important

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issues that have been addressed in recent years [1]. Thus, the number of diesel engines continuously increases each year due to its high efficiency, fuel economy, and low greenhouse gas emissions. Diesel engines prefer more than spark-ignition engines to almost all heavy applications. Therefore, the global demand for diesel fuel increases every year [2]. However, diesel engines, in contrast to spark-ignition engines, have more harmful emissions than NO_X and PM particles [3]. The emission of pollutant gases from the combustion process can have very damaging effects on the environment, humans, and the climate. Renewable fuels have a lot to gain because of their low share in carbon cycle, sustainability, and greenhouse gas emissions [4]. As fossil fuel sources are gradually decreasing, and on the other hand, they cause air pollution and global warming, alternative fuels sources for fossil fuels are essential. One of the alternative fuels is biodiesel and its combination with diesel fuel. In a study [5] on the effects of biodiesel on performance, the emissions and combustion characteristics of a direct injection diesel engine were studied. The results show that the use of biodiesel causes lower emissions of soot up to 60% and higher fuel consumption of up to 11%, compared with diesel fuel. CO emissions in B5 and B100 fuels were 9% and 32% lower than those of diesel fuel, respectively. The special fuel consumption of the biodiesel was 8.5% higher than that of diesel fuel at maximum torque. Yang et al. [6] studied the effects of adding H_2 on combustion and emissions of exhaust gases in a diesel engine. In this research, the AVL Fire software was used for numerical modeling of the engine. The results showed that the cylinder pressure and heat release rate first increased and after the addition of H_2 decreased, and the NO emission increased as PM release decreased. In another study, the impact of butanol additive on the performance and emission of Australian Macadamia biodiesel was discussed in the diesel engine [7]. The results showed that the addition of butanol to diesel-biodiesel compounds reduced the braking force (BP) and increased BSFC and BSEC energy consumption. The study revealed that the addition of butanol to diesel-biodiesel compounds reduces the production of carbon monoxide (CO), nitrogen oxide (NO_X), and PM release. In a study, the combustion simulation and features of the n-butanol/diesel fuel exhaust gases in a diesel engine were investigated [8]. The results show that the maximum combustion pressure and the temperature gradually increase, and the heat release accumulated by the addition of n-butanol decreases slightly. Special fuel consumption increases but thermal efficiency decreases. Soot mass significantly reduces, and the NOX mass initially reduces and then increases by adding n-butanol. Gharehghani et al. [9], Investigated the effect of biodiesel on fish oil waste on the characteristics of diesel combustion and emission of pollutants. The results show that fish oil biodiesel increases the pressure in cylinder and reduces the heat release rate compared to conventional diesel fuel. CO and HC concentrations for biodiesel and its mixture decrease by (27-5.2%) and (11-70%), respectively, and CO_2 and NO_X levels increase by 7.2% and 1.8%, respectively. In another study, the empirical modeling and performance of the engine and emitted biodiesel fuel from the Australian Beauty Leaf tree were investigated. Combustion engines were modeled using the AVL Fire Computational Fluid Software to predict the performance and emissions of biodiesel and diesel fuel. Experimental results showed that biodiesel fuel (B10) improves performance and significantly reduces engine exhaust emissions [10]. One of the factors having an important role today in engine performance and improved combustion and emissions is focusing on the fuel injection system and its various variables. Sayin and Canakci [4] studied the effect of injection timing on engine performance and exhaust emission of a dual diesel engine. The results showed that the NO_X and CO_2 levels decrease with the retardation of injection timing compared with the initial injection timing, and the release of HC and CO increase. On the other hand, with the advance in injection timing, the emission of HC and CO decreases, and NO_X and CO_2 emissions increase. In the study, the effect of fuel injection timing and inlet pressure on the performance of compression ignition diesel engines was investigated. The results showed that the retention time of injection increase pressure, temperature, heat release rate, cumulative heat emission, and NO_X emission. Also, the amount of soot emissions decreases with the advance in injection start time [11]. In a research study, the effect of injection timing and injection pressure on the performance and emissions of a conventional diesel engine produced by various concentrations of fish oil biodiesel blends were examined [12]. When injection timing advances, the brake power increases initially to reach the maximum value, and then decreases slightly for each type of fuel tested.

Agarwal et al. [13] studied the effect of fuel injection timing and pressure on combustion, emissions, and performance characteristics of a single-cylinder diesel engine. With the advance in injection timing, the brake thermal efficiency increases, while the brake specific fuel consumption and the exhaust gases temperature significantly reduce. The results showed that the amount of CO_2 and HC emissions decreases and NO_X emissions increase significantly with the advanced injection timing. In another study, the impact of the injection timing on the CI engine with algae oil fuel was tagged with the Taguchi technique [14]. Experimental results showed that with the advanced in injection timing, the brake thermal efficiency increases by 5.7%, and the CO and HC release decreases by 81.25 and 30%, respectively. The test of retardation of fuel injection timing showed that NO_X emissions drop by 28% at high load. As injection timing advances in high load conditions, the gas exhaust temperature and soot emissions decrease by 7.8% and 26.39%, respectively. In the research, the effect of fuel injection parameters on combustion stability and emission of PCCI engines was investigated. As the fuel injection timing advances, the HC and NO_X release slightly and soot decreases [15]. Hosmath et al. [16] studied the impact of compression ratios, CNG flow rates, and injection timings on engine performance with Hong oil methyl and CNG fuel. The results showed that increasing the compression ratio with the time of advanced injection and low CNG flow contributes to improving the performance of the diesel engine with the Hong Oil Methyl and CNG fuel during the term of increasing the thermal efficiency and reducing emissions (smoke, carbon monoxide, hydrocarbons) except nitrogen oxide (NO_X). In a study, the effect of methanol to diesel ratios and fuel injection timings on combustion, performance and emissions of a diesel engine was investigated [17]. The maximum temperature inside the cylinder reduced by the rejection of the fuel injection timing and by retarding the fuel injection timing, the HC release initially increased and then decreased, while CO emissions always increased.

In the present study, the effects of injection timing are studied on alternative fuels CI engine to obtain performance, emission, and combustion characteristics at full throttle condition. Operation parameters like brake power, brake specific fuel consumption, exhaust emissions such as CO, CO_2, and NO_x are studied and compared.

2. Methodology

The research engine in this study is a six-cylinder direct injection diesel 6068HF275 made by John Deere USA (Fig. 1). The characteristics of this engine are shown in Table 1.

2.1. Simulation Setup

In this study, the six-cylinder engine is simulated by GT-Power software to study the effect of injection timing and fuel type on performance and engine exhaust emissions. The GT-Power software is part of the GT-Suite software and gamma-technology company that emulates the engine and its accessories. Numerical calculations in this software are based on solving one-dimensional fluid dynamics equations including phenomena related to flow motion, heat transfer in the pipe, and other components of the engine. To model the engine, all parts of the engine are first introduced as a real six-cylinder engine, and then the required data are entered according to the actual engine conditions at the atmospheric pressure of 1 atm. In addition, the injected mass at per cycle engine of 96 mg and the length of the injection rate of 20 degrees is selected. The engine speed is changed from 800 to 2400 rpm, and considered injection advance at different speeds.

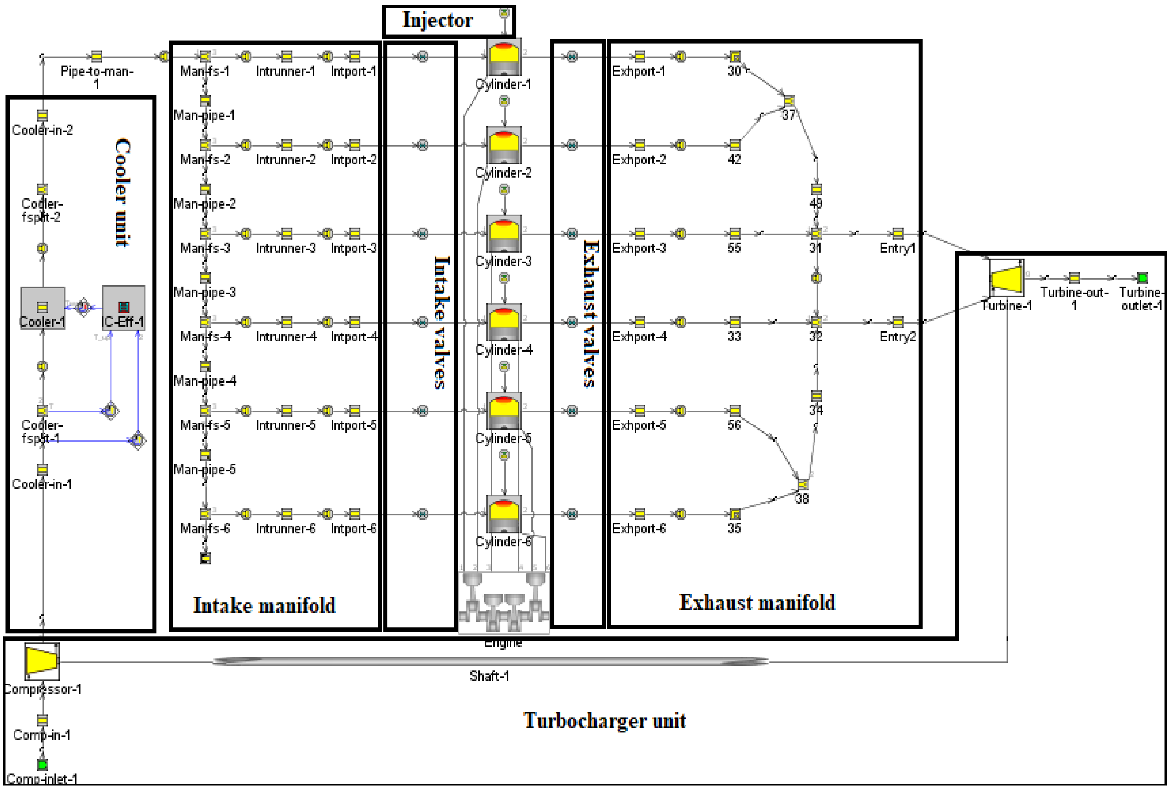

Fig. 1. Computational model of a six-cylinder, direct-injection, compression-ignition engine.

Table 1. Technical specifications of diesel engine John Deere 6068HF275.

Engine characteristics	Value
Number of cylinders	6
Displacement	6.8 L
Cylinder bore	106 mm
Stroke	127 mm
Connecting rod length	270 mm
Standard injection timing	4 BTDC
Compression ratio	17
Maximum torque	740-930 N.m @ 1400 rpm
Maximum power	129-187 kW @ 2000-2400 rpm

2.2. Modeling fluid flow in pipes

2.2.1. General governing equations

The flow model involves the solution of the Navier-Stokes equations, namely the conservation of continuity, momentum and energy equations. Which are calculated according to formulas 1 to 3. In the GT-Power software, these equations are solved in a one-dimensional fashion. This means that all the equations are in the direction of the averaging. In this study, explicit solving was used to solve the equations, in which base variables are explicitly solved in mass flow, density, and internal energy. In explicit solving, the system is divided into small volumes, in which all the splitters are subdivided into a sub volume and all tubes of one volume or more. The scalar variables (pressure, temperature, density, internal energy, enthalpy, etc.) are assumed to be uniform on the boundary of each of the underlying volumes. Vector variables (mass flux, velocity, mass fraction flux, etc.) are calculated for each boundary.

$$\frac{dm}{dt} = \sum\nolimits_{boundaries} \dot{m} \tag{1}$$

$$\frac{dme}{dt} = -P\frac{dV}{dt} + \sum_{boundaries} \dot{m}H - hA_s\left(T_{fluid} - T_{wall}\right) \tag{2}$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{voundaries} \dot{m}u - 4C_f \frac{\rho u|u|}{2}\frac{dxA}{D}}{dx} - \frac{C_p\left(\frac{1}{2}\rho u|u|\right)A}{dx} \tag{3}$$

In which $\dot{m}$ is boundary mass flux into volume, m is mass of the volume, V is volume, P is pressure, ρ is density, A is cross-sectional flow area, A_s is heat transfer surface area, e is total specific internal energy, h is heat transfer coefficient, T_{fluid} is fluid temperature, T_{wall} is wall temperature, u is velocity at the boundary, C_f is Fanning friction factor, and C_p is specific heat.

2. 2. 2. Friction loss and surface roughness effect

To calculate the coefficient of friction in these components, in addition to the Fanning friction coefficient for a smooth flow, Colebrooke equations as an improved model for turbulent flow are used; therefore, a smooth flow according is obtained as follows:

$$C_f = \frac{16}{Re_D} \qquad (Re_D < 2000) \tag{4}$$

In which C_f is Fanning friction coefficient and Re_D is Renolds number. Also, C_f for the turbulent flow is as follows:

$$C_f = \frac{1}{4}\left(4.781 - \frac{(A-4.781)^2}{B-2A+4.781}\right)^{-2} \quad Re_D > 4000 \tag{5}$$

$$A = -2\log_{10}\left(\frac{\varepsilon/D}{3.7} + \frac{12}{Re_D}\right) \tag{6}$$

$$A = -2\log_{10}\left(\frac{\varepsilon/D}{3.7} + \frac{2.51A}{Re_D}\right) \tag{7}$$

2.2.3. Heat loss and surface roughness effect

The heat transfer from fluids inside of pipes and flow split to their walls is calculated using a heat transfer coefficient. The heat transfer coefficient is calculated at every time step from the fluid velocity, the thermo-physical properties, and the wall surface roughness. The heat transfer coefficient of smooth pipes is calculated using the Colburn analogy, which is presented in Eq. (6).

$$h_g = \frac{1}{2}C_f \rho U_{eff} C_p Pr^{-\frac{3}{2}} \tag{8}$$

where U_{eff} is effective velocity outside the boundary layer and Pr is Prandtl number.

The surface roughness attribute in pipe components can have a very strong influence on the heat transfer coefficient, especially for very rough surfaces such as cast iron or cast aluminum. In this case, first, the value of h is obtained from Eq. (8), then corrected with the help of Eq. (9).

$$h_{g,rough} = h_g\left(\frac{C_{f,rough}}{C_f}\right)^n \tag{9}$$

$$n = 0.68P_r^{0.215}$$

where $h_{g,rough}$ is the heat transfer coefficient of rough pipe and $C_{f,rough}$ is Fanning friction factor of rough pipe.

2.2. Fluid properties

In the process of engine simulation, six types of fuel are examined. The GT-Suite software library contains many types of fluids as well as their specifications and features, and fluids that are not available in this library can be manually included in this software. In addition to introducing incompressible fuel specifications, the software also needs to include fuel vapor characteristics, so that in the case of evaporation, fluid specifications are predictable. The fuels used in this study include diesel, ethanol, 10% ethanol and diesel, biodiesel from soybean oil, decane, and methanol, with some important properties of the fuels used in Table 2.

3. Model validation

Today, numerical simulation has progressed greatly due to lower costs and the speed of computing systems. Hence, mathematical algorithms are able to solve real problems.

To examine and compare reality and simulation, a validation model needs to confirm the compatibility of a numerical solution to reality. To do this, it must be ensured that the numerical method exactly solves the mathematical modeling equations. In the present study, the GT-Power software is used to solve these equations to solve problems based on the one-dimensional solution of computational fluid dynamics equations.

Table 2. Important properties of used fuels.

Fuel properties	Unit	Diesel	Ethanol	Methanol	Biodiesel	Decane	Diesel-Ethanol
Density	kg m^{-3}	830	785	792	890	727	825
Heat vaporization at 298K	Mj kg^{-1}	0.25	0.92	1.17	0.35	0.36	0.317
Oxygen atoms per molecule	…	0	1	1	34.39	0	0
Hydrogen atoms per molecule	…	23.6	6	4	2	22	21.84
Carbon atoms per molecule	…	13.5	2	1	18.82	10	12.35
Lower heating value	Mj kg^{-1}	43.25	27.73	21.11	37.11	44.62	41.7
Critical temperature	K	569.4	516	513	785.87	617.8	564.06
Critical pressure	bar	24.6	6.38	79.5	12.07	21.1	22.77
Boiling point	K	463.15	351.65	337.63	624.1	447.27	452
Ignition temperature	K	527.15	707.15	738.15	636.15	483.15	545.15
Kinematic viscosity@ 40˚C	mm^2 s^{-1}	2.2	1.32	0.58	5.63	2.36	2.11
Cetane number	…	52	7	5	60.1	76	47.5

3.1. Validation of the model using experimental and simulation results

The best and easiest way to validate simulation data is to compare it with experimental data. To confirm the results of a simulation, it is essential to achieve the highest similarity between simulation data and experimental data. One of the most important issues in the simulation environment is the correct modeling, which, if not done correctly, causes a validation error. By calculating the error rate, the data obtained from the lab can be compared with the simulation data, and one can see how different they are. The equation for calculating the error rate is as follows:

$$\text{percentage error} = \frac{|\text{simulation value} - \text{experimental value}|}{\text{experimental value}} \times 100$$

In this study, the experimental results obtained from the engine test are used to validate the model. The experimental results is used for validation purposes, specific fuel consumption and engine power. Fig. 2 shows a comparison of engine power versus engine speed between experimental and simulation data between. Fig. 3 shows the deformation of brake specific fuel consumption and power as a function of speed for validation between experimental results and calculated data from simulation. As shown in the figure, the error rate is very low (< 6.12%) for experimental and simulation data, depicted using the above equation. Thus, it is acceptable for the validation of software [18-20]. Hogg studies demonstrated that a 20% error in calculation has a good agreement in evaluation and validation between experiment and numerical models [21].

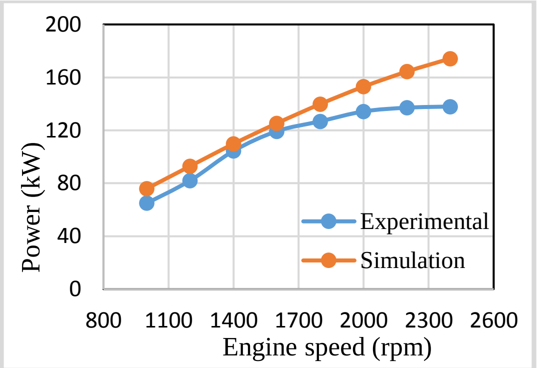

Fig. 2. Curve of engine power versus engine speed (comparison of experimental and simulation data) [22].

4. Results and discussion

4.1. Combustion characteristics

The variations in combustion delay and in-cylinder pressure in six different fuels, as a function of injection timing, at a steady speed of 1400 rpm, are shown in Figs. 4 and 5, respectively. Fig. 4 shows that the fuel injection timing from the TDC is lower, and the ignition delay increases.

When the injection takes place earlier, the pressure in the cylinder is insufficient to burn the fuel, resulting in a large amount of evaporated fuel being compressed during the combustion delay, and no combustion occurs leading to an increased ignition delay.

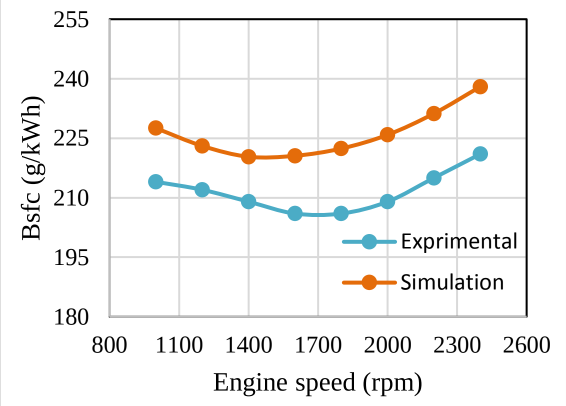

Fig. 3. Curve of brake specific fuel consumption versus engine speed (comparison of experimental and simulation data) [22].

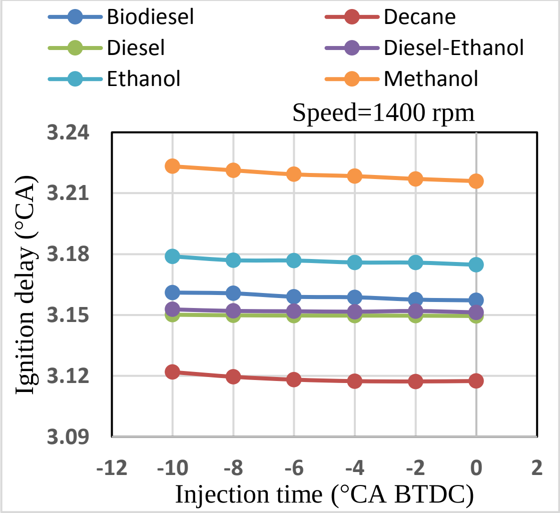

Fig. 4. Ignition delay at different injection timing.

It is known that with increased ignition temperature, combustion delays increase as a result. Given the fuel specifications in Table 2, among the fuels used in this engine, the highest ignition delay in 10 °CA BTDC with the methanol fuel is 2.33% higher than the base fuel. In contrast, the lowest ignition delay is awarded in TDC, with decane fuel which equals to 1.02% lower than the base fuel. When the fuel injection takes place later, the injection is done near the TDC, and in the compression course, the ignition delay is shortened, resulting in burning more fuel fractions in the diffusion of the combustion as a result of reducing the maximum cylinder pressure. Thus, according to Table 2, its lower heating and ignition temperature affect the pressure inside the cylinder. As shown in Fig. 5, among the various fuels, the highest pressure is from the decane fuel, and the lowest pressure is from the methanol fuel.

4. 2. Engine performance characteristics

The changes in the brake torque and brake power are shown in Figs. 6 and 7, in six different fuels, as a function of the time of spraying fuel at a constant rotational speed of 1400 rpm, respectively. The results show that advanced fuel injection timing increases the brake power and brake torque. Among the fuels used in this engine, the maximum brake power and brake torque by fuel injection at 10 °CA BTDC to the decane fuel with 6.10 % higher than baseline and low brake power and brake torque with fuel injection at TDC with 53.78% lower than baseline, is attributed to methanol fuel. At the speed of 1400 rpm, the maximum torque, according to the above results, is shown to increase the brake power of an engine with a certain ratio, since after this period, the increase in power and torque has no definite relationship and may be due to reasons of reducing the volume efficiency increases or decreases. Biodiesel is also rich in fuel with a higher oxygen content and greater combustion [23], but due to its lower thermal value, it has less brake power and torque than decane, diesel, diesel and ethanol mixture.

BSFC is defined as the ratio of fuel consumption to brake power. As shown in Fig. 8, with the retardation of the fuel injection timing, the brake specific fuel consumption is increased, which is due to the higher power generation of the fuel used in the engine [4, 24]. A fuel with a lower thermal value should be injected with more fuel to the engine, which will increase the amount of special fuel consumption.

In this study, decane fuel with a fuel injection of 10 °CA BTDC, with a reduction of 5.75%, has the lowest specific fuel consumption among other fuels. The temperature of the exhaust gases is lower under the conditions of advanced fuel injection timing. Because the primary heat release near the TDC occurs in a power curve and the earlier onset of combustion with the TDC, it increases the chance of full combustion, which provides enough time for hot gases to expand and cool before opening the exhaust valves.

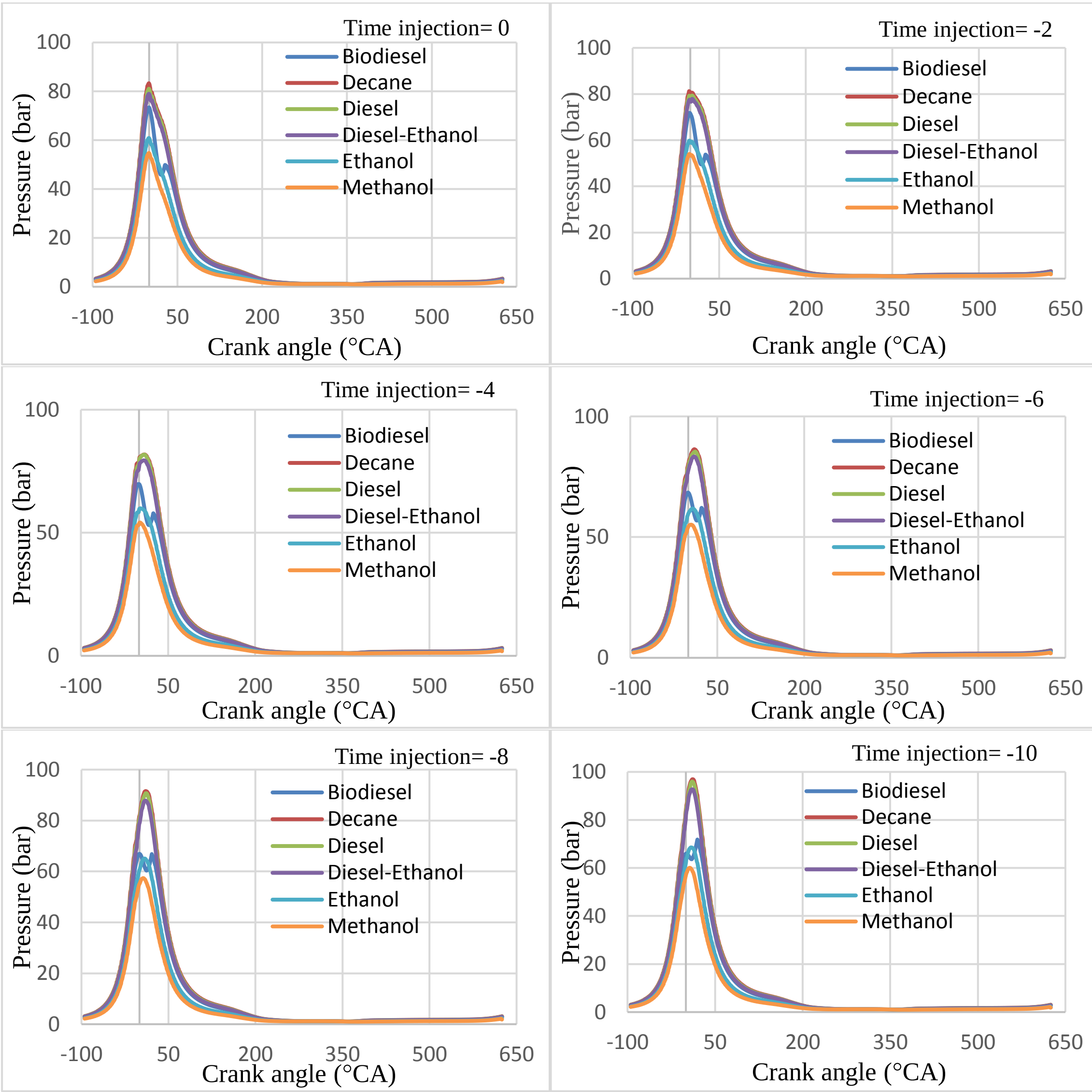

Fig. 5. Pressure in cylinder at different injection timing.

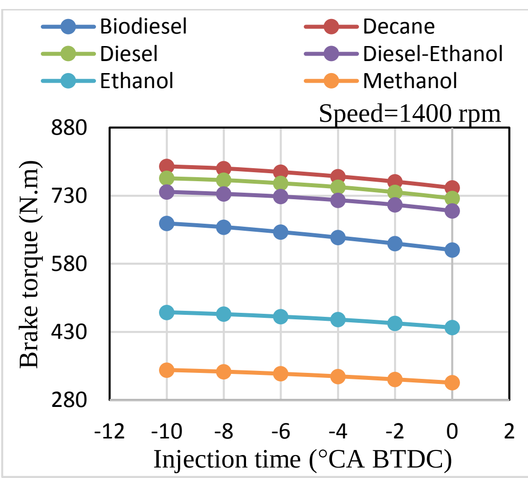

Fig. 6. Brake torque at different injection timing.

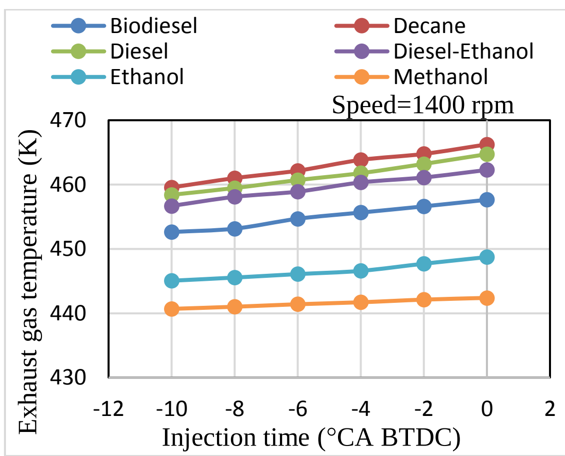

Fig. 7. Exhaust gas temperature at different injection timing.

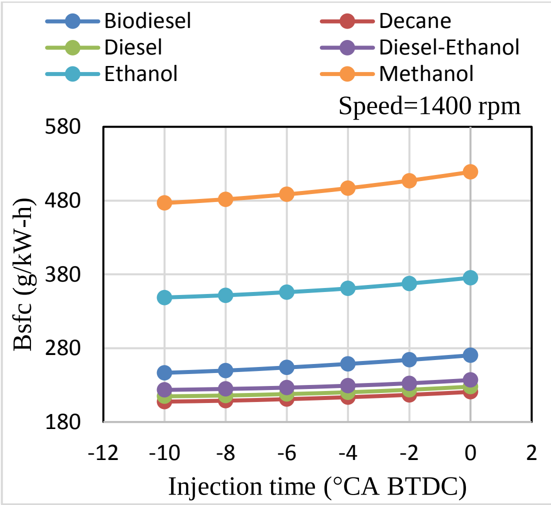

Fig. 8. Brake specific fuel consumption at different injection timing.

This increases the consumption of heat, cool down the combustion products, and thus reduces the temperature of the gas exhaust. As shown in Fig. 8, diesel fuel with injection fuel in TDC, with a 0.96% reduction, has the highest gas exhaust temperature and methanol fuel by injection fuel in 10 °CA BTDC, with a reduction of 4.56% the lowest gas exhaust temperature. The main reason that ethanol and methanol have the lowest gas exhaust temperature is the heat of vaporization [25]. Methanol and ethanol have a higher heat of vaporization than other fuels, which results in the inlet process and lower compression stage compared to other fuels, and therefore the lower temperature of the combustion chamber and also the exhaust.

4.3. Emissions characteristics

CO is a product of combustion of hydrocarbon fuels, so the emission is due to incomplete combustion. The diffusion of CO from the engine largely depends on the fuel's properties, the availability of oxygen, the fuel mix with air, and temperature and turbulence inside the combustion chamber. Therefore, CO emission is heavily dependent on the ratio of air to fuel. According to Fig. 9, CO emission increases with fuel injection retardation. Advanced injection timing improves air and fuel mixture due to the increased time available for mixing the process and increasing the oxidation process between carbon and oxygen molecules. Also, according to Table 2 and observation of the oxygen atom of biodiesel, ethanol and methanol fuels, it can be concluded that having an oxygen atom in each fuel molecule causes enough oxygen to react with carbon to convert it into CO_2, and due to complete combustion, the CO and HC are reduced. The release of CO in biodiesel fuel with fuel injection at 10 °CA BTDC is 83.89 % below the base. Fig. 9 also shows that CO emission in diesel fuel is higher than other fuels.

With reference to Fig. 10, with fuel injection retardation, HC emission increases and all fuels have lower HC emissions than diesel fuel. According to Fig. 10, HC emission in biodiesel fuel with fuel injection at 10 °CA BTDC is 64.87% lower than baseline. The reason that all fuels have lower HC emissions than diesel fuel can be due to long-term combustion delays, as more fuel is accumulated in the combustion chamber, it may result in higher emissions of hydrocarbons [26].

CO_2 is naturally occurring in the atmosphere and is a normal combustion product. Ideally, combustion of hydrocarbon fuels should only produce carbon dioxide and water (H_2O). With regard to Fig. 11 and the advanced in fuel injection timing, CO_2 emissions are reduced, and CO_2 emission increases in all fuels compared to baseline. Diesel fuel with the fuel injection at 10 °CA BTDC has the lowest amount of CO_2 among the fuels used.

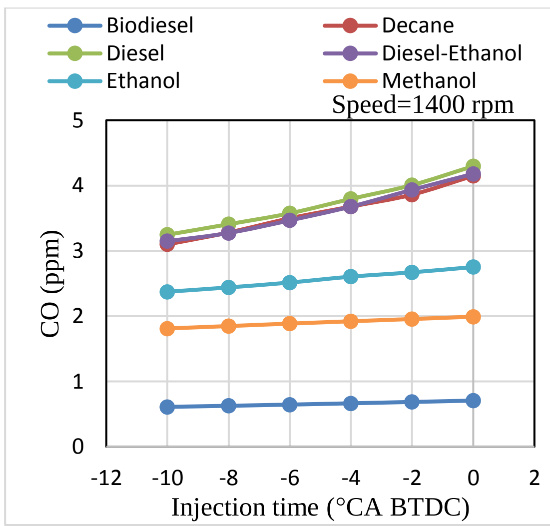

Fig. 9. CO emission at different injection timing.

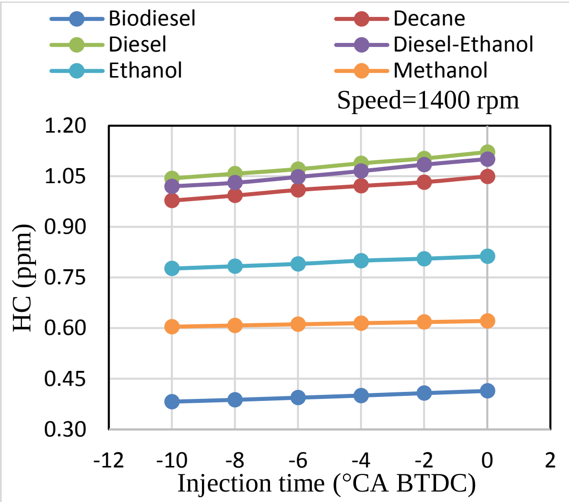

Fig. 10. HC emission at different injection timing.

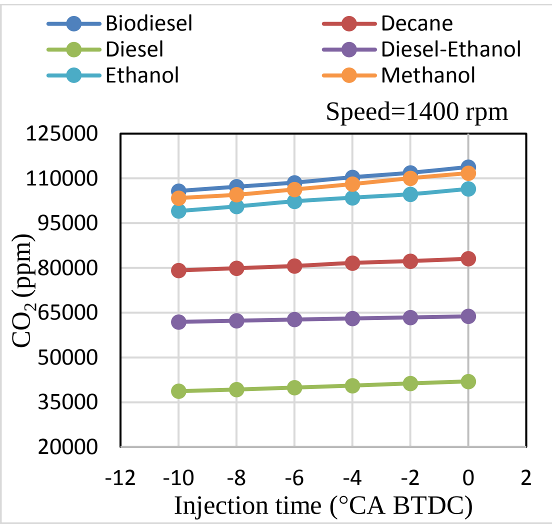

Fig. 11. CO_2 emission at different injection timing.

The greater amount of CO_2 represents the complete combustion of fuel in the combustion chamber. The CO_2 emission is not too harmful to humans, but it increases the potential for ozone depletion and global warming [27].

Most of the problems caused by compression ignition engines are related to NO_X emissions. The fuel thermal mechanism, combustion temperature, oxygen content, and the long-term availability of fuel gas are the most important factors in the formation of NO_X. With regard to Fig. 12, with the fuel injection retardation, the amount of NO_X decreases, and this is due to longer ignition delay and more time to generate NO_X, which ultimately increases NO_X emissions. The viscosity and density of fuels have an impact on NO_X emissions, and it results in NO_X emissions due to larger droplets of fuel [28]. According to Fig. 12, the highest NO_X emissions from biodiesel fuel, which increase by 125.02% by fuel injection at 10 °CA BTDC is due to the high oxygen content of this fuel. The lowest NO_X emissions from fuel injection at TDC with 11.99% decrease compared to baseline is decane, due to low fuel density compared to other fuels.

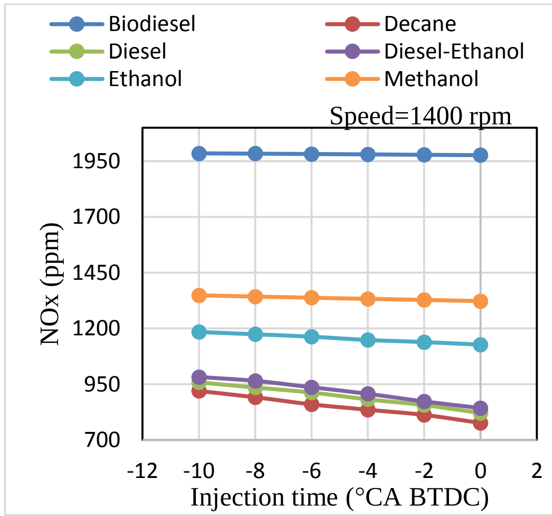

Fig. 12. NO_X emission at different injection timing.

5. Conclusions

In this research, the performance and emissions of a six-cylinder engine using different fuels and fuel injection timing are reviewed, which include the following results:

1. Power and torque values increase by advanced in injection timing and fuel type changes, use of decane fuel with start of injection of 10 °CA BTDC, has about 6.10 % more power and torque compared to base fuel.
2. The results show that with fuel injection retardation, specific fuel consumption increases, and decane fuel with fuel injection at 10 °CA BTDC has 5.75% reduction in specific fuel consumption compared to the base fuel. In addition, the lowest specific fuel consumption obtains as compared to other fuels used.
3. The results show that with retarding of injection timing, the contaminants of CO, CO_2, and HC increase and the NO_X pollutant drops. The emission of CO and HC in biodiesel fuel by

fuel injection at 10 °CA BTDC is the highest, and their amounts are 83.88 and 64.87%, respectively, compared to baseline. The decane fuel has lower CO and HC emissions than those of diesel.

4. Other results show that fuels with high viscosity and density exhibit more NO_X production; hence, the lowest amount of NO_X relates to decane fuel. Also, the highest NO_X emission from biodiesel fuel is at 10 °CA BTDC, and with a fuel injection retardation, NO_X reduces.

5. The results show that in total, decane fuel with fuel injection at 10 °CA BTDC has the best functional and pollutant characteristics among the six fuel used in this study. Therefore, this fuel can be the best alternative for diesel fuel.

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