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Multi-objective NSGA-II optimization of a compression ignition engine parameters using biodiesel fuel and exhaust gas recirculation

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Abstract

In this study an exhaust gas recirculation system was developed for a small single cylinder 4-stroke engine. Then the mathematical models to correlate responses as the engine emissions and performance characteristics to the factors, include engine load, engine speed, EGR rate and biodiesel fuel percent, were developed. Finally, by using the developed models and NSGA-II (Non-dominated Sorting Genetic Algorithm II) method, the factors were optimized. The highest decrease in NO_x emissions while using the biodiesel and EGR is 63.76% with B10 fuel blend (10% biodiesel fuel and 90% diesel fuel blend by volume) and 30% EGR rate. The highest reduction in HC emission levels while using EGR and biodiesel simultaneously, has been 54.05%. The adjusted R^2 of the proposed model for the CO, HC, NOx, Power, BSFC and smoke were 0.94, 0.91, 0.88, 0.95, 0.89 and 0.94, respectively. Results of the optimization of the engine factors with NSGA-II method has been satisfactory and the pareto front for different test conditions was proposed. The outcomes of the study revealed that the optimization should be taken into account in the development of the new policy for using of the biofuel in the internal combustion engines.

Keywords: Multi-objective optimization; EGR; RSM, Emission

Nomenclature

BSFC: Brake specific fuel consumption (g/kWh)

BX: X percent biodiesel in blend with diesel

CI: Compression ignition

CO: Carbon mono oxide (%vol)

$CO_{2(amb)}$: CO_2 measured at the ambient (%vol)

$CO_{2(exh)}$: CO_2 measured at engine exhaust (%vol)

$CO_{2(int)}$: CO_2 measured at engine intake (%vol)

CO_2: Carbon dioxide (%vol)

DPF: Diesel particulate filter

EGR: Exhaust gas recirculation

EL: Engine load (%)

ER: EGR rate (%)

ES: Engine speed (rpm)

GA: Genetic algorithm

HC: Hydrocarbon (ppm)

ICE: Internal combustion engine

LVH: Lower heating value

NO_x: Nitrogen oxides (ppm)

NSGA-II: Nondominated sorting genetic algorithm

RSM: Response surface methodology

SI: Spark ignition

PF: Pareto front

LHV: Lower heating value

P: Pitch

do: External diameter (mm)

di: Internal diameter (mm)

e: Thickness (mm)

Q_c : Cooling water heat transfer (kW)

Q_g: Recirculated gas heat transfer (kW)

T_{ci}: Inlet water temperature (°C)

T_{CO}: Outlet water temperature (°C)

T_{gi}: Inlet gas temperature (°C)

T_{gO}: Outlet gas temperature (°C)

ε: Thermal efficiency (%)

U: Cooler heat transfer coefficient (W/m^2K)

A: Area of the cooler (m^2)

C_{min} and C_{max} : Minimum and maximum values of the thermal capacity of the recirculated gas or cooling water

R^2: Coefficient of determination

1 Introduction

The growth in price of common energy sources especially in recent years, enforced human to use alternative sources of energy [1]. Then, recently, the usage of biodiesel in internal combustion engine is accepted as a reliable source of energy [2]. The most significant aspect of the biodiesel fuel is its similarity to diesel fuel. It can be used in compression ignition engine without any engine modifications. Biodiesel is a renewable source of energy and can be used without any concern about the ending of it. In addition, in most cases this fuel comes from waste materials and it would be a good way to reuse waste materials [3, 4].

Despite the advantages of the CI (compression ignition) engine and biodiesel fuels, there are some obstacles to using them with confidence. One important problem related to them is the higher emissions like NO_x and smoke than the other kind of internal combustion engines like gasoline engine [5, 6]. Then, the state of the art about this kind of fuel and engine is focused on reduction of the emissions of this kind of engine and fuels. Different technologies and methods have been proposed and developed to overcome this problem. These technologies and methods can be divided into two categories: before combustion like exhaust gas recirculation (EGR) [7, 8] and after treatment systems like diesel particulate filter (DPF)[9]. The first categories try to improve combustion quality and hence reduce the emission of the combustion. The latter try to prevent the emissions into the atmosphere. There are many commercial types of these technologies which are installed and working on advanced engines. Now, there are still challenges that are remaining in these areas. Firstly, there are many old CI engines or old designed CI engines which should be considered to use these kinds of technologies. The other challenge is the optimization of the developed technology to have as much possible improvement in their efficiency.

The most important emissions of the diesel engine are NO_x and smoke emissions [10]. Also many studies stated that the use of biodiesel will increase the NO_x emissions of CI engine [11]. The EGR system is an accepted technology to reduce NO_x [12] although there are some disadvantages such as increase in smoke emission, HC and CO emissions reported for this technology[5]. The next step of the research on the EGR system is to optimize it to have a trade-off between NO_x emissions and other emissions in different engine working conditions. Optimization is one common step in engineering problems.

The main aim of the optimization in each problem is to find the best possible solution for it [13]. Different methods to optimization can be used but first step is to find a reliable model for variables. Atmanli et al., optimized concentration of biodiesel diesel fuel blend to improve engine performance by RSM method [14]. They found 11.4% vol. as the optimal percent of biodiesel blend. In addition, they stated that by optimizing of the engine by RSM method NOx, CO and HC emissions of biodiesel-diesel blend considerably declined as 11.33%, 45.17% and 81.45%, accordingly. Dhole et al. [15] proposed a mathematical model to correlate some emissions and engine performance parameters by changing engine parameters like load and hydrogen fuel substitution. They used response surface methodology (RSM) to develop the model. Comparison of experimental data with the predicted data showed high correlation coefficients (R^2) for the various response variables. Same results about ability of RSM to predict engine emission and performance by using of injection timing and pressure is reported [16]. In addition, other researchers employed same procedure to predict engine performance and exhaust emissions. They stated that the RSM was an appropriate method for engine applications modelling [14, 17]. Yilmaz et al., [17] used two different method to modelling engine performance and emissions. They compared RSM and least-squares support vector machine (LSSVM) and found that, although LSSVM was slightly powerful than RSM method to predict engine performance and emissions, RSM method was still enough powerful to predict engine performance and emission characteristics. Finally, it is stated by Atmanli, Ileri and Yilmaz that the RSM method can be used to model and optimize percent of biodiesel fuel blended with diesel fuel based on the performance characteristics and emissions [18]. Therefore, the RSM method was used to develop the models in this study.

After modelling of the engine characteristics, optimization of the effective factor on engine performance and emissions can be done. The genetic algorithm method is an accepted method to optimize different characteristics of internal combustion engines [19-21]. This method has good capability to optimize the complicated, nonlinear and discontinuous problems [19, 22]. Deb et al. used GA to optimize the dual fuel (diesel and hydrogen) engine performance [23]. They stated that their methodology can provide the designer with more short analysis cycle time and more accurate design results than traditional optimization methods. Etghani et al. have used the GA method to optimize performance and emission of a diesel engine [24].

According to above mentioned literature, on the one hand, necessity of application of optimization method to reduce internal combustion engine drawbacks such as their emissions is undeniable. On the other hand, this optimization should not has any negative effect on engine performance. Therefore, this study is formed based on these two criteria. However, there are some other studies about optimization of engine but still application of EGR and biodiesel fuel simultaneously for small diesel engines have not been considered. To give more details, the aim of this study is to develop an EGR system for a small CI engine and optimization of its working condition by the NSGA-II method. The factors which are considered here are engine load (25, 50 and 75%), engine speed (1800, 2100 and 2400 rpm), EGR rate (0, 10, 20 and 30) and biodiesel percent (0, 5, 10 and 15%). The responses are engine emissions (CO (%), HC (ppm), NO_x (ppm), smoke (1/m)) and engine performance parameters (brake specific fuel consumption (BSFC) (g/kWh) and engine power (kW)).

2 Materials and methods

The steps of research are:

1. Developing an EGR system and EGR cooler for the small single cylinder CI engine
2. Consideration of the emission and performance parameters of the engine while using biodiesel and EGR
3. Proposing a mathematical model to correlate the engine emissions and performance parameters to the engine working condition
4. Optimization of the engine working condition to improve engine emissions and performance parameters.

2.1 Characteristics of the produced fuel

Biodiesel fuel which has been used in this study is produced in the Renewable Energy Laboratory of Tarbiat Modares University. The source of the produced biodiesel is waste cooking oil and it is produced by the stratification method. The diesel fuel used in this study is standard diesel fuel. Characteristics of the produced biodiesel fuel and used diesel fuel are shown in Table 1. As can be seen in Table 1, the viscosity and density of the biodiesel fuel are higher than diesel fuel.

Table 1. Some important characteristics of the produced biodiesel fuel along with the related standards

Characteristic	Standard test method	Allowable range	Biodiesel	Diesel	Unit
Kinematic viscosity	EN 14214	3.5 - 5	4.72	3.5	mm^2/s
Density	EN 14214	----	0.862	0.837	g/cm^3
Cloud point	ASTM D-2500		-1	-	°C
Pour point	ASTM-D97		-4	-	°C
Water and sediment	ASTM D-2709	0.05<	0.05	0.05	% vol
Free glycerol	ASTM D-6584	0.02<	0.016	-	%mass
Flash point	ASTM D-92	>130	176	55<	°C
LHV	ASTM D-240	-	39.6	43.8	MJ/kg
Cetane number	ASTM D 613	47 min	57	48	-
Oxygen content	-	-	9.45	0	%

Kinematic viscosity was measured by the SVM-3000 viscometer (Anton Paar company) and according to the ASTM D-445 standard. According to this standard, the temperature of the viscometer was set at 40 ° C and the ambient temperature of the laboratory was 25 ± 2 ° C and its relative humidity was 50 ± 5%. The flash point was measured in accordance with the ASTM D-93 standard and by means of the P611A flash point determination device manufactured by the Belgian Analysis Company and Cleveland open-cup method (COC) method. The biodiesel density was determined using the DMA 35n density Meter made by Austria's Anton Paar Company. The amount of water content in the biodiesel was measured based on the ASTM D-2709 standard and by Basic Titrino 794 Carl Fisher device made in Switzerland. The cetane number of considered fuels was measured by a CFR engine and according to the ASTM-D613 standard. LHV of the fuel was measured by Pars Paygeer PM-52 bomb-type calorimeter and according to the ASTM-D240 standard. Pour point and cloud point were measured according to ASTM D2500 and D97 respectively. To do so, the temperature of the biodiesel fuel sample in a glass tube was decreased in 1 C° intervals until a cloud was observed. The temperature in this condition was considered as cloud point. In order to measure the pour point, the process of cooling for sample was done in 3 C° until the sample cannot move when its surface was vertical for 65 seconds. The chemical composition and then oxygen content of biodiesel was measured by Clarus 580 GC-MS device. These experiment were performed in three replications and the mean results were reported for each parameter.

2.2 Test bed specifications

Specification of the considered engine is presented in Table 2. An eddy current dynamometer has been used to measure different engine performance parameters such as engine torque, engine power and speed. The maximum torque, engine speed and engine power which can be measured by this dynamometer were 80 Nm, 10000 rpm and 21 hp with 0.5–1% accuracy, respectively. Schematic view of the test bed is shown in Figure 1.

Table 2. Specifications of the evaluated engine

Model	3LD 510
Manufacturer	Lombardini, Italy
Number of cylinders	1
Stroke	90 mm
Bore	85 mm
Displacement	510 cm^3
Rated output power (@3000 rpm)	12.2 hp (9 kW)
Rated output torque (@1800 rpm)	33 Nm

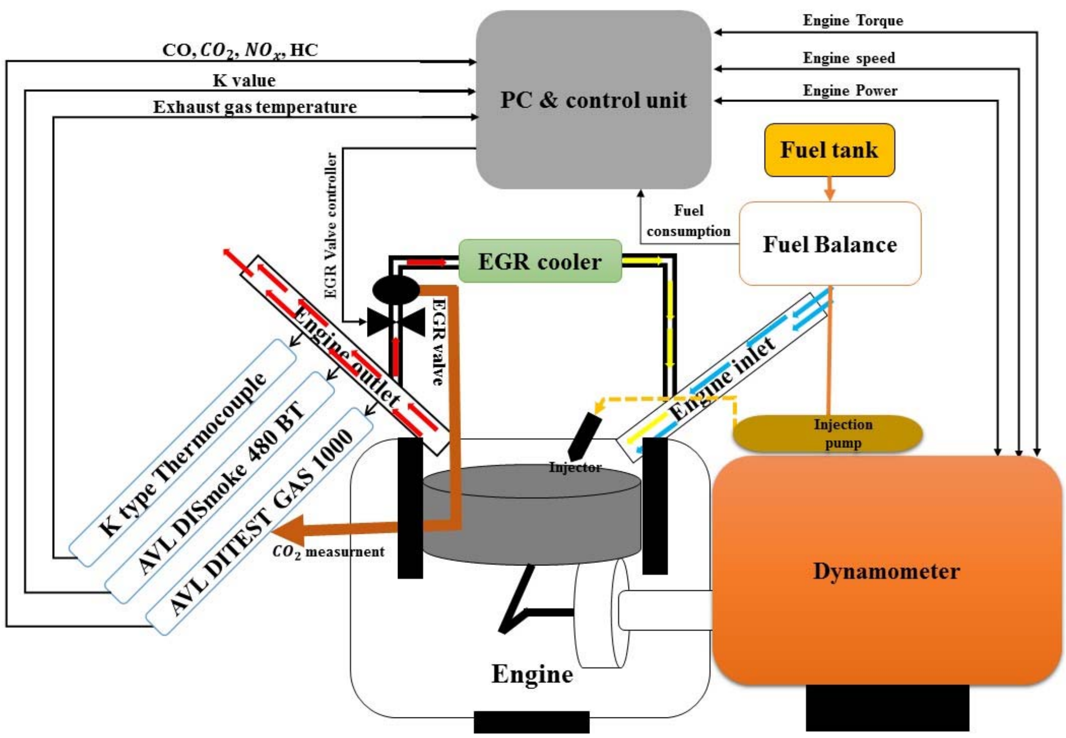

Figure 1. Schematic view of the test bed

An EGR system was added to the engine. The added EGR was a cooled EGR. The EGR cooler and other parts of the system can be seen in Figure 1. The EGR cooler was a tube-pipe cooler. This cooler has been designed and fabricated. The cross-sectional view of the pipes are shown in Figure 2**.** The EGR cooler has been designed in Solidworks software and simulated in CFX ANSYS software. The EGR cooler was fabricated according to the result of the simulation.

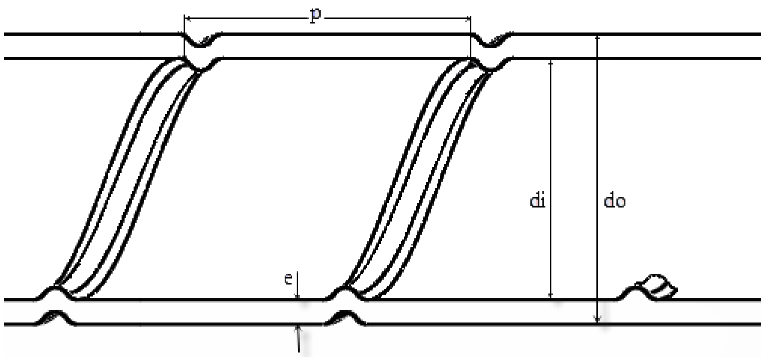

Figure 2. The cross-sectional view of the pipes and its dimensions

The pipes of the cooler were half spiral and the dimensions of them are mentioned in

Table 3 and are shown in Figure 3A.

Table 3. Dimensions of the half spiral-half smooth pipes of the EGR cooler

Length (mm)	Thickness (e) (mm)	Internal diameter (di) (mm)	External diameter (do) (mm)	The angle of the start of the spiral	Number of spiral	Pitch (p)
190	1	10	12	134	7.5	12

In addition to improve the resistance of the cooler to vibration, the baffles have been used inside the tube to support the pipes (Figure 3B). Figure 3C, D and E show the velocity, pressure and temperature distribution of recirculated gases in 0.00051 kg/h mass flow rate through the cooler.

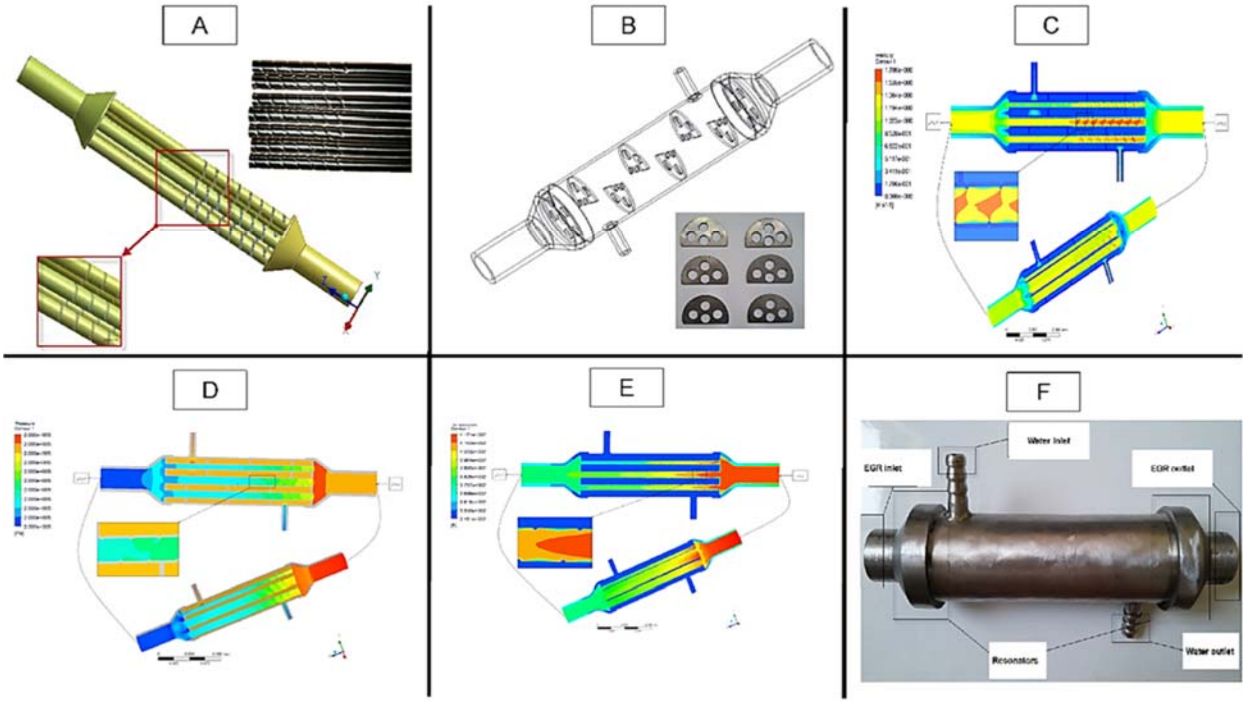

Figure 3. A: model of the pipes and created pipes; B: The tube and baffles; C, D and E: velocity, pressure and temperature distribution of recirculated gases in 0.00051 kg/h mass flow rate through cooler; D: developed EGR cooler

Figure 3F shows the fabricated EGR cooler and the inlet and outlet of the exhaust gas and cooling water.

The Number of Transfer Units (NTU) method has been used to calculate the thermal efficiency of the developed EGR cooler. The actual heat transfer of the EGR cooler can be calculated by calculating the average value of the heat transfer of the cooling water and recirculated gas.

$$Q = \frac{Q_c + Q_g}{2} \qquad \text{Eq. 1}$$

Where the Q_c and Q_g are the cooling water and recirculated gas heat transfer in kW, respectively.

$$Q_c = \dot{m}_c C_{pc}(T_{CO} - T_{ci}) \qquad \text{Eq. 2}$$

$$Q_g = \dot{m}_g C_{pg}(T_{gO} - T_{gi}) \qquad \text{Eq. 3}$$

Where T_{ci}, T_{CO} and T_{gi} , T_{gO} (°C) are inlet and outlet water temperatures and inlet and outlet gas temperatures, respectively. It is assumed that the heat losses are ignorable and so $Q_c = Q_g$. The thermal efficiency can be calculated with the following equation:

$$\varepsilon = 1 - \exp\frac{NTU^{0.22}}{C^*[\exp(-C^*NTU^{0.78} - 1)]}, NTU = \frac{UA}{C_{min}} \quad \text{Eq. 4}$$

$$\varepsilon = \frac{Q}{Q_{max}}, Q_{max} = (\dot{m}c)_{min}(T_{gi} - T_{ci}), C^* = \frac{C_{min}}{C_{max}} = \frac{\dot{m}_g C_{pg}}{\dot{m}_c C_{pc}} \quad \text{Eq. 5}$$

Which maximizes the efficiency (ε) of the real heat transfer ratio of the EGR cooler. NTU is the number of the transfer units and shows the size of the EGR cooler. U (W/m^2K) is the cooler heat transfer coefficient and A (m^2) is the area of the cooler. C_{min} and C_{max} are the minimum and maximum values of the thermal capacity of the recirculated gas or cooling water, respectively. This ratio is dependent on the value of the thermal capacity of the recirculated gas and cooling water and can be $\frac{C_c}{C_g}$ or $\frac{C_g}{C_c}$. The required temperatures to calculate the thermal efficiency has been gathered by using thermocouples.

In Figure 4, the final developed EGR cooler is shown.

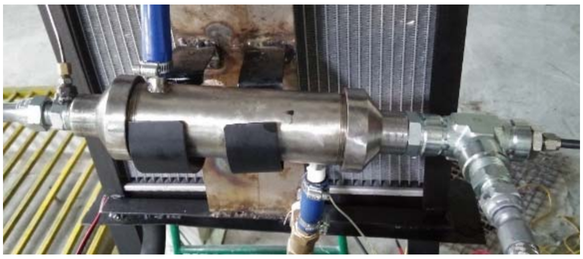

Figure 4. Homemade EGR cooler

To measure the EGR rate of the engine, the following equation was used [25]:

$$\text{EGR ratio} = \frac{CO_{2(int)} - CO_{2(amb)}}{CO_{2(exh)} - CO_{2(amb)}} \times 100 \quad \text{Eq. 6}$$

Where: $CO_{2(exh)}$, $CO_{2(int)}$ and $CO_{2(amb)}$ are the % CO_2 measured respectively at the engine exhaust, intake port and the ambient. The EGR rate is regulated by a valve.

To measure fuel consumption of the engine, a system which includes sensors, pipes and fuel tank was used. The accuracy of this system is ±1 cm^3/h.

An AVL DITEST GAS 1000 exhaust emission analyser was used to measure emission gases of the engine and AVL DISmoke 480 BT smoke opacimeter was used to measure smoke emission of the engine in terms of the K value. The accuracies of the measurements and the uncertainties in the measured results for these devices are shown in Table 4.

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Table 4. The accuracies of the measurements and the uncertainties in the measured results

Measured	Resolution	Accuracy
CO	0.01 % vol.	< 10.0 % vol.: ± 0.02% vol
		≥ 10.0 % vol:
HC	≤ 2.000 : 1 ppm vol.	< 2000 ppm vol.: ±4 ppm vol
		≥ 5000 ppm vol
		≥10000 ppm vol
NO_x	1 ppm vol.	± 5 ppm vol.
Absorption (k-Value)	0.01l/m	Measuring range: 0 ... 99.9%

Finally, a standard engine test cell was used to do the experiments. The engine was started and run in idle mode and with standard diesel fuel for a while to reach steady state condition, which is determined by the temperature of the engine oil temperature (70 °C).

2.3 Design of experiments

The factors and their level which have been considered are shown in the Table 5. Since application of the EGR system in the part-load condition is reasonable when the load selected for the experiment is in the range of 25 to 75%. The responses of the experiments were selected from the important engine performance parameters and exhaust emissions. These responses are shown in Table 6. The selected DOE (Design of Experiment) for this study is the general factorial design. There were also 3 replications for each test and in total were 432 experiments.

Table 5. Matrix of the experiments (factors)

Factor	Level 1	Level 2	Level 3	Level 4
Engine load (%)	25	50	75	-
Engine speed (rpm)	1800	2100	2400	-
EGR rate (%)	0	10	20	30
Biodiesel percent (%)	0	5	10	15

Table 6. The responses of the experiment

Response	Unit
NO_x	ppm
HC	ppm
CO	%
Smoke	1/m
BSFC	g/kWh

2.4 Response surface methodology

Response Surface Methodology (RSM) is a collection of statistical and mathematical techniques useful for developing, improving, and optimizing processes [26]. The most general use of RSM is in the specific conditions where some variables hypothetically impact some parameters or characteristic of the process. Thus, the parameters or characteristic of the process is named the response. The variables are called independent variables or factors, and they are adjusted by the researcher [16].

In response surface methodology, the empirical statistical models will be developed to explore the space of the factors. These models provide an approximation to correlate the factors to the variables. In other words, statistical models like Eq. 7 will be developed to predict factor y according to variables $x_1, x_1, \dots, x_k$.

$$y = f(x_1, x_1, \dots, x_k) + \varepsilon \qquad \text{Eq. 7}$$

The form of function f is not defined and may be very complicated and ε is the representation of the variables which may have not been considered in the f function. Generally, ε is the result of the error in response measurements, noise of background, and effect of unconsidered variables[27]. Here the factors or independent variables are engine load (%), engine speed (rpm), EGR rate (%) and biodiesel percent in fuel blend. The responses are CO (%), NO_x (ppm), HC (ppm), power (kW), BSFC (g/kWh) and smoke (1/m). The general form of the models are like this:

$$\begin{bmatrix} \text{CO (\%)} \\ \text{NO}_x\text{ (ppm)} \\ \text{HC (ppm)} \\ \text{power (kW)} \\ \text{BSFC (g/kWh)} \\ \text{smoke (1/m)} \end{bmatrix} = f(\text{engine load (\%)}, \text{engine speed (rpm)}, \text{EGR rate (\%)}, \text{biodiesel percent (\%)}) + \varepsilon \qquad \text{Eq. 8}$$

Developing of the model has been done by Design expert 8.0 software.

2.5 Multi-objective optimization

Optimization of a function can be stated as maximization or maximization of process parameters in that function according to some variables which these parameters are dependent on them [28]. In this study, engine load, engine speed, EGR rate and biodiesel percent in blend with diesel are the process parameters which should be optimized by maximizing engine power and minimizing BSFC, CO, HC, NO_x and smoke. In most optimization problems, there is a trade-off between different objective functions which should be optimized. In other words, minimization of one variable may optimize one objective function but may cause undesired changes in the other objective functions. One of the techniques offered to overcome this problem is the Pareto optimal solution which will be discussed.

Genetic algorithm which is used in this study, is one of the optimization methods which has become very popular in optimization problems. In fact, this algorithm is an evolutionary computational method inspired by evolutionary processes. Evolutionary process is an optimization problem that is based on population. Goldberg [29] compares available computational method with methods based on evolutionary computation and genetic algorithms and he concluded that this method is resistant to random oriented optimization methods. The genetic algorithm starts with an initial population and move towards an optimum population. This algorithm is a search tool which by using objective functions, connects to analytical tool. In this research, multi objective optimization methods by using genetic algorithms will be applied. Generally, this method is expressed as follows:

$$\text{Minimize}_{\vec{x}\in S}\vec{Z} = (z_1(\vec{x}), z_2(\vec{x}), \dots, z_m(\vec{x})) \qquad \text{Eq. 9}$$

$$h_j(\vec{x}) = 0, \qquad j = 1, 2, \dots, p$$

$$g_k(\vec{x}) = 0, \qquad k = 1, 2, \dots, q$$

Where $z_i(\vec{x}), i = 1, 2, \dots, m$ are objective functions and $h_j(\vec{x})$ and $g_k(\vec{x})$ are limiting functions.

Function $z_i(\vec{x}), i = 1, 2, \dots, m$ is named object or target factor and each component of $z_i(\vec{x}): R^n \to R$ is one target.

All the possible answers which is a subset of the $\vec{z}$ target vector called research space is shown by $S \subset R^n$. $h_j(\vec{x})$ and $g_k(\vec{x})$ are called constraints which will be equality or inequality constraints. Constraints will be shown by function when the number of constraints in a problem is high. $\vec{x} \in S$ is named a decision variable or vector. Each $\vec{x} \in S$ is one of the solutions of the optimization problem. Optimization problems can be expressed as a minimization problem. The maximization of function $z(\vec{x})$ is equivalent to minimization of $-z(\vec{x})$ function. Variable of this optimization problems are shown in Table 7. The different components of objective functions have been presented in Table 8.

Table 7. The variable to optimize

Parameter	Symbol	Unit	Range
EGR percent	x_1	%	[0-30]
Engine load	x_2	%	[25-75]
Engine speed	x_3	rpm	[1800-2400]
Biodiesel percent	x_4	%	[0-15]

Table 8. The objective function to optimization

Goal	Parameter	Objective function
Minimum	NO_x	f_1
Minimum	CO	f_2
Minimum	Smoke	f_3
Maximum	Power	f_4
Minimum	HC	f_5
Minimum	BSFC	f_6

Then, the optimization problem can be defined as follows:

$$\vec{f}(\vec{x}) = \left(f_1(\vec{x}), f_2(\vec{x}), \dots, f_m(\vec{x})\right) \qquad x \in R^n, n = 5, m = 5 \qquad \text{Eq. 10}$$

Or:

$$\underset{\vec{x}\in s}{\text{Minimize}}\ \overrightarrow{\vec{f}(\vec{x})} = \left(f_1(\vec{x}), f_2(\vec{x}), f_3(\vec{x}), -f_4(\vec{x}), f_5(\vec{x})\right) \qquad \text{Eq. 11}$$

There is usually a conflict among the optimization in most multi objective optimization problems. Therefore, it is difficult to attain a solution that optimizes each objective function simultaneously. The Pareto optimal is the set of the solutions to these problems. But, the concept of dominance must be introduced before defining this term.

Pareto front is a corresponding objective function values for a set of solutions which are not dominated by other solutions. The solution which are in the Pareto front cannot be improved for one objective without worsening the other ones. The main aim in each MOP problem is to find the Pareto front [23]. To represent the concept of the Pareto front for a two-objective optimization problem can be stated as follows:
A solution (x^*) is in the Pareto front if and only if the following condition is satisfied [23]:

$$f_i(x) \leq f_i(x^*) \qquad i = 1,2, \dots, n \qquad \text{Eq. 12}$$

And for at least one j, $1 \leq j \leq n$

$$f_j(x) \leq f_i(x^*) \qquad \text{Eq. 13}$$

2.6 Procedure of NSGA-II multi-objective optimization

In this study the NSGA-II evolutionary algorithm was applied [30]. The first step in evolutionary algorithms is to generate an initial population. The real number which are for the design variables expressed will be transformed to binary number, and each binary string is called an individual. An initial population will be generated by individuals which are produced randomly. The fitness for each individual is evaluated by objective function. To select and reduce the individuals, the Pareto ranking method will be used.
Then new generation of individuals will be produced by previous generation individuals or by mutation. This will be repeated till it reaches the function to predetermined value.
Special characteristics of the NSGA-II is its selection method. The domination of the individual will be checked to see if they belong to the Pareto front or if other individuals dominate them. The second Pareto front will be formed by the dominated individuals. This will be repeated until all individuals are placed in different Pareto fronts with different rankings.
In addition, the other ranking for the individual is based on their distance from their nearest neighbour. This ranking is done to have a uniformly distributed Pareto front. In the selection step, some of the individuals will be reduced and new ones will be produced by binary tournament selection, recombination and mutation operators.
The advantage of the NSGA-II algorithm is the generation of evenly distributed and high-resolution PF. The Matlab software was used for the implementation of this algorithm. The properties of the applied NSGA-II multi objective optimization is shown in
Table 9.

Table 9. Properties of the proposed NSGA-II multi objective optimization procedure to optimize engine parameters

Parameter	Value
Population type	Double vector
Population size	50
Selection function	Tournament
Crossover function	0.8
Mutation function	0.02
Stopping criteria	Generations: 400/stall generations: 100

3 Results and discussion

3.1 Model and data analysis

The first parameter is the thermal efficiency of the EGR cooler which is 63.2%. This value of the thermal efficiency is within acceptable range for an EGR cooler [31, 32].

In following step, the data were analyzed by design expert 8.0 (Statease, Minneapolis) software and ANOVA table and appropriate model were proposed for responses.

Table 10 shows the analysis of variance (ANOVA) of quadratic model for responses. A glance at the table reveals that engine load and engine speed have had a significant effect on the value of all parameters while amount of the EGR rate was only effective on the amount of the NOx and HC emissions. In addition, percent of the biodiesel blended with diesel fuel was effective on the values of the most of the parameters except power and BSFC. Furthermore, interaction effect of the engine speed and engine load has had a significant effect on all of the responses. This for interaction of the engine load*EGR rate just can be seen for the HC and NOx emissions, similar to the EGR rate main effect. Interaction effect of the engine load*biodiesel percent has had a significant effect on the CO, HC and smoke emissions. However, this significant effect can only be seen for smoke emission and power for engine speed*EGR rate and EGR rate*biodiesel percent interactions, respectively. The other important point in the Table 10 is the value of the lack of fit for the proposed model for different parameters. As it can be seen in this table, F-value for lack of fit of the CO, NOx, HC, power, BSFC and smoke models were 1.09, 1.08, 1.06, 0.98, 0.31 and 1.08, respectively. This shows lack of fits of the proposed models have not been significant and this shows validity of the proposed models to predict these parameters.

Table 10. ANOVA table for emissions and performance characteristics

		CO	NOx	HC	Power	BSFC	Smoke
Source	**df**	**F-value**	**F-value**	**F-value**	**F-value**	**F-value**	**F-value**
Model	14	875.33*	372.11*	858.34*	1820.45*	17.67*	561.69*
A-Engine load	1	3689.29*	2727.50*	3236.73*	17530.36*	99.19*	4467.56*
B-Engine speed	1	239.88*	638.91*	560.74*	54.01*	102.35*	21.19*
C-EGR rate	1	0.0598 n.s.	158.59*	48.93*	2.26 n.s.	0.6371 n.s.	3.47 n.s.
D-Biodiesel percent	1	257.60*	163.28*	565.46*	2.36 n.s.	1.35 n.s.	71.00*
AB	1	837.82*	41.30*	755.50*	22.71*	51.86*	18.50*
AC	1	0.1498 n.s.	93.14*	26.64*	0.6654 n.s.	0.0586 n.s.	3.76 n.s.
AD	1	179.02*	1.36 n.s.	159.31*	8.70 n.s.	0.0004 n.s.	94.60*
BC	1	11.61 n.s.	0.7956 n.s.	9.44 n.s.	4.82 n.s.	0.0154 n.s.	19.05*
BD	1	0.0261 n.s.	0.0795 n.s.	2.64 n.s.	136.83*	0.0356 n.s.	2.64 n.s.
CD	1	1.28 n.s.	0.6245 n.s.	7.07 n.s.	1.73 n.s.	0.2623 n.s.	5.94 n.s.
A^2	1	899.24*	638.50*	862.71*	17.73*	0.0128 n.s.	635.72*
B^2	1	60.99*	303.49*	0.4811 n.s.	166.24*	38.57*	28.70*
C^2	1	2.67 n.s.	0.2663 n.s.	27.98*	0.8830 n.s.	0.7202 n.s.	0.3001 n.s.
D^2	1	5.91 n.s.	0.0001 n.s.	2.46 n.s.	10.42 n.s.	10.57 n.s.	8.18 n.s.
Residual	417						
Lack of Fit	129	1.09 n.s.	1.08 n.s.	1.06 n.s.	0.9782 n.s.	0.3195 n.s.	1.08 n.s.
Pure Error	288						
Cor Total	431						

*significant at 95% confidence interval, n.s.: Not significant at 95% confidence interval

Some statistical parameters of the created models are shown in the Table 11. As can be seen, the correlation coefficient (R^2) and adjusted correlation coefficient for all the models are within acceptable range and near to 1.

Table 11. Models Summary Statistics

Parameter	R-Squared	Adj R-Squared	Std. Dev.	C.V. %
CO	0.95	0.94	0.135856	19.6
HC	0.92	0.91	16.22	18.83
NOx	0.90	0.88	26.98	12.0
Power	0.96	0.95	0.21	9.92
BSFC	0.90	0.89	39.34	8.93
Smoke	0.95	0.94	1.13	15.12

This parameters show the adequacy and accuracy of the model to prediction of the actual value of the parameter [15]. The small value of differences between the R square and adjusted R square signifies that the models are adequate. As can be seen in this table, the highest adjusted correlation coefficient is for power ($R^2 = 0.95$). Different models were tested to estimate the engine parameters. According to the values of the correlation coefficient for each model, a quadratic model was selected automatically by the software to predict different engine parameters.

The mathematical models for power, CO, NO_x, HC, BSFC and smoke are presented in equations (2–6), respectively. The values in these equations for variables is as their units.

$$Power = (0.04*EL) + (-7.33e\text{-}3*ES) - 0.01*ER + (0.11*BP) + (4.73e\text{-}6*EL*ES) + 3.7e\text{-}5*EL*ER - 1.13e\text{-}4*EL*BP + 5.87e\text{-}6*ES*ER - 4.77e5*ES*BP + 6.69e5*ER*BP - 7.88e\text{-}5*(EL2) + 1.80e\text{-}6*ES2 + 5.90e\text{-}5*ER2 - 7.93e\text{-}5*BP2 + 7.08; \quad \text{Eq. 14}$$

$$CO = -0.08*EL - 4.3e\text{-}3*ES - 0.02*ER -1.36e\text{-}3*BP + 2.59e\text{-}5*EL*ES + 5.48e\text{-}5*EL*ER - 6.2e\text{-}4*EL*BP + 9.2e\text{-}6*ES*ER + 3.66e\text{-}6*ES*BP - 2.35e\text{-}4*ER*BP + 5.2e\text{-}4*EL^2 + 9.02e\text{-}7*ES^2 - 4.28e\text{-}5*ER^2 + 6.08e\text{-}4 *BP^2 + 5.55 \quad \text{Eq. 15}$$

$$NO_x = 17.21*EL - 1.61*ES + 0.51*ER + 3.73*BP - 1.55e\text{-}3*EL*ES - 0.05*EL*ER - 8.8e\text{-}3*EL*BP + 4.73e\text{-}5*ES*ER - 1.48e\text{-}3*ES*BP -0.07*ER*BP - 0.1*EL^2 + 3.86e\text{-}4*(ES^2) + 0.03*ER^2 - 0.07*BP^2 + 1514.26 \quad \text{Eq. 16}$$

$$HC = (-7.49*EL) + (.096*ES) - 0.17*ER - (0.18*BP) + (2.33e\text{-}3*EL*ES) + 0.01*EL*ER - 0.06*EL*BP + 4.69e\text{-}4*ES*ER - 1.61e\text{-}4*ES*BP - 0.03*ER*BP + 0.05*(EL^2) - 2.8e\text{-}5*ES^2 - 0.02*ER^2 + 0.09*BP^2 + 8.76 \quad \text{Eq. 17}$$

$$BSFC = (-28.02*EL) + (1.5*ES) + 2.92*ER - 2.12*BP + (2.5e\text{-}3*EL*ES) + 7.07e\text{-}3*EL*ER - 0.02*EL*BP - 2.02e\text{-}3*ES*ER - 1.6e\text{-}3*ES*BP - 4.17e\text{-}3*ER*BP + 0.19*(EL^2) - 3.38e\text{-}4*ES^2 + 0.03*ER^2 + 0.37*BP^2 - 631.23 \quad \text{Eq. 18}$$

$$Smoke = (-0.36*EL) + (-8.69e\text{-}3*ES) - 0.05*ER + (6.72e\text{-}3*BP) + (8.06e\text{-}5*EL*ES) - 1.45e\text{-}4*EL*ER - (3.8e\text{-}3*EL)*BP + 5.57e\text{-}5*ES*ER - 2.74e\text{-}6*ES*BP - 5.06e\text{-}4*ER*BP + 4.17e\text{-}3*(EL^2) + 1.90e\text{-}6*ES^2 - 1.04e\ 3*ER^2 + 5.2e\text{-}3*BP^2 + 15.09 \quad \text{Eq. 19}$$

In Figure 5, the correlation between the predicted and actual value of the parameters is shown. As can be seen in this figure, most of the data are close to the 45-degree line. This means that there is a high correlation between the actual and predicted values. According to the values of the correlation coefficient for different parameters, it can be stated that the developed model can be used to predict the engine performance and emission parameters.

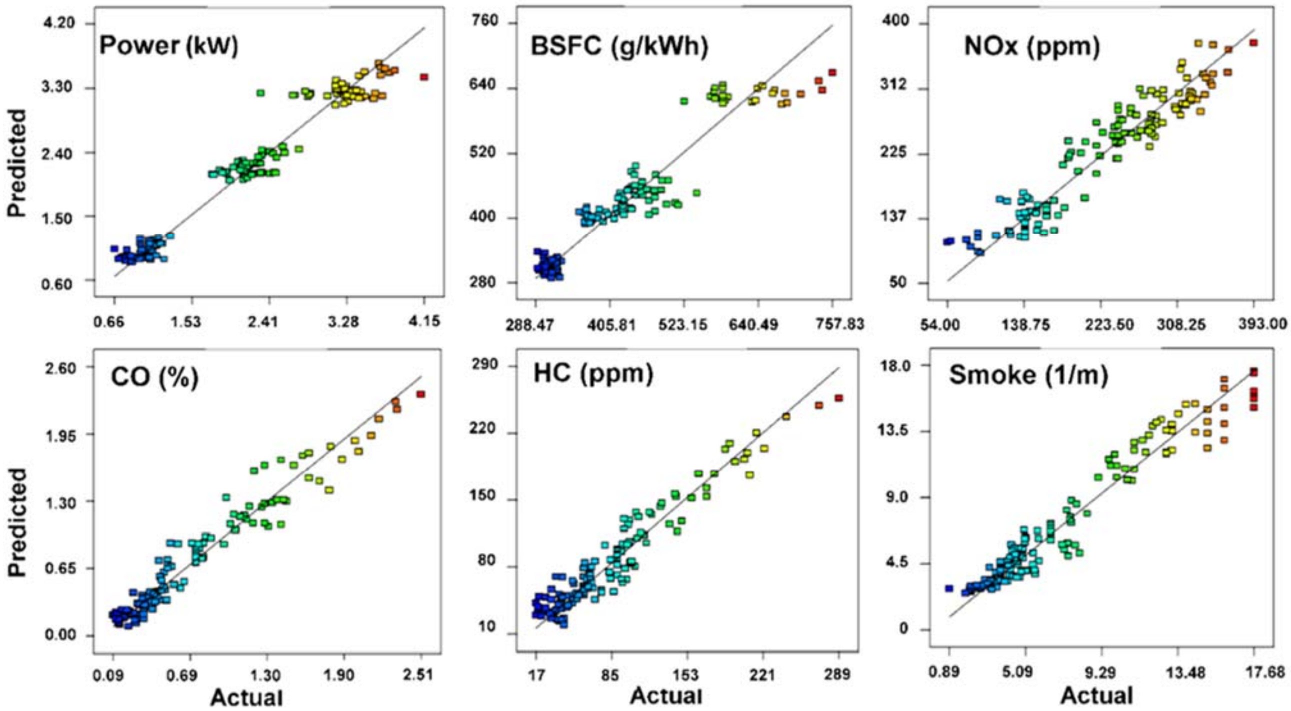

Figure 5. Experimental vs. predicted values of different responses

Optimization was conducted by application of the above proposed models. However, it may be necessary to discuss about the variation and effect of the considered factors on responses. As evaluation of the effects of the different parameters on the engine emissions and performance is not the main aim of this study, therefore in the following section, the effects of the different parameters on the engine emission and performance are presented briefly. There are many published work in this context. Therefore, in this study just the trends and reasons for them due to variation of effective parameters are discussed.

Co (carbon monoxide) emission is produced due to incomplete combustion inside the cylinder [11]. Effect of the different factors on CO emissions are shown in Figure 6 and Figure 7. As it can be seen in these figures, the amount of the CO emission decreased with increase in amount of the biodiesel blend fuel blended with diesel fuel. Oxygen content of the biodiesel fuel can be a reason for this reduction of the CO emission [33]. Application of the biodiesel fuel in the blends with diesel fuel has increased the oxygen content of the fuel-air mixture and then it has increased combustion quality [34]. The other noticeable trends are the reduction of the CO emission due to the rise of the engine speed although the highest rate of increase in amount of the CO emission is due to increase in the engine load [35]. In addition, higher dynamic viscosity of the biodiesel fuel is effective on the injection and atomization of the fuel and hence combustion quality [36]. The other important feature which can be seen in Figure 6 is a slight increase of the CO emission while using the EGR. This is in agreement with the results of the other studies [37].

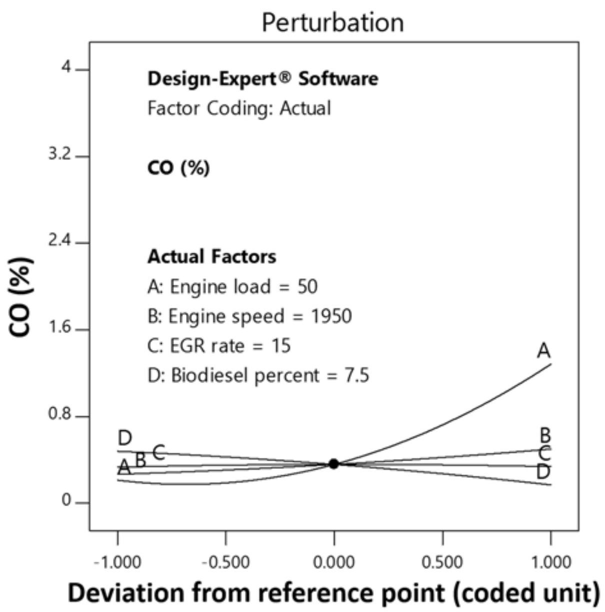

Figure 6. Perturbation graph showing the effect of each of the factors on CO emission (%)

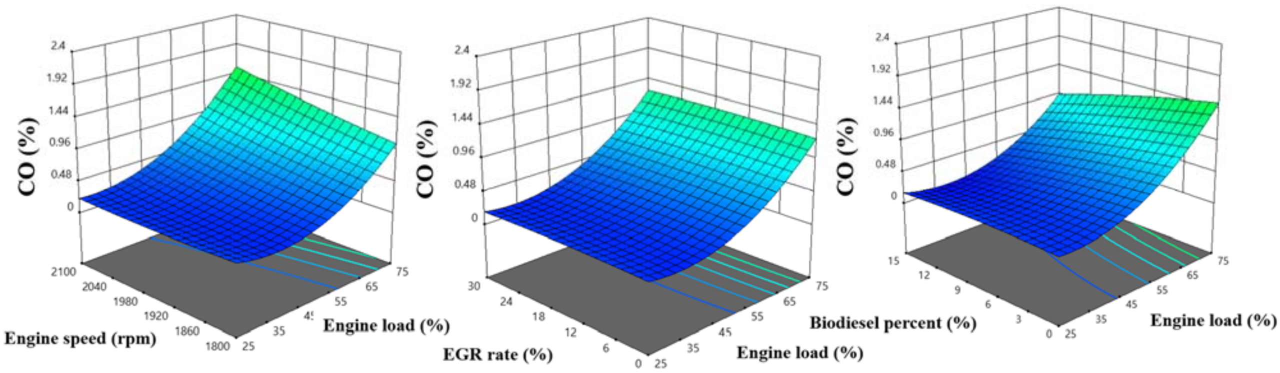

Figure 7. Variation of the CO emission vs. effective factor (statistically significant effect)

One of the most important characteristics of diesel engine is its NO_x emissions. These emissions are the disadvantage of this kind of engine and it is due to high temperature and lean combustion of the them [38]. Formation of the NO_x emission while combustion is mainly by Zeldovich mechanism [39]. Effect of the engine load, EGR rate, engine speed and biodiesel percent fuel blended with diesel fuel on NO_x emissions are shown in Figure 8 and Figure 9. As can be seen in this figures, NO_xemissions are increased initially then reaches a peak value and then falls gradually with increase in engine load and it is slightly decreased with increase in EGR rate. In addition, the NO_xemission decreased with increase in engine speed. Furthermore, the values of the NO_xemission decreased slightly with increase in biodiesel percent in the blend [40].

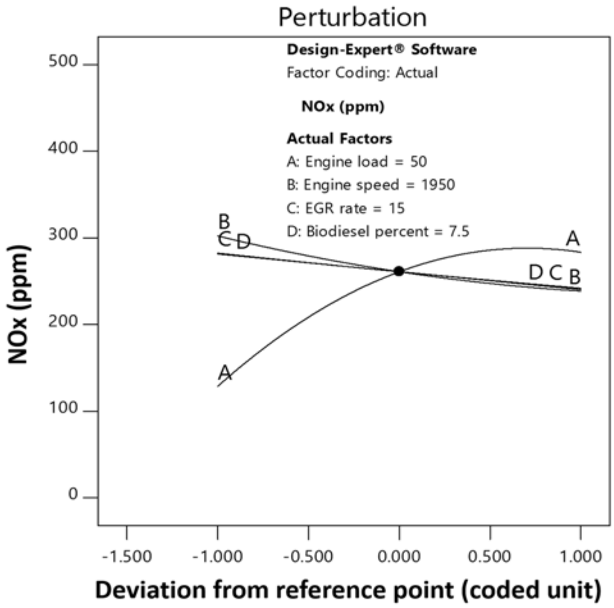

Figure 8. Perturbation graph showing the effect of each of the factors on NO_xemission (ppm)

In addition these variations of NOx emissions are shown in Figure 9. As it is discussed earlier, due to severe effect of the engine load on production of NOx emissions only interaction of the engine load*EGR rate ad engine load and engine speed have had a significant effect on NOx emissions. This is an advantage of the biodiesel fuel which is not significantly effective on NOx emissions.

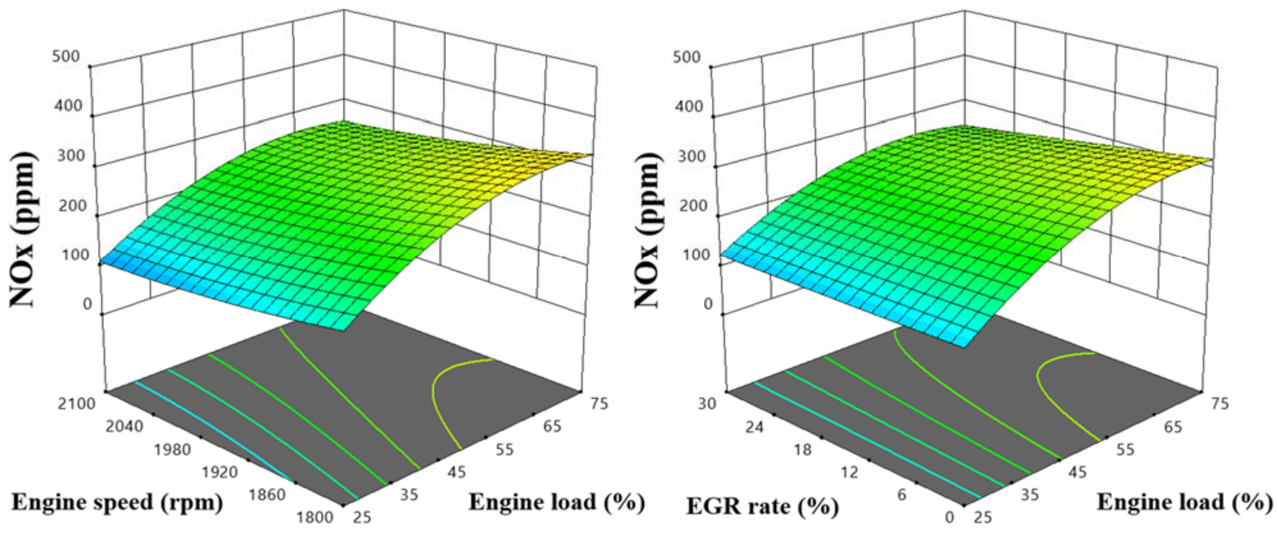

Figure 9. Variation of the NOx emissions vs. effective factor (statistically significant effect)

Effect of the EGR rate, engine load, engine speed and biodiesel fuel percent blended with diesel fuel on HC emission of the engine is shown in Figure 10. As can be seen at first glance, the HC emission increased with increase of the engine load. This increase is more noticeable in higher engine load. The other important feature is slight increase of the HC emission with increase of the EGR rate. The increase of the HC emission due to application of EGR is also stated by other researchers. Various reasons such as lower oxygen content and temperature inside the cylinder are stated as the reasons for this increase of HC emissions due to application of the EGR [41, 42]. Although application of the EGR has increased the amount of HC emissions, using the biodiesel fuel has compensated this increase by its oxygen content and improvement of the combustion quality [42]. Similar to the CO emission, HC emission production is also dependent on the combustion quality. Therefore, any parameters which reduce the quality of combustion can be the reason for increase in HC emission formation. As can be seen in this figure, the amount of HC emission was decreased gradually with increase in biodiesel fuel value. This can be explained by the biodiesel characteristics such as the reasons stated for the variation of the CO emission. Higher dynamic viscosity and density of the biodiesel fuel are reasons for the increase of the HC emission and its oxygen content is a reason for reduction of the HC emission. Hence, when using low amounts of the biodiesel fuel (B5), the dynamic viscosity and density of the biodiesel fuel are more effective than its oxygen content on the combustion quality especially in lower engine loads. This is in accordance with the results reported by other researchers [36, 43, 44].

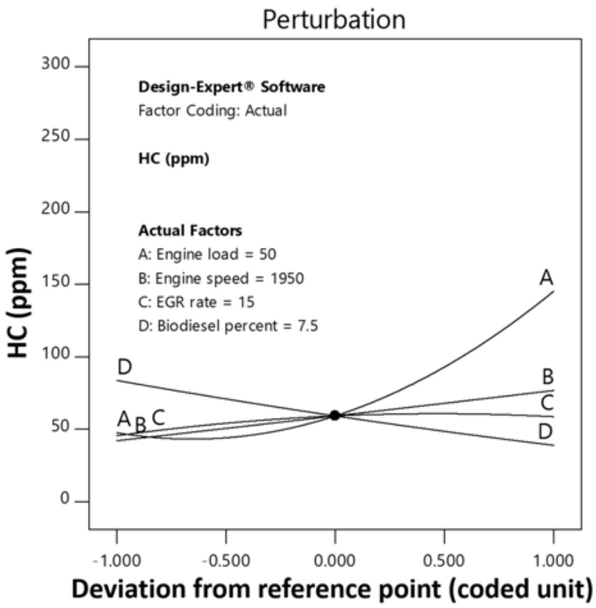

Figure 10. Perturbation graph showing the effect of each of the factors on HC emission (ppm)

According to perturbation graph for HC emission (Figure 10) and ANOVA table, values of HC emission vs. effective parameter are presented in surface graph as can be seen in the Figure 11. As it can be seen in the Figure 11, HC emission has increased with increase in engine load however the rate of this rise is lower in higher biodiesel percent, lower EGR rate and lower engine speed. These can be due to higher oxygen content when biodiesel percent is increased, lower reduction of inside cylinder oxygen content due to EGR application and increase of residual time (the time which air-fuel are in cylinder) in lower engine speed. All of these three phenomenaa can increase combustion quality and hence decrease HC formation.

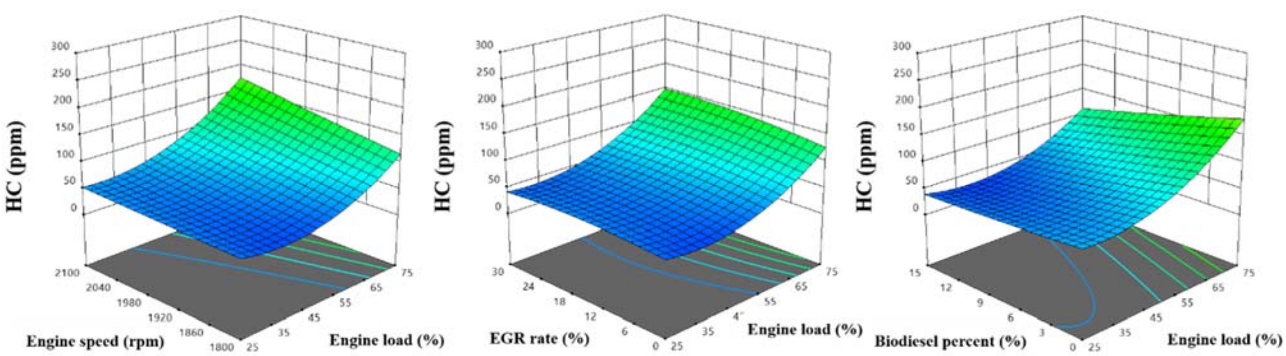

Figure 11. Variation of the HC emissions vs. effective factor (statistically significant effect)

Figure 12 (Left) shows the effects of the EGR rate, engine load, engine speed and biodiesel percent blended with diesel fuel on engine power. As can be seen in these figure, engine power is increased with increase in engine load and is decreased slightly with increase in EGR rate. It is important to note that the power of engine is decreased slightly with increase in biodiesel amount in fuel blend. This trend was also reported by other researchers [45, 46]. Higher dynamic viscosity of the biodiesel fuel in the higher engine load reduced the quality of the injection and atomization and hence combustion quality. The other reason for reduction of the engine power while using the biodiesel fuel can be its lower LHV than diesel fuel [47]. Although, power of the engine has changed due to variation in the amount of the EGR rate and biodiesel percent, they were not significantly effective on engine power. Then surface graph for the power of the engine is only presented for power of the engine vs. engine load and engine speed (Figure 12: Right).

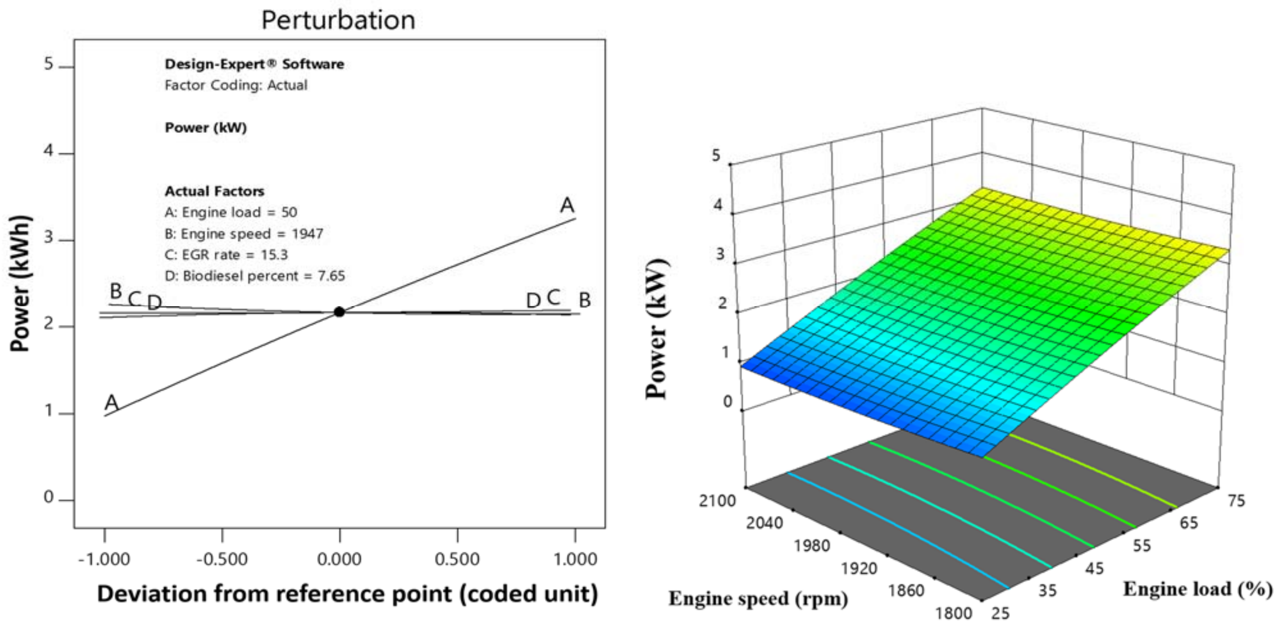

Figure 12. Left: Perturbation graph showing the effect of each of the factors on engine power (kW), Right: Variation of the engine power emissions vs. effective factor (statistically significant effect)

Figure 13 (Left) compares the BSFC (g/kWh) in various engine loads, EGR rates, engine speeds and biodiesel fuel percent blended with diesel fuel. As can be seen in this figure, the amount of BSFC is very high in lower engine loads (25%) and it is decreased with increase in engine load. This is due to higher frictional power of the engine in lower engine loads. The other noticeable point is the increase of the BSFC for higher percent of biodiesel fuel. The increase or reduction of the BSFC is reversely linked to the engine power. Thus, any reason for the reduction of the engine power will increase the engine BSFC and vice versa. The lower LHV and higher dynamic viscosity of the biodiesel fuel which have been stated as reasons for the reduction of the engine power while using the biodiesel fuel are reasons for the increase of the BSFC [45]. The other important point is almost steady state for power due to variation of the EGR rate. In addition, variation of engine BSFC vs. engine load and engine speed is shown in the Figure 13 (Right). As it can be seen in this figure and discussion about the BSFC in perturbation graph, BSFC is slightly decreased initially then it is significantly increased with increase in engine load.

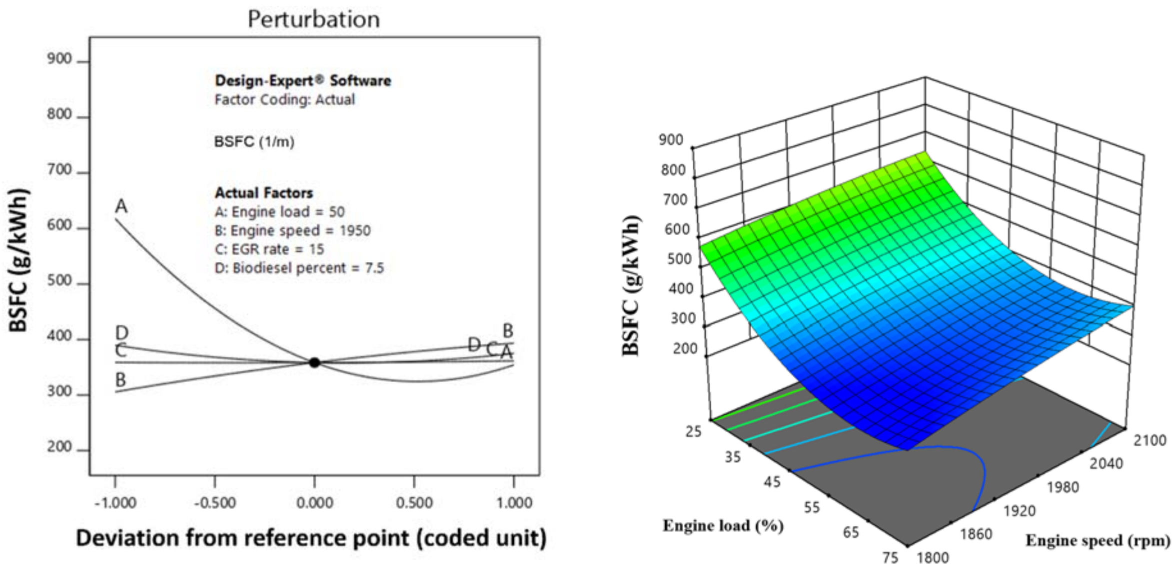

Figure 13. Left: Perturbation graph showing the effect of each of the factors on engine BSFC (g/kWh), Right: Variation of the engine BSFC vs. effective factor (statistically significant effect)

The amount of smoke emission of the engine in term of K value in various engine loads and speeds, EGR rates and biodiesel fuel percent blended with diesel fuel are shown in Figure 14 (Left). As can be seen in this figure, the smoke emission increased with increase in engine load significantly. The other significant feature is reduction of smoke emission of the engine with increase in biodiesel fuel percent in biodiesel-diesel fuel blend. Reduction of the K value while using the biodiesel fuel is due to its oxygen content which improves combustion quality [48] and its lower value of carbon atoms compared to diesel fuel [49]. In addition, the application of the EGR reduces temperature and oxygen inside the cylinder and then it has increased formation of the smoke emission and K value [35]. Looking to Figure 14 (Right), it is shown that smoke is increased gradually up to medium load (50%) and then it raised significantly and peaked at 75% engine load. The other characteristics was not significantly effective on smoke emission of the engine.

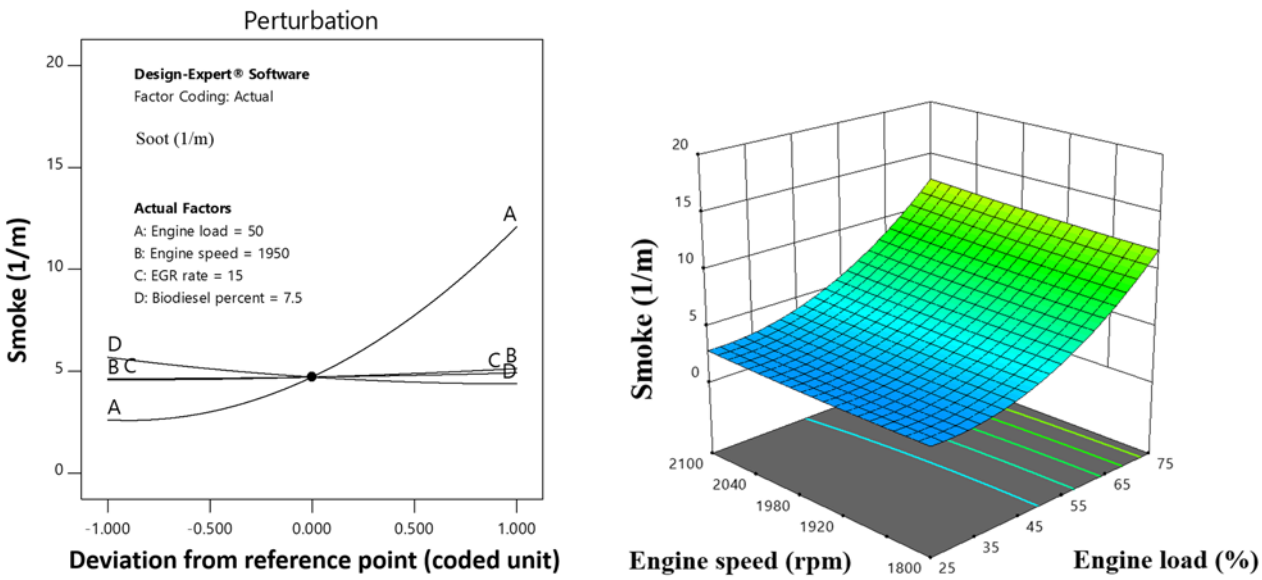

Figure 14. Left: Perturbation graph showing the effect of each of the factors on smoke (1/m),

Right: Variation of the smoke emission vs. effective factor (statistically significant effect)

3.2 Genetic algorithm optimization result

The mathematical method which was created was used as the model to optimize with multi objective optimization genetic algorithm. The optimization was run in the Matlab R2014b software and terminated after 224 generations. In multi objective optimization, changing of each parameter will change the other objective function, which is the principle of the Pareto optimal solution [23] as shown in Table 11.

There are different optimal points with different values for different variables. This is because when the software tries to optimize a variable, the values of other variables will also change. The result of the optimization with the genetic algorithm indicates that when the engine is working in the 2125 rpm at 40% engine load, in order to have the best engine operating conditions: the lowest amount of pollutants and the highest performance, the EGR ratio is 6.6% and the biodiesel used is 10.9%. In this case, the amount of CO, NO_x, UHC, power, BSFC and smoke were 0.19 (%), 215.19 (ppm), 49.01 (ppm), 0.32 (kWh), 391.83 (g/kWh) and 3.67 (1/m).

As it can be seen from these results, it seems that lower EGR ratio has a better effect on engine performance and emissions characteristics. However, EGR rate in this study was not significantly effective on performance of the engine (BSFC and power) but its increase can rise and smoke emissions while reduces NOx emissions. Then, optimization for this factor was mainly according to the engine emissions. In addition, it is interesting that the amount of the proposed optimal amount of the biodiesel percent to have optimal point for engine performance characteristics and emissions was about 11% which is a medium amount of biodiesel percent which were considered in this study (range from 5 to 15%). Approximately same amount of biodiesel blended with diesel fuel is recommended for soybean biodiesel blended with diesel fuel and water [50]. However the factors and parameters of their study were different. Although, using of biodiesel had decreased most of the emissions, it had a negative effect on engine performance and reduces engine power and increased BSFC. Then the optimized point is close to 10 percent which provides positive effect of the biodiesel fuel (due to its oxygen content) and compensates its drawbacks (due to its lower LHV, higher dynamic viscosity and density compared to diesel fuel).

Furthermore, approximately medium engine load (40%) and engine speed were the appropriate engine working point to have a proper trade-off between engine performance and emissions. This is in agreement with the result of Singh et al., [51]. They optimized engine injection timing, injection pressure, biodiesel percent blended with diesel fuel and engine load and found that the 49% engine load was the optimal working engine load in term of lowest engine emission and performance. As it is discussed in previous sections, with increase in engine load power will increase but different engine emissions such as HC emission increases, too. This means that the optimization methods have well considered the different effect of variations in the amount of the factors on responses.

The proposed value by NSGA-II method were tested experimentally on the engine and the results showed that the recommended optimal amount was close to the actual value. In addition, some characteristics of NSGA-II algorithm are shown in Figure 15. This figure includes the number of individuals vs. score, distance of individuals vs. individuals, number of individuals vs. rank of individuals, average spread in each iteration and average distance between individuals in each iteration for multi-objective optimization of the engine working conditions. The average distance among the individuals became slightly closer together and the average spread of the individual decreased with increase in iterations. These show that the proposed NSGA-II which has been applied here is capable of optimizing the engine parameters [52]. In Figure 15, the PF set is determined by the optimization process. These PF set shows the trade-off between the different objectives. The proposed values for the optimized engine operating condition point were applied and experimental results of the responses were recorded. These values, experimental and numerical, are compared in table 12. As it can be seen in this table, the maximum and minimum differences between predicted values and experimental values are 9.52 and 0.88%. These are in an acceptable range for engine optimization with high complexity of parameters and their interactions.

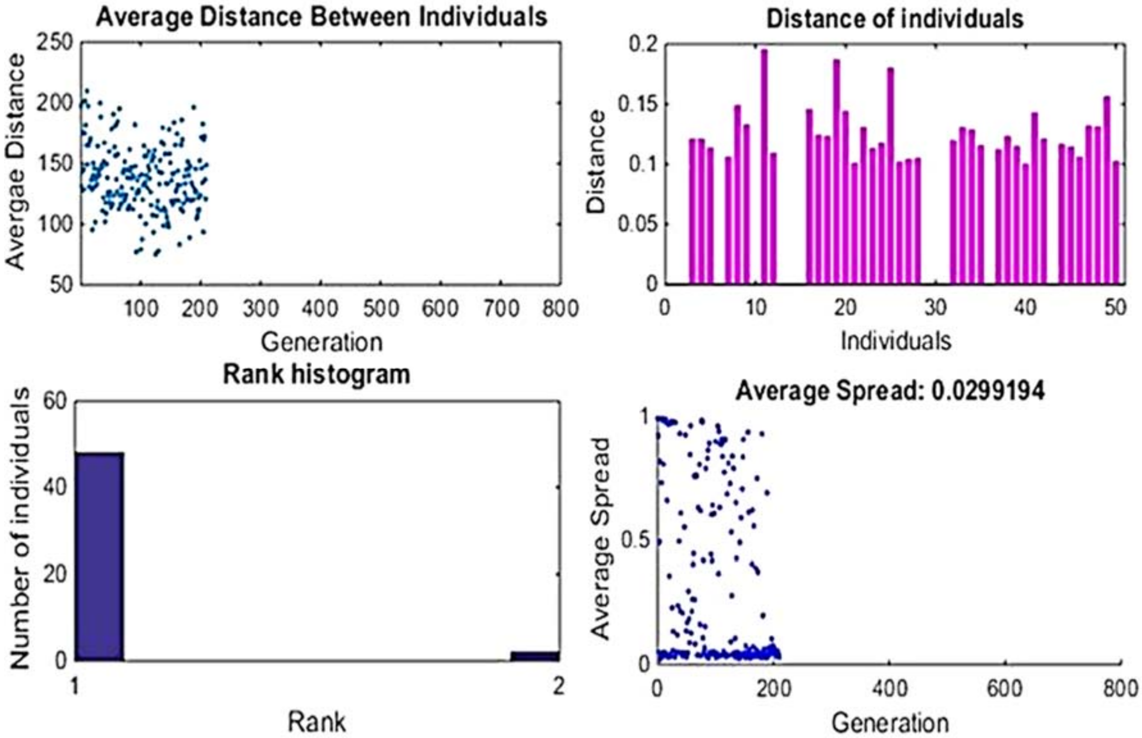

Figure 15. The results of NSGA-II optimization method

Table 12. Optimized conditions by NSGA-II method

Parameter	Unit	Experimental value	Predicted value	Error percent
CO	ppm	0.21	0.19	9.52
NO_x	ppm	214.02	215.91	0.88
UHC	ppm	52.24	49.01	6.18
Power	kWh	0.40	0.32	5.26
BSFC	g/kWh	390.83	391.83	0.26
Smoke	1/m	3.22	3.67	6.98

4 Conclusion

In this study an EGR system for a small single cylinder engine was developed and implemented. Then, the engine was run under different test conditions. The factors which were considered were engine load, engine speed, EGR rate and biodiesel blend percent in fuel blend. The performance and emission parameters of the engine have been measured and a mathematical method to correlate these parameters to the considered factor was developed. Finally, an NSGA-II has been applied to optimize the engine factor according to trade-off between considered engine performance and emission parameters. According to the results of the study, the following conclusions can be drawn:

1. HC and CO emission have increased and NO_x emissions have decreased due to application of the EGR.
2. Biodiesel fuel is effective on engine emission and performance in different ways. It increases BSFC and reduces HC and CO emissions.
3. The mathematical model can be applied to predict the engine performance and emission parameters. The adjusted R-square for the CO, HC, $\mathrm{NO_x}$, power, BSFC and smoke models were 0.94, 0.91, 0.88, 0.95, 0.89 and 0.94, respectively.
4. The NSGA-II optimization method shows a good capability to optimize the problem of the engine parameters optimization. This method can be used as a good solution to the multi objective optimization problem of the engine parameter optimization.

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