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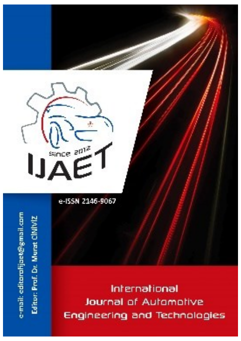

Original Research Article

Multiparametric engine optimization with application of biodiesel blends for better performance and lower exhaust gas emissions

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ABSTRACT

In this study the performance and emissions characteristics of 8-cylinder, 4-stroke, turbocharged Ford 6.7L diesel engine operating on conventional diesel fuel, different blends of soybean methyl ester (SME) and pure rapeseed methyl ester (RME) have been modelled. A multiparametric optimization was applied using Rosenbrock's method. It was found that SME B20 was a better choice among other biodiesel blends that showed good emission reduction with little performance differences compared to those of the reference diesel fuel. The optimization analysis allowed achieving 81.8% reduction in NOx emissions and 75.4% reduction in the combined NO_x and PM emissions.

Keywords: Engine performance; Exhaust gas emissions; Biodiesel blends; Multiparametric optimization

1. Introduction

Biodiesel is a renewable alternative fuel that can be used in diesel engines with little engine modifications. It can be produced from vegetable oils and animal fats including waste cooking oils through a transesterification process which separates glycerine from fat or oil and left methyl ester behind. Although diesel engines have higher efficiency than other engines used in transport, they are also heavy air pollutants. In conventional diesel combustion air pollutants are very difficult to control due to a soot-NO_x trade-off, so a reduction in one contaminant usually results in an increase in another. To reduce pollution level from diesel engines different types of biofuels have been used. Many researchers have investigated the performance and emission characteristics of diesel engines with biodiesel fuel produced from different feedstock. Özener et al. [1] studied the effect of blending of soybean methyl ester biodiesel on the performance, emissions and combustion characteristics of a diesel engine. They tested 10%, 20%, 50% and 100% biodiesel blends. The torque decreased and the brake specific fuel consumption increased with an increase in the biodiesel blend ratio. Also the increased nitrogen oxides (NOx) and carbon dioxide (CO2) emissions, and reduced carbon monoxide (CO) and unburned total hydrocarbon (THC) emissions were observed. It was shown that the biodiesel blends and pure biodiesel have a shorter ignition delays compared to the diesel fuel. Celikten et al. [2] conducted experiments

on a 4-cylinder diesel engine with three different tested fuels: diesel fuel, pure RME and pure SME biodiesel at different injection pressures of 250 bar, 300 bar, 350 bar. It was found that for all tested fuels, the engine torque and power decreased as injection pressure increased. Also smoke level and carbon monoxide (CO) emissions decreased but nitrogen oxides (NOx) emissions increased. For engine performance, combustion with soybean biodiesel showed the lowest level followed by rapeseed biodiesel and the diesel fuel showed the highest value. It was found that combustion with biodiesel had higher NOx emission than those of diesel fuel but had lower smoke level and CO emission levels for all injection pressures. NOx emission level can be controlled by using exhaust gas recirculation (EGR) system by reducing in-cylinder temperature. Palash et al. [3] mentioned that EGR is the most effective method for reducing NOx emission. It decreases about 25-75% of NOx emission with biodiesel at 5-25% EGR rate. EGR also reduces HC and CO emissions slightly, but it increases the brake specific fuel consumption and smoke emissions.
Although researchers have conducted many experiments with biodiesel fuels and compared the engine performance and emissions characteristics, none of them suggested how the engine design or operating parameters could be improved to achieve high performance and low emissions. In the present study, a computational modelling and multiparametric optimization have been conducted to analyze the performance and emissions characteristics of a commercial diesel engine fuelled with soybean methyl ester (SME) and rapeseed methyl ester (RME) and the optimum engine design and operation parameters were suggested to achieve optimum engine performance with biodiesel combustion.

2. Theory and Modeling

In this study we have used full cycle thermodynamic engine simulation programme – Diesel-RK. The program can be used for modelling direct injection diesel engines, spark ignition petrol/gas engines, dual fuel engines, opposed piston engine and etc. The typical applications include engine performance predictions, analysis and optimization of combustion, emissions, valve timing, EGR system, turbocharger, fuel injection system and piston bowl shape, and conversion of diesel engine into gas engine. Several numerical analysis of engine operation and optimization have been applied using Diesel-RK software [4-8]. Diesel-RK has a fuel spray visualization tool, multiparametric and multidimensional optimization tool and 1D & 2D parametric procedures. The program includes RK-model that is a multi-zone diesel fuel spray mixture formation and combustion model, which takes into account:

- Piston bowl shape - any geometrical shapes can be specified and saved,
- Different swirl profiles and swirl intensity,
- Injector location - central, non-central, side injection, few injectors,
- Number, diameter and direction of injector nozzles,
- Fuel properties, including biofuels and blends of biofuels with diesel oil,

Shape of injection profile including multiple injections.

2.1. Multiparametric Optimization

As mentioned earlier, multiparametric optimization is one of the advanced features in Diesel-RK [9]. The procedure of optimization uses the engine's mathematical model together with the specified goal function and restrictions to find a set of optimal design parameters.

<u>*Goal function*</u>

The efficiency parameters of an engine or its separate processes can be included in a goal [10]:

$$Z_j = Z_j(X_k) \tag{1}$$

where:

Z_j is a function of several variables.

Since the main aim of this study was to reduce the exhaust gas emissions in biodiesel fuelled engine the goal function for the optimization was set as a complex emission parameter Summary of Emissions (SE). It is stated that the complex of air pollutants is a sum of particulate matter (PM) and nitrogen oxide (NO_x) emissions and can be calculated as:

$$SE = C_{PM}\frac{PM}{0.15} + C_{NO}\frac{NO_x}{7} \tag{2}$$

where:

$C_{PM} = 0.5$, is the empiric line factor for Particulate Matter emission.

$C_{NO} = 1$, is the empiric line factor for Nitrogen Oxides emission

Independent variables

The set of design engine parameters form the vector of independent variables, X_k and it is in the restricted solutions area [10]:

$$X_{k\min} < X_k < X_{k\max} \tag{3}$$

In this study, there are six engine parameters selected as the variables for the optimization:

- X_1 is intake and exhaust valve timing (opening/closing),
- X_2 is compression ratio,
- X_3 is injector nozzle diameter,
- X_4 is injection timing,
- X_5 is injection duration,
- X_6 is EGR rate.

With these variables, the goal function becomes:

$$Z_j = Z_j(X_1, X_2, X_3, X_4, X_5, X_6) \tag{4}$$

Restrictions

Restrictions are the parameters that limit the optimal search region while searching the solution in the pool of engine parameters:

$$Y_i = Y_i(X_k) \tag{5}$$

For example, in this case the restricting parameters will be: Y_1 is power, Y_2 is specific fuel consumption (SFC) and Y_3 is volumetric efficiency. To search for an optimum of function $Z_j(X_k)$, the following restrictions have to be fulfilled [10]:

$$Y_{i\min} < Y_i(X_k) < Y_{i\max} \tag{6}$$

Generally, the goal function is a sum of Z_j, X_k and Y_i [4]:

$$F = C_{zj} \cdot \overline{Z}_j + \sum_{i=1}^{n}(C_{yi} \cdot \Delta\overline{Y}_i^2) + \sum_{k=1}^{m}(C_{xk} \cdot \Delta\overline{X}_k^2) \tag{7}$$

where:

C_{zj} is a line factor (influence coefficient) of optimized ICE parameter Z_j included into goal function; $\overline{Z}_j = Z_j / Z_{j\mathrm{mean}}$ is a relative ICE parameter Z_j related to its mean average value (e.g. Power, SFC, NOx emission, etc.); C_{yi} is a penalty factor for leaving permitted area of Y_i;

$$\Delta\overline{Y}_i = \begin{cases} \dfrac{Y_i - Y_{i\min}}{Y_{i\,\mathrm{mean}}}, & IF\ Y_i < Y_{i\min} \\ 0, & IF\ Y_{i\min} \le Y_i \le Y_{i\max} \\ \dfrac{Y_i - Y_{i\max}}{Y_{i\,\mathrm{mean}}}, & IF\ Y_i > Y_{i\max} \end{cases} \text{ is a related value of } Y_i; \tag{8}$$

C_{xk} is a penalty factor for leaving permitted area of X_k;

$$\Delta\overline{X}_k = \begin{cases} \dfrac{X_k - X_{k\min}}{X_{k\,\mathrm{mean}}}, & IF\ X_k < X_{k\min} \\ 0, & IF\ X_{k\min} \le X_k \le X_{k\max} \\ \dfrac{X_k - X_{k\max}}{X_{k\,\mathrm{mean}}}, & IF\ X_k > X_{k\max} \end{cases} \text{ is a related value of } X_k \tag{9}$$

Mean values of explanatory variables $X_{k\,\mathrm{mean}}$ and restrictions $Y_{i\,\mathrm{mean}}$ as well as penalty factors of X_k are set by the program. Specification of penalty factors for restrictions C_{yi}; maximum and minimum borders for restrictions $Y_{i\min}$, $Y_{i\max}$ and explanatory variables $X_{i\min}$, $X_{i\max}$ as well as goal function Z_j has to be made by user in the pre-processor of the program [10].

Algorithm Selection

Unfortunately, the theory of nonlinear programming does not answer the question which method is better to solve the multiparametric optimization problem and a researcher has to be guided by his/her own experience of solving problem while selecting optimization algorithm. Each algorithm allows finding solutions of optimization problems with different efficiency. For example, Monte-Carlo method is preferred to be used when the optimization problem is posed with a large number of independent variables and it is advisable to set a large number of iterations up to 1000. Due to the restricted power, specific fuel consumption and volumetric efficiency set in this case, the expected optimum will not be

far from the starting point, and hence a zero-order optimization method was used. It is advised that the first-order method be used in the case when expected optimum is far from the starting point.
As categorized under zero-order methods - on-coordinates descent method, deformable polyhedron method, Powell method and Rosenbrock's method, these four methods were used to perform multiparametric optimization. The deformable polyhedron method failed to find a local optimum, it did not converge and hence caused errors during optimization process. On-coordinates descent method and the Powell method were not preferred because the optimization results showed the exceeding values for restricted parameters. Rosenbrock's method was the only method that provided optimization results by keeping the parameters within the restricted range. Rosenbrock's method [11] proceeds by a series of stages and each stage consists of a number of exploratory searches along a set of directions. The directions are fixed for the given stage and updated from stage to stage for the make use of information obtained about the objective. In the first stage, Rosenbrock's method starts with the search of coordinate directions. It conducts searches of the directions by cycling over each in turn and then moving to new iterations that produce successful steps. The process continues until there has been at least one successful and one unsuccessful step in each search direction, and the current stage terminates after that. For the next stage, Rosenbrock's rotates the set of directions instead of repeating the search process at the same orthogonal vectors, to seize information about the objective validated during the early course of action. Rosenbrock's method imposes the condition that the set of search directions always be *n* dimensional so that the set of vectors remains independent. The function is defined as:

$$f(x,y) = (a-x)^2 + b(y-x^2)^2 \qquad (10)$$

2.2. Computational Setup

In order to perform simulation and multiparametric optimization of an engine, the engine parameters and the properties of fuels have to be specified. In this study, Ford 6.7L V-8 four stroke, turbocharged, DI diesel engine was used as a reference engine for the conversion with biodiesel application. The specifications of engine are listed in Table 1 [12].

Table 1. Engine specification

Engine Model	Ford Power Stroke V-8 4-stroke DI diesel engine
No. of Cylinders	8
Bore x Stroke	99 mm x 108 mm
Displacement	6.7 liters
Compression Ratio	16.2 : 1
Injection Pressure	2000 bar high pressure, Common Rail
Injection Nozzle	19 mm piezo actuated injectors with 8 holes
Maximum Power	223 kW @2800rpm
Maximum Torque	894 Nm @1600rpm
Valve Timings IVO/IVC (°CA)	15 bTDC/40 aBDC
EVO/EVC (°CA)	60 bBDC/15 aTDC

Table 2. Fuel properties

Property	Diesel No. 2	SME B20	SME B40	SME B100	RME B100
C mass fraction	87	84.96	82.97	77.31	78
H mass fraction	12.6	12.45	12.3	11.88	13.5
O mass fraction	0.4	2.591	4.73	10.81	8.5
Density @ 323K (kg/m^3)	830	841	852	885	874
Viscosity @ 323K (Pa.s)	0.003	0.003343	0.003677	0.00463	0.00692
Low heating value (MJ/kg)	42.5	41.18	39.89	36.22	37.1
Cetane number	48	48.68	49.37	51.3	39
Specific heat of vaporization (kJ/kg)	250	265.8	281.2	325	325

3. Results and Discussion

The simulations were conducted using diesel fuel first in order to obtain the reference values for further comparison of engine performance characteristics with other simulated fuels such as SME B20, SME B40, SME B100 and RME

B100. Under the same engine conditions, the diesel fuel was replaced by other simulated fuels and the results are shown in Figures 1 and 2. Figure 1 illustrates the variation of engine power with a range of engine speed for all simulated fuels and Figure 2 shows the brake torque of the engine using all simulated fuels. It can be seen that SME B20 has the best performance among all biodiesel blends. However, the peak power at 2800 rpm decreased about 23% for SME B100 and the peak torque at 1600 rpm decreased about 16% for the same fuel compared to the reference diesel fuel.

The engine is designed in such a way that it is most efficient between 1600 RPM, as shown in Figure 2, and 2800 RPM, as shown in Figure 1. That means that the valve timing and camshaft profiles were made in such a way that the engine "breathes" best between those speeds. That's why the torque is maximum in that region. Another thing is that as the rpm increases, it gets harder and harder to get the optimal amount of air and fuel into the cylinder and burn it at the optimal rate. The faster the engine revolutions, the less time there is to suck in, compress, burn and blow out. Hence, the engine power and torque decrease after the relevant rpms.

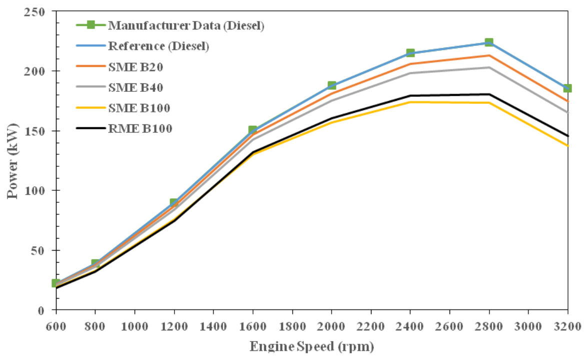


Figure 1. Variation of engine power with a range of engine speed

Figure 3 shows the brake specific fuel consumption for the simulated fuels. The brake specific fuel consumption is a ratio of the engine fuel consumption and the engine power. It measures how efficiently an engine is using the fuel supplied to produce work. It is found that the brake specific fuel consumption increases with the increasing percentage of biodiesel blends. The same trend was obtained by most of the researches [14-17]. The increase in brake specific fuel consumption with biodiesel fuels is due to the combined effects of the higher fuel density and lower heating value. The higher density of biodiesel has led to more fuel being injected for the same injection pressure, thereby increasing the specific fuel consumption [18]. The lower heating value of biodiesel will require more fuel to be injected into the combustion chamber to maintain a constant power output [19]. Xue [18] reviewed that the fuel consumption becomes higher when the engine is fueled with biodiesel because it is required to compensate the loss of heating value of biodiesel. Figure 3 shows that in the engine rpm range between 1600 rpm (max. torque) and 2800 rpm (max. power) the maximum specific fuel consumption for SME B100 at 1600 rpm it showed about 13% increase and at 2800 rpm it showed about 32% increase compared to the reference diesel fuel. This trend is consistent as the density of SME B100 is the highest among other biodiesel blends used in this study, as shown in Table 2.

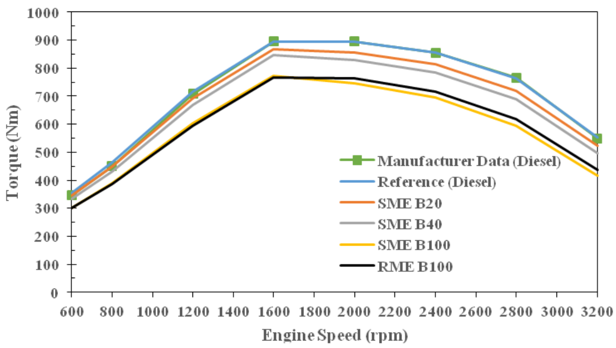


Figure 2. Variation of brake torque with a range of engine speed

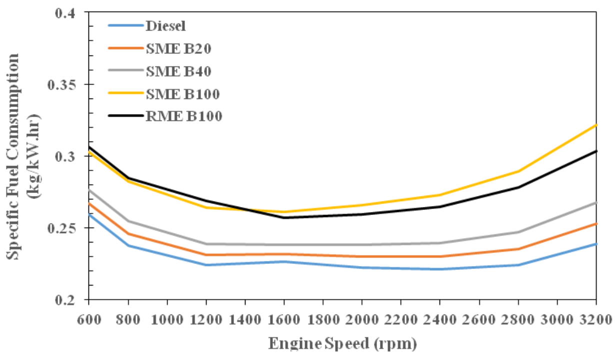


Figure 3. Variation of brake specific fuel consumption with a range of engine speed

Figure 4 shows the heat release rate for simulated fuels. It was shown that diesel fuel has the highest peak of heat release rate and the combustion process is slightly advanced for SME B20 and B40 compared to diesel fuel. Although it is difficult to estimate the rate of change of the heat release for different biodiesel blends due to the transient nature of the heat release affected by the mixing and combustion of fuels with different physical properties, we can at least estimate the pick of the heat release in the diffusion combustion zone. As shown in Figure 4, the pick of heat release at 370 CA degree for RME B100 decreased about 20% compared to the reference diesel fuel.

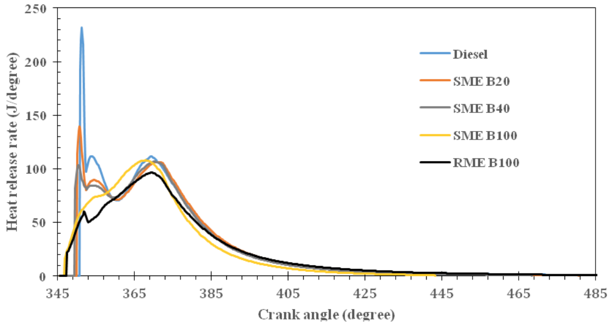


Figure 4. Variation of heat release rate at max. torque at 1600 rpm

Figure 5 illustrates the variation of average cylinder temperature with crank angle for the simulated fuels. It shows that the biodiesel has lower cylinder temperature compared to that of diesel fuel. The lower cylinder temperature occurs with the increase of biodiesel percentage in the fuel blend. It was found that the maximum in-cylinder temperature for RME B100 was 10% lower than that of diesel reference fuel and in-cylinder temperatures for other biodiesel blends were between these limits.

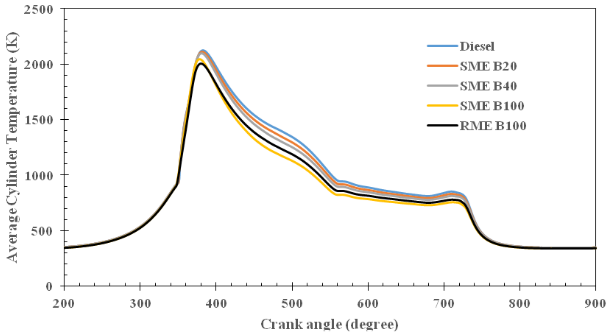


Figure 5. Variation of average cylinder temperature at max. torque at 1600 rpm

Figure 6 shows the NO_x emission for the simulated fuels. The NO_x emission for each biodiesel is higher than that for diesel fuel. Higher NO_x emission was occurred with the increased percentage of biodiesel in the blend. The biodiesel molecule contains oxygen to react with the nitrogen resulting in a higher amount of NO_x formation [20].

Figure 7 shows the PM emission for the simulated fuels. The soybean biodiesel SME provides positive impact on PM emission. The same trends were previously obtained by Nabi [16] and Borgelt [20].

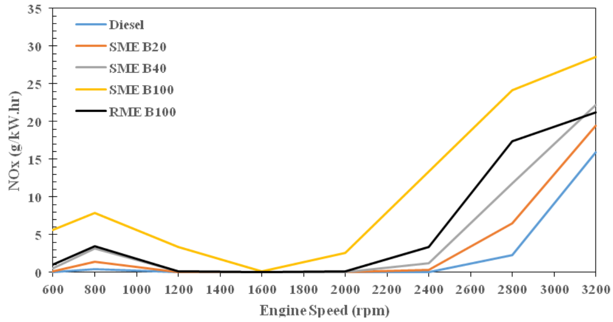


Figure 6. Variation of NO_x at different engine speed

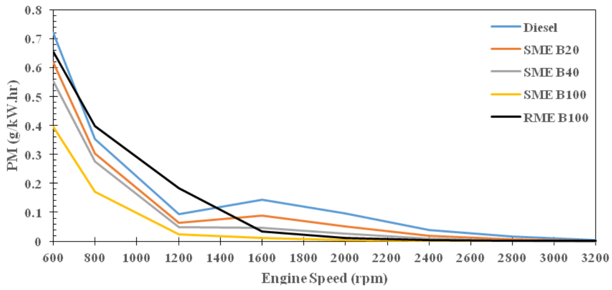


Figure 7. Variation of PM with a range of engine speed

3.1. Multiparametric Optimization

Rosenbrock's method was used for engine multidimensional optimization to find out the optimum variables such as intake and exhaust valve timing (opening/closing), compression ratio, injection timing, injection duration, injector nozzle diameter and exhaust gas recirculation ratio in order to achieve the lowest emissions and specific fuel consumption levels.

Intake and exhaust valve timing

During optimization analysis, it was found that the optimized values of the intake and exhaust valve opening/closing timings were different for every rpm. Since the investigated engine did not have variable valve timing mechanism, the valve opening/closing timing were set as: intake valve opening - 15° bTDC, intake valve closing - 40° aBDC, exhaust valve opening - 60° bBDC and exhaust valve closing - 15° aTDC.

Compression ratio

Table 3 shows the engine optimization results for the compression ratio at different engine speeds. It can be seen that compression ratio 15.7 at 1600 rpm and 2800 rpm has the lowest summary emissions. Although 600 rpm has the lowest summary emissions at compression ratio 17.2, it has lower volumetric efficiency. In addition, the compression ratio has to be constant for all rpm therefore by considering all the results at each rpm, compression ratio 16.2 was used as the final optimization parameter.

Table 3. Optimization results for compression ratio

	Power (kW)	Specific Fuel Consumption (kg/kW.hr)	Volumetric Efficiency	Summary Emissions (g/kW.hr)
600 rpm (Idling)				
16.2 CR (Base line)	21.629	0.26631	0.92147	1.8736
15.7 CR (Optimized)	21.612	0.26652	0.9216	1.8853
17.2 CR (Optimized)	21.615	0.26648	0.917	1.8269
1600 rpm (Maximum Torque)				
16.2 CR (Base line)	146.97	0.23149	0.92344	0.27334
15.7 CR (Optimized)	146.74	0.23186	0.92477	0.26313
17.2 CR (Optimized)	147.33	0.23092	0.92004	0.28913
2800 rpm (Maximum Power)				
16.2 CR (Base line)	214.06	0.2345	0.94669	0.88801
15.7 CR (Optimized)	213.94	0.23464	0.94849	0.8631
17.2 CR (Optimized)	213.65	0.23495	0.94221	0.92745

Table 4. Optimization results for injector nozzle diameter.

	Power (kW)	Specific Fuel Consumption (kg/kW.hr)	Volumetric Efficiency	Summary Emissions (g/kW.hr)
600 rpm (Idling)				
0.19 mm (Base line)	21.650	0.26605	0.92116	1.8731
0.18 mm (Optimized)	21.701	0.26542	0.91803	1.6342
1600 rpm (Maximum Torque)				
0.19 mm (Base line)	146.99	0.23147	0.92330	0.27356
0.18 mm (Optimized)	146.87	0.23164	0.92350	0.27868
2800 rpm (Maximum Power)				
0.19 mm (Base line)	214.05	0.23452	0.94648	0.88573
0.18 mm (Optimized)	214.02	0.23455	0.94684	0.88714

Injector nozzle diameter

The correlation between a nozzle diameter and a cylinder diameter is presented in Diesel-RK and it was stated that for perspective high-speed diesel engines with cylinder bore less than 150 mm, the nozzle diameter can be reduced by 0.1 to 0.15 mm. Since the diameter of cylinder of Ford diesel V-8 engine is only 99 mm, and after the deduction, the minimum nozzle diameter in this case will be 0.15 mm and maximum nozzle diameter will be 0.19 mm. Those values were used as the solution definition range in the multiparametric optimization. Table 4 shows the optimization results for injector nozzle diameters. It can be observed that the best emission reduction occurs with the 0.19 mm nozzle diameter.

Injection timing and injection duration

Al-Dawody and Bhatti [21] mentioned that the most reduction in NO_x emission with biodiesel fuel can be achieved by retarding the injection timing and increasing the nozzle diameter. The results of optimized injection timing and duration are recorded and listed in Table 5. At 2800 rpm, the injection timing retarded from 24.5°CA to 23°CA has reduced the NOx and SE emissions by 6.2% and 5.8% respectively.

Table 5. Comparison of baseline and optimized injection timing and duration

Engine Speed	Injection Timing (degree bTDC)		Injection Duration (crank angle)	
	Base line	Optimized	Base line	Optimized
600 rpm	6	6	22	20.5
800 rpm	7.5	7.5	23	21.5
1200 rpm	11	11	25	23.5
1600 rpm	14.5	14.5	22	25
2000 rpm	18	18	17	20
2400 rpm	21	19.5	16	16
2800 rpm	24.5	23	16	16
3200 rpm	25	23.5	15	15

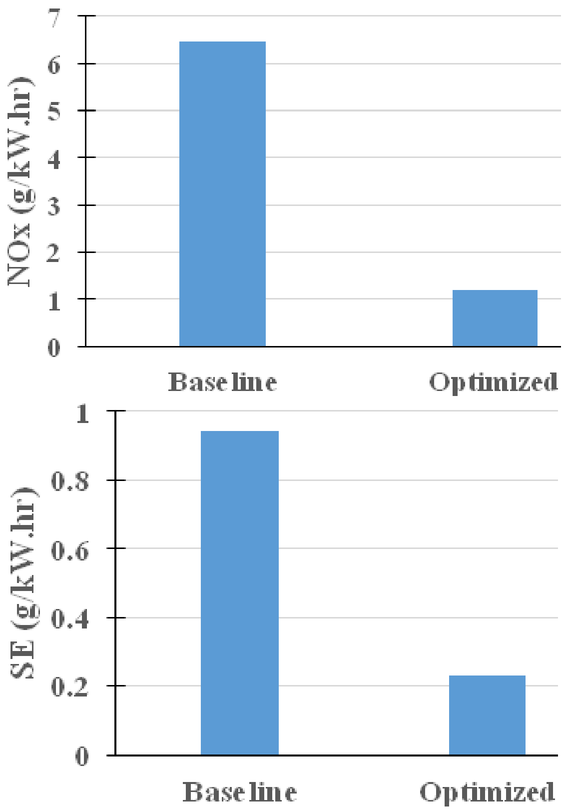


Figure 8. Comparison of NO_x and Summary Emissions for SME B20 at 2800 rpm

Exhaust gas recirculation ratio (EGR)

With the EGR system applied to the engine, a portion of exhaust gas is recirculated back to the engine cylinder which causes a rich air-fuel mixture. The EGR ratio can be calculated by the following formula: $EGR = M_{EGR} / (M_{AIR} + M_{EGR})$, where M_{EGR} is the mass flow of recirculated exhaust gas and M_{AIR} is the air mass flow through the engine cylinders. The optimization gives a good reduction in NO_x and summary emissions with the increase of EGR ratio from 0 to 0.004. Figure 8 shows the optimized emissions results at 2800 rpm with the SME B20 biodiesel. Figure 9 shows the comparison of brake specific fuel consumption of SME B20 before and after the optimization.

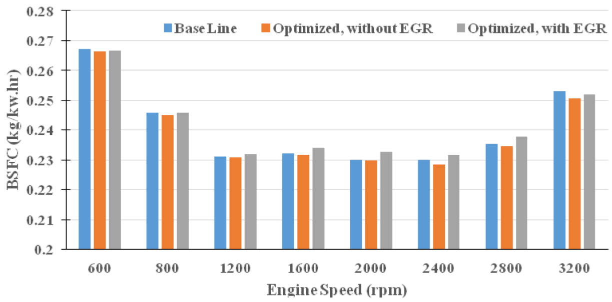


Figure 9. Comparison of optimized BSFC for SME B20 at different engine speed

Therefore, using Rosenbrock's optimization method it was possible to achieve 81.8% reduction in the NO_x emission and 75.4% reduction in the Summary Emissions (SE) compared to those of the baseline for SME B20 at 2800 rpm with compression ratio-16.2, nozzle diameter-0.19 mm, injection timing-23° bTDC, injection duration-16 CA° and EGR ratio-0.004.

3.2. Piston Bowl Analysis

Table 5 shows the engine performance and emissions results at 2800 rpm with different types of piston bowls. Kuleshov [22] stated that deep piston bowls are preferable for low boosting pressure engines with a small cylinder bore due to a longer spray tip penetration. However, if the BMEP is high, the deeper piston bowl geometry causes excessive overlap of near-wall flow zones formed by adjacent sprays leading to the reduction of the engine performance. As shown in Table 5, the Mexican hat is the preferred piston bowl with better engine performance compared to the baseline piston bowl configuration with the deepest bowl geometry. Although the NO_x emission from Mexican hat increases compared to that of the base line, the PM emission significantly decreases, and hence, it causes a reduction in summary emissions (SE).

Table 5. Engine performance and emission results at 2800 rpm with different piston bowl configurations

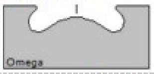
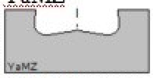
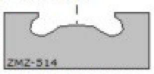
Type of Piston Bowl	Bowl Diameter (mm)	Bowl Depth (mm)	Power (kW)	Torque (Nm)	SFC (kg/kWh)	NOx (g/kWh)	PM (g/kWh)	SE (g/kWh)
Base line 	32.2	17.7	211.16	720.20	0.23773	1.1802	0.019131	0.23236
Mexican Hat 	35.4	13.2	211.62	721.78	0.23721	1.2389	0.01479	0.22629
YaMZ 	29.3	17	211.19	720.32	0.23769	1.2943	0.014017	0.23163
ZMZ-514 	31.5	15.9	211.23	720.43	0.23765	1.191	0.017952	0.22998
AVL-528 	33.1	16.3	211.06	719.87	0.23784	1.1885	0.018843	0.2326

Table 6. Engine performance and emission comparison

Performance	Ford 6.7L V-8 (Diesel No. 2)	Optimized (SME B20)	Difference
Power (kW)	223	221.62	0.62%
Torque (Nm)	763	721	5.5%
SFC (kg/kW.hr)	0.2243	0.23721	1.3%

Finally, the comparison of engine performance and emissions for base line Ford 6.7L V-8 engine and optimized engine are shown in Table 6. Ford 6.7L V-8 engine with diesel fuel complies with Euro V emission standards [23] with NO_x emission - 2.3g/kWh and PM emission - 0.015g/kWh. By applying multiparametric optimization and using Mexican Hat piston bowl configuration, both NO_x and PM emissions were reduced showing better results than those for Euro V standard.

Figure 10 shows the comparison of indicator pressures for reference fuel and optimized conditions with biodiesel blends. It can be seen that the pressure decreases with the use of biodiesel compared that of diesel. Among the biodiesel blends, pure SME biodiesel combustion showed the highest pressure rise, followed by SME B40 and then SME B20. Pure rapeseed methyl ester (RME) biodiesel showed the lowest in-cylinder pressure.

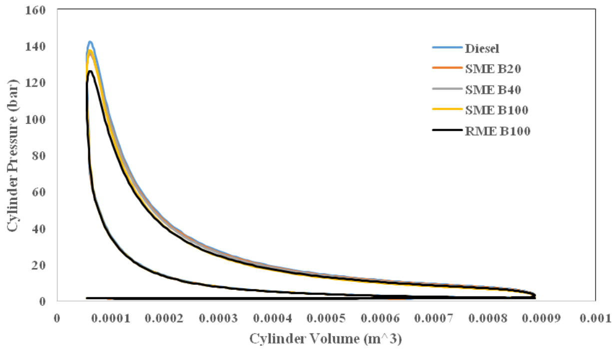


Figure 10. Indicator diagram at max. torque of 1600 rpm

4. Conclusion

From this study, the conclusions can be drawn as follows:

1. The increase of biodiesel percentage in the fuel blend causes higher specific fuel consumption.
2. Engine combustion with biodiesel fuel causes higher level of NO_x emission compared to that of diesel fuel.
3. Multiparametric optimization of injection timing, injection duration and EGR rate allowed reducing NO_x emission and

summary emissions with slight increase in brake specific fuel consumption.

4. With increased exhaust gas recirculation rate, NO_x emissions and summary emissions decreased 81.8% and 75.4% respectively.

5. With the applied multiparametric optimisation technique and piston bowl analysis, the engine exhaust gas emission characteristics were further improved with small sacrifice in the engine performance.

Combined multidimensional optimization and piston bowl analysis allowed improving engine emissions level beyond Euro V standards.

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