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Analyzing the Effect of Fuel Injection Timing and Injection Duration on Performance and Emissions in Diesel Engines

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Abstract

Fossil-based fuels have been decreasing all over the world and the energy requirements of power generation systems are increasing. Diesel engines are one of the most common and efficient power generation systems. However, national and international organizations impose various restrictions to reduce emissions from diesel engines. Reducing emissions and improving performance in diesel engines are one of the most common areas of work. Improvement of performance and emission parameters is possible with changes in alternative fuels, engine geometry and fuel injection systems.

In this study, the effects of different injection timing and injection duration on diesel engine performance and exhaust gas emissions have been investigated with computational fluid dynamics (CFD) method. The numerical study has been carried out on a 4-stroke, single cylinder test engine and the results have been compared with the experimental results obtained from the literature. As a result of the study, combustion chamber speed profiles, fuel mass change, temperature, pressure, chemical heat release rate has been determined for the standard operating conditions. Parametric study has been performed for different injection timing and injection duration of engine. For each case, performance and emissions parameters have been determined. Analysis results has compared, and the most suitable injection parameters has been introduced.

Keywords: *Diesel Engine, Computational Fluid Dynamics (CFD), Injection Timing, Combustion.*

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1. Introduction

Diesel engines are widely used because of high reliability and highly efficient energy conversion. The use of fossil fuels in diesel engines has also led to the creation of undesirable harmful emissions. These undesirable harmful emissions have caused acid rain, global warming and many environmental problems [1–3]. Many countries and international organizations have made mandatory restrictions to reduce emissions such as Carbon Monoxide (CO), Carbon Dioxide (CO_2), Nitrogen Oxides (NO_x), Hydrocarbon (HC) and Particular Matter (PM) from diesel engines [4–6]. In the scope of MARPOL Annex 6, especially in marine diesel engines, restrictions have been applied by International Maritime Organization (IMO) to reduce emissions caused by greenhouse gas effects.

Restrictions on NO_x emissions have been determined as 17, 14.4 and 3.4 grams per kWh on ship diesel engines on January 1, 2000, 2011 and 2016, respectively. In diesel engines, NO_x emissions can be reduced with engine operating parameters, engine design parameters or after treatment systems. Engine operating parameters cause a decrease in emissions as well as performance. The injection timing is an operation parameter that directly affects engine performance and emissions. Injection timing varies based on the fuel specification, inlet air pressure and temperature, compression ratio, injection systems, engine speed and combustion chamber design [7]. Many studies have been carried out demonstrating the effect of the fuel injection timing on the diesel engine. Hagos et al. [8] have experimentally investigated the effects of start of injection timing (SOI) on the performance, combustion and emissions on the SI engine fueled with a hydrogen-rich synthesis gas. Fan et al. [9] have examined the effects of injection timing and nozzle angle on various blends formation and combustion performance in 3D model. The study has been conducted for 25 different cases. Wang et al. [10] have numerically analyzed the effect of ignition position and fuel injection timing on the engine performance and combustion in a direct-injection natural gas engine. Pham et al. [11] have examined both experimentally and numerically the effects of variable fish oil-biodiesel blends on the common-rail diesel engine performance and emission characteristics at different injection timing and injection pressure. How et al. [12] have studied the effects of injection timing and split injection strategies on emission and combustion characteristics. This study deals with parametrically that relation with variable start of injection timing and multiple injection schema for B20 and B50 biodiesel blends. Wamankar et al. [13] have analyzed the effect of injection timing on a DI diesel engine with synthetic fuel blends. It has been observed that the early injection timing has increased the cylinder pressure and engine performance. Ashok et al. [14], Gong et al. [15], Natarajan et al. [16] and Deep et al. [17] have investigated the effects of injection timing, injecting pressure and alternative fuel blends on diesel engine.

This study numerically examines the effects of injection timing and injection duration on the diesel engine performance, emission and combustion characteristics through computational fluid dynamics (CFD). In the previous studies, generally while the parameters such as alternative fuels, injection angle and injection pressure have been examined, in this study injection duration and injection timing have been investigated as a combination. Computations of fluid mechanics in internal combustion engines put a strain on including parameters such as spray dynamics, chemical reactions and turbulence [18]. CFD method, which is widely used in modeling of heat-flow problems in recently, is an alternative to experimental studies in terms of time and cost [19]. Accurate emissions and engine performance characteristic can be predicted through appropriate chemical reduction mechanisms in combustion processes. Numerical results obtained by this study has been compared with the experimental data from the literature.

2. Materials and Methods

2.1. Numerical Investigation

In this study, a single cylinder, 4 stroke diesel engine is modeled in 3D to obtain chemical and physical properties. With 3D-CFD model, all the processes that occur during combustion have been examined [20]. The numerical model has been developed by means of commercial software Ansys-FORTE [21]. The numerical study has been performed for the operating conditions of the diesel engine in Table 1 at a rotation speed of 2200 rpm and a torque of 16 Nm [22]. The aim of the numerical analysis is to enable the investigation of diesel engine performance and emissions parameter in different injection durations and injection timing conditions.

Chemical reactions and thermodynamic properties should be defined to determine combustion products and performance characteristics in diesel engines [23]. N-tetradecane reduction mechanism developed by the University of Wisconsin Engine Research Center (ERC) has been used. The reduction mechanism consists of 35 species and 76 reactions [24]. The diesel fuel has been selected as tetradecane ($C_{14}H_{30}$) to analyses the fuel atomization, vaporization and the mixing fuel with air. It is very important analyze that the combustion process in the diesel engines is controlled by the fuel spray dynamics. Choosing the right spray model allows you to model the combustion process correctly. Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) and gas-jet model spray breakup models have been used for the analysis [25]. In internal combustion engines, air is compressed in the combustion chamber and has high turbulence. The k-ε Re-Normalization Group (RNG) turbulence model using Reynolds Averaged Navier-Stokes equations (RANS) has been used in the analysis [25]. 3D Mesh structure has been carried out engine combustion chamber as domain. The computational model described in Figure 1 with a 90-degree sector mesh consists of nozzle, cylinder head, liner and piston parts.

The numerical analysis must have mesh independency to improve the accuracy of the CFD simulations. The mesh independent solutions are entirely dependent on the mesh selected to resolve the fluid flow and the turbulence model that is chosen to illustrate the physics of the problem. For this reason, the correct mesh structure must be applied the to demonstrate reliability of the analysis. In this study, 117840, 202095 and 320500 element numbers have been analyzed.

Table 1. *Basic Engine Parameters*

Parameter	Units	Value
Basic Engine Model	-	Antor 6LD400
Number of Cylinders	-	1
Bore × Stroke	mm	86 x 68
Compression Ratio	-	18
Displacement	mm^3	395
Combustion Chamber Type	-	Mexican Hat
Diesel Fuel Injection Type	-	Direct Injection
Maximum Power	kW	5.4 (3000 rpm)

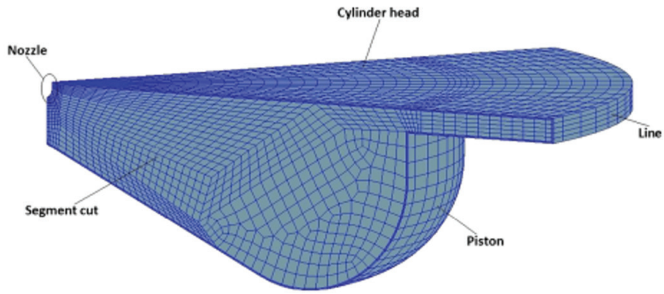


Figure 1. *Computational Domain*

The cylinder pressure for three different mesh number is shown in Figure 2. The results have demonstrated that the analysis is independent from computational grid.

In addition, Table 2 has shown that the boundary conditions and injection parameters of diesel engine. As nozzle diameter and cone angle has affect combustion performance, these parameters have been determined according to experimental test engine. The solutions of elementary gas-phase chemical kinetics have been enabled by chemkin solver which is used as combustion model. Combustion model has allowed identification thermodynamics properties, equation of state and chemical and chemical production rates [27].

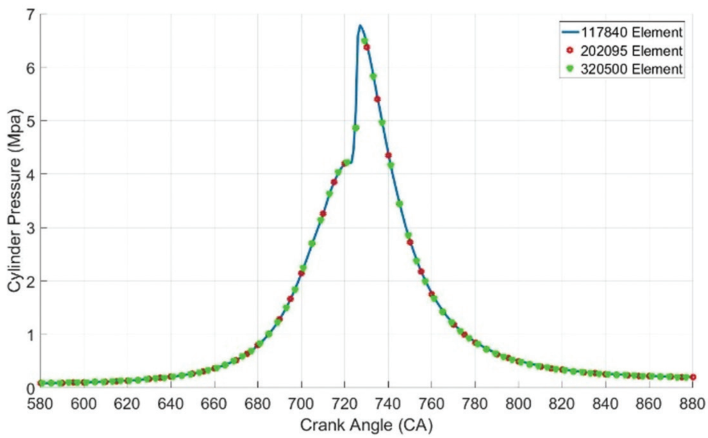


Figure 2. *Mesh Independence Test*

Table 2. *Boundary Condition*

Parameter	Units	Value
Inflow Droplet Temperature	K	368
Mean Cone Angle	-	20°
Nozzle Diameter	mm	0.24
Inlet Temperature	K	293.15
Inlet Pressure	bar	0.8
Combustion Model	-	Chemkin

The computational model is designed based on the opening and closing times of the intake and exhaust valves. The analysis includes compression and combustion processes between 565 crank angles (IVC) and 880 crank angles (EVO) [28]. Dimensionless representation of the intake, exhaust and piston movement has been shown as in Figure 3.

The numerical analysis of the diesel engine is usually carried out in the case where both the intake and exhaust valves are closed. For an engine calculation using a sector mesh, the work will be calculated for the compression stroke and the expansion stroke (from –180° ATDC to 180° ATDC). In such a case, the calculation is typically carried out from intake valve closure (IVC) to exhaust valve opening (EVO) [21]. The P-V curve between IVC and EVO is integrated directly using the following equation.

$$W_{from\ 540\ ATDC\ to\ IVC} = P_{ivc}\ x\ (V_{ivc} - V_{BDC}) \tag{1}$$

$$W_{from\ EVO\ to\ 900\ ATDC} = 0.5\ x\ (P_{ivc} - P_{EVO})\ x\ (V_{BDC} - V_{EVO}) \tag{2}$$

where, P_{IVC} the combustion chamber pressure when intake valve closed, P_{EVC} the combustion chamber pressure when exhaust valve open, V_{IVC} the combustion chamber volume when intake valve closed, V_{BDC} the combustion chamber volume when bottom dead center and V_{EVO} the combustion chamber volume when exhaust valve closed. Therefore, for the partial combustion chamber model, the total work and power expression is as follows.

$$W = W_{from\ 540\ ATDC\ to\ IVC} + W_{from\ IVC\ to\ EVO} + W_{from\ EVO\ to\ 900\ ATDC} \tag{3}$$

$$P = WxN/(60xn_{rev}) \tag{4}$$

where, N is engine speed (revolution/min), n_{rev} is number of revolutions per engine cycle [21].

2.1. Governing Equations

A gas-phase flow motion is governed using the Navier-Stokes equations. The continuity equation is based on conservation of mass. The rate of change in the volume of the arbitrarily selected unit control volume in the combustion chamber is equal to the total mass flow at the control volume limits. Following equations over all species yields the continuity equation for the whole gas flow [29].

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho U_j)}{\partial x_j} = \grave{S}_k \tag{5}$$

where, ρ, U and $\grave{S}_k$ are fluid density, velocity and source terms due to fuel injection respectively.

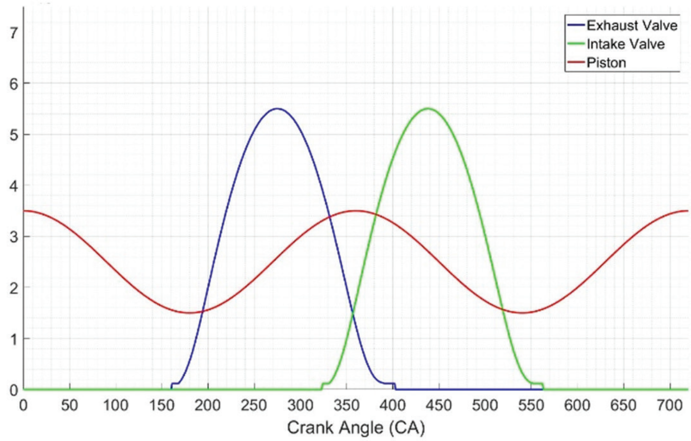


Figure 3. *Intake Valve, Exhaust Valve and Piston Movement Profile*

The momentum or Navier-Stokes equation for combustion chamber control volume is

$$\frac{\partial(\rho U_i)}{\partial t} + \frac{\partial(\rho U_i U_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + F_i^s + F_i^b \quad (6)$$

where p, τ and F_i^s and F_i^b is combustion chamber pressure, viscous stress tensor, spray induced source term and the body force which is equal to ρg.

The energy equation for control volume thermodynamics properties can be expressed as follows.

$$\frac{\partial(\rho e)}{\partial t} + \frac{\partial(\rho e U_j)}{\partial x_j} = -P\frac{\partial(\rho U_j)}{\partial x_j} + \frac{\partial \rho j_j}{\partial x_j} + Q^s + Q^c \quad (7)$$

where e, J_j, Q^s and Q^c are sensible energy, the heat flux arised from heat conduction and enthalpy diffusion and the source terms which derived from injection, respectively.

3. Results and Discussions

Under the specified boundary conditions, the effects of injection timing and injection duration on engine performance and emissions have been determined. Numerical analysis should be verified by experimental setup or experimental data which has been received from the literature. So, the results obtained by numerical analysis have been compared with experimental data.

3.1. Model Validation

In this section, numerical model has been compared with experimental model discussed with literature. Comparisons have been made at 2200 rpm speed and 16 Nm torque. The cylinder pressure obtained from the numerical analysis has been confirmed by the experimental results. Experimental analysis results used to verify the numerical analysis have been obtained from the Ph.D. Thesis [22]. This study has investigated the effects of partial premixed fuel charge effects on the emissions and performance parameters of Antor 6LD400 diesel engine. Numerical and experimental data are demonstrated in Figure 4. Experimental data obtained from the experimental results have been created by using statistical and filtering methods within ± 0.6% accuracy.

Cylinder pressures obtained from experimental and numerical data during combustion and the engine performance parameters for the boundary conditions are shown in Table 3.

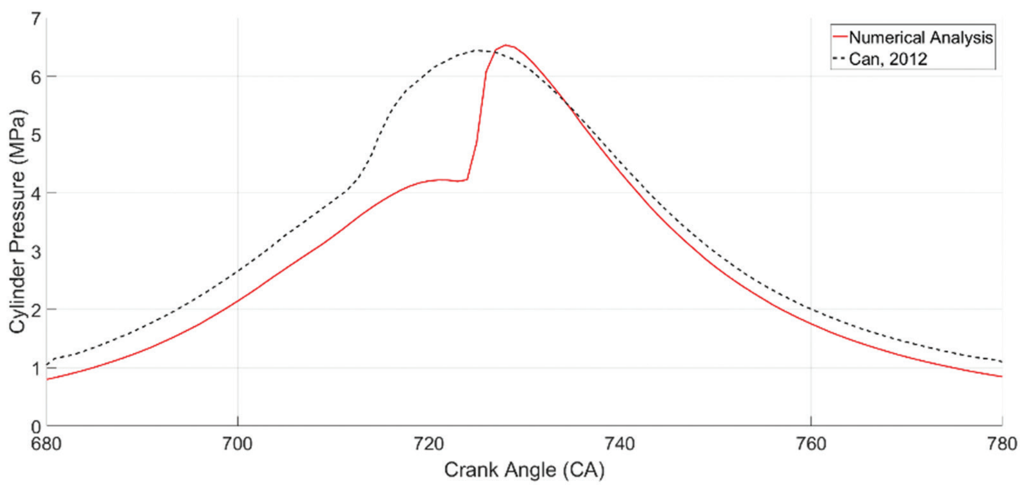


Figure 4. *Experiment and Numerical Results Comparisons in terms of the In-Cylinder Pressure*

Table 3. *Motor Performance Parameters for Injection Timing*

Parameter	Units	Numerical Results	Experimental Results (Can,2012)
Power	kW	3.65067	3.6858
Torque	N.m	15.8472	16
Specific Fuel Cons.	g/kWh	247.68	225
NO_X	ppm	539	545
CO	% Volume	0.292	0.312

The power, torque, specific fuel consumption, NO_x and CO emission characteristics which has been calculated through numerical analysis were compared with the experimental study as in above table. According to the comparison results, it is observed that the maximum deviation is 8.9% with specific fuel consumption. comparisons have been done with the results of the combustion analysis for 2200 rpm, 16 N.m, 24° CA injection timing under operation conditions.

3.2. Combustion Characteristics

At first, numerical analysis was carried out for different injection timings and injection durations of fuel. In the study, temperature, fuel concentration and performance parameters were determined for different injection timings and injection durations. Combustion chamber temperature, pressure, chemical heat release rate and unburned hydrocarbon amount were determined. These parameters have expressed combustion characteristics. In Figure 5, combustion performance parameters have been demonstrated.

Chemical heat release rate, combustion chamber pressure, combustion chamber temperature, and unburned hydrocarbons are shown to vary with the crank angle (CA) for different injection timing for 20° CA injection duration. Reliable estimation of the combustion process depends on correct determination of performance parameters such as ignition delay, flame, temperature and pressure [30]. Injection timing directly

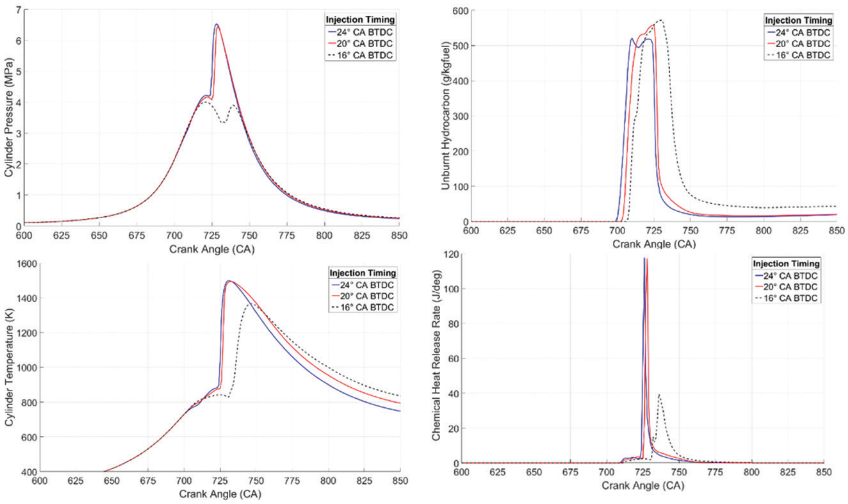


Figure 5. *Motor Performance Parameter for 20° CA Injection Duration*

affects the ignition delay of fuel. Optimum injection timing improves air-fuel mixture quality. Analysis results have indicated that the maximum combustion performance has designated for 20° CA BTDC injection timing. It can be observed that the cylinder pressure has been significantly affected by injection timing. Maximum cylinder pressure has risen with early injection timing. As a matter of fact, the compression temperature and pressure will be higher near the TDC. Therefore, this should be considered in determining the time of fuel injection. Table 4 has demonstrated the motor performance parameter results that has been realized in the same fuel amount and properties.

According to results of various performance characteristics shown in Table 4, the engine power has been determined to be the highest for the 20° BTDC injection timing and the lowest for the 16° BTDC injection timing. The combustion efficiency in diesel engines is based on the ratio of the amount of energy chemically released to the maximum amount of fuel that can be supplied. For 16° CA BTDC injection timing, it was determined that the performance parameters are decreased due to the lean and inefficient combustion of the fuel in the combustion chamber. Fuel-air mixing is a substantial for diesel engine performance parameters. Air fuel does not mix sufficiently with injection timing delay. Insufficient mixture causes unburned hydrocarbons in the exhaust gas. The decrease in combustion efficiency negatively affects the temperature and the amount of the heat release.

Some of the fuel taken into the cylinder is stored by hydrocarbon formation sources during the normal spreading process of the flame. Therefore, they do not participate in the burning event. Some of these unburned hydrocarbons are oxidized in the cylinder in the next combustion process. Some of the hydrocarbons that are not oxidized but remain in the cylinder leave the cylinder. The unburnt hydrocarbons remaining in cylinder mix with fuel-air blends. Remaining unburnt hydrocarbon amount causes a decrease in engine performance. So, to determine mass fraction of hydrocarbons in cylinder and to select suitable injection timing is important for diesel engine performance parameters. In diesel engine performance parameter changes during combustion processes. The fluctuation has transformed into steady state behavior in progress of time. According to in Figure 6, the amount of unburned hydrocarbon remained constant after about 800° CA. Combustion chamber fuel mass fraction been obtained from the numerical analysis at 800° CA which is shown in Figure 6. For the same injection duration of the fuel, it has been observed that the unburned mass fuel fraction has been at most 16° CA injection timing and the least for the 20° CA injection timing.

3.3. Emissions Characteristics

The NO_x and CO emissions are shown in Figure 7. Considering temperature and pressure in Figure 5, it appears that emissions parameters are affected

Table 4. *Motor Performance Parameters for Injection Timing*

Parameter	Units	24° BTDC	20° BTDC	16° BTDC
Power	kW	3.65	3.89	3.52
IMEP	MPa	0.50	0.54	0.49
Lower Heat Value	MJ/kg	44.52	44.52	44.52
Total Chemical Heat Release	J	423.7	454.8	421.3
Combustion Efficiency	-	0.69	0.746	0.691

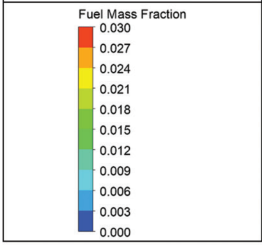
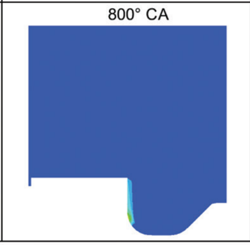
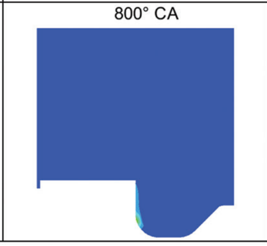
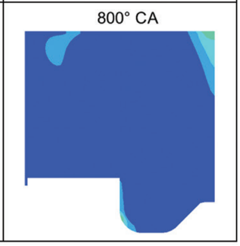
Fuel Mass Fraction	Injection Timing		
	24° CA BTDC	20° CA BTDC	16° CA BTDC
			

Figure 6. *Fuel Mass Fraction in 800° CA for 20 CA Injection Duration*

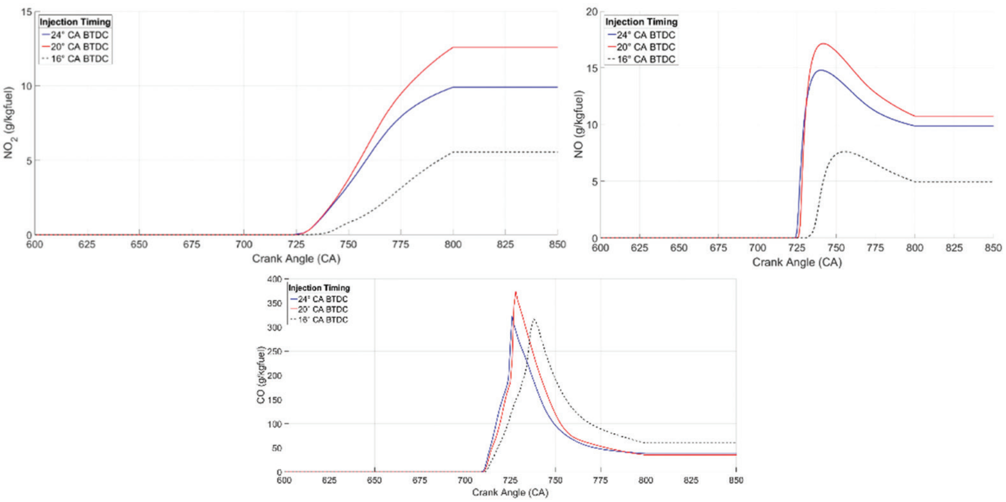


Figure 7. *NO_x and CO Emissions for 20 CA Injection Duration*

combustion temperature and pressure during operation conditions. Although NO_x emissions comprise of high combustion temperature, CO emissions arise from poor engine performance based on low temperature.

It is seen that CO emissions has increased with decreased combustion efficiency. CO emissions has occurred primarily when fuel has not burnt completely. This was due to longer ignition delay and poor mixture formation. However, it is clearly noticeable that if injection timing selected in early CA, cylinder CO emissions would decrease [31]. According to Figure 8, the in case of combustion chamber temperature rises, NO_x emissions increases, and CO and unburned hydrocarbons decreases. Temperature and NO_x emissions indicated in the figure at 800° CA. This crank value is the final phase of the combustion process and the chemical components have reached equilibrium. The emission characteristics of diesel engines depend on combustion temperature and pressure. Most of NO_x emissions are formed from NO. As shown in Figure 8, NO_x emissions have been observed to increase in parallel with the temperature. NO_2 emissions has consist of NO emissions at approximately 750°-800° CA interval. In this case it can be explained with thermal NO which is formed by the combination of

N_2 and O_2 atoms at high temperatures. With the decrease in the combustion chamber temperature, NO emissions decrease and turn into NO_2. With the temperature combustion, although NO_x formation has been raised, CO emissions have been decreased due to partly non premixed and incomplete combustion. The increase in CO is due to the reduction of combustion efficiency and the unburned hydrocarbons. In Figure 9, the relation between CO and temperature has been shown. It has been observed that the CO mass fraction is less than it is in the early injection timing. As a result of, injection timing has affected NO_x and CO emissions. For the early injection timing, CO emissions has been determined as the minimum while NOx is maximum.

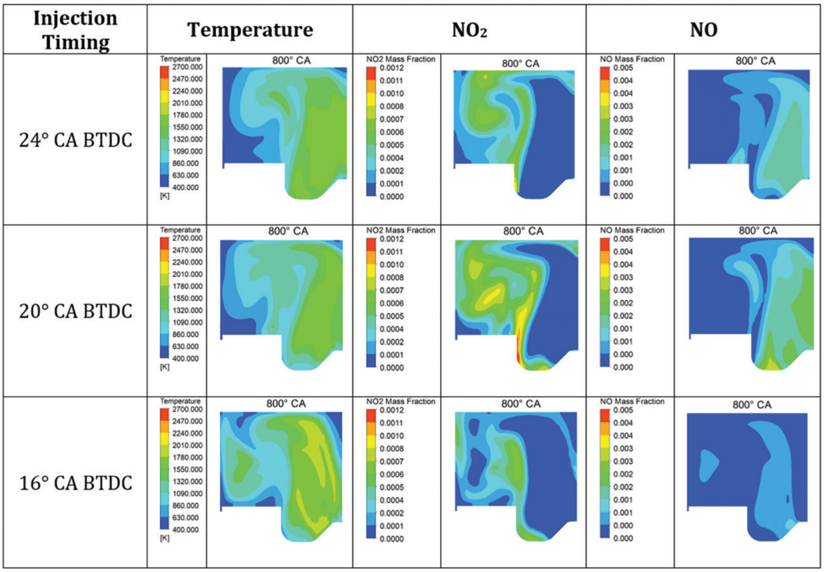


Figure 8. *Combustion Chamber Temperature, NO_2 and NO Emissions*

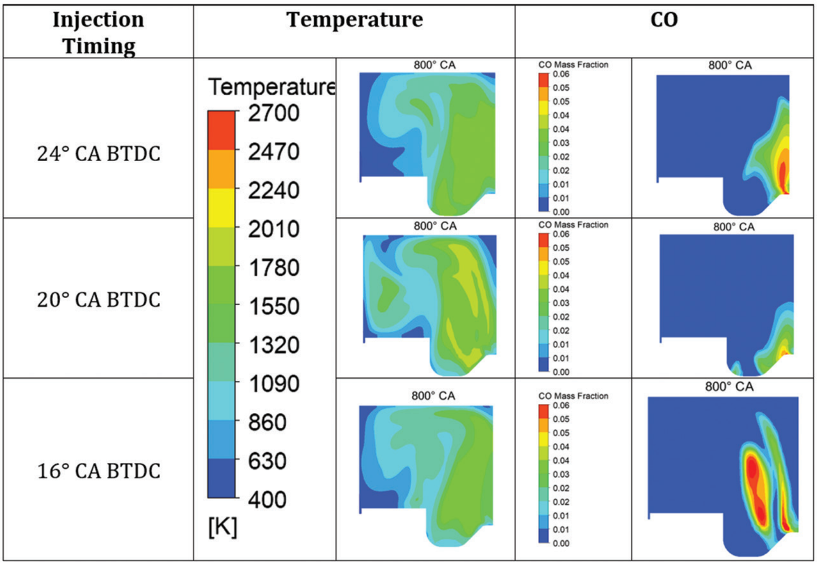


Figure 9. *Combustion Chamber Temperature and CO Emissions*

3.4. The Effect of Injection Duration on the Emission and Performance

Another issue investigated in this study is the effect of injection duration on engine performance and emissions. Different injection duration has been examined for different injection timing. The boundary conditions for different injection duration are given in Table 5. Nine different parameters have been evaluated for three different injection duration.

For each injection timing, three different injection duration have been analyzed at constant boundary conditions. Exhaust gas emissions and performance parameters have been calculated under the same boundary. In order to ensure efficient combustion of diesel engine, the injection timing and injection duration must be optimally adjusted. Injection duration which should be selected considering the injection timing is significant for combustion and performance parameters. Since the injection timing directly affects the ignition delay of the fuel, selecting both injection timing and injection duration properly will prevent loss of efficiency. Spraying the same amount of fuel for different boundary conditions affects fuel injection pressure. With increasing injection duration, injection pressure is decreased inside the cylinder. with increasing injection pressure, maximum value of pressure is increased inside the cylinder. Increasing of spraying pressure raises the fuel speed and air mixture formation in delay phase of combustion. This provides more mixture for pre-mixing phase. Therefore, if injection duration is very high, engine performance parameters adversely affected.

For different injection durations and injection timing, combustion chamber temperature is given with nine varied contour plots in Figure 10. The maximum cylinder temperature has been determined for 20° CA BTDC injection timing and 20° CA injection duration. Because the engine performance parameters are affected by combustion temperature and pressure. Minimum combustion temperature has been obtained for 16° CA BTDC injection timing and 25° CA injection duration compared to other injection parameters. Table 6 shows that motor performance parameters depend on injection duration. According to Table, maximum engine power and IMEP is determined for 20° CA injection duration. The combustion results show that the combustion chamber temperature value is clearly parallel to the engine performance parameters. In addition, NO_X emissions have been determined to be minimum for 25° CA injection duration. It is clearly seen that NO_x emissions rise with rising combustion chamber temperature and CO decreases.

Table 5. *Boundary Condition for Different Injection Duration*

Boundary Condition	Units	Boundary Values		
Injection Timing	-	24° CA BTDC	20° CA BTDC	16° CA BTDC
Number of Nozzles	-	4	4	4
Spray Direction	-	14°	14°	14°
Injection Temperature	K	368	368	368
Mean Cone Angle	-	20	20	20
Turbulence Model	-	k-ε RNG	k-ε RNG	k-ε RNG
Injection Duration	-	15° CA 20° CA 25° CA	15° CA 20° CA 25° CA	15° CA 20° CA 25° CA

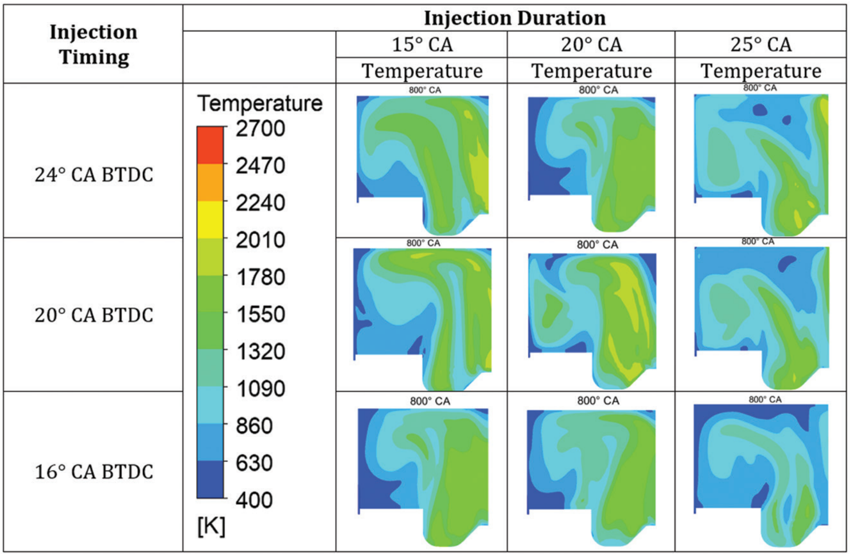


Figure 10. *Combustion Chamber Temperature for Different Injection Duration*

Table 6. *Performance and Emissions Parameter for 24° CA BTDC Injection Timing*

Performance Parameter	Units	Performance Values		
Injection Duration	-	15° CA	20° CA	25° CA
Power	kW	3.18	3.65	3.44
IMEP	MPa	0.44	0.5	0.47
CO Emissions	ppm	1178	1135	1446
NO_x Emissions	ppm	515	539	481

The performance and emission parameter are shown in Table 7 for 20° CA BTDC injection timing. The numerical analysis results show that maximum NOx emissions, IMEP and engine power have been obtained for 20° CA injection duration.

Table 8. demonstrate the performance and emissions parameters for the 16° CA BTDC injection timing. Emissions and performance parameters differ from other injection timing characteristics. Maximum performance parameters and NO_x emissions has been found for the 15° CA injection duration. In addition, for the 16° CA BTDC injection timing, performance and emissions parameter have been determined as most close to each other.

Table 7. *Performance and Emissions Parameter for 20° CA BTDC Injection Timing*

Performance Parameter	Units	Performance Values		
Injection Duration	-	15° CA	20° CA	25° CA
Power	kW	3.42	3.89	3.50
IMEP	MPa	0.47	0.54	0.48
CO Emissions	ppm	1468	1237	2063
NO_x Emissions	ppm	474	624	422

Table 8. *Performance and Emissions Parameter for 16° CA BTDC Injection Timing*

Performance Parameter	Units	Performance Values		
Injection Duration	-	15° CA	20° CA	25° CA
Power	kW	3.49	3.42	3.43
IMEP	MPa	0.48	0.46	0.47
CO Emissions	ppm	1325	2108	1939
NO_x Emissions	ppm	399	283	313

4. Conclusion

In this study, the effects of a single cylinder, four stroke diesel engine injection parameters on engine performance and emissions have been investigated numerically. The engine performance and emissions were compared for injection timings of 24° CA BTDC, 20° CA BTDC and 16° CA BTDC. It has been determined that the power is the maximum value for the 20° CA BTDC injection timing. When the amount of exhaust gas emission is considered, it is observed that CO amount is at least, and NO_x amount is highest due to the highest combustion efficiency of 20° CA BTDC injection timing.

Additionally, the effect of injection duration on diesel engine combustion and emissions have been investigated. The effect of 15° CA, 20° CA and 25° CA injection durations have been investigated for different injection timing. As a result of the study, it was observed that the highest power and combustion efficiency value was obtained with the 20° CA BTDC injection timing and 20° CA injection duration.

In this study, it was determined that combustion efficiency and exhaust gas emissions depend on injection timing. In order to achieve high combustion efficiency, injection timing must depend on optimal crank angle. It was observed that combustion temperature and pressure affect the engine performance and emissions parameters. Increasing cylinder temperature increases NO_x emissions while reducing CO emissions. Injection timing and injection duration should be selected at suitable crank angle. If the injection timing coincides with earlier or later crank angle timing, engine performance and emission parameters are affected negatively.

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