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Investigation of Performance and Emissions in a DISI Engine when Using Ethanol Blends as Fuel

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Abstract

Nowadays the major problem of automobile is extravagant of fuel consumption in transportation affected in many ways as currency and fuel economy dissipative including with environmental impurity. Those belong to environmental quality declination and lack of fossil fuel especially in transportation that uses lots of fuel mostly consumed by motor vehicle. So the opportunity to save currency and improve environmental quality through fuel use reductions is clear by using ethanol as renewable resources from benefits its properties together with direct injection spark ignition (DISI) engines.

The objective of this research presents effect of ignition timing on performance and emissions when fueled with ethanol. The four cylinder and four stroke of direct spark ignition engine (DISI engine) is tested and controlled spark timing by standalone electronic control unit. To test performance and emissions were conducted by using difference purity ethanol blended in gasoline (E10, E20 and E85) and gasoline (E0), variation of spark timing and stoichiometric A/F ratio which measured power, torque, break specific fuel consumption (BSFC) and break specific energy consumption (BSEC) at 2000 rpm of low and medium loads. The results showed that when compare with gasoline, Fuel blends with higher content of ethanol causes ignition timing to advance 3° CAD which obtained 26.79% more torque together with emissions have reduced likewise; carbon monoxide-31.03% and hydrocarbon-42.86%. However, carbon dioxide is higher than 4.86% due to quantities of fuel blended with ethanol has lower heating value.

***Keywords**:* Ethanol, Ignition Timing, Direct Injection Spark Ignition Engine, DISI

1. Introduction

With contribution to performance of engine, the fuel consumption in transportation field is an important reason to realize for propelling the motor vehicles. In particular, the total energy consumption in the world depends on the remaining fossil fuels. Using new development technologies from a direct injection spark ignition engine and renewable oxygenated fuels such as ethanol are considered as the most suitable solution for the future.

To investigate the ethanol characterization and optimization to an engine have been studied by many researchers. P. Ornman [1] found that ethanol has started to vaporize slower than ethanol but when terminated, ethanol was fully evaporated in to vapor phase faster than that of gasoline because of gasoline has more light and heavy fraction that ethanol. Hence, the lighter components leads to evaporated early at the beginning stage while ethanol wasn't start. After that, heavy fraction in gasoline which are comprised of various higher carbon atoms regardless to RVP properties still remained while ethanol which is comprised of only one component, lower carbon atom than the gasoline, already completely changed into the vapor phase.

The effect of ethanol-unleaded gasoline blends on engine performance and exhaust emission was studied by Hasan [2], 4-stroke SI-engine was fueled with gasoline and gasohol with various load and speed (1000 to 4000 rpm) at 75% of full throttle. The results founded higher percentage of ethanol increased the brake power, brake specific fuel consumption, thermal and volumetric efficiency. Carbon-monoxide (CO) and hydrocarbon (HC) were decreased 46.5% when compared with gasoline. However, carbon-dioxide was increased up to 7.5%.

Türköz and et al. [3] studied the use of E85 in the engine with 9.2:1 of compression ratio and advancing ignition timing from original gasoline timing. This study operated at wide open throttle ranging speeds from 2250, 2500, 3000 3500 to 4000 rpm, respectively. The results found that engine torque and power were increased when using E85 advanced ignition timing of 4 degree crank angle and causes minor effect of carbon-dioxide and carbon-monoxide reduction. On the contrary, nitrogen oxide, hydrocarbon and brake specific consumption were increased [4, 5].

However, using ethanol blends together with a DISI engine is interesting due to high octane number fuel and the optimization of spark advance which influence to better performance and reduction of emissions. Hence, investigation the effect from properties of ethanol and DISI engine operations is essential to develop insightful information.

The objective of this study has focused on the sensitivity of gasoline compare with ethanol as alternative fuel together with engine modification by adjust an ignition timing and control a lambda. The advantages from the results are discussed on performance and emissions in order to optimize the DISI engines when using ethanol as a fuel.

2. Experimental Apparatus

2.1 Fuel

Compared with gasoline, using ethanol as a fuel leads to benefits by outstanding properties for instance high research octane number (RON) which causes better anti-knock quality from increased of compression ratio that obtained better volumetric efficiency, the stoichiometric air fuel ratio is lower due to the fact that it has oxygen content inside its molecule leads to complete of combustion and reduce emissions from surplus of oxygen to react remains. However, ethanol must injected more quantities because of lower heating value to achieve same of output energy and cold start problem from high initial boiling point (IBP) when compare with gasoline as shown in table 1.

Table. 1 Fuel Properties [6, 7]

Fuels properties	Gasoline	E10	E20	E85	Ethanol
Formula	C_4 to C_{12}	$CH_{2.043}O_{0.015}$	$CH_{1.63}O_{0.065}$	$CH_{2.822}O_{0.425}$	C_2H_5OH
MW. [g/mol]	100 - 105		88.12	50.60	46.70
Carbon [mass%]	85-88	86.70	79.85	55.36	52.20
Hydrogen [mass%]	12 - 15	13.2	12.88	12.89	13.1
Oxygen [mass%]	0	1.94	7.54	31.75	34.70
Density, kg/l, at 15°C	0.72-0.77	0.7608	0.7645		0.79
RVP at 37.8°C, kPa	48-103	59.60	58.30	35-70	15.90
LHV, MkJ/lkg	44.00	40.97	40.60	29.50	26.90
HOV, kJ/kg	305.00	-	-	610-762.5	840.0
RON	92.0-98.0	96.0	98.3	101.6	107.0
MON	80.0-90.0	84.1	84.6	91.1	89
Stoichiometric air/fuel ratio	14.70	14.05	13.51	9.87	9.03
Distillation temperature, °C					
Initial boiling point, IBP	35	39.7	42.1	46.9	77.6
10 vol%	51.5	52.5	54.3	68.6	77.8
50 vol%	78.2	79.1	70.5	77.6	77.9
90 vol%	154	153.6	150.9	78.5	78
End boiling point	197.3	186.1	182.8	79.7	80

2.2 Engine

Gasoline direct injection (GDI) or direct injection spark ignition (DISI) engine is a variant of fuel injection employed in modern engines. The gasoline is highly pressurized and injected directly into the combustion chamber of each cylinder, as opposed to conventional multi-point fuel injection in which fuel is injected in the intake port. In some applications, gasoline direct injection enables a stratified fuel charged (ultra-lean burn) combustion for improving fuel efficiency and reducing emission levels at low load.

The in-line 4 cylinders with 1,834 cm^3 displacement of DISI engine is operated in this experiment which shown in table 2.

Table. 2 Engine Specifications

Model	MITSUBISHI 4G93 GDI
Type	In-Line OHV, DOHC 16 Valve
Number of Cylinder	4
Displacement (cm^3)	1,834
Compression Ratio	12 : 1
Bore x Stroke (mm)	81.0 x 89.0
Maximum Output	96 kW @ 6000 rpm
Maximum Torque	177 Nm @ 3750 rpm

2.3 Engine Dynamometer

The "Tokyo Plant 150 PS Model" engine dynamometer interfaced with in-house program has measured the results of this experiment likewise; power, torque, brake specific fuel consumption as shown in Fig. 1.

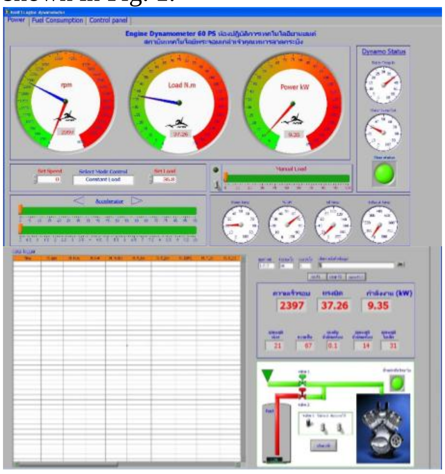

Fig. 1 Engine dynamometer with in-house controller

2.4 Electronic Control Unit (ECU)

The DTA fast S60 Pro is standalone electronic control unit of experimental engine which controls quantity, duration of fuel injection and spark timing with precision resolution as presented in Fig. 2.

Fig. 2 DTA fast S60 Pro user interface

3. Methodology

A DISI engine has tested at 2000 rpm with 20% and 40% of load which represent as light and middle load condition. Gasoline and ethanol blended fuels (E10, E20 and E85) are used in this experiment which runs on homogeneous charge by variation of ignition timing from 18° to 36° bTDC. All conditions are controlled at $\lambda \approx 1$ by Innovate Lm-2 oxygen wide band sensor and 87°C of engine coolant. Emissions are examined from exhaust pipe directly and evaluated by KOENG KEG-500 gas analyzer illustrated in Fig. 3 Moreover, performances for instance power, torque, brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) have measured due to understanding the effect from different fuel and conditions of engine.

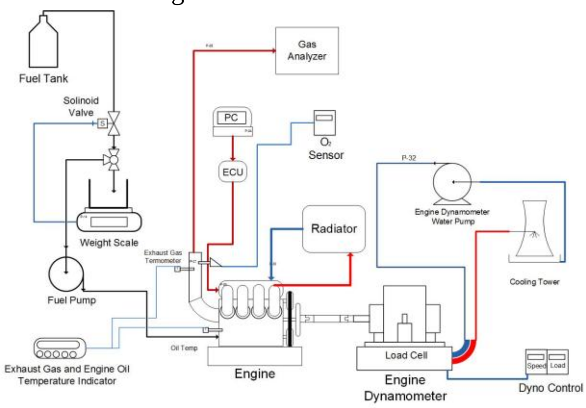

Fig. 3 Schematic diagram of experimental setup

4. Results and Discussion

4.1 Engine Performance Characteristics

4.1.1 Brake Torque

The results are discussed at the maximum break torque (MBT), the optimum timing which explained as the minimum advance of ignition timing. The gasoline (E0) was found at CAD (crank angle degree) of 27° followed with E10, E20 and E85 at 30° 30° and 33° bTDC, respectively.

When compared with E0, using ethanol blended with gasoline E85, E20, and E10 increased the engine torque 19.53%, 6.09% and 2.09% at 20% of load and 26.79% , 14.70% and 6.46% at 40% of load respectively regarding to fuel mass at the same air-fuel ratio where illustrated in Fig. 4-6. The increment tendency of brake torque is discussed as follow.

Firstly, ignition timing impacts to break torque as advanced ignition timing leads to early process of combustion in the cycle. This extends the residence time and the reaction of partial oxidation during the combustion stroke. Peak pressure and high temperature are resulted from the combustion of most fuel when the piston reaches closer to top dead center while the volume of cylinder becomes smaller which is an idealistic crank angle of combustion. Hence, thermal efficiency is increased [8]. Nevertheless, too advanced spark timing lets the abundance of gas burnt by the time of piston going up. Then, the net power will be lessened from the work used to compress gas.

Conversely, the highest pressure and temperature is decreased with retarded ignition timing due to inadequate time between top dead center and ignition timing to complete chemical reaction. Then, several of fuels are burnt after top dead center known as post-reaction [9]. Hence, incomplete combustion is assorted with retarded ignition timing.

Lastly, the LHV of fuel blended with ethanol is lower than that of gasoline. The plenty of ethanol injected into the engine test should be about double amount of gasoline for achieving the same heat energy output which mounted up in torque and power [9]. Moreover, higher latent heat of vaporization causes air intake temperature in combustion chamber lower because of evaporation of alcohol which leads to increased volumetric efficiency together with too delay of ignition timing from high octane number. Hence, advanced ignition timing is required [3]. A higher flame speed of oxygenated fuels reduce combustion duration equate to gasoline which causes shorter flame propagation. Thus, the rate of energy releases rapidly which decreases heat loss from the engine due to insufficient time for heat to depart the cylinder through heat transfer to engine coolant [10].

From this result founded that to increase 12.63% of ethanol's RON causes 3 degrees of advance spark timing. This experimental result is identical with other researches [11] and [12] as augmentation of octane number or conversely ignition delay. Then, using ethanol with early ignition is essential.

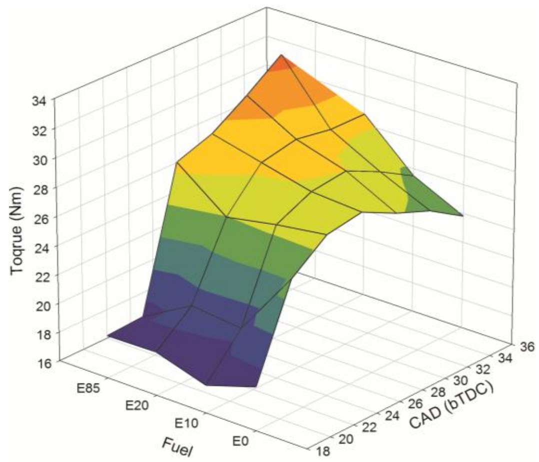

Fig. 4 Torque from DISI Engine at 20% load

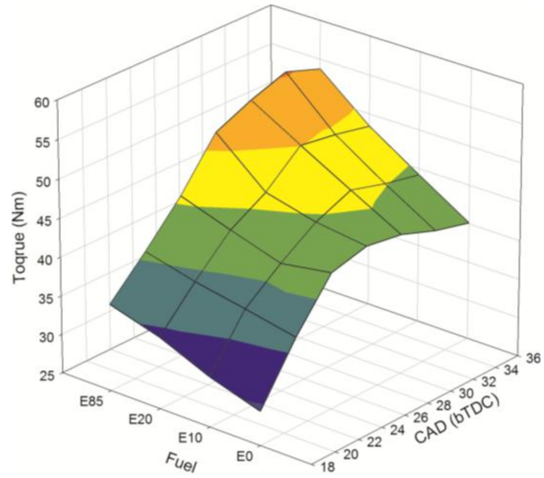

Fig. 5 Torque from DISI Engine at 40% load

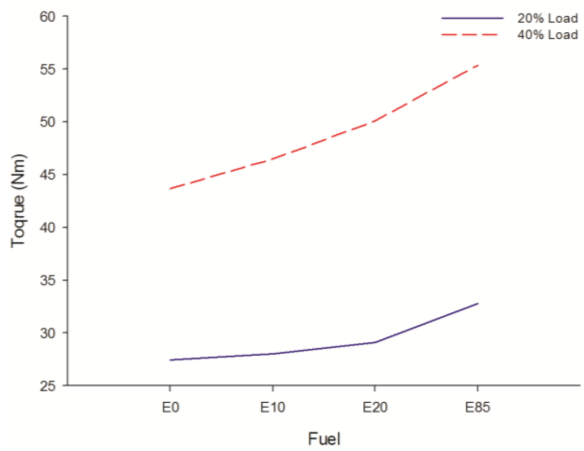

Fig. 6 Torque trends at optimum ignition timing

4.1.2 Brake Specific Consumption and Brake Specific Energy Consumption

Fig. 7-8 portrays the BSFC of the test fuels at diverse of engine load. It showed that at optimal timing, gasoline has the lowest BSFC when compared to other ethanol gasoline blends. At the 20% of load BSFC increased with E10, E20 and E85 by 4.00%, 7.12% and 15.92% whereas at 40% of load ethanol additions in gasoline increased 4.48%, 6.70% and 34.98% with E10, E20 and E85 respectively.

The heating value of fuel is an important factor which affects brake specific fuel consumption (BSFC) of engine. This increasing relies on the percentage of ethanol in fuel where ethanol has heating value less than gasoline around 30% [13]. More ethanol blended in the fuel to produce same heat energy at similar condition resulted to the increment of BSFC.

Fig. 9 shows the comparison of fuel consumption at different load. The result indicates that higher load causes less fuel consumption due to lower of pumping loss from throttle plate is open wider.

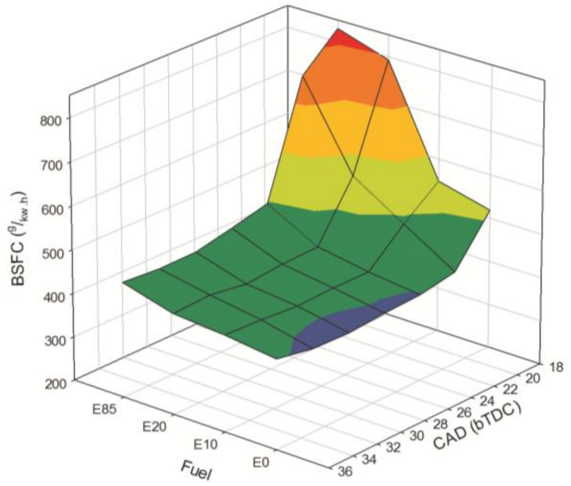

Fig. 7 Brake specific fuel consumption from engine at 20% load.

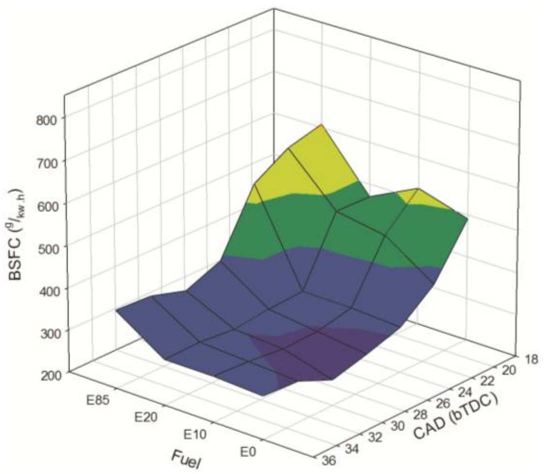

Fig. 8 Brake specific fuel consumption from engine at 40% load.

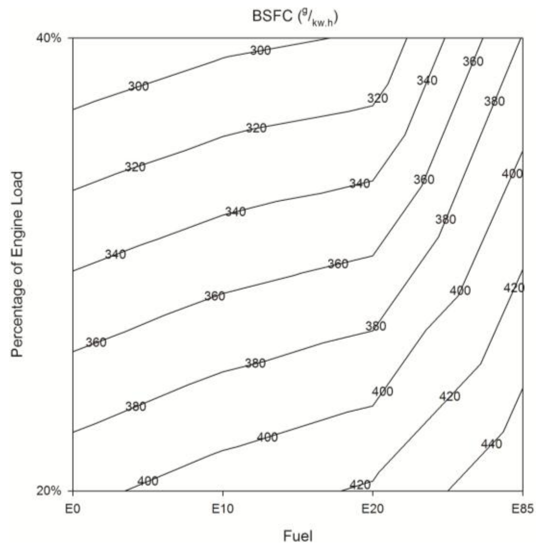

Fig. 9 Tendency of brake specific fuel consumption at optimum ignition timing

Brake specific energy consumption (BSEC) is described as the quantity of energy consumed per unit power developed in a unit of time. Described in briefly, BSEC is how efficiently of energy obtained from its fuel. With the same condition as above BSFC, Fig. 10-12 depict that ethanol blends cause the decreased brake specific energy consumption by 3.16%, 1.16% and 22.28% with E10, E20 and E85 at 20% load and around 2.71%, 1.27% and 9.50% with E10, E20 and E85 respectively of 40% load. This reverse trend was investigated due to lower calorific value with increasing in the percentage of ethanol blends.

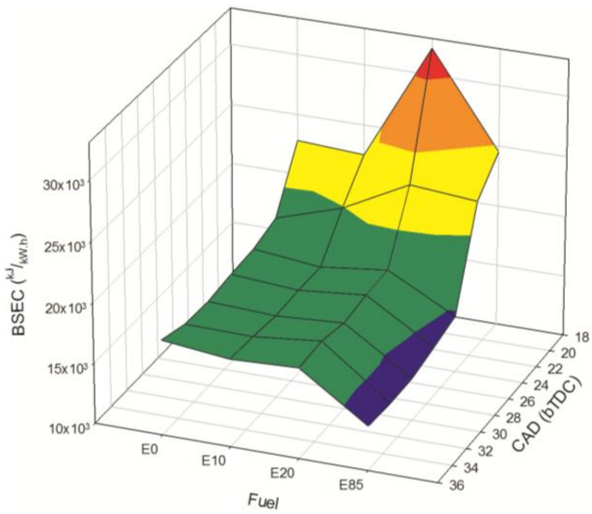

Fig. 10 Brake specific energy consumption from engine at 20% load.

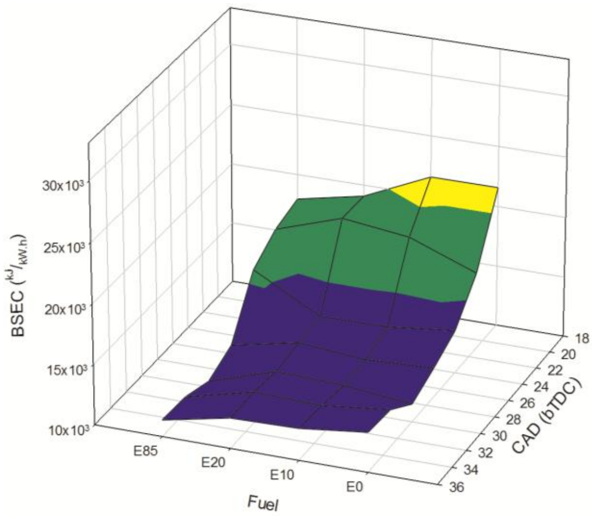

Fig. 11 Brake specific energy consumption from engine at 40% load.

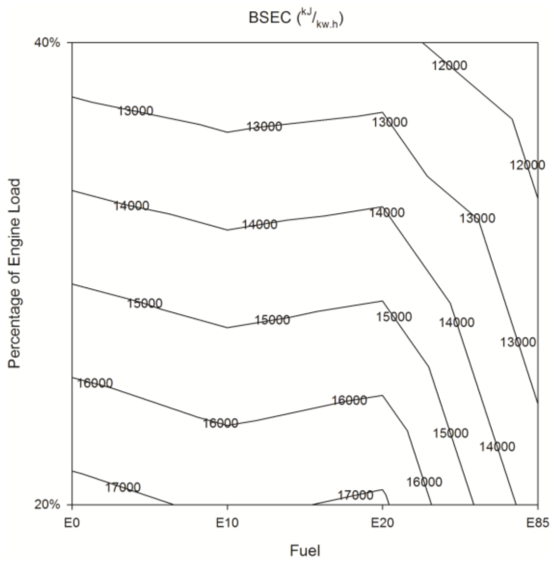

Fig. 12 Brake specific energy consumption trends from engine at optimum ignition timing

4.2 Emissions

4.2.1 CO and HC emission

Carbon monoxide (CO) and hydrocarbon (HC) are influenced by oxygen which available during combustion. So, there are anticipated to reduce as the mixture becomes leaner.

Using ethanol blends reduce HC as 38.97%, 53.31% and 59.19% of 20% load along with 21.73%, 34.18% and 42.86% of 40% load at optimal ignition timing with E10, E20 and E85 respectively as shown in Fig. 11(a). Fig. 11(b) shows the reduced CO of 26.47%, 44.12% and 50.00% of 20% load along with 8.62%, 27.59% and 31.03% of 40% load at optimal ignition timing with E10 E20 and E85 respectively.

The lower CO and HC than those of gasoline due to oxygen content of ethanol in blended fuel mount up the ratio of oxygen to fuel in over-rich zone. The actual of air-fuel ratio becomes stoichiometric whereas increasing of ethanol content in fuel contributes to more complete combustion which leads to decrease soot formation [14].

To consider the impact of timing, advance in ignition timing is important strategy when using ethanol due to higher of octane number and latent heat of vaporization compare to gasoline which cause prolong of ignition delay from heat absorption. Then, reduction of those emissions from the short duration between the terminate of fuel injection and ignition timing contributes to improvement of flame propagation from increasing of turbulence intensity and high mixture stratification which defined as the mixture near the spark plug which is locally rich and retains a constant overall lean air-fuel ratio [15-17].

However, to retard ignition timing from optimum timing decreases flow intensity in the cylinder and lessens in the mixture stratification which mount up the fraction of unburned hydrocarbon in the region of lean mixture as well as CO is increased from incomplete combustion.

Lastly in term of engine load, those emissions are increased. The more of engine load, the more fuel is injected. Thus, flame speed is important factor for completing combustion of rich mixture conditions together with high engine load. This study showed outstanding property of ethanol that lets higher flame speed to assist this process complete and to reduce the duration of combustion. The high blend ratio of ethanol can reduce emissions effectively.

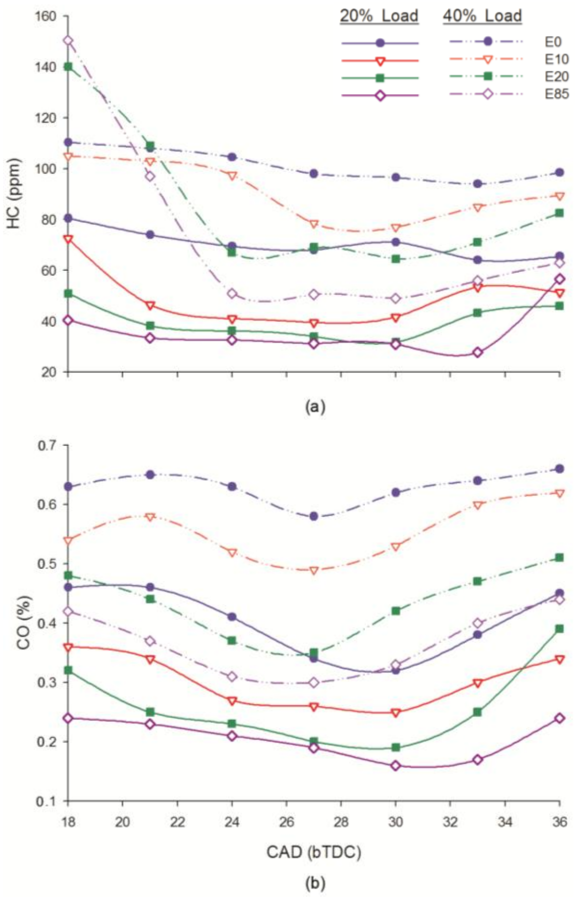

Fig. 11a Carbon monoxide (CO) and Fig.11b Hydrocarbon (HC) trends from DISI engine affected by variation of fuel from E0 to E85 and ignition timing from 18° to 36° bTDC

4.2.2 CO_2 Emission and O_2

As seen in fig. 12a, ethanol blended with gasoline increase CO_2 emissions as 0.08%, 0.64%

and 2.25% of 20% load coped with 3.84%, 4.70% and 4.86% of 40% load as shown in fig. 12a following with O_2 as 18.39%, 27.59% and 32.18% of 20% load together with 17.14%, 28.57% and 32.38% of 40% load as shown in fig. 12b. Even though, the fuel quantities of ethanol blended is more than gasoline but, the oxygen content in ethanol lets excess air to combust residue emissions and enhance complete combustion which leads higher of CO_2 [9].

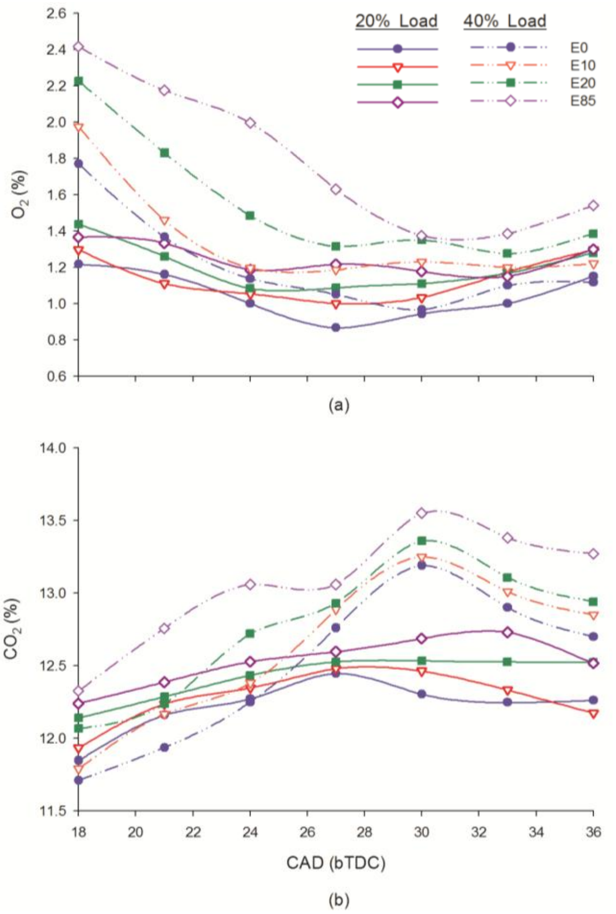

Fig. 12a Carbon dioxide (CO_2) and Fig, 12b oxygen (O_2) trends from DISI engine by variation of fuel and ignition timing

5. Conclusion

1. To advance ignition timing brings higher torque and lower emission from complete combustion whereas retard of ignition timing is contrary from post-reaction and lower peak pressure which occurred in expansion process.

2. Gasoline blended with ethanol increases brake torque from quantity that injected and reduces carbon-monoxide and hydrocarbon by excess air of oxygen content in its fuels. However, brake specific consumption is higher than gasoline due to lower heating value. Conversely, brake specific energy consumption is lower because more energy is released than gasoline.

3. High octane number of ethanol causes too ignition delay from higher heat absorption. Then, higher flame speed of ethanol together with advancing ignition timing is suitable from fuel blended with ethanol for complete combustion.

This experiment will conduct the DISI engine suitably using ethanol blended with gasoline and utilizing the benefits of spark timing which adjusted together with outstanding properties of ethanol to operate on stratified mode with less emission in the next study.

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