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Operating range, combustion, performance and emissions of an HCCI engine fueled with naphtha

Samet Çelebi[a], Can Haşimoğlu[b], Ahmet Uyumaz[c,*], Serdar Halis[d], Alper Calam[e], Hamit Solmaz[d], Emre Yılmaz[a]

[a] Sakarya University of Applied Science, Arifiye Vocational High School, Depart. of Automotive Techn., 54330 Sakarya, Turkey
[b] Sakarya University of Applied Science, Faculty of Technology, Depart. of Mech. Eng., 54300 Sakarya, Turkey
[c] Burdur Mehmet Akif Ersoy University, Faculty of Eng. and Archit., Depart. of Mech. Eng., 15030 Burdur, Turkey
[d] Gazi University, Faculty of Technology, Depart. of Automotive Eng., 06500 Ankara, Turkey
[e] Gazi University, Technical Education Vocational High School, Depart. of Machine and Material Tech., 06760 Ankara, Turkey

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ABSTRACT

In this study, it was aimed that the effects of naphtha on homogeneous charged compression ignition (HCCI) combustion, performance, exhaust emissions and operating range were researched experimentally in an HCCI engine. The test engine was able to operate with HCCI mode between 800 and 2000 rpm and $\lambda = 1.61$–$\lambda = 2.93$ lambda values range at constant inlet air temperature of 60 °C. Pure n-heptane and naphtha, N25, N50 and N75 fuel blends were used as test fuels. Experiments showed that knocking tendency reduced with the increase of lambda. Similarly, more stable combustion was obtained with the addition of naphtha into n-heptane due to higher octane number. Test results showed that HCCI combustion was delayed when naphtha fraction increased in the test fuels. MPRR was obtained as 15.6 bar/°CA and 11.9 bar/°CA with n-heptane and naphtha respectively at $\lambda = 2$ and 1000 rpm. ITE increased from 29% to 37% with naphtha according to n-heptane at $\lambda = 2$ and 1000 rpm. On the contrary, HC and CO increased from 331 ppm to 411 ppm and 0.051% to 0.075% with naphtha compared to n-heptane at $\lambda = 2$ and 1000 rpm, respectively. It was also seen that Naphtha showed wider HCCI operating range according to n-heptane especially knocking zone.

1. Introduction

Highly efficient and environmentally friendly combustion modes are researched by the scientists due to strict emission regulations and cleaner atmosphere. At this point, HCCI has been attracted by higher thermal efficiency and leaner operation by the researchers. Nevertheless, HCCI combustion that gives higher thermal efficiency with lean mixture is needed to be improved in view of higher HC and CO. HC that unburned fuel molecules are formed as a result of extinguishing the flame in the combustion chamber. Likely, CO is a product that is formed due to incomplete combustion with unsufficient oxygen and temperature during combustion. Many studies have been carried out by scientists to reduce these emissions and increase engine performance from the internal combustion engines [1,39–41]. Many studies have been carried out to reduce soot (PM) and NO_x (Nitrogen oxide) emissions from CI engines. On the other hand, spark ignition engines must be operated near the stoichiometric ratio resulting in richer operation and harmful emissions. To overcome these difficulties, HCCI combustion mode that can be applied as low temperature combustion to internal combustion engine has been developed [2,42,43,51,52]. HCCI combustion provides high fuel efficiency under partial load conditions, minimizes soot and NO_x emissions with efficient energy conversion [3,44]. In HCCI combustion modes, a homogeneous mixture is formed as in spark ignition (SI) engines. This mixture is auto-ignited with the increase of temperature and compression within the cylinder as in compression-ignition engines [4,5,45–47]. One of the most important advantages of HCCI engines is the ability to work with many fuels (diesel, methanol, ethanol, fusel oil, natural gas, propane, and butane etc.) which have different chemical properties [4,6,48,49]. On the contrary, there are some disadvantages in HCCI engines such as limited operating range, timing control of combustion, high carbon monoxides (CO) and unburned hydrocarbons [7,8,50]. Some methods such as different intake manifold pressure [9], exhaust gas recirculation (EGR) [10], octane number, intake air inlet temperature [11–14],

* Corresponding author.
E-mail addresses: scelebi@subu.edu.tr (S. Çelebi), canhasim@subu.edu.tr (C. Haşimoğlu), auyumaz@mehmetakif.edu.tr (A. Uyumaz), serdarhalis@gazi.edu.tr (S. Halis), acalam@gazi.edu.tr (A. Calam), hsolmaz@gazi.edu.tr (H. Solmaz), emreylmz@subu.edu.tr (E. Yılmaz).

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Nomenclature

IMEP	Indicated Mean Effective Pressure
CC	Combustion Chamber
ITE	Indicated Thermal Efficiency
MTBE	Methyl Tertiary Butyl Ether
SOC	Start of Combustion
CI	Compression Ignition
LTC	Low temperature combustion
MPRR	Mean Pressure Rise Rate
N25	%75 n-heptane + %25 Naphtha
N50	%50 n-heptane + %50 Naphtha
N75	%25 n-heptane + %75 Naphtha
HRR	Heat Release Rate
PR	Premixed Ratio
SFC	Specific Fuel Consumption
PRR	Pressure Rise Rate
SI	Spark Ignition
RON	Research Octane Number
°CA	Crank Angle Degree
λ	Lambda
COV_{imep}	Correlation of Variation
TDC	Top Dead Center
HCCI	Homogeneous Charged Compression Ignition

variable compression ratio and different valve lift mechanisms are used to control combustion in HCCI engines [15–17].

Kim and Lee [18] used EGR, as a control mechanism for HCCI combustion and investigated combustion characteristics and emissions of partial HCCI engine by supplying premixed fuel such as n-heptane, diesel, and gasoline. It was reported that using diesel premixed fuel, decreasing of NO_x and soot can be obtained by increasing the premixed ratio (PR). They found that gasoline premixed fuel shows more reductions of NO_x and soot emissions compared to other premixed fuels. Maurya et al. [19] analyzed the effects of air temperature on emissions, thermal efficiency, and combustion parameters in an HCCI engine fueled with ethanol at different lambdas. It was found that the maximum combustion efficiency of 97.45% and the maximum gas exchange efficiency of 97.47%. In addition, the maximum indicated thermal efficiency (ITE) was obtained 44.78% at the air temperature of 393 K while lambda was 2.5. It was found that NO_x emissions were too low in all HCCI operating conditions. However, CO emissions increased and HC (Hydrocarbon) emissions decreased, as lambda increased.

In HCCI engines, since the auto-ignition occurs at nearly constant volume in the cylinder, it causes sudden heat and pressure rise in the combustion chamber. In HCCI combustion, the temperature of homogeneous charge mixture is increased to auto-ignition temperature to initiate the combustion because there is no ignition mechanism. Homogeneous charge is ignited suddenly and simultaneously towards the combustion chamber. Hence, pressure rise rate increases so much and undesirable pressure waves are seen during combustion process. This effect is seen as knock in HCCI engines. Chemical reactions govern the combustion process depending on the thermodynamic properties of the mixture taken into the cylinder. Therefore, misfiring and knocking problems occur under partial and full load conditions that limit the operating range of HCCI engines [20]. Recently, scientists have investigated the effects of operating conditions and fuels that can be used in the HCCI engine [21–24]. Gharehghani [21] performed an experimental study to determine the operating range of HCCI engine fueled with methanol, natural gas and ethanol fuels having different octane numbers. The results showed that natural gas is suitable for HCCI combustion with high intake air temperatures. However, methanol and ethanol are seen to have potential as alternative fuel at low intake air temperatures and lean mixture conditions (at low loads), when HCCI engine is operated with richer mixture (at high loads),

Sudheesh and Mallikarjuna [25] investigated the usage of diethyl ether as combustion improver in an HCCI engine with biogas. Both biogas and diethyl ether were taken into the cylinder by intake manifold and homogeneous mixture was obtained. The diethyl ether which provides the highest thermal efficiency for each different load case, was experimentally researched. It was stated that SOC can be controlled and more stable combustion was seen for each different loads. They found that brake thermal efficiency (BTE) increased 3.48%, and 9.21% compared to dual-fuel and SI modes, respectively. Hou et al. [26] investigated the effects of the addition of methanol, Methyl Tertiary Butyl Ether (MTBE) and ethanol which have high octane number to n-heptane from 10% to 60% ratios on combustion phase and combustion efficiency in an HCCI engine. Combustion phasing was delayed and the heat release rate (HRR) increased compared to the use of pure n-heptane, in case the blend of high octane fuel and n-heptane were used. It was found that start of combustion (SOC) was delayed at a similar rate and the maximum heat release shifted around top dead center (TDC), As methanol, ethanol and MTBE ratios increased in the fuel mixtures. Yılmaz [27] examined the effects of fuels obtained by mixing fusel and diesel in certain ratios on characteristics of combustion, engine performance and emissions at different speeds and loads. Similar in-cylinder pressures were observed with the usage of fusel oil blends compared to neat diesel at full load. It was stated that fusel oil fraction in the fuel blend increased, specific fuel consumption (SFC) increased. Moreover, it was reported that CO increased but soot and NO_x decreased compared to diesel as the fusel oil increased in the fuel blend. Scientists have researched the effects of octane number on combustion performance of the multiple premixed compression ignition. It was stated that the low octane gasoline such as naphtha (RON 66) had advantages in terms of emissions and efficiency without intake heating due to the sufficient ignitability and the high volatility [28–31]. In addition, the production of low octane gasoline is more economical, and it can be easily produced in a more environmentally friendly way [15]. Some researchers have demonstrated that low reactivity naphtha fuels can achieve high fuel efficiency with low NO_x and soot emissions [32–35]. Zhang et al. [36] evaluated the effects of light and heavy naphtha fuels on engine performance and combustion in a six cylinder heavy duty diesel engine. It was stated that the naphtha fuels exhibited quite low soot and NO_x emissions compared to ultra low sulfur diesel. Results showed that naphtha-2 (RON 69) exhibited longer ignition delay and lower soot emission than naphtha-1 (RON 59) due to the lower reactivity. Naphtha that low-octane (RON 64.5) has more than 90% paraffinic content. It is one of the first products obtained during refining of the crude oil. Naphtha is economical and attractive fuel for low temperature combustion (LTC) modes where auto-ignition is the primary control mechanism. The auto-ignition characteristics of a low octane fuel light naphtha were investigated in a study by Javed et al. The light naphtha ignition delay was measured in a rapid compression machine. They stated that the multi-component surrogate matched the light naphtha ignition delay more closely than the primary reference fuel surrogate at low temperature rapid compression machine conditions [16]. In distinct study, Hao et al. [37] investigated the effect of using low octane gasoline in the gasoline compression ignition engines in view of greenhouse gas emissions and energy consumption. It is found that the low octane fuel leads to a 22.8% reduction in greenhouse gas emissions and 24.6% reduction in energy consumption compared to the SI engine.

Naphtha is seen to prevent knocking tendency with mixing n heptane. Because knocking is critical problem to provide stable HCCI combustion. In addition, naphtha has higher RON and density compared to n-heptane. Operating range can be also enlarged in misfiring and knocking zone in HCCI engines using naphtha. On the other side, misfiring and knocking have received great attention in the research literature, because controlling of combustion phasing is needed to be investigated and comprehensive approach should be performed on this issue. There is limited number of studies with naphtha on detailed combustion analysis to demonstrate advantages or disadvantages in HCCI combustion. Controlling combustion of homogeneous charged compression ignition (HCCI) has long been a question of great interest in a wide range of fields. No previous study has given sufficient and detailed consideration with naphtha fuel blends on HCCI combustion characteristics. In this study, the effects of naphtha fuel mixtures on combustion characteristics such as in-cylinder pressure, heat release rate (HRR), CA10, combustion duration, indicated thermal efficiency (ITE), specific fuel consumption (SFC), brake torque and power output, hydrocarbons (HC), carbon monoxides (CO) emissions and HCCI operating range were investigated and compared pure n-heptane.

2. Experimental setup and procedures

In the experiments, single cylinder, four stroke and port injection system HCCI test engine was used. The test engine was the Ricardo Hydra model. Test engine was transformed into the HCCI engine from the spark ignition engine. List of the specifications of test engine is shown in Table 1. Further, the HCCI test bench is shown in Fig. 1.

The air/fuel mixture was prepared in the intake manifold by using the port injection fuel system. The amount of fuel to be injected is set via the control unit. There are 100 equal divisions on the potentiometer where the amount of fuel injection is adjusted. The potentiometer was adjusted in positions 1, 2, 4, 6, 8 and 10 respectively and the engine was started at constant speed. While the fuel tank was placed on the precision balance, fuel consumption was determined by the time of 120 s. The fuel consumption was formulated with the fuel injection characteristic obtained.

All equipment were calibrated before the experiments started. The test motor was connected to an electric type dynamometer capable of absorbing 30 kW of power. The brand of the dynamometer was McClure. The intake air heater device was located on the intake manifold, just before the fuel injection system. The inlet air temperature was measured by a K type thermocouple where it placed between the air heater and the injection system. Air inlet temperature was set on the control panel. In the experiments, the original air heater system was used. The Kistler 6121 model piezoelectric pressure sensor was used for determining the in-cylinder pressure. The crank position versus the in-cylinder pressure was determined using an encoder with a precision of 0.36 °CA. Some of the specifications of encoder are shown in Table 2.

In-cylinder pressure was collected with Cussons P4110 combustion analyzer. The data was then transferred to computer with National Instruments USB6259 model data acquisition card. In-cylinder pressure signals were recorded to the computer with data from the encoder. The average of 50 consecutive in-cylinder pressure data were used in all operating conditions. When the test engine was operated in HCCI mode, the cooling water and lubricating oil temperature of the test engine was expected to stabilize. While the lubricating oil at 65 °C, the cooling water temperature was fixed at 75 °C. These temperatures were defined as operating temperatures. The test engine was operated firstly in SI mode. When the lubricating oil and cooling water temperature reached operating conditions, the ignition system was closed. Thus, HCCI combustion was started. In the experiments, neat light naphtha, n-heptane, N25, N50 and N75 fuels were utilized. The N25 was constituted from 75% n-heptane and 25% light naphtha, N50 was constituted from 50% n-heptane and 50% light naphtha while the N75 was constituted from 25% n-heptane and 75% light naphtha mixture by volume. Some of the specifications of test fuels are shown in Table 3.

During the operation, the exhaust gases were recorded with the Bosch BEA350 exhaust gas analyzer. The exhaust gas analyzer measures NO, HC, CO_2, O_2 and CO emissions. However, lambda was determined by the Brettschneider formula. 50 consecutive pressure data were averaged to determine in-cylinder pressure. For thermodynamic analysis, a code was prepared by using MATLAB Simulink program. By using MATLAB Simulink code, ITE, heat release rate, SOC, in-cylinder pressure, combustion duration and indicated mean effective pressure were determined. HRR was determined using the first rule of thermodynamics. For this reason, mass, and gas leaks during one cycle are disregarded. In order to calculate the HRR, the heat transfer from the cylinder wall was calculated. The HRR versus to the crank angle was calculated with Eq. (1).

$$\frac{dQ}{d\theta} = \frac{n}{n-1}P\frac{dV}{d\theta} + \frac{1}{n-1}V\frac{dP}{d\theta} + \frac{dQ_{heat}}{d\theta} \tag{1}$$

here, V and P express to cylinder volume and cylinder pressure, respectively. dQ expresses to net heat output. n refers to ratio of specific heat values and $d\theta$ expresses to change of crank angle. $\frac{dQ_{heat}}{d\theta}$ expresses to heat transfer from cylinder to cooling system.

Thermal efficiency was calculated with Eq. (2).

$$\eta_T = \frac{W_{net}}{\dot{m}_{\text{naphtha}} \times Q_{LHV\ \text{naphtha}} + \dot{m}_{\text{n - heptane}} \times Q_{LHV\ \text{n - heptane}}} \tag{2}$$

here and express to fuel consumption of n-heptane and naphtha per cycle and expresses to net work. However, and express to calorific values of naphtha and n-heptane fuels. The net work was calculated with Eq. (3).

$$W_{net} = \int PdV \tag{3}$$

The indicated mean effective pressure values depends on cylinder volume, engine speed and number of cylinders. IMEP is one of the simplest definitions used to express engine efficiency. The IMEP is calculated as given in Eq. (4). Here, V_{stroke} expresses to cylinder swept volume [38].

$$IMEP = \frac{W_{net}}{V_{stroke}} \tag{4}$$

3. Results and discussion

The role of the usage of high-octane number fuel in HCCI combustion has received great attention across a number disciplines. To date, the problem of misfiring and knocking has received scant attention in the research literature. So, there is urgent need to address the controlling of combustion problems caused by HCCI. Previous studies have failed to demonstrate any significant advantages of using naphtha in view of detailed combustion analysis. Fig. 2 shows the effects of lambda with test fuels on HRR and in-cylinder pressure. It was seen that HRR and in-cylinder pressure decreased with the increase of lambda for each test fuel. In addition, combustion was retarded with the increase of lambda due to lower fuel concentration for each test fuel. No resistance to knocking for n-heptane caused to undesirable auto-ignition characteristics as seen in Fig. 2-a. Knocking tendency reduced with the addition of naphtha into n-heptane due to higher octane number. There is no sharp variations between on maximum in-cylinder pressure with test fuels. It was also seen that HCCI combustion could not be achieved with pure naphtha at higher lambda.

CA10 can be assumed start of combustion. CA10 refers to crank

Table 1
The test engine specifications.

Test Engine	Ricardo Hydra
Cylinder number	1
Max. engine speed [rpm]	5400
Compression ratio	13/1
Valve lift [mm]	Exhaust lift 3.5, Intake lift 5.5
Stroke × bore [mm]	88.90 × 80.26
Max. power output [kW]	15

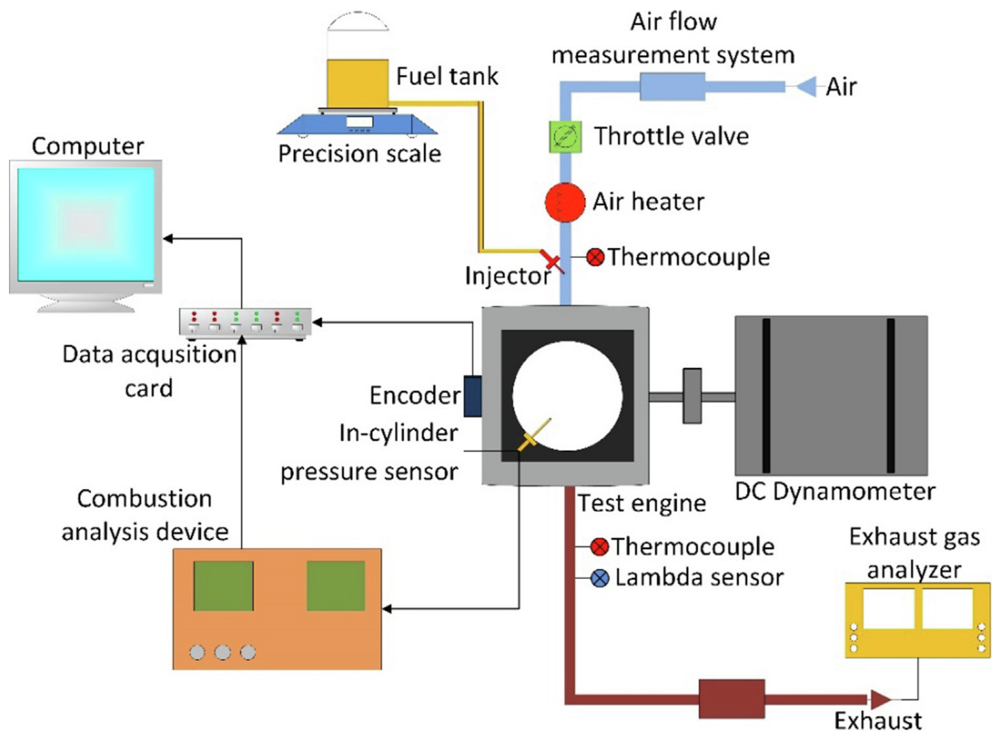

Fig. 1. Schematic of HCCI test bench.

Table 2
Properties of encoder.

Encoder	Opkon
Supply voltage	5 V
Operating temperature	253 to 353 K
Shaft diameter	8 mm
Body diameter	50 mm
Output type	Line driver
Pulse number per rotation	1000
Max. operating speed	4000 rpm

Table 3
Properties of naphtha and n-heptane fuels.

	n-Heptane	Light naphtha
Calorific value (kJ/kg)	44,566	43,360
RON	0	68
Boiling point (°C)	97–98	152
Density (kg/cm^3 at 20 °C)	0.68	0.72
Viscosity (cSt)	0.57	0.5

angle where 10% of charge mixture completed to combust. CA10 is highly affected by the conditions at the end of compression stroke. Fig. 3 compares experimental data on variations of CA10 and combustion duration with test fuels. Looking at Fig. 3-a, it is clear that CA10 increased with the increase of lambda. Lower heat is released to the combustion chamber with leaner mixtures. Hence, self-ignition chemical reactions could not occur easily. Consequently, CA10 is delayed. From this data, it was seen that naphtha showed the highest CA10 due to higher octane number. Moreover, what stands out in the figure is the increase of lambda with the increase of naphtha in the fuel blends.

Released heat decreases in the combustion chamber during self-ignition with naphtha due to lower calorific value. So, lower heat in the combustion chamber prevents to occur oxidation reactions. The most striking factor is that naphtha shows higher resistance to auto-ignition owing to higher octane number. It can be also explained that boiling point is higher than n-heptane. Higher boiling point causes to show cooling effect during vaporization. This situation also decreases the heat gradient in the combustion chamber to start the self-ignition. Overall, higher boiling point of naphtha caused to decrease in-cylinder heat and start of combustion was retarded.

Fig. 3-b shows combustion duration versus lambda with test fuels. As expected, combustion duration increased with the increase of lambda. Completion of auto-ignition reactions takes a long time with leaner charge mixtures. Heat energy remains lower due to lower fuel molecules during combustion. Interestingly, combustion duration decreased with the addition of naphtha at a given lambda. The shortest combustion duration was observed with n-heptane. The longer combustion duration was determined with N50.Combustion duration was obtained as 34.2 °CA and 35.3 °CA at $\lambda = 2$ with n-heptane and naphtha respectively. No knocking resistance of n-heptane resulted in the shortest combustion duration. The addition of naphtha caused to show resistance owing to higher octane number according to n-heptane.

The results of the CA50 and ITE are summarized in Fig. 4. CA50 defines the crank angle where the 50% of charge mixture completed to combust. CA50 increased with the increase of air fuel ratio as seen in Fig. 5-a. Similar tendency was realized for CA50 like CA10. The highest CA50 was computed with naphtha while the lowest CA50 was determined with N25. Higher octane number of naphtha caused to delay CA50. ITE increased with the increase of lambda while it decreased after a specific lambda value for each test fuel. Lower ITE was obtained although combustion is achieved with leaner mixture (λ greater than 2). It can be mentioned that oxygen concentration reduces compared to other lambda values with $\lambda = 1.6$ or 1.7 and 1.8. This phenomena causes to prevent complete combustion with lack of oxygen in the combustion chamber. It is also possible to emphasize that fuel molecules can react easier with oxygen molecules with higher lambda. Oxidation reactions can occur towards the combustion chamber resulting in higher combustion efficiency. Likely, released heat decreased in the presence of combustion of richer mixtures. Thus, ITE decreased. The highest ITE was computed as 38.9% at $\lambda = 2.1$ with naphtha. This is a remarkable outcome. ITE increased about 27% with naphtha compared to n-heptane at $\lambda = 2$. For higher ITE, CA50 should be nearly

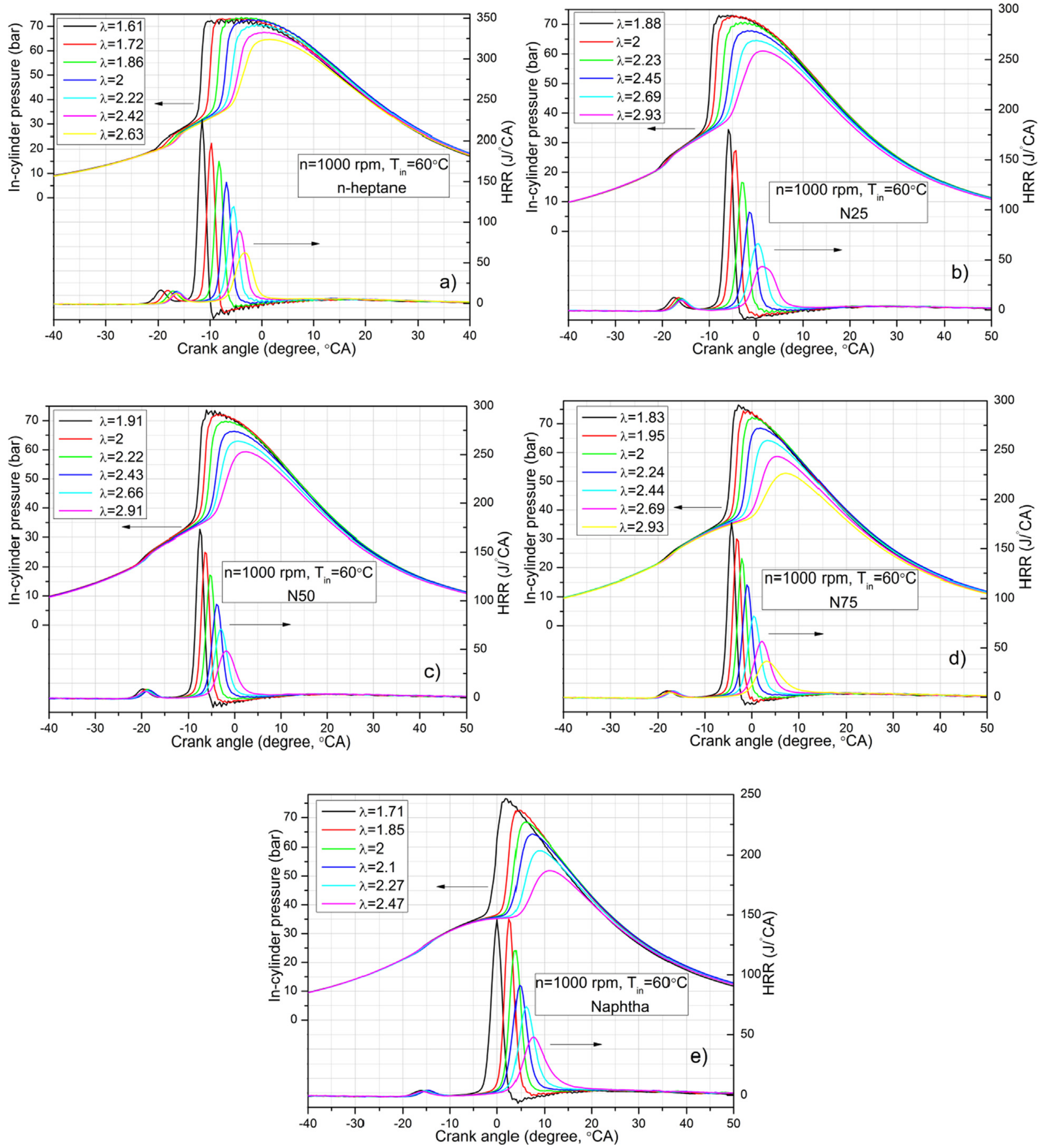

Fig. 2. The effects of lambda with fuels on cylinder pressure and HRR.

after TDC (5-10°CA). There was a significant positive correlation between CA50 and ITE as seen Fig. 4.

Cyclic variations are caused by thermodynamically properties, trapped residual gases at the end of compression stroke, heat transfer to the cylinder wall. This parameter directly reflects the stable combustion. Reasonable limit value for stable operation is accepted 10% in the internal combustion engines [38]. Since, there is no mechanism on self-ignition in HCCI combustion, evaluation of stable operation will be evident. The results obtained from the experiments for cyclic variations of IMEP are displayed in Fig. 5. COV_{imep} decreased as lambda increased. At richer mixtures, local rich zones can be formed in the combustion chamber. In addition, some fraction of fuel molecules could not be ignited because of lack of oxygen molecules and insufficient auto-ignition temperature. Fuel molecules which could not be ignited remain for the next cycle and mix with the fresh charge. In-cylinder temperature and pressure history vary cycle by cycle. Higher COV_{imep} was obtained with N25 and N50 especially with richer mixtures and limit value were exceeded. The addition of naphtha caused to decrease cyclic variations. High octane number of naphtha resulted in more stable combustion. So, cyclic variations were reduced.

COV_{imep} was determined as 5.88% and 6.45% with naphtha and n-heptane at $\lambda = 2$ and 1000 rpm. It can be mentioned that more uniform structure can be obtained in the combustion chamber with leaner mixtures. The formation of local rich and lean regions decrease and residual gases transferred to the next cycle reduce. This aspect resulted in lower cyclic variations. Pressure rise rate (PRR) defines the in

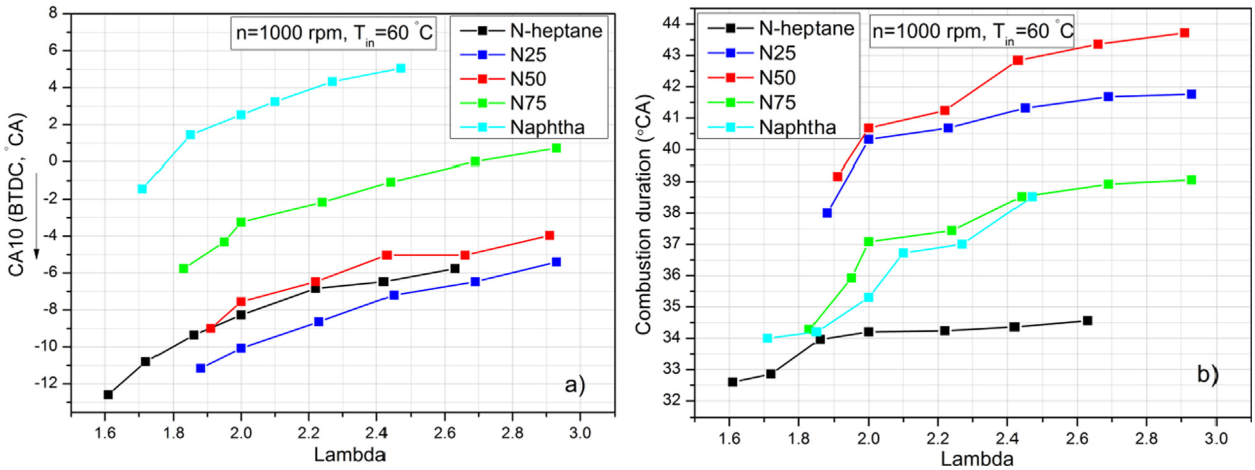

Fig. 3. The variations of CA10 and combustion duration.

cylinder pressure applied to the piston and crankshaft for each crank angle variation. Higher pressure rise rate causes to apply high amount of force to engine bearing and crankpin. This effect reduces the life of engine parts. From the Fig. 6 below we can see the variations of MPRR versus lambda with test fuels. MPRR decreased with lambda increase for all test fuels. Heat energy and maximum in-cylinder pressure decreased with lean charge mixtures owing to lower fuel concentration. This result also decreases MPRR. The highest MPRR was obtained with n-heptane at richest charge mixture. The most surprising aspect of Fig. 6 is that naphtha presented lower MPRR compared to other test fuels. For a specific lambda, the addition of naphtha resulted in lower MPRR. Having higher octane number of naphtha is the most remarkable reason for lower MPRR.

IMEP is an indication parameter to define performance. Fig. 7 shows the IMEP values of 50 successive cycles with test fuels. In Fig. 7, there is clear trend of decreasing on IMEP values with reduced naphtha addition in the fuel blends. Maximum IMEP was computed with naphtha. The most striking result to emerge from IMEP data is that naphtha presented higher IMEP than n-heptane in spite of lower heating value. The lowest IMEP was obtained with N50. Moreover, higher density of naphtha caused to obtain higher IMEP because of higher fuel molecules by mass. Knocking resistance increased when the naphtha was used and more controllable HCCI combustion was achieved.

In the current study, full load characteristics were also studied. Test engine was run at different engine speed and wide open throttle. The results obtained from full load characteristics with test fuels can be seen in Fig. 8. When Fig. 8-a is examined, brake torque increased and then

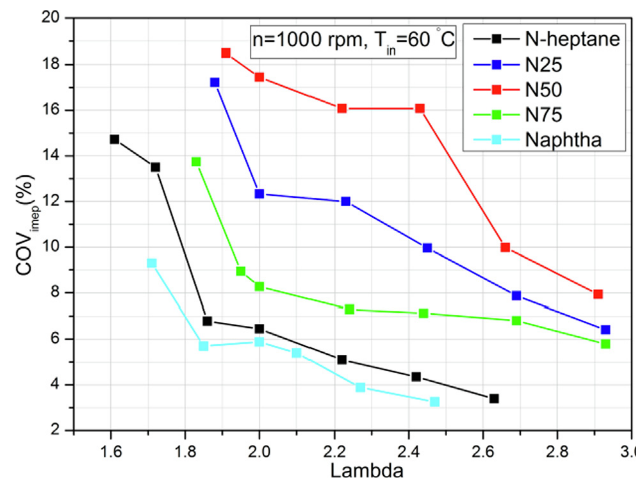

Fig. 5. COV$_{imep}$.

decreased with the increase of engine speed for all test fuels. The more surprising correlation is that test engine could not be operated on HCCI mode at higher engine speed with the decrease of naphtha. HCCI combustion was just achieved between 800 and 1200 rpm with pure naphtha. Time is limited for complete combustion in view of crank angle at higher engine speed. Moreover, higher boiling and octane number of naphtha could not be ignited due to misfiring. Maximum

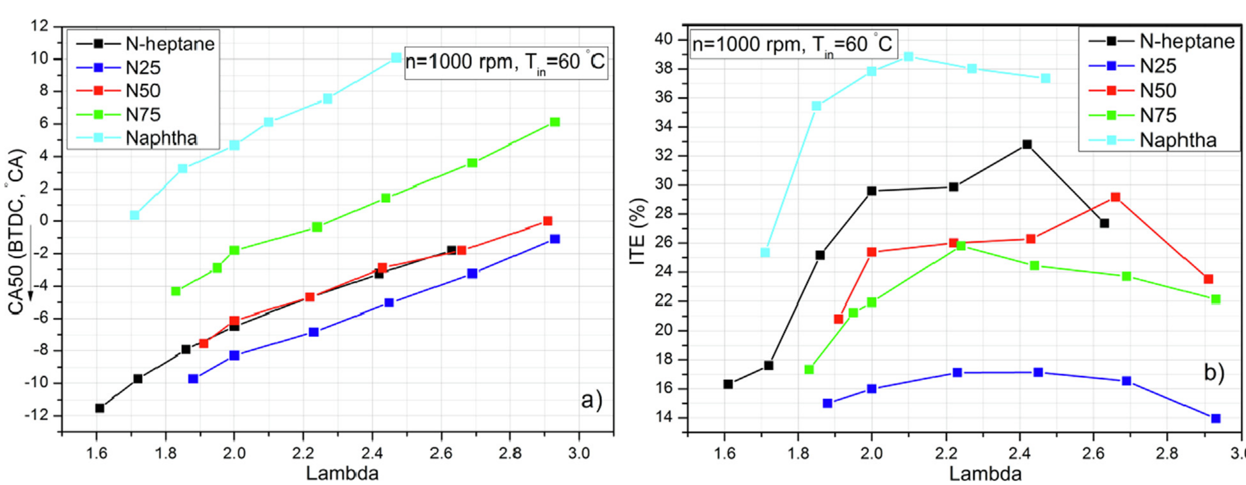

Fig. 4. The variations of CA50 and ITE.

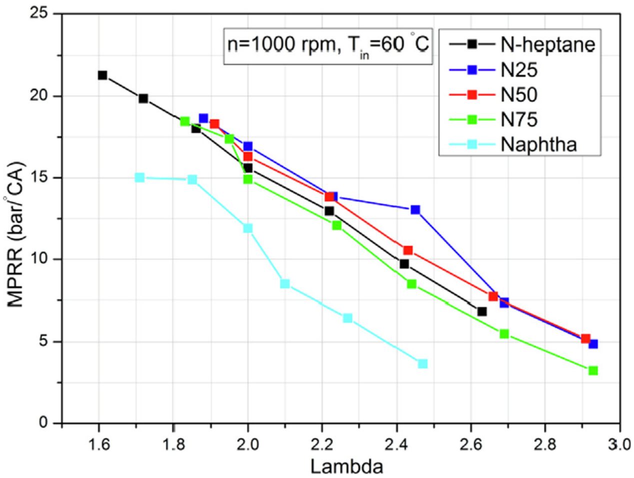

Fig. 6. MPRR variations versus lambda.

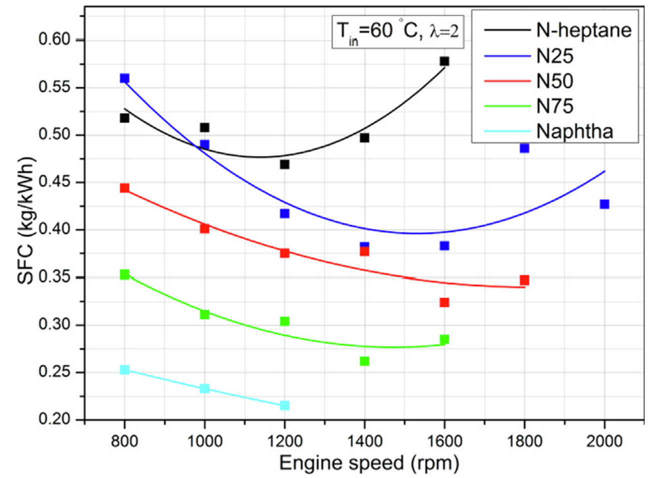

Fig. 9. The variations of SFC.

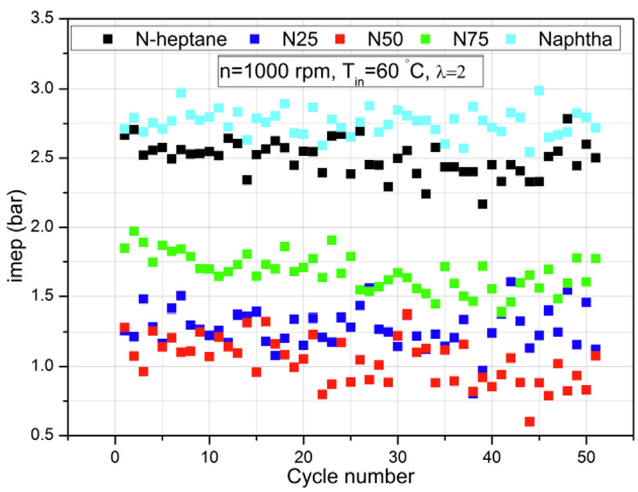

Fig. 7. IMEP values of 50 successive cycles with test fuels.

brake torque was measured as 8.55 Nm where as 4.7 Nm at 1200 rpm with naphtha and n-heptane respectively. higher density and reasonable heating energy of naphtha caused to increase brake torque. Power output increased with the increase of engine speed. Power also increased with the increase of naphtha in the fuel blends. This is significant outcome that engine performance improved with naphtha.

Fig. 9 depicts the SFC versus engine speed with test fuels at $\lambda = 2$. SFC defines the consumed fuel fraction for unit power production in a specified time. SFC decreased with the increase of engine speed, because homogeneity of charge mixture improves and gas leakages decreases. On the contrary, SFC increased at higher engine speeds owing to limited time for combustion, frictional and heat losses due to higher piston speed. Minimum SFC was computed 0.215 kg/kWh at 1200 rpm with naphtha. Although naphtha has lower calorific value and higher density, lower SFC was obtained compared to n-heptane. Combustion phasing can be controlled with naphtha due to higher octane number and decided auto-ignition reactions. SFC decreased by about 45% with naphtha compared to n-heptane at 1200 rpm.

Fig. 10 shows the HC and CO variations versus engine speed with test fuels at $\lambda = 2$. There is no significant difference on HC emissions except for N75 and naphtha according to engine speed. Lower heating value of naphtha decreases the heat in the combustion chamber. In addition, more time is required in order to complete combustion of naphtha due to higher boiling point. Heat is absorbed in order to reach auto-ignition temperature. Hence auto-ignition condition is deteriorated. HC is formed in cool regions and piston cavities in the combustion chamber. HC was measured 335 and 434 ppm with n-heptane and naphtha respectively at 1200 rpm. CO variations versus engine speed is seen in Fig. 10-b. CO increased with the increase of naphtha at a given engine speed. The highest CO was measured with N75 and naphtha. It can be pointed out that CO reduced with the increase of engine speed due to better homogeneity. CO increased 57.6% with naphtha

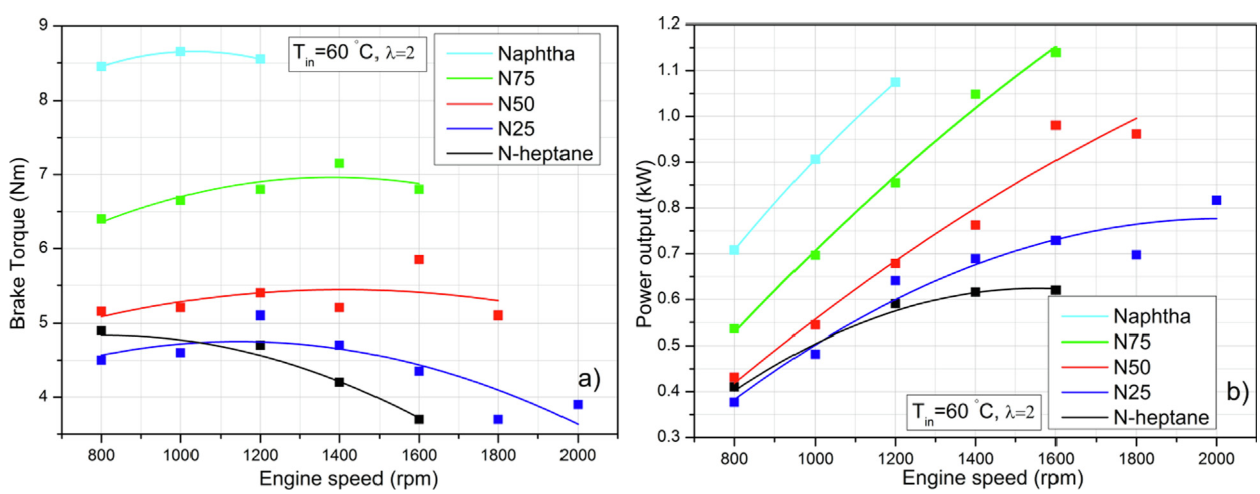

Fig. 8. Brake torque and power output versus engine speed.

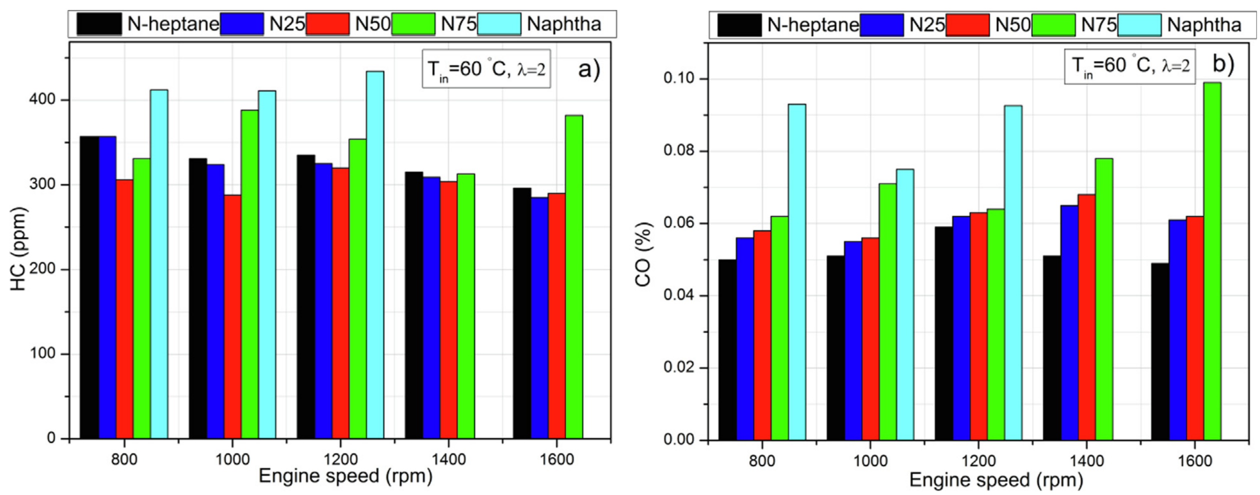

Fig. 10. HC and CO variations versus engine speed with test fuels.

compared to n-heptane at 1200 rpm and $\lambda = 2$. Incomplete combustion product CO also increased due to higher density of naphtha.

Fig. 11 displays the experimental data on CO and HC emissions. It was clearly seen that HC increased with the increase of lambda for all test fuels. Fuel concentration decreases and unburned HC normally decreases with higher lambda. Flame goes out especially near the cylinder wall owing to cooler surfaces. At these regions, oxidation reactions could not occur and HC is formed. It was also seen that HC increased with the addition of naphtha for a specific lambda. The lowest HC was measured with N50. HC increased about 24% and 17% with naphtha and N75 compared to n-heptane at $\lambda = 2$. Similar trend was seen on CO. The addition of naphtha caused to increase CO at a given lambda. In-cylinder temperature decreases with leaner charge mixture. It does not reach to auto-ignition temperature. Hence, CO is formed at low in-cylinder temperature. CO was measured 0.051%, 0.071% and 0.075% for n-heptane, N75 and naphtha respectively at $\lambda = 2$ and 1000 rpm.

Several reports have shown that alternative fuel caused to expand HCCI operating range. So, this study also set out with the aim of importance of naphtha in extending the HCCI operating range. Fig. 12 refers the HCCI operating range with fuel blends. Two important challenge called misfiring and knocking restrict the HCCI operating range as seen in Fig. 12. Fig. 12-a was formed at 1000 rpm engine speed to determine operating range. When Fig. 12-a is observed, it was clearly seen that HCCI combustion was achieved with leaner mixtures compared to n-heptane. Higher octane number and density allowed to achieve HCCI combustion for naphtha fuel blends with higher lambda. It can be also said that in-cylinder temperature increased much due to higher boiling point of naphtha at the end of compression time. This is the key role for HCCI combustion process. As it is known, HCCI operating range was extended in knocking zone due to higher octane number. HCCI combustion was achieved at higher engine speed with fuel blends according to n-heptane. N25 and N50 showed reasonable performance at higher engine speed in view of stable HCCI combustion. similarly, naphtha caused to expand HCCI operating range in knocking zone.

4. Conclusions

The presents study was designed to determine the effects of naphtha on HCCI performance, combustion, emissions and operating range. This research has demonstrated, for the first time, that HCCI combustion was achieved with naphtha/n-heptane fuel blends at leaner mixtures compared to pure n-heptane. HCCI combustion was achieved with higher engine speed with N25 and N50 test fuels. Another implication was observed that ITE increased 29% to 37% with naphtha compared to n-heptane at $\lambda = 2$ and 1000 rpm. Combustion was retarded with the addition of naphtha due to higher octane number. Combustion duration also increased with fuel blends. An additional finding is higher brake torque with naphtha according to n-heptane. Power and brake torque increased from 0.591 kW to 1.074 kW and 4.7 Nm to 8.55 Nm with naphtha compared to n-heptane respectively at 1200 rpm and $\lambda = 2$

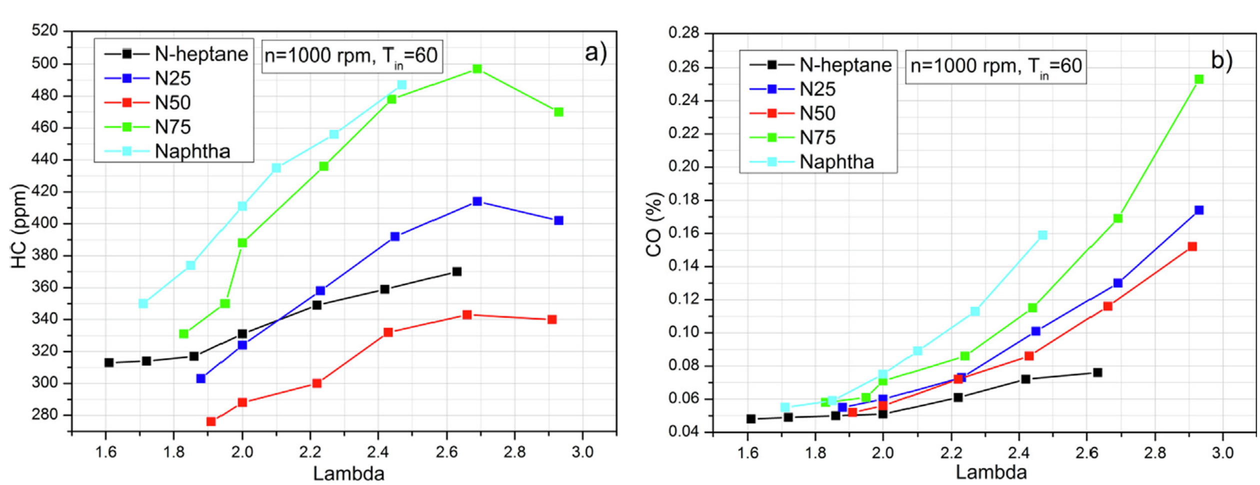

Fig. 11. HC and CO variations versus lambda at 1000 rpm with test fuels.

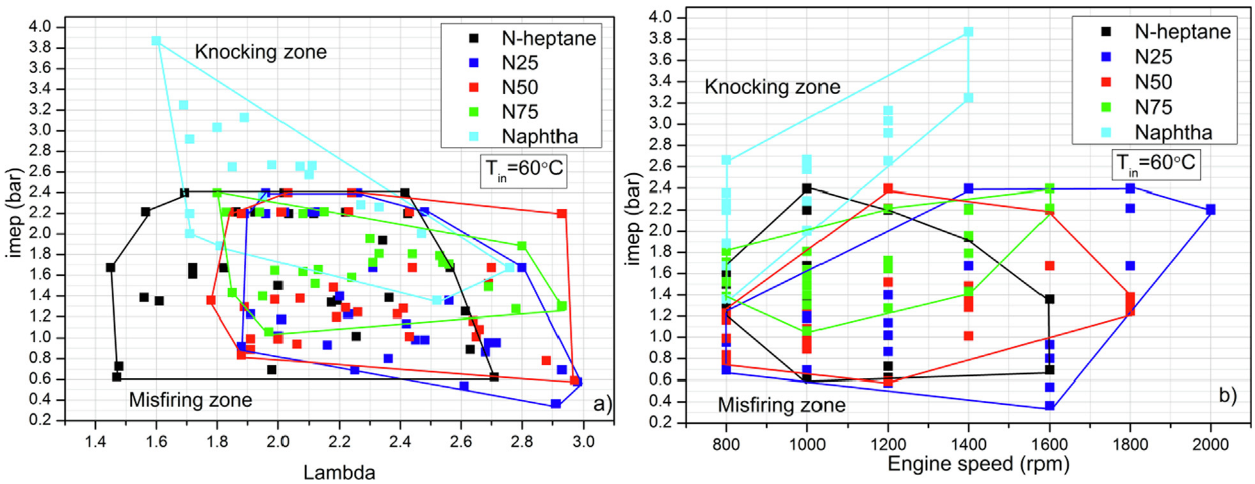

Fig. 12. HCCI operating range with fuel blends.

respectively. On the contrary, naphtha fuel blends were unable to improve CO and HC emissions. CO and HC increased 24% and 47% at $\lambda = 2$ with naphtha compared to n-heptane. The present investigation showed that naphtha can be effectively utilized in HCCI engine. These findings also showed that HCCI operating range can be extended with naphtha fuel blends.

CRediT authorship contribution statement

Samet Çelebi: Conceptualization, Data curation, Investigation, Methodology, Software, Visualization, Writing - review & editing. **Can Haşimoğlu:** Conceptualization, Investigation, Methodology, Supervision, Writing - review & editing. **Ahmet Uyumaz:** Resources, Validation, Visualization, Writing - original draft, Writing - review & editing. **Serdar Halis:** Investigation, Project administration, Resources, Software, Visualization. **Alper Calam:** Formal analysis, Investigation, Project administration, Validation, Writing - original draft, Writing - review & editing. **Hamit Solmaz:** Data curation, Formal analysis, Investigation, Methodology, Resources, Software, Supervision, Validation, Visualization, Writing - original draft. **Emre Yılmaz:** Conceptualization, Data curation, Formal analysis, Investigation, Methodology, Project administration, Resources, Software, Supervision, Validation, Writing - original draft, Writing - review & editing.

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.

References

[1] Azad AK, Rasul MG, Khan MMK, Sharma SC, Bhuiya MMK. Recent development of biodiesel combustion strategies and modelling for compression ignition engines. Renew Sustain Energy Rev 2016;56:1068–86.
[2] Onishi S, Jo SH, Shoda K, Jo PD, Kato S. Active thermo atmosphere combustion – A new combustion process for internal combustion engines. SAE Trans 1979;88:1851–60.
[3] Xingcai L, Yuchun H, Linlin Z, Zhen H. Experimental study on the auto-ignition and combustion characteristics in the homogeneous charge compression ignition (HCCI) combustion operation with ethanol/n-heptane blend fuels by port injection. Fuel 2006;85:2622–31.
[4] Yao M, Zheng Z, Liu H. Progress and recent trends in homogeneous charge compression ignition (HCCI) engines. Prog Energy Combust Sci 2009;35:398–437.
[5] Khaliq A, Triverdi SK, Dincer I. Investigation of a wet ethanol operated HCCI engine based on first and second law analyses. Appl Therm Eng 2011;31:1621–9.
[6] Khandal SV, Banapurmath NR, Gaitonde VN. Performance studies on homogeneous charge compression ignition (HCCI) engine powered with alternative fuels. Renew Energy 2019;132:683–93.
[7] Das P, Subbarao PMV, Subrahmanyam JP. Control of combustion process in an HCCI-DI combustion engine using dual injection strategy with EGR. Fuel 2015;159:580–9.
[8] Pacheco AF, Martins MES, Zhao H. New European Drive Cycle (NEDC) simulation of a passenger car with a HCCI engine: Emissions and fuel consumption results. Fuel 2013;111:733–9.
[9] Djermouni M, Ouadha A. Comparative assessment of LNG and LPG in HCCI engines. Energy Procedia 2017;139:254–9.
[10] Neshat E, Saray RK, Parsa S. Numerical analysis of the effects of reformer gas on supercharged n-heptane HCCI combustion. Fuel 2017;200:488–98.
[11] Kozarac D, Taritas I, Vuilleumier D, Saxena S, Dibble RW. Experimental and numerical analysis of the performance and exhaust gas emissions of a biogas/n-heptane fueled HCCI engine. Energy 2016;115.
[12] Halis S, Nacak C, Solmaz H, Yilmaz E, Yucesu HS. Investigation of the effects of octane number on combustion characteristics and engine performance in a HCCI engine. J Therm Sci Tech 2018;38(2):73–84.
[13] Li G, Zhang C, Zhou J. Study on the knock tendency and cyclical variations of a HCCI engine fueled with n-butanol/n-heptane blends. Energy Convers Manage 2017;133:548–57.
[14] Uyumaz A. An experimental investigation into combustion and performance characteristics of an HCCI gasoline engine fueled with n-heptane, isopropanol and n-butanol fuel blends at different inlet air temperatures. Energy Convers Manage 2015;98:199–207.
[15] Kim K, Wang Z, Wang B, Shuai S, Yang H, Bae C. Load expansion of naphtha multiple premixed compression ignition (MPCI) and comparison with partially premixed compression ignition (PPCI) and conventional diesel combustion (CDC). Fuel 2014;136.
[16] Javed T, Nasir EF, Ahmed A, Badra J, Djebbi K, Beshir M, et al. Ignition delay measurements of light naphtha: A fully blended low octane fuel. Proc Combust Inst 2017;36:315–22.
[17] Badra J, Viollet Y, Elwardany A, Im HG, Chang J. Physical and chemical effects of low octane gasoline fuels on compression ignition combustion. Appl Energy 2016;183:1197–208.
[18] Kim DS, Lee CS. Improved emission characteristics of HCCI engine by various premixed fuels and cooled EGR. Fuel 2006;85:695–704.
[19] Maurya RK, Agarwal AK. Experimental study of combustion and emission characteristics of ethanol fuelled port injected homogeneous charge compression ignition (HCCI) combustion engine. Appl Energy 2011;88:1169–80.
[20] Polat S, Yucesu HS, Kannan K, Uyumaz A, Solmaz H, Shahbakhti M. Experimental comparison of different injection timings in an HCCI engine fueled with n-heptane. Int J Auto Sci Tech 2017;1:1–6.
[21] Gharehghani A. Load limits of an HCCI engine fueled with natural gas, ethanol, and methanol. Fuel 2019;239:1001–14.
[22] Hasan MM, Rahman MM, Kadirgama K. A review on homogeneous charge compression ignition engine performance using biodiesel-diesel blend as a fuel. Int J Auto Mech Eng 2015;11:2199–211.
[23] Ganesh D, Nagarajan G. Homogeneous charge compression ignition (HCCI) combustion of diesel fuel with external mixture formation. Energy 2010;35:148–57.
[24] Azad AK, Rasul MG, Bhatt C. Combustion and emission analysis of Jojoba biodiesel to assess its suitability as an alternative to diesel fuel. Energy Procedia 2019;156:159–65.
[25] Sudheesh K, Mallikarjuna JM. Diethyl ether as an ignition improver for biogas homogeneous charge compression ignition (HCCI) operation - An experimental investigation. Energy 2010;35:3614–22.
[26] Hou YC, Lu XC, Zu LL, Ji LB, Huang Z. Effect of high-octane oxygenated fuels on n-heptane fueled HCCI combustion. Energy Fuels 2006;20:1425–33.
[27] Yilmaz E. Investigation of the effects of diesel-fusel oil fuel blends on combustion, engine performance and exhaust emissions in a single cylinder compression ignition engine. Fuel 2019;255:115741.
[28] Yang H, Shuai S, Wang Z, Wang J. Fuel octane effects on gasoline multiple premixed compression ignition (MPCI) mode. Fuel 2013;103:373–9.

[29] Yang H, Shuai S, Wang Z, Wang J, Xu H. New premixed compression ignition concept for direct injection IC engines fueled with straight-run naphtha. Energy Convers. Manage. 2013;68:161–8.
[30] Yang H, Shuai S, Wang Z, Wang J. High efficiency and low pollutants combustion: Gasoline multiple premixed compression ignition (MPCI). SAE Int 2012. 2012-01-0382.
[31] Wang B, Yang H, Shuai S, Wang Z, He X, Xu H, Wang J. Numerical resolution of multiple premixed compression ignition (MPCI) mode and partially premixed compression ignition (PPCI) mode for low octane gasoline. SAE Int 2013. 2013-01-2631.
[32] Akihama K, Kosaka H, Hotta Y, Nishikawa K, Inagaki K, Fuyuto T, Iwashita Y, Farrell JT, Weissman W. An investigation of high load (Compression ignition) operation of the "Naphtha Engine"-A combustion strategy for low well-to-wheel CO_2 emissions. SAE Int J Fuels Lubricants 2008. 2008-01-1599.
[33] Chang J, Kalghatgi G, Amer A, Viollet Y. Enabling high efficiency direct injection engine with naphtha fuel through partially premixed charge compression ignition combustion. SAE Tech. Pap. 2012, 2012-01-0677.
[34] Leermakers C, Bakker P, Somers L, Goey P, Johansson B. Commercial naphtha blends for partially premixed combustion. SAE Int J Fuels Lubricants 2013;6:199–216.
[35] Viollet Y, Chang J, Kalghatgi G. Compression ratio and derived cetane number effects on gasoline compression ignition engine running with naphtha fuels. SAE Int. J Fuels Lubricants 2014;7:412–26.
[36] Zhang Y, Voice A, Tzanetakis T, Traver M, Cleary D. An evaluation of combustion and emissions performance with low cetane naphtha fuels in a multicylinder heavy-duty diesel engine. J Eng Gas Turbine Power 2016;138. 102805 (1-10).
[37] Hao H, Liu F, Liu Z, Zhao F. Compression ignition of low-octane gasoline: Life cycle energy consumption and greenhouse gas emissions. Appl Energy 2016;181:391–8.
[38] Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill; 1988.
[39] Solmaz H, Kocakulak T. Buji ile Ateşlemeli Motor Kullanılan Seri Hibrit Elektrikli Bir Aracın Modellenmesi. Proceedings on International Conference on Technology and Science 2018.
[40] Kocakulak T, Solmaz H. Ön ve son iletimli paralel hibrit araçların bulanık mantık yöntemi ile kontrolü ve diğer güç sistemleri ile karşılaştırılması. Gazi Üniversitesi Mühendislik Mimarlık Fakültesi Dergisi 2020;35(4):2269–86. https://doi.org/10.17341/gazimmfd.709101.
[41] Kocakulak T, Solmaz H. HCCI MENZİL ARTTIRICI MOTOR KULLANILAN SERİ HİBRİT BİR ARACIN MODELLENMESİ. Gazi Üniversitesi Fen Bilimleri Dergisi Part C: Tasarım ve Teknoloji 2020;8(2):279–92. https://doi.org/10.29109/gujsc.670564.
[42] Calam A, Aydoğan B, Halis S. The comparison of combustion, engine performance and emission characteristics of ethanol, methanol, fusel oil, butanol, isopropanol and naphtha with n-heptane blends on HCCI engine. Fuel 2020;266. https://doi.org/10.1016/j.fuel.2020.117071.
[43] Calam A. Effects of the fusel oil usage in HCCI engine on combustion, performance and emission. Fuel 2020;262. https://doi.org/10.1016/j.fuel.2019.116503.
[44] Uyumaz A, Aydoğan B, Calam A, Aksoy F, Yılmaz E. The effects of diisopropyl ether on combustion, performance, emissions and operating range in a HCCI engine. Fuel 2020. https://doi.org/10.1016/j.fuel.2019.116919.
[45] Solmaz H. A comparative study on the usage of fusel oil and reference fuels in an HCCI engine at different compression ratios. Fuel 2020;273. https://doi.org/10.1016/j.fuel.2020.117775.
[46] Aydoğan B. An experimental examination of the effects of n-hexane and n-heptane fuel blends on combustion, performance and emissions characteristics in a HCCI engine. Energy 2020;192. https://doi.org/10.1016/j.energy.2019.116600.
[47] Aydoğan B. Experimental investigation of tetrahydrofuran combustion in homogeneous charge compression ignition (HCCI) engine: Effects of excess air coefficient, engine speed and inlet air temperature. J Energy Inst 2020;93(3):1163–76. https://doi.org/10.1016/j.joei.2019.10.009.
[48] Aydoğan B. Combustion characteristics, performance and emissions of an acetone/n-heptane fuelled homogenous charge compression ignition (HCCI) engine. Fuel 2020. https://doi.org/10.1016/j.fuel.2020.117840.
[49] Ardebili SMS, Taghipoor A, Solmaz H, Mostafaei M. The effect of nano-biochar on the performance and emissions of a diesel engine fueled with fusel oil-diesel fuel. Fuel 2020. https://doi.org/10.1016/j.fuel.2020.117356.
[50] Yılmaz E. A Comparative Study on the Usage of RON68 and Naphtha in an HCCI Engine. Int J Auto Sci Technol 2020;4(2):90–7. https://doi.org/10.30939/ijastech.721882.
[51] Yeşilyurt MK, Doğan B, Erol D. Experimental assessment of a CI engine operating with 1-pentanol/diesel fuel blends. Int J Auto Sci Technol 2020;4(2):79–89. https://doi.org/10.30939/ijastech.708517.
[52] Örs İ, Sayın B, Ciniviz M. A comparative study of ethanol and methanol addition effects on engine performance, combustion and emissions in the SI engine. Int J Auto Sci Technol 2020;4(2):56–69. https://doi.org/10.30939/ijastech.713682.