

Effect of Ethanol on Knock in Spark Ignition Gasoline Engines

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ABSTRACT

This study examines the effects of ethanol content on engine performances and the knock characteristics in spark ignition gasoline engine under various compression ratio conditions by cylinder pressure analysis, visualization and numerical simulation. The results confirm that increasing the ethanol content provides for greater engine torque and thermal efficiency as a result of the improvement of knock tolerance. It was also confirmed that increasing the compression ratio together with increasing ethanol content is effective to overcome the shortcomings of poor fuel economy caused by the low calorific value of ethanol. Further, the results of one dimensional flame propagation simulation show that ethanol content increase laminar burning velocity. Moreover, the results of visualization by using a bore scope demonstrate that ethanol affects the increase of initial flame propagation speed and thus helps suppress knock.

INTRODUCTION

In recent years, as public interest in global warming has increased and crude oil prices have risen, the use of bioethanol as a fuel for motor vehicles is spreading worldwide. In countries such as the United States, Brazil, and Sweden, ethanol-blended gasoline products, ranging from E10 with low ethanol concentration (gasoline containing 10% ethanol by volume) to E85 with high concentration, as well as E100 are already being sold at regular gas stations after making arrangements for transportation and storage of the fuel. In addition, U.S. motor vehicle manufacturers are now developing Flexible Fuel Vehicles (FFVs) that run on any ethanol mixture ratio up to 85%.

Bioethanol produced from plant sources such as sugar cane and corn has the merit of being renewable and CO₂ neutral. It also has many advantages in terms of engine combustion. For example, because it contains oxygen, it can suppress the generation of CO and soot. Also, with its high octane number, it can be used as an anti-knock agent. Although it has problems such as poor

fuel economy (in this paper, "fuel economy" is defined as the distance traveled per unit volume of fuel) due to its low heating value and theoretical air fuel ratio as well as poor cold-start performance due to the low vapor pressure, the use of bioethanol as an alternative to gasoline is expected to increase because, provided the ethanol mixture ratio in gasoline is low, the fuel can easily be used in regular gasoline vehicles with only minor modifications.

Hence, this study investigated the effect of various ethanol mixture ratios in gasoline on combustion and anti-knock performance of spark ignition gasoline engines under different compression ratio conditions by means of numerical analysis, visualization analysis, and cylinder pressure analysis, and confirmed the effect of ethanol addition and higher compression ratios on engine performance.

EXPERIMENTAL APPARATUS AND TEST CONDITION

Table 1 shows the engine specifications⁽¹⁾. As the test engine, a single-cylinder engine with a displacement of 0.325 L that had been constructed based on the Suzuki M13A was used. The combustion chamber is in a pent-roof shape, having squish areas at its four corners, which is shown in Figure 1(a). The spark plug is set slightly off the center of the combustion chamber to the exhaust side. Compression ratio ε is adjustable up to 15.0 by changing squish height, and in this study, tests were conducted under conditions of $\varepsilon=9.5$, 11.0, 13.9, and 15.0. As for the piston, a flat piston was used when ε was 9.5 and 11.0, and a convex piston, which is shown in Figure 1(b), was used when ε was 13.9 and 15.0. When the convex piston is used, the top surface of the piston will intrude into the combustion chamber on the head side by about 4 mm at $\varepsilon=15.0$ and 3.5 mm at $\varepsilon=13.9$.

The devices⁽¹⁾ that were used in the experiment are as follows. The in-cylinder pressure was measured using a pressure sensor (6517B, KISTLER) placed on the spark plug. The intake air volume and fuel consumption

were measured using a laminar flow meter (LFE 50B, SOKKEN) and a mass flow meter (735S, AVL) respectively, and A/F was calculated based on these values. In addition, for backing up the A/F measurement and verifying exhaust gas components, an exhaust gas analyzer (MEXA-7100D, HORIBA) , where a heated filter was attached to the measurement pipe, was used.

Next, Table 2 shows the test conditions. Testing was conducted at engine speeds of 1500, 2000, 3000, and 4000 rpm, and with full load and half load. The engine intake air was controlled to the temperature of 25°C and humidity of 50% by a temperature/humidity controller installed at the inlet. Furthermore, engine oil and coolant were circulated by an electric pump installed on the outside of the engine. The flow rate and temperature of the coolant were controlled to 25 L/min and 85°C, respectively. The temperature of oil was controlled to be within the range of 85°C to 90°C.

Table 1. Engine specifications

Engine	4 valves, Single cylinder
Displacement	0.325 L
Bore×Stroke	78×69.5 mm
Compression Ratio	9.5, 11.0, 13.9, 15.0
Piston	Flat ($\varepsilon=9.5$), Convex
Intake Valve Opening	19° BTDC
Intake Valve Closing	62° ABDC
Exhaust Valve Opening	48° BBDC
Exhaust Valve Closing	24° ATDC
Fuel preparation	Port fuel injection

Table 2. Test condition

Engine speed	1500, 2000, 3000, 4000 rpm
Load	Full load, Half load
Intake air temperature	25°C
Intake air humidity	50%
Coolant temperature	85°C
Coolant flow rate	25 L/min
Oil temperature	85 to 90°C

TEST FUEL

Table 3 shows the composition and properties of the blend fuel used in the test. In testing, surrogate gasoline

with research octane number (RON) 91, in which iso-octane, normal heptane, and toluene were blended at the ratio of 55:15:30 by volume, was used as the base fuel (E0) so as to eliminate variations of gasoline composition and properties as well as to make comparisons with the results of detailed chemical kinetic calculations. From this base fuel and absolute ethanol with RON108, E3 (97% base fuel and 3% absolute ethanol), E10, E20, E40, E60, E80, and E100 fuels were produced to investigate the effect of different ethanol mixture ratios on the combustion and anti-knock performance of spark ignition gasoline engines. Table 3 indicates that as ethanol mixture ratio increases, the heating value per unit mass and theoretical air fuel ratio decrease. Since all the tests were conducted using the theoretical air fuel ratio, the heating value injected per cycle was almost the same between E0 and E100, as shown in Table 4.

Table 4. Ratio of heating value injected per cycle

Fuel	Ratio of heating value against E0
E3	1.000
E10	1.001
E20	1.003
E40	1.006
E60	1.010
E80	1.014
E100	1.019
RON100	1.005

EFFECT OF DIFFERENT ETHANOL MIXTURE RATIOS ON IGNITION DELAY TIME

Figure 2 shows ignition delay times of the ethanol blend fuel, which were obtained by zero-dimensional calculation using CHEMKIN-II. The fuel composition of the surrogate gasoline used for the calculation is given in Table 3. As the chemical reaction scheme, a combination of gasoline surrogate reaction mechanism⁽²⁾ (number of chemical species: 57; number of reactions: 383), proposed by Golovichev et al., and ethanol reaction mechanism⁽³⁾ (number of chemical species: 57; number of reactions: 383), proposed by Marinov et al., was used. It had been confirmed that the gasoline surrogate reaction mechanism and the ethanol reaction mechanism are capable of reproducing the values

Table 3. Test fuel property

Fuel	C8H18 vol %	C7H16 vol %	C7H8 vol %	ethanol vol %	H/C	O/C	Density kg/m ³	LHV MJ/kg	Cp kJ/kgK	Stoich. A/F	MN g/mol
E0	55	15	30	0	1.85	0.0	739.0	43.05	1.98	14.54	103.54
E3	53.35	14.55	29.1	3	1.87	0.01	740.4	42.54	1.99	14.37	99.59
E10	49.5	13.5	27	10	1.92	0.03	743.6	41.36	2.03	13.95	91.49
E20	44	12	24	20	2.01	0.07	748.2	39.70	2.07	13.75	82.06
E40	33	9	18	40	2.19	0.15	757.4	36.43	2.16	12.23	68.25
E60	22	6	12	60	2.41	0.24	766.6	33.25	2.25	11.12	58.61
E80	11	3	6	80	2.68	0.36	775.8	30.14	2.34	10.04	51.52
E100	0	0	0	100	3.00	0.50	785.0	27.10	2.42	8.98	46.07
RON100	–	–	–	–	1.74	0.00	749.8	42.87	2.42	14.41	93.55

measured by Gauthier et al.⁽⁴⁾ and those measured by Natarajan et al.⁽⁵⁾, respectively, with high precision.

The initial pressure was set at 20 atm and 55 atm, and calculations were made by increasing the ethanol mixture ratio from 3%, 10%, 20%, 50%, 75%, to 100% against the reference E0. The figure demonstrates that, under any of these conditions, the ignition delay time tends to become longer in the cool-flame dominant region through the negative temperature-coefficient region as the ethanol mixture ratio increases. Figure 3 shows unburnt gas temperatures and pressure history obtained by three-dimensional combustion calculation⁽¹⁾ in which an improved Flamelet model and an improved Shell model are coupled. Each value represents a value at point A in figure 1(a), located adjacent to the intake valve, where the transition from low-temperature oxidation reaction to hot-flame reaction was observed in the calculation. Calculations were made using the engine speed of 1500 rpm and the full load and by changing the compression ratio to 9.3, 9.8, and 10.3. The fuel used was E0. The initial conditions and boundary conditions were determined by one-dimensional calculation using GT-Power. As the condition of ignition timing, it was advanced to the optimum ignition timing for the calculation (ignition timing immediately before knock). The figure confirms that, in the range between compression top dead center and 20-30° ATDC (immediately before knock occurs), the in-cylinder pressure changes from 2 MPa to 4 MPa, and the unburnt gas temperature changes from 700 K to 900 K. It is therefore predicted that knock particularly depends on the negative temperature coefficient region in Figure 2. Furthermore, since this temperature region generates results that are extremely sensitive to differences in the ethanol mixture ratio, it is also predicted that the increase of ethanol mixture ratio significantly affects the anti-knock performance of engines.

Figure 4 shows the reaction path of ethanol. In ethanol, similarly to paraffin components such as normal heptane, after hydrogen is abstracted by OH radical, oxygen is attached. The transition state of the intermolecular hydrogen abstraction reaction of $\text{OOCH}_2\text{CH}_2\text{OH}$ radical thus generated takes the form of a five-membered ring. Five-membered ring provide large strain energy, which increased the activation energy. This prevents the intermolecular hydrogen abstraction reaction from occurring, and no chain branching reaction will take place. Thus, a path where CH_3CHO , HCHO , etc. are generated by chain propagation reaction, which then produces one radical, will be the main path⁽⁶⁾. Consequently, when substances, such as ethanol, whose number of carbon atoms in small and whose transition state does not take the form of a six-membered ring with small strain energy like that of paraffin components, are mixed with gasoline, the ignition delay time will become longer. Furthermore, another possible cause of longer ignition delay time is the consumption of the OH radical produced from normal heptane through the hydrogen abstraction reaction in ethanol.

EFFECT OF DIFFERENT ETHANOL MIXTURE RATIOS ON LAMINAR BURNING VELOCITY

Figure 5 shows the results of one-dimensional calculation for laminar burning velocity conducted using the PREMIX function of CHEMKIN-II. For the calculation, Golovichev et al.'s gasoline surrogate reaction mechanism and Marinov et al.'s ethanol reaction mechanism were used in the same way as described above. For simulating the conditions of unburnt gases at the initial and late stages of combustion in the cylinder, calculations were made at 20 atm and 700 K as well as 55 atm and 750 K. In addition, for reference, laminar burning velocity of iso-octane, normal heptane, toluene, and ethanol measured at 304 kPa and 450 K by Farrell et al.⁽⁷⁾ are shown in Figure 6. It can be confirmed from Figure 6 that the laminar burning velocity of ethanol is higher than that of the other fuels. Figure 5 predicts that the addition of ethanol to gasoline will increase the laminar burning velocity. Furthermore, at 20 atm and 700 K, the laminar burning velocity of E20 is higher than that of E0 by about 4%.

Therefore, verification tests were conducted by visualization of the engine cylinder to assess the extent of the effect of increased ethanol mixture ratio on the actual initial flame. Figure 7 shows a schematic of the measuring devices. The initial flame was measured using a bore scope installed between intake and exhaust valves at the rear of the combustion chamber on the cylinder head side. Images thus obtained were recorded by a high-speed video camera (Phantom V7.1, Vision Research) through an image intensifier. As shown in Figure 7, the bore scope was directed towards the center of the combustion chamber and set so that the focus would be on the proximity of the spark plug at a viewing angle of 45°. As shown in Figure 8, its visualized range was a circle of diameter about 30 mm, and the spark plug electrode was located directly above the visualized range. The depth of field was about 1 mm. With regard to the 2 mm grids shown in the figure, perspective correction has been performed, and the captured images received similar correction.

Figure 9 shows a 10-cycle average image of the initial flame captured by the bore scope. The test was conducted using the conditions given in Table 5. The figure confirms that, after ignition, the flame spreads around 9° ASOI (After Spark Of Ignition) more widely under the conditions of E20. This difference becomes even larger at 12° ASOI.

In order to verify this trend quantitatively, the flame-propagation distance in the X1 and X2 directions in Figure 8 were determined based on the recorded images. The distance component in the normal direction in relation to the piston surface was not obtained as the flame would contact the piston at the initial stage of image recording. The result is shown in Figure 10. In the figure, while almost the same value are indicated in the X1 direction, the flame-propagation distance at the same

timing is longer under the conditions of E20 in the X2 direction. Since fuel is injected into the intake port in the engine used in the test, assuming that the intake air-fuel mixture volume, in-cylinder turbulence, and other states will not be changed by the fuel composition, it can be said that these differences in the flame propagation are primarily caused by differences in the laminar burning velocity of ethanol.

The effect of this improvement of the laminar burning velocity is observed not only at the initial and main stages of combustion but also at the late stage of combustion, where the flame front approaches the wall surface with weak turbulence and the flame-propagation speed becomes lower. Rapid combustion of unburnt gases made possible by improved laminar burning velocity is desirable from the standpoint of anti-knock performance as well.

Table 5. Test condition of visualization

Engine speed	750 rpm
Compression ratio	11.0
Piston	Convex
Fuel	E0, E20
Excess air ratio	1.0
Throttle position	Closed
Spark timing	10.0° BTDC

COMBUSTION ANALYSIS USING AN ACTUAL ENGINE

First, the effect of ethanol blending on the combustion is described. Figure 11 shows the in-cylinder pressure, RHR(Rate of Heat Release), and MBF(Mass Burn Fraction) at the matched CA50(the crank angle at which 50% of total heat release occurs) with $\varepsilon = 11.0$ and $\varepsilon = 13.9$. The fuel used was E20, and the engine speed was 1500 rpm. The figure confirms that as the compression ratio increases, the combustion duration slightly increases. The possible causes of this include a decrease of the turbulence around the ignition timing and increase of the cooling loss due to the effect of the convex piston.

Figure 12 shows the pressure and RHR at the knock-limit ignition timing with $\varepsilon = 9.5$, 11.0, and 13.9. The fuel used was E0, and the engine speed was 1500 rpm. The knock limit was determined based on the amplitude of waves obtained by bandpass filtering on the in-cylinder pressure. The amplitude threshold was set at 2 bar. The figure shows that due to the delay in the ignition timing that occurs in conjunction with increase of the compression ratio, combustion becomes slower as the compression ratio is increased. In addition, under the condition of $\varepsilon = 13.9$, heat release can be observed from around 10° BTDC, before ignition, to 2° ATDC, at the ignition timing. This heat release is presumably caused by low-temperature oxidation reaction since it is decreased by increasing the ethanol mixture ratio from

3%, 10%, to 20% under the condition of $\varepsilon = 13.9$ as shown in Figure 13.

Furthermore, Figure 12 confirms that, under the condition of $\varepsilon = 11.0$, there is a region around 13° ATDC where the RHR starts to stagnate. Since this trend is also observed in the results for E20 under the condition of $\varepsilon = 15.0$, as shown in Figure 14, as well as for commercially-available high octane gasoline with RON100, it is presumed that it is not due to the fuel, but depends on the combustion chamber shape, etc. we predicted that this combustion stagnation is caused by extinction of flame due to its interference with the piston.

Next, Figure 15 shows the relationship between CA50 and IMEP(Indicated Mean Effective Pressure) at different compression ratios. The fuel used was E20, and the engine speed was 1500 rpm. For the condition of $\varepsilon = 15.0$, the result for E100 is also indicated. Since the crank angle where the IMEP reaches a peak when ε is 11.0 or above was not obtained, it is difficult to make accurate predictions. Nevertheless, it can be seen from the figure that the CA50 that shows the maximum IMEP is almost the same for different compression ratios. In addition, similar results were obtained under conditions between 2000 rpm and 4000 rpm.

Figure 16 shows the relationship between CA50 and IMEP when the ethanol mixture ratio is changed from 0% to 20% under the conditions of $\varepsilon = 9.5$ and 13.9. The engine speeds were 1500 rpm and 3000 rpm. The result confirms that the relationship between CA50 and IMEP remains unchanged even when the ethanol mixture ratio is changed. Furthermore, Figure 17 shows the relationship between CA50 and IMEP when the ethanol mixture ratio is changed from 0% to 100% under the condition of $\varepsilon = 15.0$. This result also confirms that changes in the ethanol mixture ratio do not affect the relationship between CA50 and IMEP. Moreover, similar results were obtained for other driving conditions of 2000 rpm and 4000 rpm. Caton et al. demonstrated the relationship between pressure peak location and IMEP for E10 and E85 using a CFR engine under the conditions of $\varepsilon = 9.0$, excess air ratio $\lambda = 0.9$, and engine speed = 900 rpm, and indicated that the pressure peak location and IMEP would change depending on the ethanol mixture ratio⁽⁸⁾. However, the results of our study did not confirm this. It is possible that since they conducted the test at low engine speed and rich-burn state, the flame-propagation speed and cooling loss easily affected the result, thus generating such difference. Still, as can be seen in the result of comparing the log P-V plot between E0 and E100 under the conditions of $\varepsilon = 15.0$ and engine speed = 3000 rpm as shown in Figure 18, when the IMEP is the same, there is no large difference in the pressure between E0 and E100. For this reason, the CA50 that shows the maximum IMEP or the pressure peak location is unlikely to significantly change. Furthermore, there is some doubt about the result of their study, in which their data

indicated that the pressure peak location changes between E0 and E85 by as much as 10°CA.

RESULTS OF PERFORMANCE TEST

This section describes the engine performance observed when the ethanol mixture ratio was changed from E0, E3, E10, to E20 for compression ratio $\varepsilon = 9.5, 11.0, 13.9$, and 15.0.

Figure 19 shows IMEP plotted against ignition timing. The engine speeds were 1500 rpm and 3000 rpm. It can be seen that as the compression ratio increases, the anti-knock performance deteriorates, resulting in delays in the ignition timing. On the other hand, the figure confirms that when ethanol is added, knock is suppressed, advancing the ignition timing, and that in conjunction with this, the IMEP also increases.

Figure 20 shows the knock-limit ignition timing and the IMEP when the ethanol mixture ratio is changed using the same conditions. The figure confirms that ethanol demonstrates excellent anti-knock performance and that as the ethanol mixture ratio is increased, the ignition timing advances significantly. Furthermore, the amount of advance in the ignition timing in relation to ethanol mixture ratio remains almost the same even when the compression ratio is changed. Moreover, by suppressing knock by increasing the ethanol mixture ratio, the IMEP exceeds the standard for $\varepsilon = 9.5$ under any compression ratio. Under the condition of 3000 rpm, since the ignition timing can be advanced to the proximity of MBT (Minimum Advance for Best Torque) by increasing the ethanol mixture ratio even under the condition of $\varepsilon = 15$, it is possible to obtain the IMEP that exceeds $\varepsilon = 11$ and $\varepsilon = 13.9$ under the condition of $\varepsilon = 15$. On the other hand, under the condition of 1500 rpm, where the time per a crank angle becomes longer and anti-knock performance becomes worse compared with 3000 rpm, the ignition timing for $\varepsilon = 15$ largely deviated from the MBT and did not exceed the IMEP for $\varepsilon = 11$ even when the ethanol mixture ratio was increased to 20%.

Figure 21 shows the BMEP(Brake Mean Effective Pressure) and BSFC(Brake Specific Fuel Consumption) for 1500 rpm and 3000 rpm. At $\varepsilon = 9.5$, advancement to the proximity of MBT is possible even when E0 fuel with low RON is used, and the BMEP hardly increases even when the ethanol mixture ratio is increased. For this reason, the specific fuel consumption becomes significantly worse. On the other hand, at $\varepsilon = 11$ and $\varepsilon = 13.9$, since the knock-limit ignition timing for E0 largely deviates from the original MBT, the BMEP can be increased by increasing the ethanol mixture ratio and suppressing knock, which improves the specific fuel consumption. However, at $\varepsilon = 15$, where the anti-knock performance extremely deteriorates, the specific fuel consumption that was obtained under the condition of $\varepsilon = 9.5$ cannot be achieved even by increasing the ethanol mixture ratio to 20%.

Next, the engine performance observed when the ethanol mixture ratio is changed from E0 to E100 under the condition of $\varepsilon = 15.0$ will be shown for each engine speed. Figure 22 shows the IMEP against the ignition timing and Figure 23 shows the BSFC that is standardized under the conditions for E0. The figure confirms that, under any of these conditions, the amount of advance in the ignition timing becomes smaller when the ethanol mixture ratio exceeds E20, and from E60 to E100, there is no advancement. As a result, the BSFC worsens when the ethanol increases. The reason for the anti-knock performance remaining unchanged when the ethanol mixture ratio is increased to 60% or above is still being investigated, but since the Arrhenius plot for ignition delay time, shown in Figure 2, presents no effect of changes in ethanol mixture ratio in the negative temperature coefficient region through the blue-flame dominant region on the high-temperature side, it can be presumed that increases in the temperature and pressure of unburnt gases caused by higher compression ratios and advanced ignition timing as well as ignition delay time properties affect the result.

On the other hand, the brake thermal efficiency graph, shown in Figure 24, under the condition of $\varepsilon = 15.0$, reveals that a thermal efficiency largely exceeding $\varepsilon = 9.5$ can be obtained by increasing the compression ratio by adding ethanol. In terms of thermal efficiency, ethanol is an extremely effective means of making the engine highly efficient.

Next, the effect on combustion variation of making the ethanol blend fuel leaner is described. Figure 25 shows Cov.(Coefficient of variation) of IMEP at 1500 rpm, at full load and partial load, and with BMEP=400 kPa. Under any of these conditions, no remarkable difference is observed between E0 and E20. Furthermore, similar results were obtained for other engine speeds and load conditions.

Finally, Figure 26 compares the brake thermal efficiency and fuel economy at partial load of BMEP=400 kPa. Since there is no effect of knock at half load even when ε is 13.9, thermal efficiency can be improved by higher compression ratios even for E0. Thus, the figure confirms that, regardless of whether ethanol is added or not, compared with $\varepsilon = 9.5$, both thermal efficiency and fuel economy can be slightly improved at $\varepsilon = 13.9$. Since increasing the ethanol mixture ratio lowers the heating value per unit mass, when compared at the same compression ratio, the thermal efficiency is the same but the fuel economy worsens. Nevertheless, since a comparison between the conventional conditions of $\varepsilon = 9.5$ for E0 and $\varepsilon = 13.9$ for E10 shows that the fuel economy is improved for E10, it can be said that ethanol is also an effective means of enhancing the fuel economy of engines.