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Performance, emissions and exergy analysis of a HCCI engine working with two different research octane number fuels

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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) is one of the candidates of combustion modes that promise the world an efficient combustion process as compared to the conventional. In this study, the combustion characteristics and exhaust gas emissions rate of a gasoline engine working on HCCI mode are experimentally investigated using a single-cylinder, four strokes, and water cooled research engine. The engine tests are performed with two different research octane number (RON) fuels: gasoline 95 and gasoline 98. The variations of the brake specific fuel consumption (BSFC), the exhaust gas emissions and the exhaust gas temperature with the engine load represented by the indicated mean effective pressure (IMEP) are compared for both fuels. In addition, in-cylinder pressure, heat release rate, energetic and exergetic efficiencies, exhaust exergy rate, heat transfer exergy rate and destruction exergy rate are compared. It was found that BSFC, exhaust gas emissions (except CO) and exhaust gas temperature were lower with gasoline 95. The heat release rates and its peak were obtained to be higher with gasoline 95. No effects on energetic efficiency, exergetic efficiency and exhaust exergy destruction were found. However, exhaust exergy rate and heat transfer exergy rate were found to be affected by fuel octane number.

NOMENCLATURE

$\dot{E}$	energy rate (kW)
$\dot{Ex}$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
H	heating value (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
$\bar{R}$	universal ideal gas constant (kJ/kg·K)
s	specific entropy (kJ/kg·K)
$\dot{S}$	entropy production rate (kW/K)
T	temperature (^{o}C or K) or torque (N.m)
$\dot{W}$	work rate or power (kW)
Y	mole fraction (%)

Greek symbols

ε	specific flow exergy (kJ/kg)
Ψ	exergetic efficiency (%)
η	thermal efficiency (%)
η_{ite}	indicated thermal efficiency (%)
η_{vol}	volumetric efficiency (%)
ω	angular velocity (rad/s)
φ	chemical exergy factor

Subscripts

0	reference (dead) state
cw	cooling water
dest	destruction
e	environment
ex	exhaust
in	input
out	output
u	lower

Abbreviations

AFR	air fuel ratio
BSFC	brake specific fuel consumption
CAD	crank angle degree
EGR	exhaust gas recirculation
EVC	exit valve close
EVO	exit valve open
HC	hydrocarbon
HCCI	homogenous charge compression ignition
SI	spark ignition
IMEP	indicated mean effective pressure
IVC	inlet valve close
IVO	inlet valve open
NVO	negative valve overlap
PM	particulate matter
RON	research octane number
SI	spark ignition
TDC	top dead center
WOT	wide open throttle

1. INTRODUCTION

The modern society has absolute dependence on internal combustion engines with both types of spark ignition (SI) and compression ignition (CI) in uses such as transportation performed by millions of vehicles on road and at sea, and power generation engines to reach living standard. This dependency forced researchers across the world to improve their performance and fix its operational negatives. HCCI has a high efficiency of CI engine and very low emissions of SI engine. For the advantages of HCCI relative to SI engines, it has less fuel consumption, more efficient on their performance and less nitrogen oxides (NO_x) emissions. Furthermore, unthrottled part load operation eliminates pumping losses leading to improved fuel economy over SI engines. In addition, compared with CI engines, HCCI engines have lower emissions of particulate matter (PM) and NO_x due to its lower flame temperature [1-6]. Although recent

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investigations on HCCI combustion appear promising, several problems can appear.

A great deal of research has been carried out to understand and overcome the various problems associated with the operation of HCCI engines [7-13]. Yang et al. [14] observed higher fuel consumption with HCCI than SI engines.

The stability of HCCI combustion can be improved by controlling the start of combustion [15-17], which is advantageous of low fuel consumption, low emission and widely operation region.

The improvement of HCCI combustion efficiency has also been investigated by various researchers [11,18-21]. The improved efficiency comes from three sources: the elimination of throttling losses, the use of high compression ratios, and a shorter combustion duration because it already works with homogeneous premixed charge and is not necessary for a flame to propagate across the cylinder.

Different strategies have been investigated to improve fuel conversion efficiency and extend the operating load of HCCI engines [22-29] including negative valve overlap (NVO) [22,23], variable compression ratio [24,25] and split injection [26-28]. In addition, the dual-fuel-injection system resulted in a high load extension; retarded the ignition timing, the combustion duration remained longer and the maximum cylinder pressure reduced considerably, which is necessary for HCCI high load extension [29].

Energy and exergy analyses play a very essential role in identifying efficiencies of all thermal systems and deducting its potential for further improvements. Exergy analysis is based on the second law of thermodynamics. In recent years, it has been widely used for simulation, analysis and performance assessment of various types of thermal systems [30]. In this regard, even though significant numbers of energy and exergy analyses have been published for many kinds of thermal systems, the number of studies on internal combustion engines is relatively low.

Most of the publications are concerning with the application of the second-law of thermodynamics to internal combustion engines [31-33]. In regard of types of internal combustion engines, i.e. SI, CI (direct or indirect injection), turbocharged or naturally aspirated, a comprehensive review was made to collect the finding of various researchers in the field over the last 40 years by [34]. A comprehensive review pertaining to fundamental studies on thermodynamic irreversibility and exergy analysis in the processes of combustion of gaseous, liquid and solid fuels was made by [35]. Energy and exergy analysis of Otto and Diesel engines for previous studies from 1963 to 2008 was reviewed by [36].

The destruction of availability (exergy) during combustion processes has been examined either for an adiabatic, constant volume system [37] as a function of temperature, pressure, and equivalence ratio for octane–air mixtures or as a function of operating and design parameters for a SI engine [38]. The effect of varying dead state temperatures on exergy efficiency of internal combustion engine was investigated by various investigators [30, 39-41]. The exergy cost analysis [36], and sustainability assessment of a diesel engine as well as the exergy potential of the exhaust gas [42] were studied while energetic and exergetic analyses of T56 turboprop engine at various power loading operation modes were done [43]. A conceptual wet ethanol operated HCCI engine was proposed by Khaliq et al. [44] to shift the energy balance in favor of ethanol. Energy and exergy analyses were applied to this engine. Khaliq and Trivedi [45] also performed first and second law analyses of a new combined power cycle based on wet ethanol fuelled HCCI engine and an organic Rankine cycle using a computational analysis.

As can be seen from the literature survey, no studies on performance comparison of gasoline 95 and gasoline 98 as well as their exergetic assessment based on the operational data have appeared in the open literature to the best of the authors` knowledge. This provided the main motivation behind this contribution.

In this regard, the main objectives of the present research are to: (i) investigate the effect of fuel octane number on the performance and emissions of a gasoline HCCI engine experimentally, (ii) undertake a parametric study for investigating the effect of working parameters on the engine performance, and (iii) calculate energy and exergy efficiencies of the HCCI engine studied.

2. EXPERIMENTAL SETUP

The engine used in the experiments is a single-cylinder, four stroke, water cooled SI research engine. Engine specifications are given in Table 1. A schematic of the engine setup is illustrated in Fig 1. The cylinder head is featured by a centrally mounted fuel injector and pent roof combustion chamber with four valves. The injection system is a spray guided type commercially available from Bosch. The air aspirated into the engine is controlled by an electronically operated throttle valve and the engine was operated in a wide open throttle (WOT) conditions. The engine crankshaft is connected to an eddy current dynamometer (Froude Hoffman, AG30) with a maximum power of 30 kW. The engine performance and exhaust gas emissions data are logged using Texcel V10. In-cylinder pressure was measured by a water cooled piezo-electric pressure sensor (Kistler 6061B). Intake and exhaust pressure were measured by a piezo-resistive pressure sensors (Kistler 4045A10). Signals from all pressure sensors were coupled to a charge amplifier (Kistler 4618A0), and were subsequently recorded using AVL’s Indicom software V1.6 based data acquisition system. Off-line post processing was then carried out to extract the heat release and other combustion parameters. In addition to in-cylinder pressure, intake manifold and exhaust manifold pressure data and gasoline direct injection voltage profiles were also recorded by Indicom. The fuel was injected directly into the cylinder by using a Bosch fuel injector. The fuel injection was done with a d-Space-

Micro Auto-box ECU (Engine Control Unit), which provides accurate control of the injection timing as well as the injection duration. The fuel consumption was measured by using AVL fuel consumption measurement system (AVL KMA 4000). The air fuel ratio (AFR) ratio was calculated using ECM AFR meter and the reading from such meter was based on amount of air flow rate in the intake manifold along with the amount of fuel injected without taking into account the already existing gases in the engine cylinder.

Table 1 Specifications of the tested engine.

Engine type	Single Cylinder, 4 stroke
Bore / Stroke	80.5 mm / 88.2 mm
Displacement	448.985 cm^3
Compression ratio	10.5:1
Connecting rod	141.0 mm
Aspiration	Natural
IVO / IVC	451° / 589°
EVO / EVC	131° / 269 °
Valves lift	3.6 mm (HCCI)

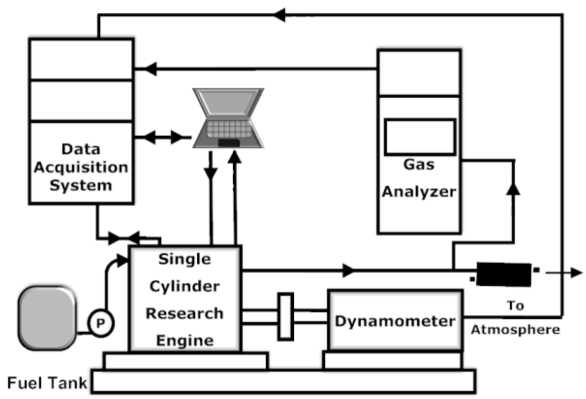

Fig. 1. Schematic of engine test cell.

The exhaust gas emissions such as carbon monoxide (CO), carbon dioxide (CO_2), unburned hydrocarbons (HC) and NOx were measured using a Horiba exhaust gas analyzer (MEXA-7170DEGR). MEXA-7170DEGR is a measurement unit consists of four analyzer modules (Horiba AIA-72X, Horiba MPA-720, Horiba FIA-725A and Horiba CLA-720A) controlled by a PC acting as the main control unit. CO and CO_2 concentrations are measured by a Horiba AIA-72X analyzer, HC concentration is measured by a Horiba FIA-725A Flame Ionization Detector and NO/NOx concentrations are measured by a Horiba CLA-720A chemiluminescence analyzer.

The HCCI combustion mode was achieved by trapping sufficient amounts of residual gases in the cylinder by using the low lift and short duration of intake and exhaust camshafts, as indicated in Table 1. As exhaust valves were closed early in the exhaust stroke, the cylinder pressure was increased by compression of the remained residuals trapped in the cylinder and then decreased as the piston started to descend, resulting in a re-compression and re-expansion process around top dead center (TDC) of the intake. Although the engine had a low compression ratio of 10.5:1, auto-ignition was achieved by employing two strategies: NVO and split injection as shown in Fig 2. The percentage of internal exhaust gas recirculation achieved was strongly dependent on the speed of the engine, valve timing, throttle opening, temperature and pressure of the mixture just before exhaust valve opens (EVO). With the camshaft of the tested engine and at a speed of 2500 rpm under WOT conditions, an internal exhaust gas recirculation of 50-60% could be achieved. The injection was divided into two stages namely: pilot and main injections. Pilot injection was the first stage of injection with the highest percentage of the fuel by mass compared to the main injection. Pilot injection occurred after exhaust valve closes (EVC) in the NVO region during the compression of the trapped exhaust gases. The injected fuel was thought to react with chemical radicals already present in the exhaust gas, but did not auto-ignite. Main injection of the fuel occurred just before the TDC with the lowest percentage of the fuel by mass is injected during this stage.

The compression of air-fuel-exhaust gases mixtures raised the temperature inside the cylinder to a degree enough to auto-ignites the mixture. The engine was run with two different RON fuels of 95 and 98. No fuel or air preheating was performed and all experiments were undertaken at the normal room temperature and the experiments were carried out at a constant speed of 2500 rpm.

3. ENERGY AND EXERGY ANALYSIS

3.1 Energy analysis

For a steady-state open thermodynamic system [41]:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} + \dot{W} = \sum \dot{m}_{out}\, h_{out} - \sum \dot{m}_{in}\, h_{in} \quad (2)$$

where Eqs. (1) and (2) are mass and energy balance for a control volume of a general thermodynamic open system, respectively. The subscripts in and out represent inlet and exit (output) states, respectively while $\dot{Q}$ denotes the heat rate, $\dot{W}$ the work rate, $\dot{m}$ the mass flow rate, and h the specific enthalpy. These balances describe what is happening to a system with an instantaneous time between two instants of time.

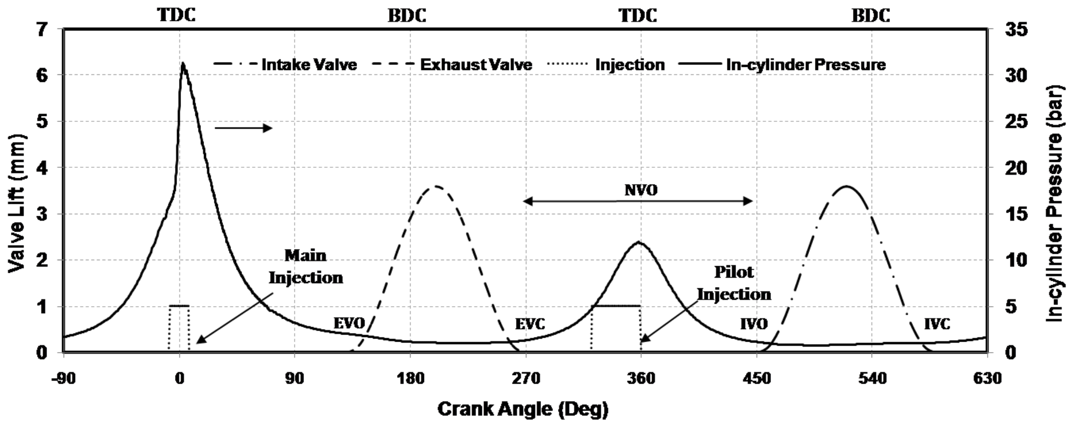

Fig. 2. Valve events along with a sample of in-cylinder pressure for HCCI operation.

For a complete cyclic process where the initial and final states of the system are identical, the accumulation terms in all the balances are zero [46]. Mass, energy, and exergy rate equations are generally applied to steady-state engine test procedures. Transient engine cycles include many steady-state full and partial load conditions and their linear transitions. Therefore, it is possible to apply the balance equations onto the transient engines cycles; however it requires integrations over the whole cyclic time period [41].

Net work rate of the control volume is calculated with experimental data using the following equation where ω is the angular velocity and T is the torque.

$$\dot{W} = \omega T \quad (3)$$

Energy input rate to the control volume is calculated as follows:

$$\dot{E}_{fuel} = \dot{m}_{fuel} H_u \quad (4)$$

where H_u is the lower heating value and $\dot{m}_{fuel}$ is the mass flow rate of the fuels.

The heat losses of the control volume are evaluated as the differences between the energy input rate and net work rate by

$$\dot{Q}_{loss} = \dot{E}_{fuel} - \dot{W} \quad (5)$$

Thermal efficiency of the control volume is calculated as the ratio of the net work rate to the fuel energy input rate from

$$\eta = \frac{\dot{W}}{\dot{E}_{fuel}} \quad (6)$$

3.2 Exergy analysis

Exergy analysis is widely gaining acceptance over traditional energy methods in both industry and academia because it clearly tells about the efficient utilization of resources (fuels) and explains the benefits of sustainable energy and technology.

Exergy balance of a control volume can be obtained using Eq.7 [41]:

$$\dot{Ex}_{heat} + \dot{Ex}_W = \sum \dot{m}_{in}\,\varepsilon_{in} - \sum \dot{m}_{out}\,\varepsilon_{out} - \dot{Ex}_{dest} \quad (7)$$

where $\dot{Ex}_{heat}$ is the exergy transfer rate associated with heat transfer at temperature T, $\dot{Ex}_W$ is the exergy work rate, $\dot{m}$ is the mass flow rate, ε is the specific flow exergy and $\dot{Ex}_{dest}$ is the exergy destruction (irreversibility) rate.

Exergy rate due to heat transfer from the cooling water, T_{cw} to the environment is defined by

$$\dot{Ex}_{heat} = \sum\left(1 - \frac{T_0}{T_{cw}}\right)\dot{Q} \quad (8)$$

where $\dot{Q}$ is the output heat rate from the engine to the environment through the cooling water of the engine, which is given by

$$\dot{Q} = \dot{m}_{fuel} H_u - \left(\dot{Ex}_W + m_{out}\,\Delta h_{out}\right) \quad (9)$$

where Δh = h-h_0, while h and h_0 is enthalpies of the exhaust gases at measured exhaust temperature h_0 is enthalpies of the exhaust gases at reference (dead) state temperature. In there, m_{out} is the total mass of exhaust gas species and Δh_{out} is calculated from the sum of enthalpy differences (h-h_0) of all exhaust gas species.

Net exergy work rate is equal to the net energy work rate as follows:

$$\dot{Ex}_W = \dot{W} \quad (10)$$

Input exergy rate, includes only chemical exergy which can be described by

$$\dot{Ex}_{in} = \dot{m}_{fuel}\,\varepsilon_{fuel} \quad (11)$$

where ε_{fuel} is the specific exergy of the fuel and can be defined as follows:

$$\varepsilon_{fuel} = H_u \varphi \quad (12)$$

where φ is the chemical exergy factor, which is described for non sulphur fuels as follows [47]:

$$\varphi = 1.0401 + 0.1728\,\frac{h}{c} + 0.0432\,\frac{o}{c} \quad (13)$$

where "h" is the hydrogen mass rate in %, "c" is the carbon mass rate in %, "o" is the oxygen mass rate in %. This equation is taken from [47], h", "c" and "o" values of the fuels must be obtained first. (They can be analyzed at chemical laboratories). Then, the values can be used in Eq. 13.

Output exergy, which is the so-called exhaust exergy, contains thermo-mechanical (or physical) and chemical exergies. Specific thermo-mechanical (physical) exergy can be defined by

$$\varepsilon_{tm} = (h - h_0) - T_0(s - s_0) \quad (14)$$

where s is the specific entropy, and the subscript "0" denotes the dead (reference) state.

Chemical exergy of the exhaust gases were calculated using Eq. 15. Mole fractions of the exhaust gases were calculated by balancing the real combustion equations of the fuels by means of the emission measurement.

$$\varepsilon_{chem} = \bar{R}T_0 \ln\frac{y}{y^e} \quad (15)$$

where $\bar{R}$ is general gas constant (N_2, O_2, CO_2, H_2O, CO …), T_0 is the environment (dead state) temperature, y is the mole fraction of exhaust gas component. And y^e is the mole fraction of the component given under the definition of the environment.

Physical and chemical exergies of the exhaust gases must be calculated for the entire exhaust component (N_2, O_2, CO_2, H_2O, CO …). Then, the total exhaust exergy can be calculated as follows:

$$\dot{Ex}_{ex} = \sum \dot{m}_i \varepsilon_{top\,i} \quad (16)$$

where $\dot{Ex}_{ex}$ is the exhaust exergy, $\dot{m}_i$ is the total real mass rates of the combustion products and $\varepsilon_{top\,i}$ is the specific flow exergy of total of combustion products.
Exergetic efficiency can be found as

$$\Psi = \frac{\dot{Ex}_w}{\dot{Ex}_{in}} \quad (17)$$

4. RESULTS AND DISCUSSION

Figure 3 shows the speed-load map of engine operation in the HCCI mode. In HCCI combustion, low load limit is widely defined by high value of coefficient of variance (COV) of indicated mean effective pressure (IMEP) and high load limit is defined by knocking combustion (represented by pressure rise rate). Therefore, tests were subjected to the following criteria, pressure rise rate should not exceed 10 bar/CAD, and COV of IMEP should be less than 5%. In Fig. 3, the upper operating limit is the knocking limit above which the knock criterion is not met and the lower limit is the stability limit characterized by high cycle-by-cycle variations due to unstable combustion. The unstable combustion is the result of low residual gas temperatures and longer heat transfer time available in case of low speeds, thereby cooling the charge. Experimental tests have been done in the engine speed range of 2000 to 3500 rpm and the IMEP range of 1.5 to 3.3 bar. Most of the operation tests were done at 2500 rpm, which was found to be the best suited speed to do the experiment at various loads.

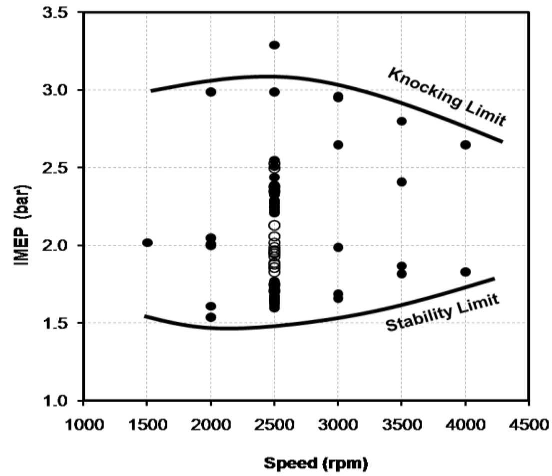

Fig. 3. Speed-Load map for HCCI combustion. Open symbols for gasoline 98 and solid symbols for gasoline 95.

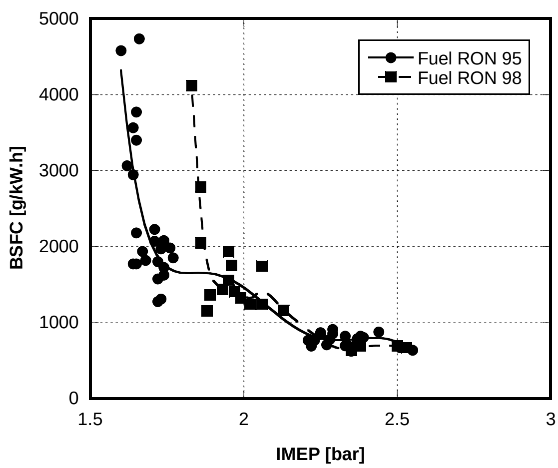

Fig. 4. Variation of BSFC with IMEP.

Figure 4 indicates a variation of the brake specific fuel consumption (BSFC) with IMEP for gasoline 95 and gasoline 98. Generally, it can be seen that, at low loads, the BSFC for gasoline 95 is lower than that obtained with gasoline 98, as shown in Fig. 3. This reveals a poor utilization of gasoline 98 due to mainly the lower temperature resulting in a slower combustion rate as will be observed from the results of the heat release rate analysis. On the other hand, at a high load, the improvement of gasoline 98 utilization leads to a relevant improvement of the BSFC. The BSFC with gasoline 98 tends to converge with the one under gasoline 95.

Figures 5 through 8 show a comparison of CO, CO_2, NOx and HC emissions rate of HCCI combustion mode per the brake power as a function of the engine load, respectively, when fuelled with both examined fuels. It can be observed that all emissions with gasoline 95 are lower than that with gasoline 98 except CO, which is initially lower for gasoline 95 at low engine loads, but tend to be higher than with gasoline 98 at higher loads. Hydrocarbon (HC) and CO emissions generally increase with delays in combustion timing because the temperatures achieved in the cylinder are too low for complete oxidation (CO oxidation requires roughly 1500 K [48]).

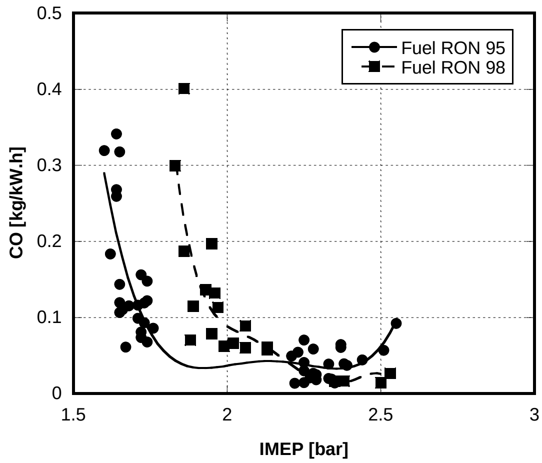

Fig. 5. Variation of CO emission with IMEP.

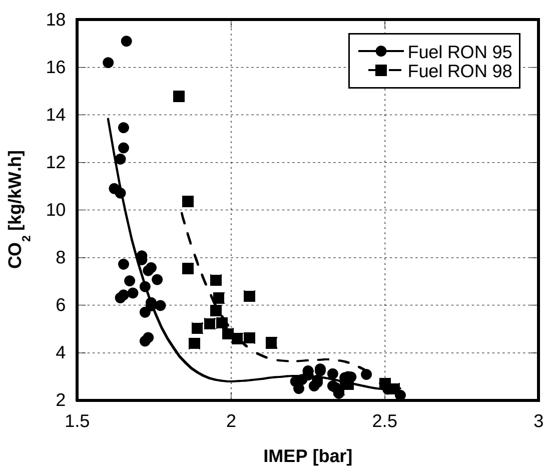

Fig. 6. Variation of CO_2 emission with IMEP.

As a result of the lower combustion temperatures, the concentration of OH radicals, which are critical for the oxidation process, are insufficient for achieving complete combustion. Early combustion timing (before TDC) could also have the same effect in lowering the combustion temperatures.

The trend for NO_x emissions, shown in Fig. 8, may be mainly due to lower combustion temperature with gasoline 95 and this trend can be partially explained in terms of exhaust gas temperatures, which are plotted in Fig. 9.

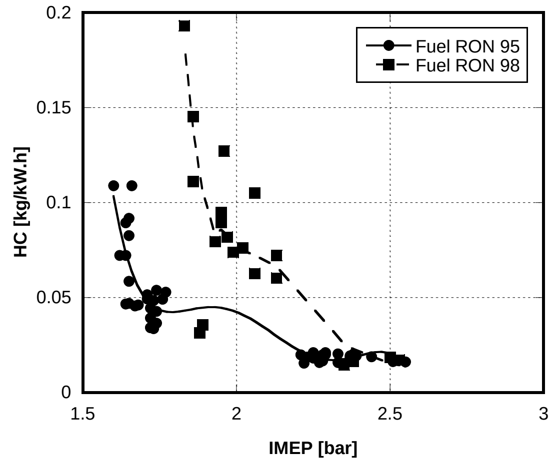

Fig. 7. Variation of HC emission with IMEP.

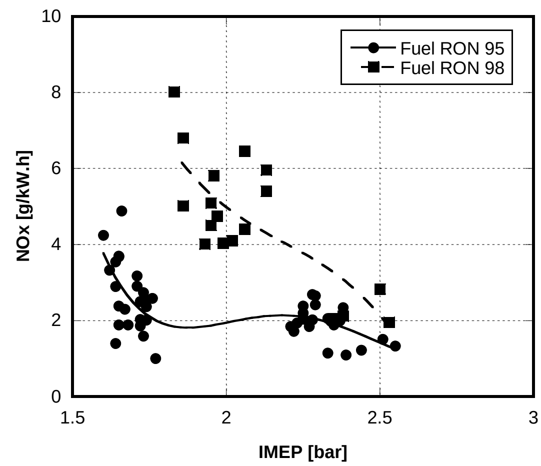

Fig. 8. Variation of NO_x emission with IMEP.

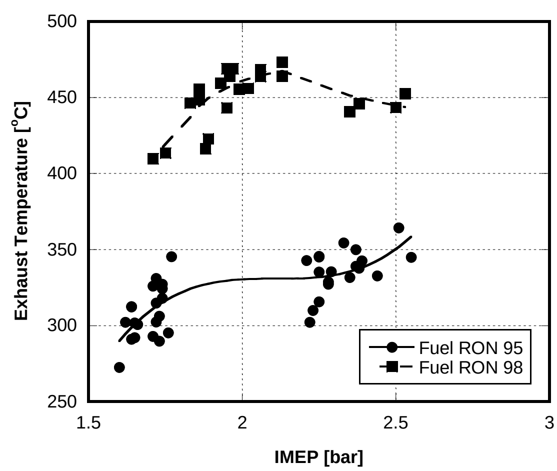

Fig. 9. Variation of exhaust gas temperature with IMEP.

Figure 9 illustrates the variation of exhaust gas temperature with IMEP. It can be observed that the exhaust gas temperature for gasoline 98 is higher compared to gasoline 95. Increasing the fuel octane number retarded the peak heat release rate as will be observed from the results of the heat release rate analysis. Since the fixed valve timings were used, the duration from the start of ignition to expel of exhaust gases was shorter for gasoline 98 when compared to gasoline 95. This resulted in higher exhaust gas temperatures with gasoline 98.

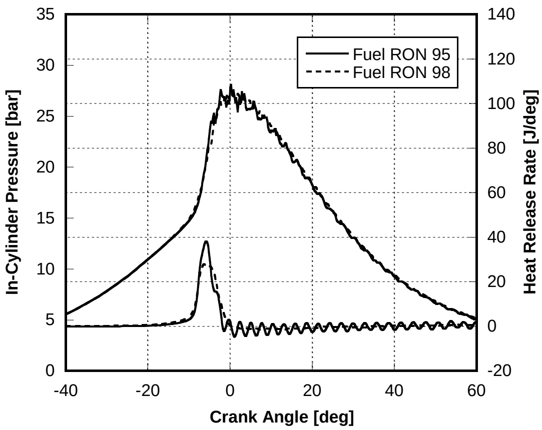

Fig. 10. In-cylinder pressure and heat release rate as a function of crank-angle at an IMEP of 1.70 bar.

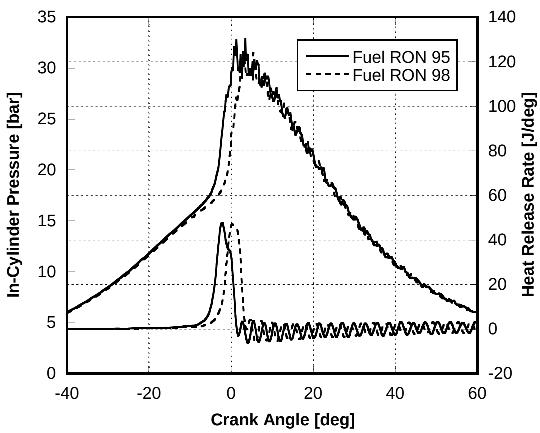

Fig. 11. In-cylinder pressure and heat release rate as a function of crank-angle at an IMEP of 2.37 bar.

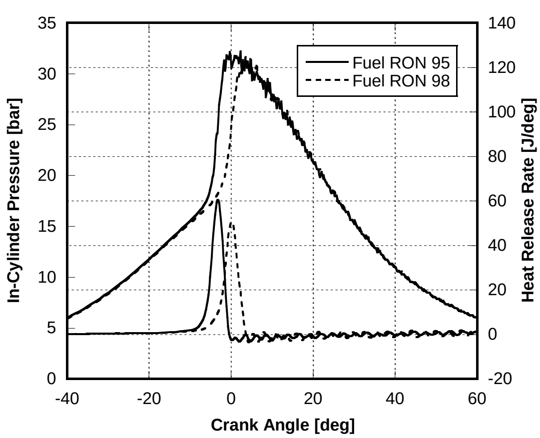

Fig. 12. In-cylinder pressure and heat release rate as a function of crank-angle at an IMEP of 2.60 bar.

Figures 10-12 show the variations of in-cylinder pressure and heat release rate with crank-angle for both fuels examined at IMEP values of 1.7, 2.37 and 2.6 bar. It can be seen that gasoline 95 had higher heat release rates and the peak rate of heat release was advanced before TDC compared to gasoline 98. However, gasoline 98 was observed to undergo smoother combustion than gasoline 95, as seen from in-cylinder pressure curves. It can be seen that the gasoline-air mixture has one stage auto-ignition (high temperature reaction). At engine load of 1.7 bar, combustion of both fuel starts nearly 10 CADs before TDC and the peak of heat release rate is higher for gasoline 95 (38 J/CAD compared to 27 J/CAD for gasoline 98). When engine load increases to 2.6 bar, the start of combustion is retarded to 8 and 6 CAD before TDC and the peak of heat release rate increases to 61 and 52 J/CAD, correspondently for gasoline 95 and 98, respectively. It is generally accepted that an HCCI combustion process with its auto-ignition timing is controlled primarily by chemical kinetics. Furthermore, intermediate and radical species present in the residuals at IVC may accelerate the chemical processes preceding the main heat release event for gasoline 95. However, advancing the ignition timing before TDC will increase the compression effort and in turn result in a reduction in net indicated thermal efficiency. The best combustion timing depends on a variety of factors and likely varies depending on the operating conditions. Following the study of Flowers et al. [49], the use of TDC combustion timing has been assumed as the best ignition timing.

As the temperature of the exhaust gas stream of gasoline 98 is higher than that of gasoline 95, it is important to investigate the availability of the exhaust energy of gasoline 98 compared to gasoline 95.

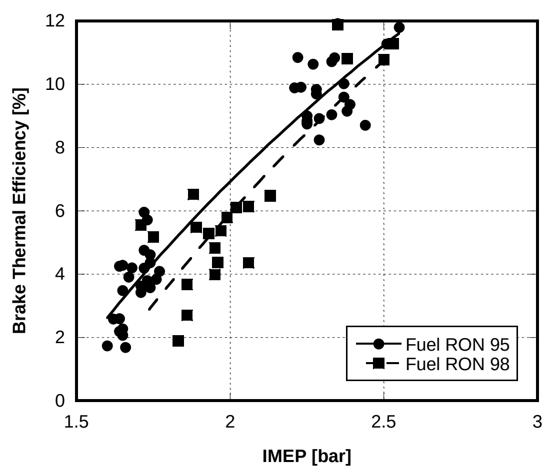

Fig. 13. Brake thermal efficiency versus IMEP.

Exergy is a thermodynamic concept that measures the maximum amount of useful work that can be extracted when a system is brought into equilibrium (thermal, mechanical and chemical) reversibly with the standard environment (T_0 = 298.15 K, P_0 = 1.013 bar) conditions. As a consequence of its great potential for optimization of systems and processes, in the last decades, availability analysis has received increased attention in the literature [30-47].

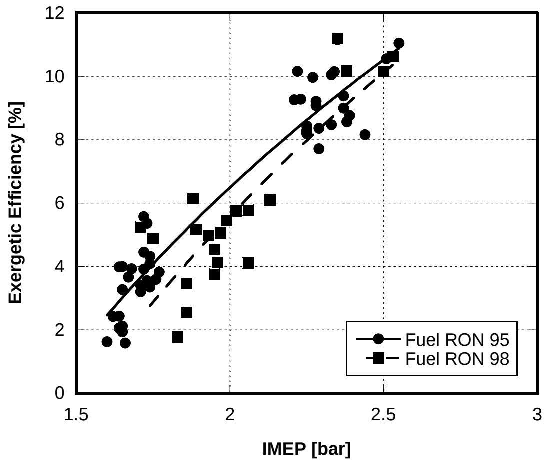

Fig. 14. Exergetic efficiency versus IMEP.

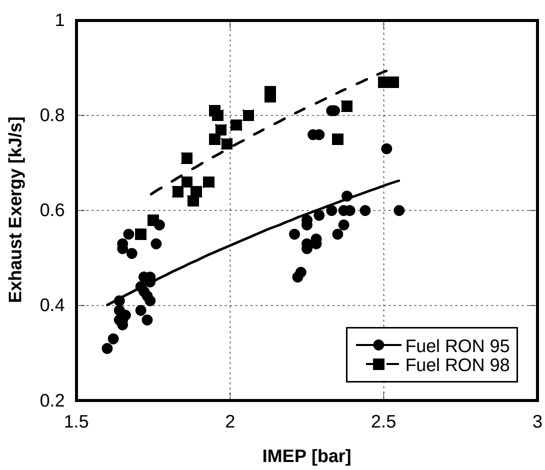

Fig. 15. Exhaust exergy versus IMEP.

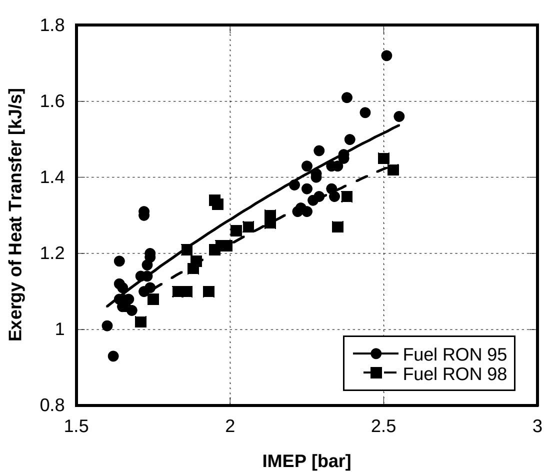

Fig. 16. Heat transfer exergy versus IMEP.

Figs. 13-17 provide the effect of using these two fuels in the HCCI engine operating mode. In general, it is observed that the energetic efficiency, the exergetic efficiency, the exhaust exergy rate, the heat transfer exergy rate and the destruction exergy rate increase with the increase in the engine load. For the exergetic efficiency and heat transfer exergy rate, gasoline 95 is higher than gasoline 98 whereas it is lower in case of exhaust exergy rate and exergy destruction. These results were due to mainly the different values obtained for exhaust gases temperatures.

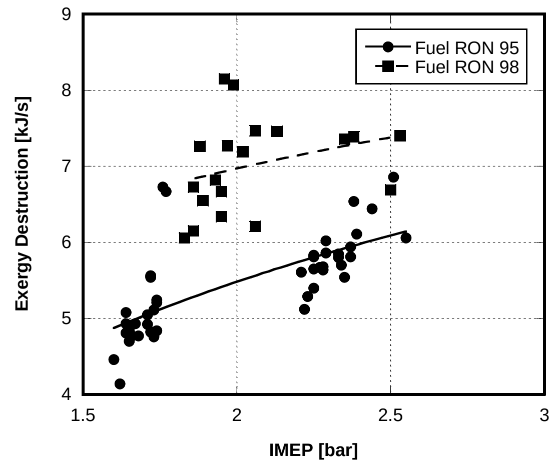

Fig. 17. Exergy destruction versus IMEP for.

4. CONCLUDING REMARKS

Homogeneous charge compression ignition is an alternative engine combustion mode to the conventional engines because it has less fuel consumption, more efficient on their performance, less NO_x and PM emissions and fuel-flexibility.

In this study, engine tests were performed to investigate the effect of fuel octane number on the performance and emissions of a gasoline engine working on HCCI mode. The following concluding remarks may be drawn from the results of the present study:

a) At low loads, the BSFC for gasoline 95 is lower than that obtained with gasoline 98. At higher loads, however, the BSFC for gasoline 98 tends to converge with that of gasoline 95.
b) Exhaust gas emissions rate with gasoline 95 is lower than that with gasoline 98 except CO, which is initially lower for gasoline 95 at low engine loads, but tend to be higher than that with gasoline 98 at higher loads.
c) Exhaust gas temperature for gasoline 98 is higher compared to that of gasoline 95.
d) Gasoline 95 has higher heat release rates and its peak rate of heat release was advanced before TDC compared to gasoline 98. However, in-cylinder pressure curves show that gasoline 98 was undergoing smoother combustion than gasoline 95.

e) The different fuel octane number has no effect on energetic efficiency, exergetic efficiency and exhaust exergy destruction. Conversely, it has been found that the different fuel octane number has an effect on exhaust exergy rate and heat transfer exergy rate. It results from the different values obtained for exhaust gases temperatures.

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