Applied Energy 330 (2023) 120349

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Optimizing the effect of a mixture of light naphtha, diesel and gasoline fuels on engine performance and emission values on an HCCI engine

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HIGHLIGHTS

- The highest desirability value of 97.68% was found with created model by using RSM.
- Optimum fuel mixtures, injection duration and RPM for HCCI were found using RSM.
- Gasoline-naphtha mixture gave better results than diesel-naphtha mixture.

ARTICLE INFO

Keywords:
Naphtha
Diesel
Gasoline
Response surface method
Performance
Emission

ABSTRACT

In this study, the effects of using naphtha-diesel and naphtha-gasoline fuel mixtures were investigated using experimental and statistical methods on homogeneous charge compression ignition (HCCI) engine performance and emission values. Response Surface Method (RSM) was used as a statistical method. Fifty-two experiments were specified in the experimental set created by the RSM central composite design method, which were carried out under the same conditions. As a result of the experimental study, effective torque, brake-specific fuel consumption (BSFC), CO, HC, and NO emissions values were examined. The experimental results extrapolation equations were obtained with the RSM method, and the contour graphs were drawn. Optimum input parameters were determined to obtain the targeted output parameters. After the optimization, it was seen that the usage of gasoline fuel gave better results than diesel. Among the optimum input parameters, the gasoline ratio was determined as 0.297 %, the injection duration was 11.97 ms, and the engine speed was 1325.21 rpm. Effective torque was 12.79 Nm, BSFC value was 589.79 g/kWh, CO emission value was 0.11 %, HC emission value was 519.86 ppm, and NO emission value was 0 ppm depending on the optimum input parameters.

1. Introduction

Today, with the increase in the number of vehicles using internal combustion engines, serious rise has been observed in environmental pollution problems [1]. With the aim of reducing these problems, updates are made on emission regulations every day [2–4]. Due to the mandatory of emission updates, it is seen as an indispensable condition for engine manufacturers to continue research and development studies on internal combustion engines [5,6]. At the beginning of these studies are the reduction of engine fuel consumption values and emission values [7,8]. One of the main objectives of the production of engines suitable for this purpose is to ensure homogeneous combustion in the cylinder. In order to achieve this goal in the best way, the concept of HCCI engine has come to the fore and its research continues [9,10].

High homogeneity is achieved in the cylinder with the HCCI combustion mode. However, significant improvements are observed in NOx emission and specific fuel consumption values [11,12]. High thermal efficiency is obtained in HCCI combustion mode [13,14]. The engine operating with the HCCI combustion principle has disadvantages such as low operating speed range, inability to reach high torque values, difficulty in controlling the combustion timing, and excess HC and CO emissions [15–18]. In order to overcome these problems, studies are

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https://doi.org/10.1016/j.apenergy.2022.120349
Received 13 October 2022; Received in revised form 6 November 2022; Accepted 14 November 2022
Available online 26 November 2022

carried out on methods such as increasing of intake manifold pressure, intake air temperature, compression ratio, changing valve timing, using alternative fuels and (exhaust gas recirculation) EGR application [19–22].

Maurya et al., examined the effects of thermal efficiency, combustion parameters, and emissions by changing intake air temperature in an HCCI engine operating at different air excess coefficients with ethanol. It has been reported that the highest thermal efficiency is obtained as 44.78 % in conditions where air excess coefficient (lambda) is 2.5 and intake air temperature is 393 K. It has been observed that NO_x emissions are at minimum levels under all operating conditions. However, it was observed that CO emissions increased whereas HC emissions decreased with increasing lambda [23]. Kim and Lee, studied EGR application to control HCCI combustion and their effects on partial HCCI engines by premixing gasoline, diesel and *n*-heptane fuels [24]. Gharehghani, conducted experimental research to determine the operating range of the HCCI engine working with ethanol, methanol, and natural gas fuels. It has been reported that natural gas fuel is suitable for HCCI combustion at high intake air temperatures. It has been reported that ethanol and methanol fuels have the potential to be used in HCCI engines with lean and rich mixtures at low intake air temperatures [25]. Hou et al., investigated the effect of 10 % to 60 % adding ethanol, methanol, and methyl-tertiary-butyl ether (MTBE) on the HCCI combustion phase and efficiency. It has been reported that the addition of high-octane fuel when compared to 100 % *n*-heptane, retards the HCCI combustion phase and increases the heat release rate. It was observed that as the ratio of ethanol, methanol, and MTBE in the fuel mixtures increased, the start of combustion was similarly delayed and the maximum heat release shifted to the region close to the TDC [26]. Yılmaz investigated the effects of diesel and fusel oil fuel mixtures on combustion properties, engine performance, and emissions. The use of fusel oil mixtures gave similar in-cylinder pressure values with 100 % diesel at full load conditions. The specific fuel consumption increased with the increase of fusel oil in fuel mixtures [27].

Structural and operating parameters of HCCI engine directly affect engine performance, efficiency and emission values [28,29]. Experimental studies are required to examine these effects. With the increase of input parameter values, time and high-cost problems arise. In addition, the analysis of the results becomes difficult and there is a need for optimization. For this reason, statistical methods are used. The experimental study process is managed with these statistical methods, providing advantages in terms of both cost and time, as well as analyzing the results and input parameter values [30–32]. Some methods such as artificial neural networks (ANN), taguchi, and Response Surface Method (RSM) can be used in experimental studies on internal combustion engines [33–35]. In addition to the automotive sector, the RSM method has a wide range of research field such as energy, chemistry, food industry, engineering, production, and system design [36–38]. High-accuracy results are obtained in studies on the internal combustion engine with the RSM method [39]. For this reason, the RSM method was used in this study.

In the literature, there are studies on naphtha-gasoline and naphtha-diesel fuel mixtures and HCCI engines. These studies have been examined in a detail. It is seen that there is no study related to the effects of the use of naphtha-gasoline and naphtha-diesel fuel mixtures and optimization on the HCCI engine output parameters. In this study, the performance and emission values of naphtha-gasoline and naphtha-diesel fuel mixtures were investigated by experimental and statistical methods and the optimization of the input parameters was carried out in a test setup with an HCCI engine. Performance and emission values of naphtha-gasoline and naphtha-diesel fuel mixtures in HCCI mode were obtained experimentally for 52 different situations determined by the RSM technique. Obtained experimental data were used as RSM input parameters. Optimum conditions were determined according to the input parameters utilized in HCCI mode using the RSM technique.

2. Material and method

This study researched the effects of different fuel mixtures on the HCCI engine performance and emissions. Experimental and statistical methods were used in the research process. The test was conducted on the HCCI engine test system. Two different types of fuel mixtures, naphtha-diesel, and naphtha-gasoline, were formed in different concentrations. Statistical evaluation of experimental data was made with the response surface method.

2.1. Test setup

The experimental part of this study was carried out in the HCCI engine test setup environment. The schematic representation of the test setup is given in Fig. 1.

There is a four-stroke, Lombardini brand, LDA 450 model test engine on the test setup. Test engine technical specifications are given in Table 1.

The CUSSONS brand test device, which can absorb 10 kW at 4000 rpm as seen in Fig. 1. The speed control can be performed via the control panel on the test setup, and the engine load and exhaust temperature values can also be measured.

Light naphtha, gasoline, and diesel fuels were used in the experiments. The characteristic properties of the fuels used in the experiment are given in Table 2. Light naphtha diesel and gasoline blends were injected into the inlet manifold by an electronic injection system adapted to the test engine. Start and end of injection were controlled electronically depending on the crankshaft angle with high precision. Crankshaft position information is provided with the help of an encoder.

BOSCH/BEA350 brand emission device was used for measurement of CO, CO_2, HC, O_2 and lambda values during the test. The measurement range and tolerance values of the emission measuring device are given in Table 3.

Simultaneous measurement of engine torque and speed was carried out in the experiments. Engine power output is calculated by substituting the engine torque and speed values in Equation (1) [40]. In Equation (1), T and n represent engine torque and engine speed respectively.

$$Pe = \frac{T.n}{9549} \tag{1}$$

The net work value is calculated by integrating the in-cylinder pressure (P) with respect to stroke volume. This process is given in Equation 2 [41].

$$W_{net} = \int_{CA0^o}^{CA180^o} PdV \tag{2}$$

The BSFC value is calculated by Equation (3). In the equation, m_f indicates the amount of fuel consumed in the test process, and W_{net} indicates the effective amount of work [42].

$$BSFC = \frac{m_f}{W_{net}} \tag{3}$$

2.2. RSM

Increasing the number of input and response parameters and high cost and time issues can be a problem in experimental studies. In such experimental studies, improvements are needed in terms of cost and time. These improvements can be achieved with hybrid methods created by using experimental and statistical methods together. In addition to providing time and cost improvements with hybrid methods, the optimization of the parameters can also be achieved [43].

In this study, the effect of different fuel mixtures on the HCCI engine on performance and emission values under different input parameters and the optimization were carried out using the response surface

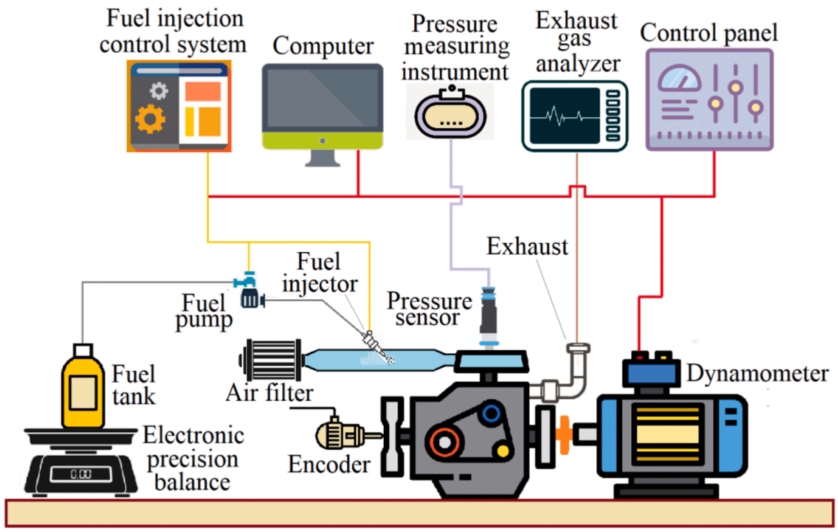

Fig. 1. Representation schematic of the engine test unit.

Table 1
Test engine properties.

Parameter [unit]	Value
Number of cylinders	1
Bore [mm] × Stroke [mm]	85 × 80
Compression ratio	17.5/1
Injector spray pressure [bar]	183
Valve timing [°BTDC]	25
Maximum torque [Nm-rpm]	28.5–1700
Maximum power [kW]	5.5

Table 2
Test fuels properties.

	Light naphtha	Gasoline	Diesel
Auto ignition temperature [°C]	280–470	246–280	210
Density [kg/cm^3 at 20 °C]	72	730	840
Boiling point [K]	32–402	210	453–653
Research octane number	64.5	95	–
Cetane number	–	–	60
Low heating value [kJ/kg]	48,500	46,400	42,550

Table 3
Gas analyser measurement range and calculated uncertainties.

Measurement	Range	Resolution	Accuracy %
CO [% vol.]	0– 10	0.001	±0.01
CO_2 [% vol.]	0– 18	0.01	±0.05
HC [ppm]	0 – 9999	1	±0.01
O_2 [% vol.]	0– 22	0.01	±0.04
Lambda	0.5 – 9.999	0.001	±0.0001
NO [ppm]	0 – 5000	≤ 1	±0.1
Smoke Opacity [%]	0–100	0.1	±0.1

method (RSM). Design Expert 11 software environment was used for The RSM method. Design of the experimental set implemented with Face-centered composite design (FCCD) [44]. By using ANOVA (analysis of variance), optimum input parameters and response parameter values depending on these parameters were determined.

Naphtha-diesel and naphtha-gasoline fuel types were indicated as categorical input parameters. Mixed fuel concentration ratios (0 to 20 %), fuel injection duration (11 ms-12 ms), and engine speed (1100 rpm to 1500 rpm) were defined as numerical input parameters. The experimental set created as a result of the FCCD design has a total of 52 experimental contents. Experiments of these data sets were carried out in the same test environment, under equal conditions. The obtained data were entered into the Design expert 11 environments. Numerical input parameters limit and alpha values are given in Table 4.

The cubic model is used for all parameters [45]. The equation of the Cubic model is given in equation (9) [46]. In the equation, x_i is the input variable $\hat{y}$ is the response, β_0 is the constant coefficient,β_i, β_{ii} and β_{iii} is the regression coefficient, and β_{ij} is the Cubic coefficient. If the p-value is<0.05 in the ANOVA created for the Cubic model, it means that the variable parameter is "significant" [47,48]. If the P-value is greater than 0.05, it can be considered "not-significant" [49]. The size of the F-value in the ANOVA table shows how much that variable parameter affects the response parameters.

$$\hat{y} = \beta_0 + \sum_{i=1}^{k}\beta_i x_i + \sum_{i=1}^{k}\beta_{ii}x_i^2 + \sum_{i=1}^{k}\beta_{iii}x_i^3 + \sum_{i=1}^{k}\sum_{j=1,i<j}^{k}\beta_{ij}x_i x_j \tag{4}$$

Another parameter that should be mentioned in the RSM method is the R2 and adjusted R2 values. R2 value greater than 0.9 improves the similarity between the actual data and the estimated data [50]. In addition, the difference between the R2 value and the adjusted R2 value is required to be < 0.2 [51].

3. Results and discussion

By providing the contour visuals and model equations independently, the effective torque, BSFC, CO, HC, and NO emission values derived depending on the input parameters have been examined. The results of the optimization carried out with the RSM method and the desirability ratio values are given.

3.1. Performance and emission results depending on input parameters

The counter graphs of the effective torque value depending on the variable parameters of fuel injection time, naphtha-diesel and naphtha-

Table 4
Independent variables levels.

Independent variables	Codes	Levels				
		−α	−1	0	+1	−α
Diesel/gasoline ratio [%]	A	0	0	10	20	20
Fuel injection duration [ms]	B	11	11	11.5	12	1600
Engine speed [rpm]	C	1100	1100	1300	1500	1500

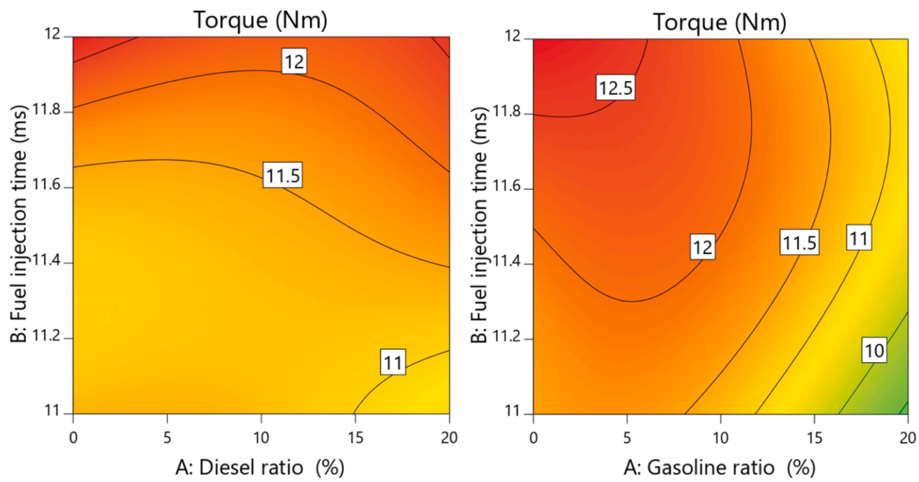

Fig. 2. The effect of fuel injection duration and diesel-gasoline ratio in light naphtha on effective torque.

gasoline fuel concentration ratio are given in Fig. 2. With an increase in injection time for both fuel types, an increase in the effective torque value was observed. This increase is caused by the cylinder being filled with more fuel while maintaining a consistent injection pressure value for a longer injection duration. Increases in the in-cylinder pressure and effective torque are directly proportional to the amount of fuel injected into the cylinder. The increase in the diesel ratio in the naphtha-diesel blended fuel did not result in a consistent change in the effective torque value. The effective torque value decreased with an increase in diesel fuel concentration at low injection times. With an increase in the gasoline ratio in the naphtha-petrol fuel, a steady decline in the effective torque value was seen. This is assumed to be caused by reduced calorific value of gasoline. The amount of heat that will be produced by the burning of the fuel put into the cylinder will grow as the concentration of gasoline with lower calorific value does, and this will happen simultaneously with a decrease in pressure value. It is possible to see a drop in torque value in this situation.

The counter graphs of the effective torque value depending on the variable parameters of engine speed, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 3. When using a naphtha-diesel fuel mixture, an increase in the effective torque value is observed as the engine speed increases.

ANOVA results for the effective torque value are given in Table 5. According to the ANOVA result of the effective torque response parameter, it was concluded that the model was significant. When the input parameters were evaluated separately, it was observed that the diesel/gasoline ratio in the light naphtha fuel variable parameter was not significant, while the fuel injection duration and engine speed variable parameters were significant. When the P-value value is examined, it is seen that the engine speed variable parameter has the highest effect on the effective torque response parameter.

The model created to calculate the effective torque value depending on different variable parameters is given in Equation (5).

$$\begin{aligned} T_{eff}\ (Nm) = {} & 11.687 - 0.219A + 0.444B + 1.584C + 0.331D + 0.115AB - 1.137AC - 0.491AD + 0.384BC \\ & -0.0312BD - 0.740625CD - 0.311A^2 + 0.0266B^{22} - 1.214C^2 - 0.0281ABC - 0.0125ABD - 0.872ACD \\ & +0.1125BCD + 0.397A^2B - 1.772A^2C - 0.411A^2D - 0.525AB^2 - 0.349B^2D - 0.164C^2D \end{aligned} \tag{5}$$

The counter graphs of the BSFC value depending on the fuel injection duration, naphtha-diesel and naphtha-gasoline fuel concentration ratio variable parameters are given in Fig. 4. In case of an increase in diesel and gasoline fuel concentrations, a slight increase in BSFC values is expected due to the decrease in the lower heating values of the blended fuel. It was observed that there was a significant increase in the BSFC value with the increase of the blend fuel gasoline concentration ratio parameter. No regular increase or decrease was observed in BSFC values depending on fuel injection duration and blended fuel diesel concentration ratio parameters.

The counter graphs of the BSFC value depending on the variable parameters of engine speed, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 5. An improvement in BSFC value was observed with the increase in diesel fuel ratio. With the increase of gasoline concentration, BSFC value improved at low engine speeds, while an increase was observed at high engine speeds.

ANOVA results for the BSFC value were given in Table 6. According to the ANOVA result of the BSFC response parameter, it was concluded that the model was significant. When the input parameters are evaluated separately, it has been observed that the diesel/gasoline ratio in light naphtha and fuel injection duration are not significant, while the fuel injection duration parameters are significant. When the P-value value were examined, it was seen that the engine speed has the highest effect on the engine speed response parameter.

The model created to estimate the BSFC value depending on the HCCI engine variable parameters is given in Equation (6).

$$\begin{aligned} BSFC\ (g/kWh) = {} & 632.092 + 8.433A + 3.212B - 112.551C - 28.97D - 44.196AB + 129.786AC + 74.674AD \\ & -55.888BC - 25.754BD + 96.101CD + 38.385A^2 + 13.776B^2 + 108.148C^2 - 34.56ABC - 33.741ABD \\ & +113.664ACD - 42.802BCD - 59.520431621715A^2B + 183.716A^2C + 49.584A^2D + 90.084AB^2 \\ & +45.374B^2D + 22.2C^2D \end{aligned} \tag{6}$$

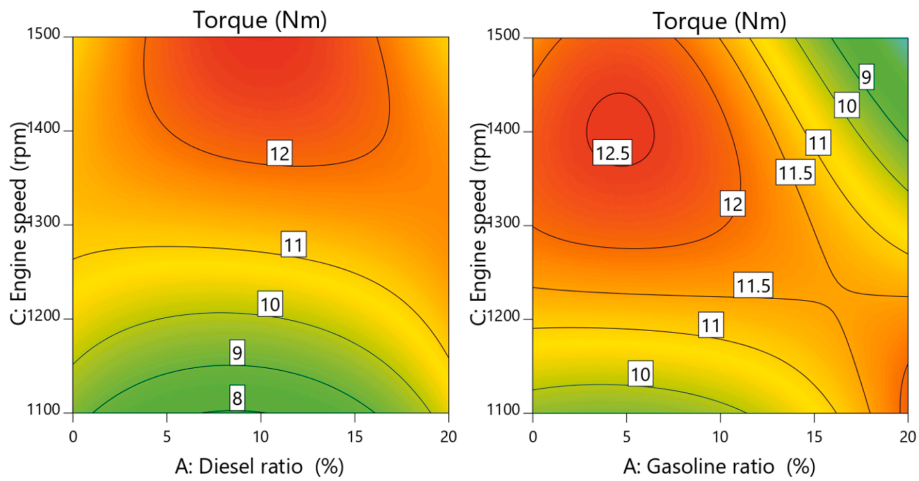

Fig. 3. The effect of engine speed and diesel-gasoline ratio in light naphtha on effective torque.

CO emission comes out as a result of incomplete combustion of fuel. It is expected that an increase in CO emissions will be observed with the increase of fuel injection duration. However, this situation does not appear to be stable. It is thought that the reason of this situation is to occur combustion more effectively in the cylinder with the increase in the injection duration. The counter graphs of the CO emission value depending on the variable parameters of fuel injection duration, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 6. It has been observed that the CO emission value is at least 0.012 % in the use of naphtha-diesel fuel and 0.085 in the use of naphtha-CO fuel. It is seen that the increase in gasoline ratio in naphtha fuel causes more CO increases than diesel fuel.

$$
\begin{aligned}
CO\ (\%) = {} & 0.107 + 0.0275A + 0.00175B + 0.0423C - 0.0123D - 0.00312AB + 0.0628AC + 0.0345AD \\
& -0.057BC - 0.00925BD + 0.0339CD + 0.0693A^2 + 0.036B^2 + 0.0459C^2 + 0.00475ABC - 0.01ABD \\
& +0.0445ACD - 0.00462BCD - 0.0591A^2B + 0.1A^2C + 0.0269A^2D + 0.0415AB^2 + 0.00821B^2D \\
& +0.0166C^2D
\end{aligned}
\tag{7}
$$

As engine speed increases, the time is shortened in order to mix fuel and air in the cylinder. However, the formation of a homogeneous mixture is somewhat prevented. The counter graphs of the CO emission value depending on the variable parameters of engine speed, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 7. As the engine speed increased, the CO value increased in both fuel mixtures. Depending on the engine speed, more CO emission changes were observed in the naphtha-gasoline fuel mixture.

ANOVA results for the CO emission value were given in Table 7. According to the ANOVA result of the CO emission response parameter, the model was found to be significant. When the input parameters were evaluated separately, it has been observed that the diesel/gasoline ratio and fuel injection duration variable parameters are not significant, while the fuel injection duration parameters are significant. When the P-value value is examined, it is seen that the engine speed variable parameter has the highest effect on the engine speed response parameter.

The model created to estimate the CO emission value depending on the HCCI engine variable parameters is given in Equation (7).

Table 5
Effective torque ANOVA results.

Source	Sum of Squares	Mean Square	F-value	p-value	Remarks
Model	116.56	5.07	28.67	<0.0001	significant
A-Diesel/ gasoline ratio	0.1914	0.1914	1.08	0.3070	Not significant
B- Fuel injection duration	0.7877	0.7877	4.46	0.0438	significant
C-Engine speed	10.04	10.04	56.81	<0.0001	significant
Residual	4.95	0.1768			
Cor Total	121.51				

One of the main reasons leading to the formation of HC emissions is the low end-of-combustion temperature. In HCCI combustion mode, the final combustion temperature is lower than that of diesel and gasoline engines. Therefore, higher HC emissions occur. The counter graphs of the HC emission value depending on the variable parameters of fuel injection duration, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 8. No stable change was observed in HC emission according to fuel injection duration. Reduction was observed on HC emissions as the ratio between diesel fuel and naphtha fuel is increasing. An increase in HC emissions is observed as the gasoline ratio increases.

The counter graphs of the HC emission value depending on the variable parameters of engine speed, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 9. It was concluded that there was no stable change in HC value depending on engine speed in the use of the naphtha-diesel fuel mixture. In the case of a naphtha-gasoline fuel mixture usage, the HC value increased with the increase of engine speed. The reason for this situation is considered to be lower temperature at the end of combustion because of the deterioration of homogeneity due to engine speed.

ANOVA results for HC emission value are given in Table 8. The HC emission was obtained according to the results of the ANOVA of the response parameter. When the input parameters were evaluated separately, the diesel/gasoline ratio in light naphtha and fuel injection duration was “not significant” and the fuel injection duration was “significant”. When the P-Value value is examined, the highest effect on the engine speed response parameter provides the engine speed variable

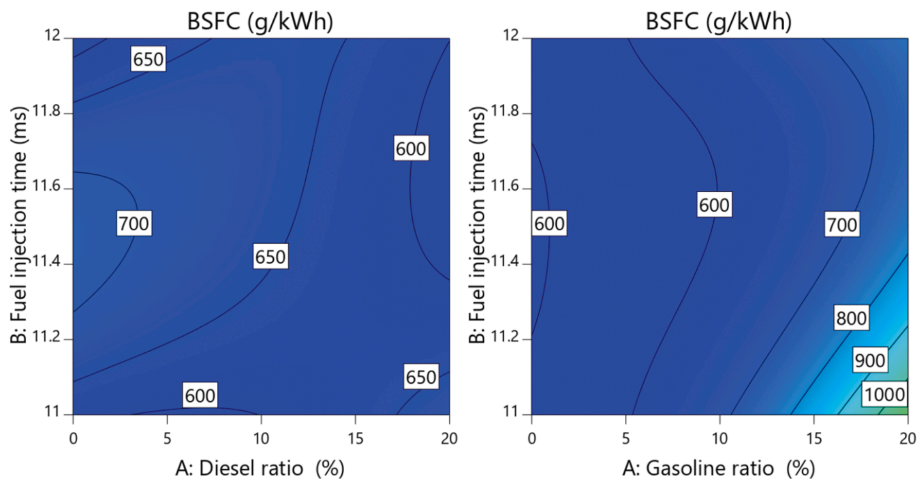

Fig. 4. The effect of fuel injection duration and diesel-gasoline ratio in light naphtha on BSFC.

parameter.

The model used to estimate the HC emission value depending on the HCCI engine variable parameters is given in equation (8).

$$HC\,(ppm) = 434.239 - 4.499A + 6B + 94.5C + 34.016D + 0.0625AB + 153.43AC + 142.85AD - 19.812BC - 7.0499BD + 108.35CD + 106.04255319149A^2 + 66.542553191489B^2 + 51.542C^2 + 2.437ABC - 8.812ABD + 138.312ACD - 7.687BCD - 35.1875A^2B + 108.687A^2C + 36.936A^2D + 102.312AB^2 + 44.436B^2D + 45.436C^2D \tag{8}$$

Due to the low temperature at the end of combustion in HCCI combustion mode, NO emission formation is very low. The counter graphs of the NO emission value depending on the variable parameters of engine speed, naphtha-diesel and naphtha-gasoline fuel concentration ratio are given in Fig. 10. It is observed that the amount of NO emission gradually decreases with the increase in engine speed. When the HCCI engine is operated at the speed range of 1100–1500 rpm, it is observed that a maximum of 30 ppm NO emission was obtained when a naphtha-diesel fuel mixture is used, whereas maximum of 20 ppm NO emission when a naphtha-gasoline fuel mixture is used.

ANOVA results for NO emission value were given in Table 9. According to the results of the ANOVA of the NO emission response parameter, the model was obtained as significant. When the input parameters were evaluated separately, diesel/gasoline mixture ratio in light naphtha and fuel injection duration variable parameters were "not significant" and fuel injection duration parameters were "significant". When the P-Value value is examined, the highest effect was seen on the engine speed response parameter provides the engine speed variable parameter.

The model created to estimate the NO emission value depending on the HCCI engine variable parameters is given in Equation (9).

3.2. Optimisation results

HCCI engine variable parameters were optimized by the RSM method. After optimization, the optimal input parameters and result values dependent on these input parameters were obtained. Optimization criteria and results are given in Table 10. Among the response parameters, it is aimed that the effective torque value is maximum and BSFC, CO, HC and NO emission values are minimum. From the variable parameter values after the optimization, the gasoline ratio was determined as 0.297 %, the fuel injection duration was 11.97 ms and the engine speed was 1325.21 rpm. Depending on the input parameters obtained after the optimization, the effective torque was determined as 12.79 Nm, BSFC value is 589.79 g/kWh, CO emission value is 0.11 %, HC emission value is 519.86 ppm, and NO emission value is 0 ppm.

The desirability rate value strengthens the accuracy of statistical data obtained from RSM. The closer the desirability rate value is to 1, the more it strengthens the study's accuracy [52]. Desirability rate values can be given individually or in combination. Individual and combined desirability rate values were given in Fig. 11. The red columns in the figure represent the input parameters, and the blue columns represent the response parameters. Individual desirability rate values were determined as effective torque 1, BSFC 1, CO 1, HC 0.889, and NO 1. The combined desirability rate value was determined as 0.976844. Directly targeted values were achieved in the effective torque, BSFC, CO and NO response parameters. In the HC response parameter, although the targeted minimum value was 286 ppm, 519.86 ppm was obtained. For this reason, it is thought that the individual desirability rate value of HC emission deviates from 1 and is 0.889. The high individual and

$$NO\,(ppm) = 1.164 + 0.25A + 0.5B - 15.75C - 0.68617021276596D - 0.6875AB + 1.4375AC - 1.5625BC - 0.0499BD + 1.2CD - 4.409A^2 - 4.159B^2 + 11.09C^2 + 0.687ABC + 0.06249ABD - 0.3125ACD - 0.3125BCD + 1.5625A^2B + 13.437A^2C + 2.494A^2D - 1.687AB^2 + 2.244B^2D - 4.505C^2D \tag{9}$$

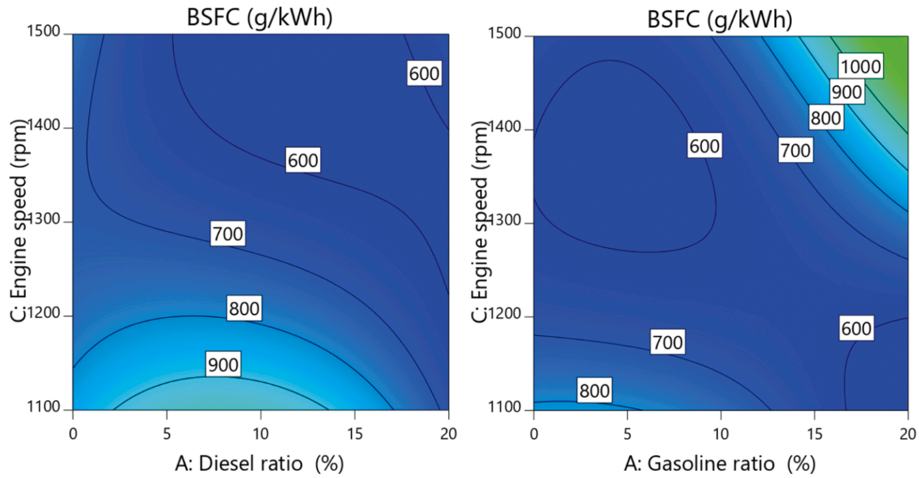

Fig. 5. The effect of engine speed and diesel-gasoline ratio in light naphtha on BSFC.

Table 6
ANOVA results for BSFC.

Source	Sum of Squares	Mean Square	F-value	p-value	Remarks
Model	1.686E6	73309.73	17.43	<0.0001	significant
A-Diesel/ gasoline ratio	284.46	284.46	0.0676	0.7967	Not significant
B- Fuel injection duration	41.27	41.27	0.0098	0.9218	Not significant
C-Engine speed	50671.55	50671.55	12.04	0.0017	significant
Residual	1.178E5	42062.98			
Cor Total	1.804E6				

combined desirability rate values strengthen the accuracy of the study.

4. Conclusion

In this study, 52 experiments were conducted in an HCCI engine fuelled with diesel-light naphtha and gasoline-light naphtha mixtures. This experimental study determined the most suitable optimized operating parameters by applying the RSM technique. Naphtha-diesel and naphtha-gasoline fuel type inputs were made as categorical input parameters. Mixed fuel concentration ratio (0 % to 20 %), fuel injection duration (11 ms-12 ms) and engine speed (1100 rpm to 1500 rpm) were defined as numerical input parameters. As the most appropriate value according to these categorical and numerical parameters;

Firstly, it has been determined that gasoline-light naphtha is a more suitable fuel mixture than gasoline-light naphtha and diesel-light naphtha fuel mixtures. Here, it was determined that the most suitable gasoline-light naphtha fuel mixture was the ideal mixture ratio of 0.297 % gasoline 99.703 light naphtha mixture. It can be deduced from that the fuel mixture does not have a significant effect.

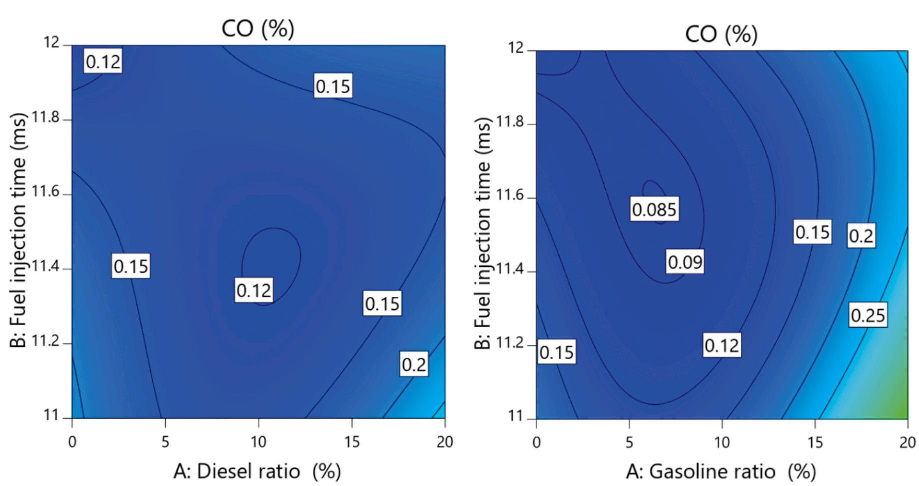

Fig. 6. The effect of fuel injection duration and diesel-gasoline ratio in light naphtha on CO emissions.

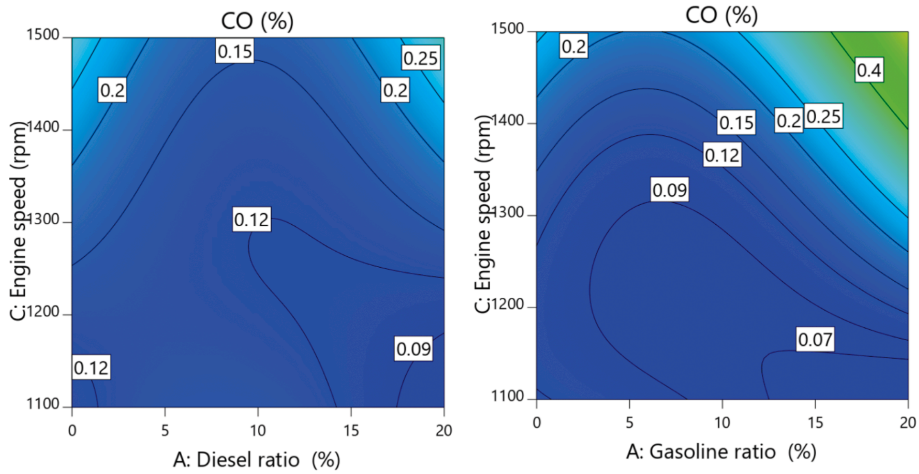

Fig. 7. The effect of engine speed and diesel-gasoline ratio in light naphtha on CO emissions.

Table 7
ANOVA results for CO.

Source	Sum of Squares	Mean Square	F-value	p-value	Remarks
Model	0.9162	0.0398	47.15	<0.0001	significant
A-Diesel/ gasoline ratio	0.003	0.003	3.58	0.0689	Not significant
B- Fuel injection duration	0	0	0.0145	0.905	Not significant
C-Engine speed	0.0072	0.0072	8.5	0.0069	significant
Residual	0.0237	0.0008			
Cor Total	0.9399				

According to the optimization result of the experimental study for the engine speed in the range of 1100–1500 rpm, it was determined that the most suitable engine speed was 1325.21 rpm. It has been observed that the engine speed is the most effective parameter among the input parameters.

Among the input parameters such as engine speed, fuel injection duration and fuel mixtures, engine speed was found to be significant for all parameters in the ANOVA analysis. It has been determined that the fuel mixtures are not significant and the fuel injection duration is significant for the effective torque. When the desirability values were examined according to the results of the ANOVA analysis, it was seen that the HC emission was 0.889 and all other values were 1. It was found that model predicted the results with 100 % accuracy except for HC emission. The desirability value of the HC emission was found to be acceptable.

CRediT authorship contribution statement

Samet Çelebi: Conceptualization, Data curation, Investigation, Methodology, Software, Visualization, Writing – original draft, Writing – review & editing. **Tolga Kocakulak:** Conceptualization, Data curation, Formal analysis, Investigation, Methodology, Project administration, Resources, Writing – original draft. **Usame Demir:** Conceptualization, Data curation, Investigation, Methodology, Project administration, Resources, Writing – original draft, Writing – review & editing. **Gökhan Ergen:** Conceptualization, Data curation, Investigation, Visualization. **Emre Yilmaz:** Conceptualization, Data curation, Investigation,

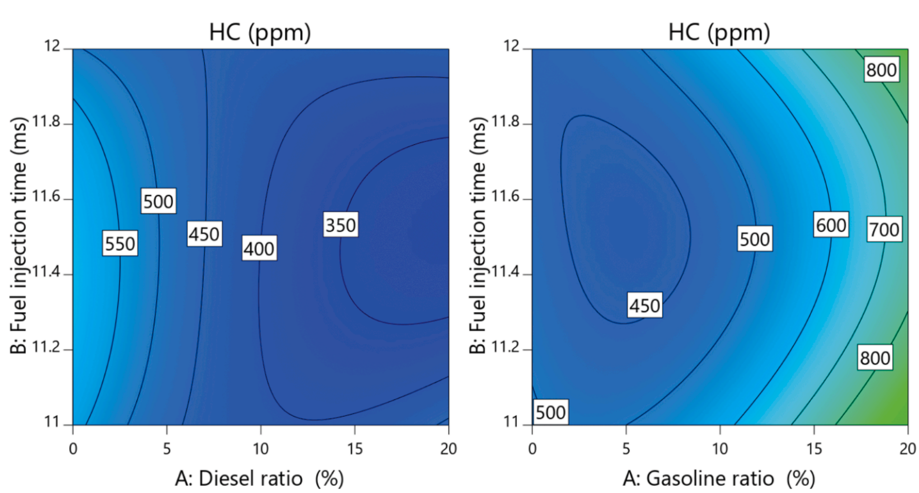

Fig. 8. The effect of fuel injection duration and diesel-gasoline ratio in light naphtha on HC emissions.

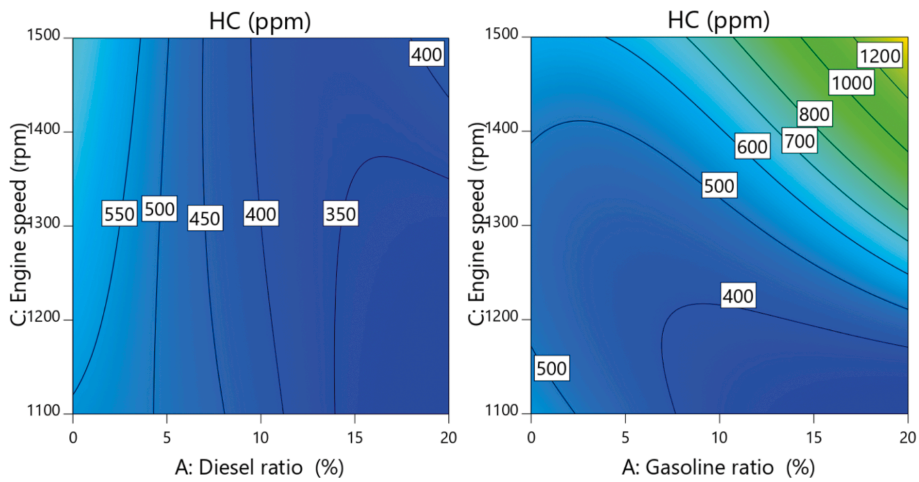

Fig. 9. The effect of engine speed and diesel-gasoline ratio in light naphtha on HC emissions.

Table 8
ANOVA results for HC.

Source	Sum of Squares	Mean Square	F-value	p-value	Remarks
Model	3.201E6	1.392E5	21.01	<0.0001	significant
A-Diesel/ gasoline ratio	81	81	0.0122	0.9127	Not significant
B- Fuel injection duration	144	144	0.0217	0.8838	Not significant
C-Engine speed	35721	35721	5.39	0.0277	significant
Residual	1.854E5	6622.96			
Cor Total	3.386E6				

Table 9
ANOVA results for NO.

Source	Sum of Squares	Mean Square	F-value	p-value	Remarks
Model	2136.44	92.89	5.86	<0.0001	significant
A-Diesel/ gasoline ratio	0.25	0.25	0.0158	0.9009	Not significant
B- Fuel injection duration	1	1	0.0631	0.8035	Not significant
C-Engine speed	992.25	992.25	62.63	<0.0001	significant
Residual	443.62	15.84			
Cor Total	2580.06				

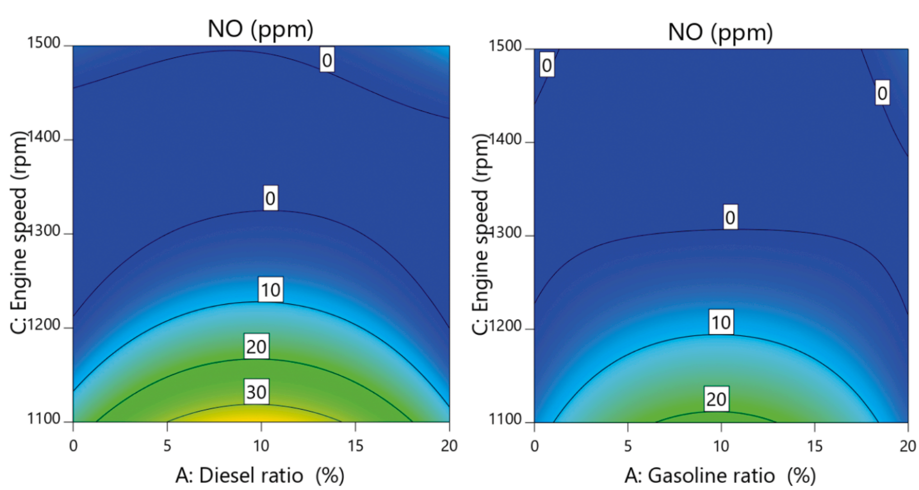

Fig. 10. The effect of engine speed and diesel-gasoline ratio in light naphtha on NO emissions.

Table 10
Criteria and results of optimization.

Parameter	Target	Limits		Optimized input and response parameters	Unit
		Lower	Upper		
A-Diesel/ gasoline ratio	In range	0	20	0.297 (Gasoline)	%
B- Fuel injection duration	In range	11	12	11.97	ms
C-Engine speed	In range	1100	1500	1325.21	rpm
Effective torque	Maximize	3.55	13.025	12.79	Nm
BSFC	Minimize	602.405	1859.23	589.79	g/ kWh
CO	Minimize	0.087	0.855	0.11	%
HC	Minimize	286	1820	519.86	ppm
NO	Minimize	0	46	0	ppm

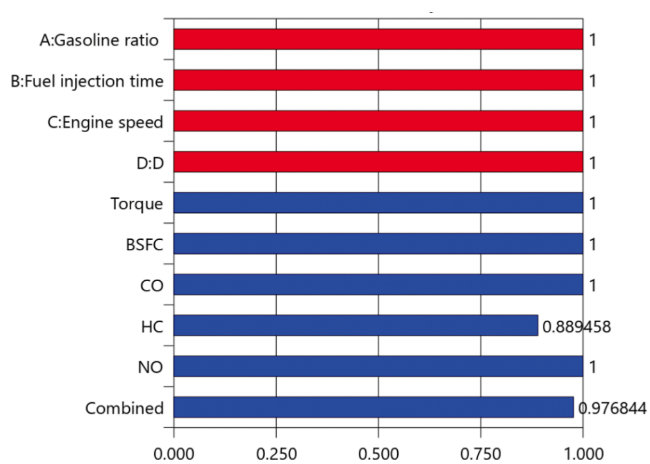

Fig. 11. Optimization results in desirability ratio values.

Methodology, Software, Visualization, Writing – original draft.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

The authors do not have permission to share data.

Acknowledgments

This study was supported by the Research Fund of the Sakarya University of Applied Science Project Number: 082-2022. All authors would like to thank the individuals who were involved in making this work possible.

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