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Title: Effect of low load combustion and emissions on fuel dilution in lubricating oil and deposit formation of DI diesel engines fueled by straight rapeseed oil

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Abstract: The objective of this study is to apply neat biomass fuel to a DI diesel engine and investigate the effect of in-cylinder gas flow and combustion on the deposit formation and the fuel dilution in lubricating oil. The study focuses on the low load combustion and emissions considering that low load exhaust contain much unburned fuels and the unburned fuels are the source of the deposit formation and the fuel dilution. Piston configuration and swirl velocity were altered in the engine test. The engine was fueled by neat rapeseed oil. The test was carried out through the four hours continuous engine operation with keeping low load. After the operation, state of deposit formation and fuel dilution in lubricating oil were investigated. Results indicate that Re-entrant piston which creates strong reverse squish and high swirl forms the deposit annular on the piston top. Toroidal piston easily produces deposit on the undersurface of cylinder head. The deposit in the cavity accumulates where initial rapeseed oil spray impinges regardless of piston types. The carbonization of the deposit is promoted on the wall surface where the burned gas with high temperature and high velocity comes into contact. It is important to avoid extremely strong reverse squish to the cylinder liner in order to control the fuel dilution. The deep-bowl chamber changes the direction of reverse squish from the cylinder liner direction to the cylinder head direction. The low velocity outflow from the piston cavity reduces the adhesion of unburned fuel on the cylinder liner, resulting in the smaller amount of unburned fuel scraped off by a piston ring.

1. Paper title

Effect of low load combustion and emissions on fuel dilution in lubricating oil and deposit formation of DI diesel engines fueled by straight rapeseed oil

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Figure

List of figures

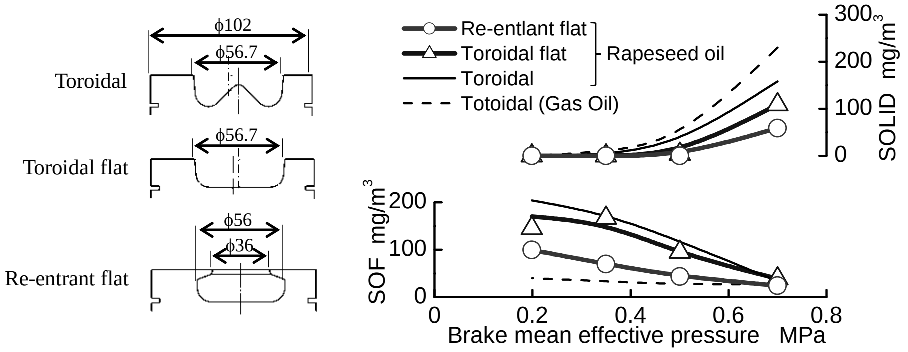


Fig.1. Reduction of particulate emissions by means of combustion chamber configurations [26].

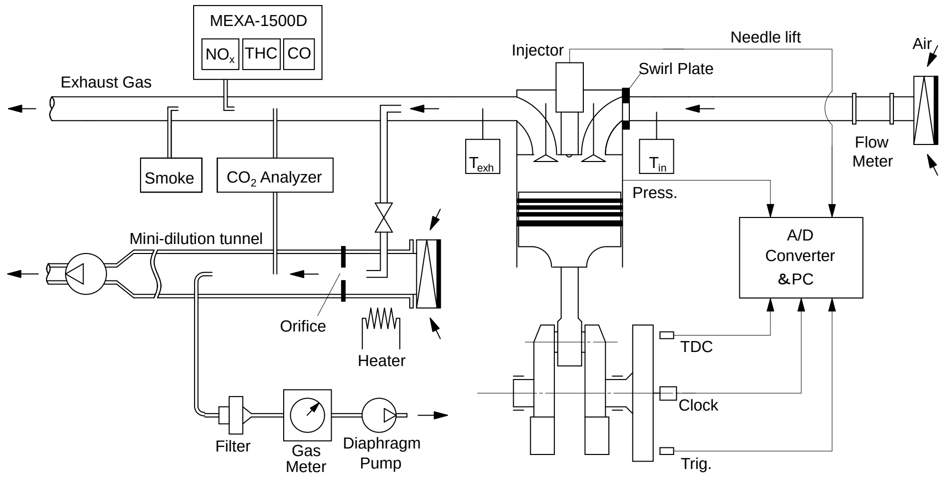


Fig.2. Experimental setup.

Table 1
Engine specifications.

Engine type	Single cylinder, Natural aspiration, Direct injection
Bore x Stroke	102mm x 105mm
Displacement	857cc
Compression ratio	17
Swirl ratio, r_s	2.2 and 3.1
Piston type	(See Table 2 and Fig. 3)
Fuel injection	Mechanical type (Nozzle opening press.=19.6MPa)
Injection nozzle	150^o - 4 x 0.29mm

Table 2
Tested piston specifications and swirl ratio.

Type	Outlet dia. D_{out} mm	Bowl dia. D_{in} mm	Bowl depth H mm	Aperture ratio D_{out}/D	Aspect ratio D_{in}/H	Swirl ratio r_s
R35	36	53.4	22.1	0.353	2.416	2.2
		58.4	19.7		2.964	2.2
						3.1
		63.4	17.7		3.582	2.2
R45	46	55	19.7	0.451	2.792	2.2
						3.1
Toroidal flat	56.7		18.1	0.556	3.133	2.2
						3.1
	61.7		15.2	0.605	4.059	2.2
	66.7		12.9	0.654	5.171	2.2
(Ref.)Toroidal base	56.7		19.7	0.556	2.878	2.2

D:Outer diameter of the piston

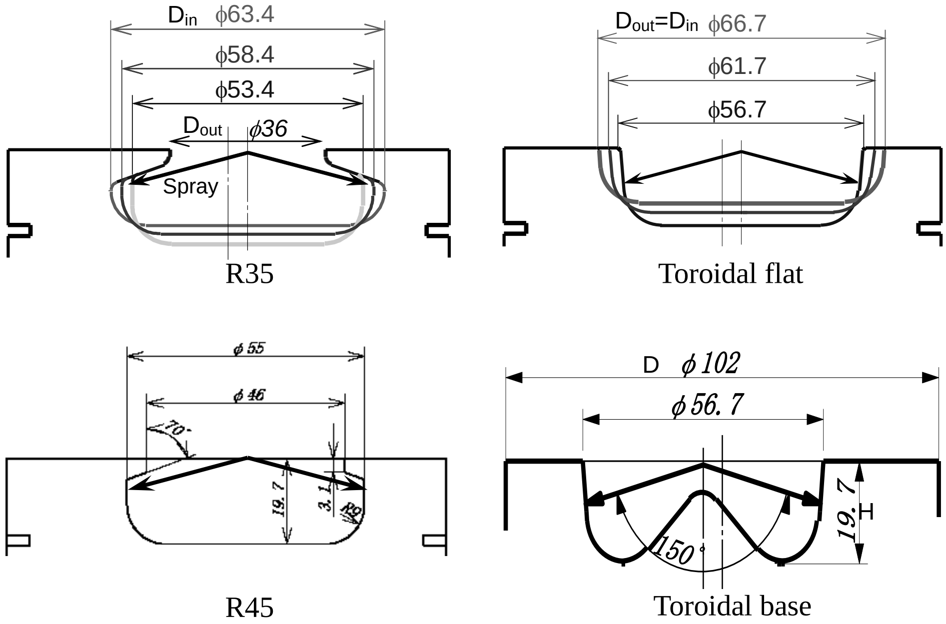


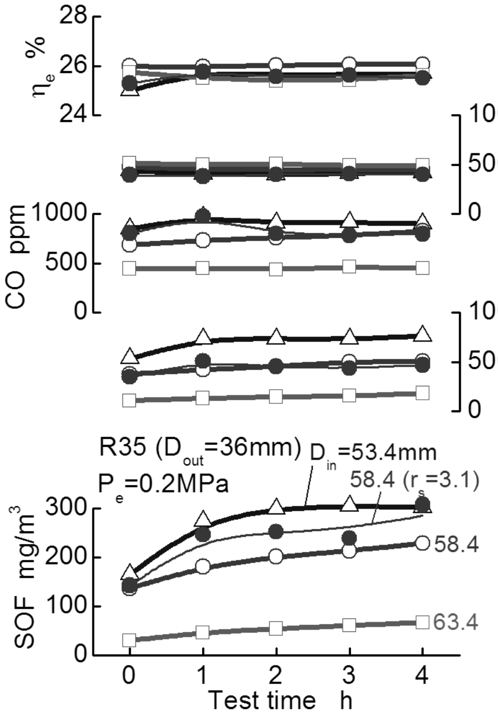
Fig.3. Piston configurations.

Table 3
Test condition.

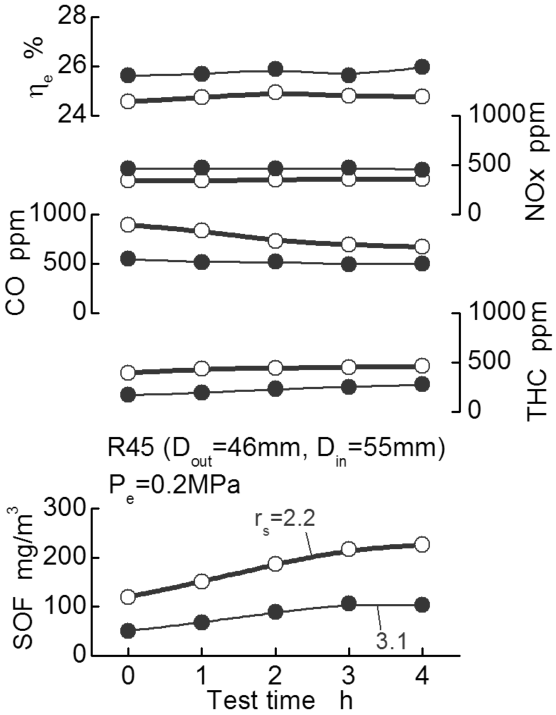
Fuel	Rapeseed oil
Engine speed	1800rpm
Injection timing	-5^{o}ATDC(Dynamic)
Engine load, P_e	0.2MPa(BMEP)
Engine operation	continuous four hours

Table 4
Fuel properties.

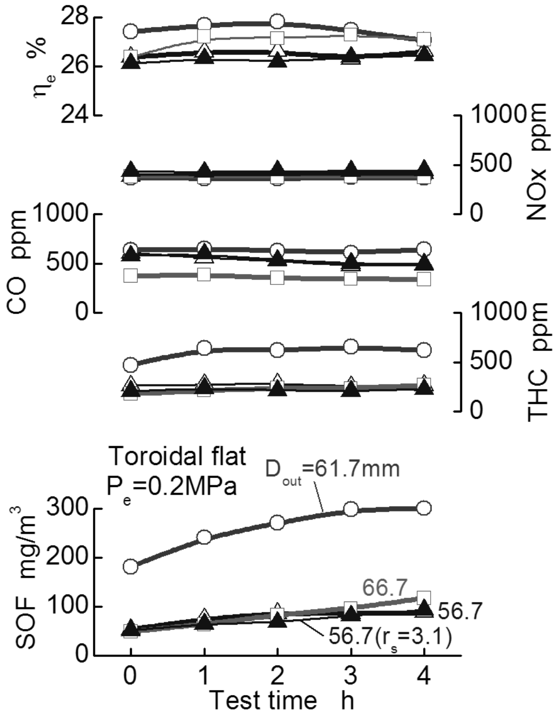
		Rapeseed oil	Gas oil
Density	kg/m^3@288K	911	837
Kinematic viscosity	cSt@303K	47.8	3.53
Distillation, T90	^{o}C	(>400)	325
Lower heating value	MJ/kg	38.1	42.7
Oxygen content	wt%	11.1	0



(a) R35



(b) R45



(c) Toroidal flat

Fig.4. Change in exhaust emissions. (a) R35. (b) R45. (c) Toroidal flat.

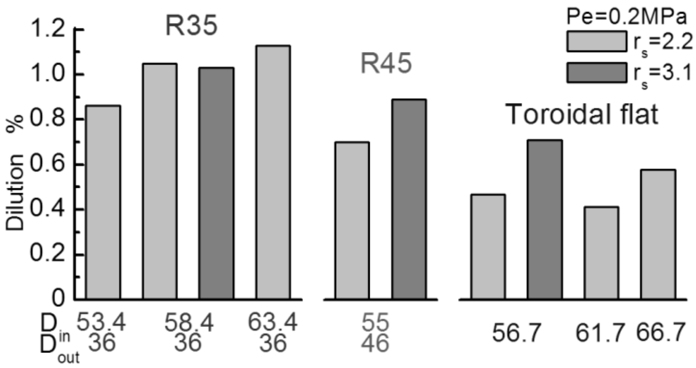


Fig.5. Comparison of fuel dilution in lubricating oil.

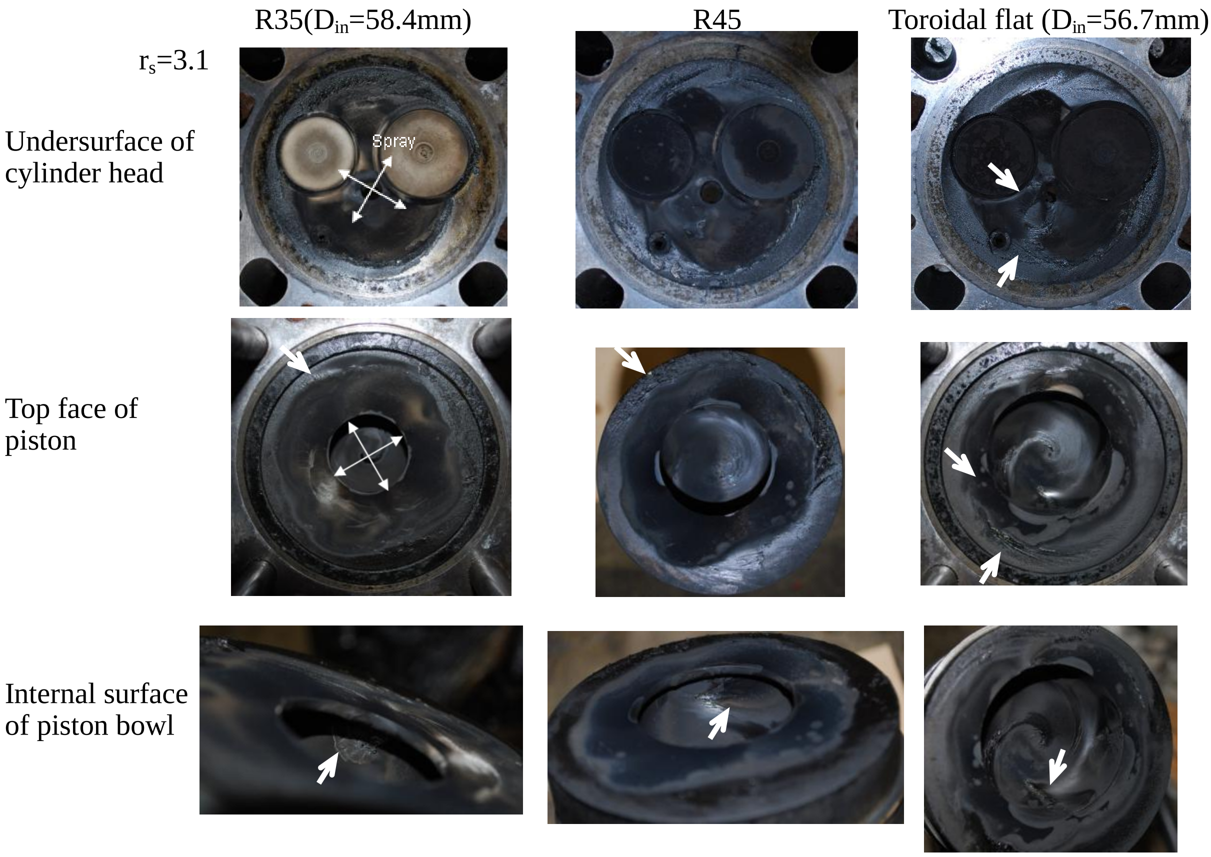


Fig.6. Deposit formation on the wall surface.

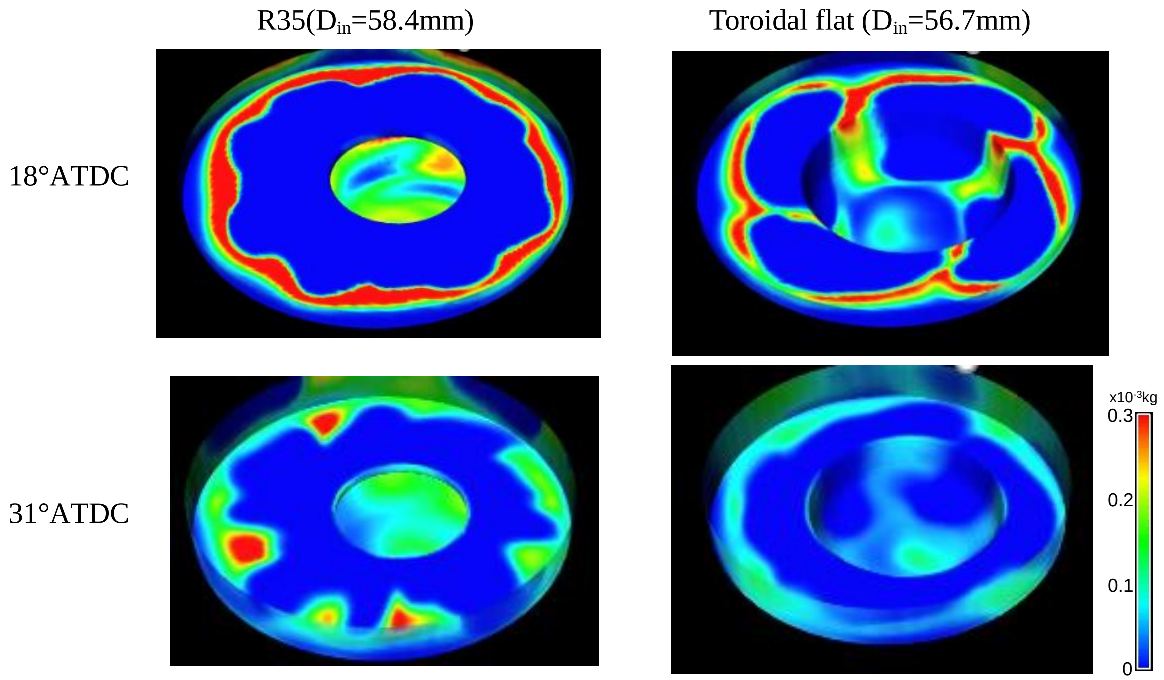


Fig.7. CFD analysis of fuel distribution near the top surface of piston.

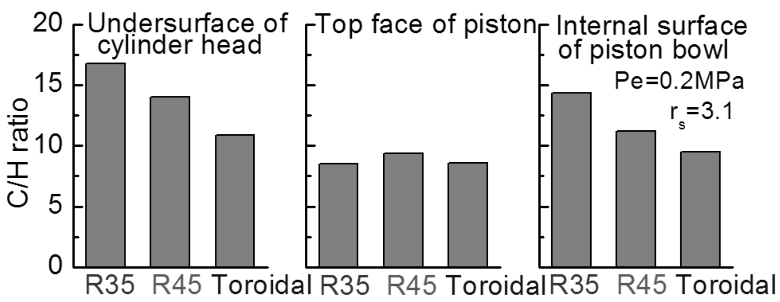
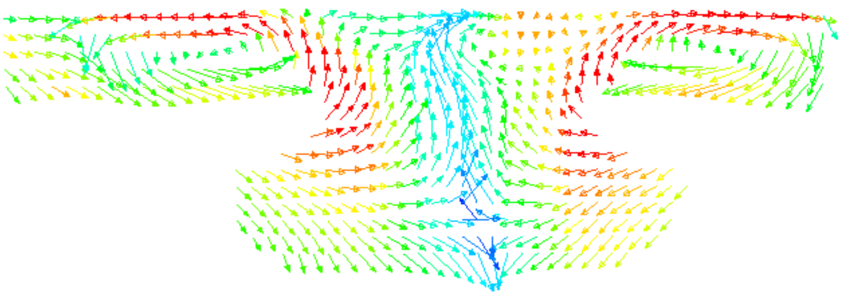
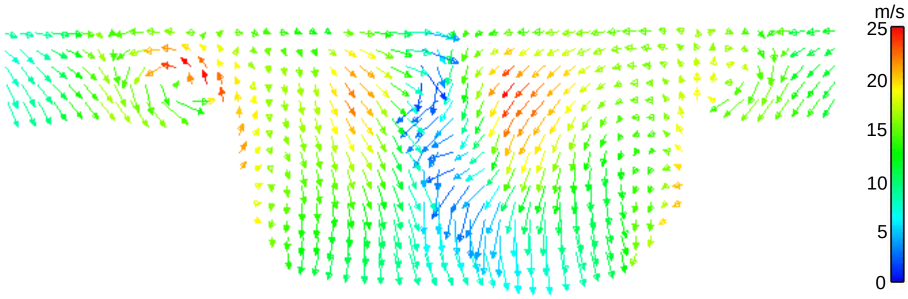


Fig.8. Elementary analysis (C/H ratio) of deposit.



R35(D_{in}=58.4mm)



Toroidal flat (D_{in}=56.7mm)

Fig.9. Analysis of gas flow (31°ATDC).

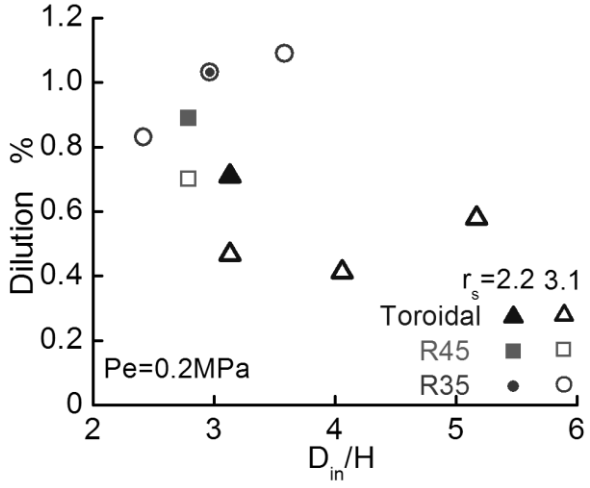
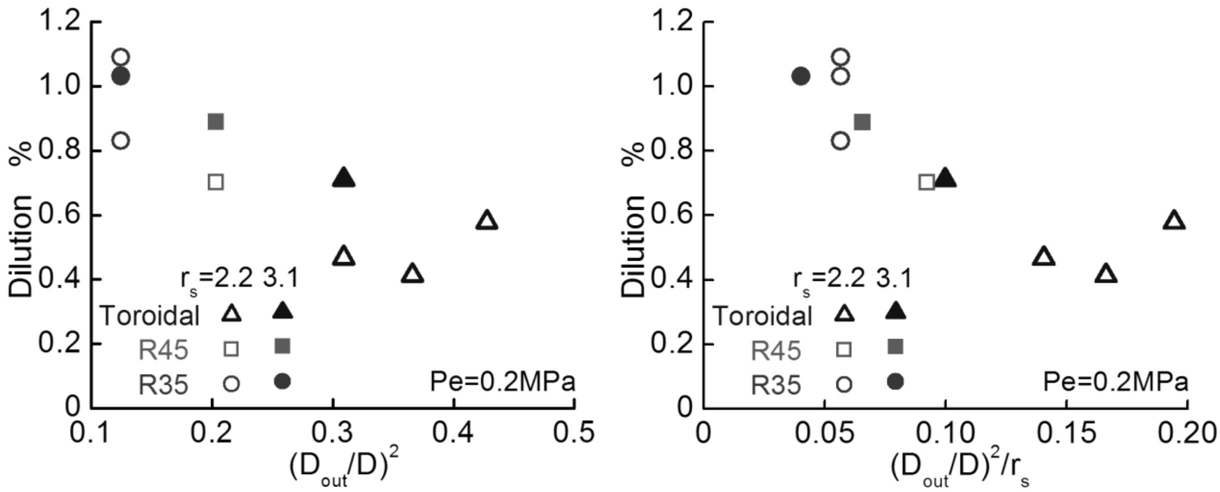


Fig.10. Relation between aspect ratio and fuel dilution in lubricating oil.



(a) Effect of aperture ratio

(b) Effect of aperture ratio and swirl ratio

Fig.11. Effect of gas velocity on fuel dilution.
(a) Effect of aperture ratio. (b) Effect of aperture ratio and swirl ratio.

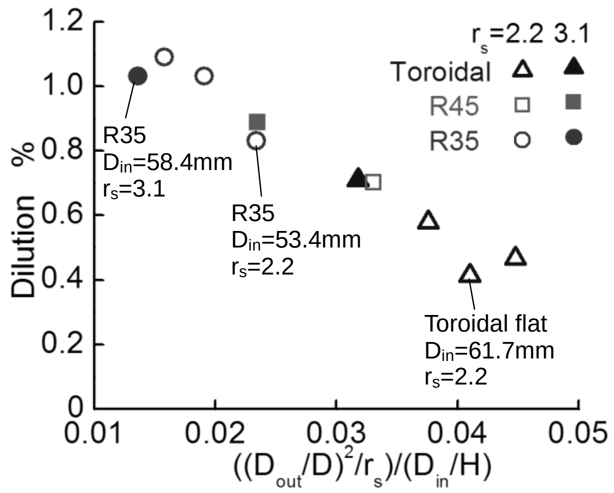


Fig.12. Effect of gas flow on fuel dilution.

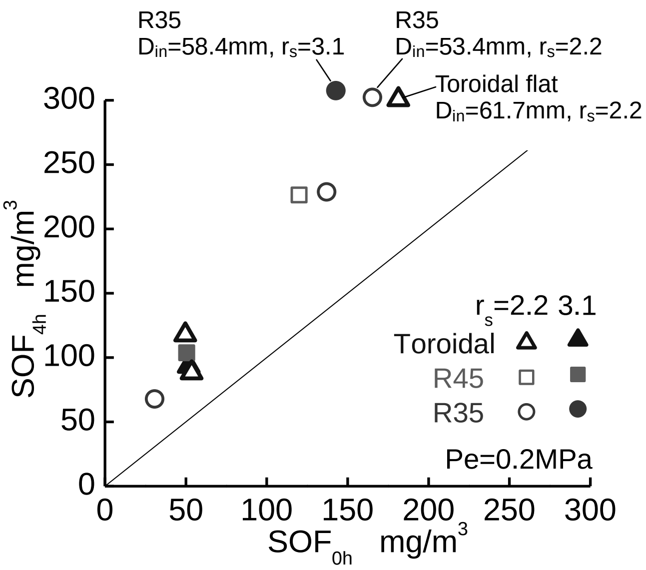
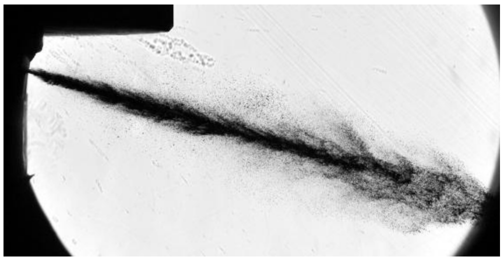


Fig.13. SOF emission before and after four hours continuous operation.

(a) without deposit



(b) after deposit formation

Fig.14. Change in spray characteristics before and after deposit formation in the nozzle hole [20]. (Injection pressure is 70MPa. Images are captured at 0.5ms aSOI under ambient condition of 298K and 1MPa.) (a) without deposit. (b) after deposit formation.

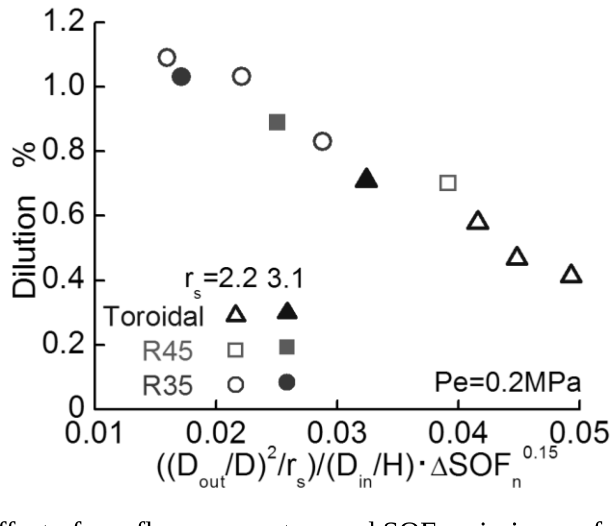


Fig.15. Effect of gas flow parameters and SOF emission on fuel dilution.

Effect of low load combustion and emissions on fuel dilution in lubricating oil and deposit formation of DI diesel engines fueled by straight rapeseed oil

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Abstract

The objective of this study is to apply neat biomass fuel to a DI diesel engine and investigate the effect of in-cylinder gas flow and combustion on the deposit formation and the fuel dilution in lubricating oil. The study focuses on the low load combustion and emissions considering that low load exhaust contain much unburned fuels and the unburned fuels are the source of the deposit formation and the fuel dilution. Piston configuration and swirl velocity were altered in the engine test. The engine was fueled by neat rapeseed oil. The test was carried out through the four hours continuous engine operation with keeping low load. After the operation, state of deposit formation and fuel dilution in lubricating oil were investigated. Results indicate that Re-entrant piston which creates strong reverse squish and high swirl forms the deposit annular on the piston top. Toroidal piston easily produces deposit on the undersurface of cylinder head. The deposit in the cavity accumulates where initial rapeseed oil spray impinges regardless of piston types. The carbonization of the deposit is promoted on the wall surface where the burned gas with high temperature and high velocity comes into contact. It is important to avoid extremely strong reverse squish to the cylinder liner in order to control the fuel dilution. The deep-bowl chamber changes the direction of reverse squish from the cylinder liner direction to the cylinder head direction. The low velocity outflow from the piston cavity reduces the

adhesion of unburned fuel on the cylinder liner, resulting in the smaller amount of unburned fuel scraped off by a piston ring.

1. Introduction

Biomass fuel is substitute for petroleum and has a carbon neutral property. The fuel can contribute to improve energy security and prevent global warming. Therefore, it has been studied on application of a biomass fuel to a diesel engine as a carbon neutral alternative fuel [1-7]. The majority of the researches apply fatty acid methyl ester as biomass fuel. According to these researches, exhaust emission, in particular, high load particulate emission can be reduced by using biomass fuel with a direct injection (DI) diesel engine. An et al. [8] and E et al. [9] investigated combustion and emissions of diesel engine fueled by biodiesel at partial load conditions. They showed that the thermal efficiency is reduced at lower engine loads and the efficiency is improved at higher engine loads. At lower engine loads, the CO emissions increase with the increase of the biodiesel blend ratio. Liu et al. [10] and E et al. [11] further developed a skeletal mechanism and predicted soot formation characteristics for biodiesel blends. They indicated that the kinetic viscosity is a very important factor to the combustion of biodiesel fuels. In most of these researches, biomass fuel was esterified generally and used as fatty acid methyl ester as mentioned above. The esterification brings a biomass fuel property close to gas oil, which makes it easy to use the alternative fuel with an existing engine. However, the esterification requires an industrial plant and the treatment process of the by-product, which raises the fuel cost. In this kind of situation, it is desirable to apply a biomass fuel directory to a DI diesel engine. Concerning the researches supplying neat biomass fuels to DI and pre-chamber type diesel engines, engine performance and emissions have been investigated from the past [12-15]. These researches showed that applying neat biomass fuel instead of gas oil can improve emissions at high load. Biomass fuel is oxygenated fuel, which is favorable at high load that needs much oxygen. However, low load emission will deteriorate more than the case of using fatty acid methyl ester. This is because neat biomass fuel has the high kinetic viscosity and high evaporation temperature characteristics as compared with the fatty acid methyl ester.

Moreover, researches of the past [12, 16-19] pointed out that deposit formation is one of the problems when a biomass fuel is directly used with a DI diesel engine. Kawasaki et al. [16] evaluated the deposit formation in the combustion chamber and investigated the effect of fuel injector deposit on engine performance and emissions. They suggested that deposit formation in the nozzle hole affects engine performance and emissions. Biomass fuel has poor atomization and evaporation characteristics due to high distillation temperature with high viscosity [20-21]. These characteristics results in deposit formation. Once the deposit is formed in the fuel injector hole, the spray characteristics changes, and subsequently, combustion and emission get worse. When neat biomass fuel is applied to a DI diesel engine, some of the liquid phase spray impinges on the wall of piston chamber without fully evaporation. Further, in the case of neat biomass combustion, burned gas contains some amount of unburned fuel in particular under low load condition. The unburned component cools off on the wall surface and stick to the wall, forming deposit.

Another problem in applying neat biomass fuel to diesel engines is fuel dilution in lubricating oil

[22-25]. Kitazaki, et al. [22] suggested the mechanism of the fuel dilution as follows. The fuel which sticks on the wall surface of cylinder liner is scrapped off by a piston ring, and then the fuel gets mixed with the lubricating oil. Sem [25] indicated that the deposit formation is affected by type of lubrication oil. When the fuel is diluted in the lubricating oil, the lubrication performance deteriorates and causes the abrasion and ghosting of the slide parts.

Both deposit formation and fuel dilution in lubricating oil affect the engine durability greatly. It is necessary to reduce them with keeping low emission. As for the emissions, soluble organic fraction (SOF) emissions and unburned fuel are much produced at low load as compared with higher load when the biomass fuel is applied [13]. These emissions may be the source of the deposit formation and the fuel dilution. It is also likely that the in-cylinder gas flow during combustion affects the adhesion of unburned fuel on the wall surface of cylinder liner as well as the emission itself. However, it is not clear the influence of combustion and emissions on the deposit formation and the fuel dilution. And furthermore, there are few researches on detail investigation of the effect of engine combustion on deposit formation and fuel dilution.

This study investigates the effect of in-cylinder gas flow and combustion on the deposit formation and the fuel dilution focusing on low load operation that produces much unburned fuels. Authors previously attempted to improve emissions from a DI diesel engine when applying neat rapeseed oil as biomass fuel [13, 26]. Figure 1 shows one of the results [26]. Generally, as compared with gas oil (indicated by the broken line), rapeseed oil combustion (the thin line without plots) reduces particulate emission (PM) at high load, mainly insoluble fraction (SOLID), but soluble organic fraction (SOF) increases with the lowering of the load. We presumed that combustion with high SOF emission is closely related to the increase of deposit. Furthermore, burned gas with high SOF emission also contains the component similar to the original fuel. When such burned gas comes to the cylinder liner, fuel component adheres to the cylinder liner. Then, the component is scrapped off by a piston ring, resulting in the increase of fuel dilution in lubricating oil. Based on such supposition, we tried to reduce SOF emission at low load. Rapeseed oil spray is hard to be atomized and evaporated in the low temperature atmosphere at low load. The poor spray characteristics causes SOF emission. We attempted to utilize in-cylinder gas flow by modifying piston configuration as shown in Fig.1 so as to improve the atomization and evaporation of rapeseed oil spray. The flat bottom cavity can elongate the spray penetration and atomization after the wall impingement. Re-entrant shape can create high turbulence in the chamber, which supports the mixing of fuel with air and also avoiding the wall impingement itself in the cavity. Both techniques finally result in the reduction of SOF emission at low load as shown in the figure. In particular, re-entrant piston with a flat bottom offers the greatest effect of the reduction.

This study applied the above combustion with low SOF emission and investigated the mechanism of deposit formation and fuel dilution affected by combustion. The experiment was conducted by continuous four hours engine operation with fueling straight rapeseed oil and keeping engine load at low level. The long-time engine operation formed deposit and caused fuel dilution in lubricating oil. The piston configuration and swirl velocity were altered to change the in-cylinder gas flow and combustion.

2. Experimental method

The test engine was a four-stroke single cylinder naturally aspirated direct-injection diesel engine (YANMAR NFD-170). Table 1 shows the engine specifications. This study used two classes of swirl ratio, r_s, that is original swirl ratio of 2.2 and higher swirl ratio of 3.1.

This study also changed piston configurations. Detail of piston specifications is shown in Table 2 and Fig.2. The piston geometries are designed based on our previous research [26] as shown in Fig.1. Toroidal base for reference is the original piston geometry with this engine. Toroidal flat has a flat bottom which promotes evaporation and atomization of rapeseed oil spray after the wall-impingement at low load. R35 and R45 are re-entrant type pistons with a flat bottom. Re-entrant piston can generate high turbulence and improve particulate emissions at high load as well as low load. The aperture ratio D_{out}/D which is the ratio of cavity-exit diameter D_{out} to piston diameter D is about 35% for R35 and 45% for R45. R35 and Toroidal flat have three types respectively with different bowl diameter D_{in}. There are 7 types of pistons used in this study. All pistons have the same volume of piston bowl with different bowl depth H. The spray arrows in Fig.2 indicate the spray injecting direction at TDC (Top dead center).

Table 3 shows the test condition of the engine. Rapeseed oil was supplied to the engine. Table 4 compares fuel property between rapeseed oil and gas oil. Rapeseed oil is oxygen-containing fuel with high viscosity and high distillation temperature. Figure 3 shows the schematic diagram of experimental setup. Exhaust emissions such as NOx, CO and THC were measured by an engine exhaust gas analyzer (HORIBA: MEXA-1500D). A mini-dilution tunnel and filters were employed for sampling particulate emissions. A part of exhaust gas was introduced to a mini-dilution tunnel. The particulate concentrations were determined by measuring the filter weight before and after the sampling. The soluble organic fraction (SOF) was extracted by Soxhlet extraction method using dichloromethane as solvent. In this way, particulate matter (PM) is separated into insoluble fraction (SOLID i.e. dry soot and sulfate) and SOF. The measurement errors of the analyzer with this test were within 15ppm for NOx, 30ppm for CO, 10ppm for THC and 20mg/m^3 for SOF.

The engine was operated for four hours with keeping the engine speed at 1800rpm and the low load of 0.2MPa BMEP (Brake mean effective pressure, Pe). The test procedure is as follows. Before the continuous running, the inside of the engine, that is the undersurface of the cylinder head, cylinder liner, piston bowl and injection nozzle was cleaned up. New lubricating oil was supplied to the engine. During the four hours operation, exhaust emissions were measure at hourly intervals. After the four hours operation, the lubricating oil was extracted and the mass concentration of the fuel included in the oil was measured. The concentration is defined as fuel dilution rate. The dilution measurement was conducted by means of FTIR (Fourier transforms infrared spectrometer, Measurement instrument: PerkinElmer: Spectrum100). The area of peak waveform of ester bond with wavenumber of 1746cm^{-1} provides the fuel dilution rate in lubricating oil [22]. Deposit formed in the engine was sampled and the mass ratio of carbon to hydrogen in the deposit was measured by an organic elementary analyzer (J-SCIENCE LAB: MICRO CORDER JM10).

CFD calculation was employed to support the analysis of the experimental results through the data of gas flow and fuel distribution in the cylinder. A commercial software CONVERGE was used. The RNG k-ε model and KH-RT breakup model are applied to simulate turbulence and droplets behavior. O'Rourke and Amsden model is applied to simulate behavior of wall film. Splash model is based on Weber number, film thickness and viscosity. Adaptive mesh size is based on 0.04mm. The calculation was performed in the case of R35 (D_{in}=58.4mm, r_s=2.2) and Toroidal flat (D_{in}=56.7mm, r_s=2.2) pistons.

3. Results and discussion

3.1. Exhaust emissions, deposit formation and fuel dilution in lubricating oil

Figure 4 shows the histories of emissions during four hours continuous engine operation at low load of Pe=0.2MPa. Test parameters are piston configuration and swirl ratio. It can be seen from the figure that NOx, CO and THC emissions and brake thermal efficiency η_e change little for four hours under every condition. η_e changes within 0.8% for every condition. The R35 piston with D_{in}=53.4mm shows the greatest increase of CO, THC concentrations. The increases are 141ppm and 231ppm, respectively. On the contrary, SOF emissions increase greatly with the lapse of time. SOF emissions increase over 65% for four hours operation under every condition. The greatest change in SOF is from 165.4 to 305.2mg/m^3 for R35 with D_{in}=534mm and the minimum change is from 53.6 to 88.3mg/m^3 for Toroidal flat with D_{out}=56.7mm. The increase is more remarkable when initial SOF emission is high. It is important to keep an eye on low load SOF emission when using rapeseed oil as fuel.

Figure 5 indicates the fuel dilution in lubricating oil after the engine operation in Fig.4. The fuel dilution rate is corrected by the same injection quantity as the condition of Toroidal flat, D_{in}=56.7mm and r_s=2.2. We confirmed in advance that, at high load condition, the fuel dilution rates for all conditions were in the ranges from 0.15 to 0.27%, which are low levels as compared with those at low load condition. This is because high load produces less unburned fuel as compared with low load. As seen from the Fig.5, the fuel dilution is in the order of R35>R45>Toroidal flat. Re-entrant pistons which cause high turbulence and high squish flow produce high fuel dilution rate. Larger bowl diameter D_{in} produces higher dilution rate for R35 piston. Toroidal piston also has similar trend between fuel dilution and D_{in}, although the data of D_{in}=61.7mm shows small amount of dilution. Further, the three conditions of D_{in}=53.4mm for R35, r_s=2.2 for R45 and D_{in}=61.7mm for Toroidal flat offer comparatively low dilution rate. These conditions produce higher SOF emissions in Fig.4. There may be some relation between SOF emission and fuel dilution. This will be discussed later.

Figure 6 shows the examples of deposit formation after the four hours running at low load. The bold white arrows in the figure indicate the position where deposit is formed greatly. Comparing the deposits on the undersurface of cylinder head, Toroidal flat forms thick deposit and the deposit is accumulated in the direction of injecting fuel spray. This is considered as follows. The injection timing with this test was -5^{o}ATDC which is the timing near TDC and Toroidal flat piston has wide outer diameter of piston bowl. Under this condition, part of initial spray is affected by the gas flow in the chamber and the spray tends to contact to the undersurface of cylinder head and form the deposit on the face. Deposit formation at the top face of piston differs depending on piston type. R35 and R45 pistons form deposit circular on the outer side of piston top. On the other hand, Toroidal flat accumulates deposit in the direction of the outflow of four sprays. Deposit formation on the internal surface of piston cavity indicates that, regardless of piston type, deposit is accumulated where initial sprays impinge. This means that, in spite of small injection quantity at low load, some amount of rapeseed oil spray impinge on the wall without fully evaporation. This is because rapeseed oil has weak evaporation characteristics and the ambient temperature is low at low load condition. It is important to avoid the wall impingement at low load when using rapeseed oil.

The deposit formation in Fig.6 is analyzed by CFD calculation. Figure 7 shows fuel distribution on the wall surface at expansion stroke of 18^{o}ATDC and 31^{o}ATDC. The distribution indicates the fuel mass in the calculation mesh on the wall. At the expansion stroke, burned gas containing some amount of unburned fuel flows from the piston cavity to the squish area and impinges on the cylinder liner. It can be seen from the figure that R35 has a ring-shaped high density fuel distribution on the piston top and Toroidal flat has high density fuel distributions at the direction of the spray outflow. These distributions resemble the deposit formation on the top face of piston shown in Fig.6. R35 piston has small size bowl-outlet diameter, which creates strong reverse squish and high swirl. This kind of gas flow makes unburned fuel distribution ring-shaped on the outer side of piston top and produces deposits on this location. Moreover, in the piston cavity, high density fuel distribution is seen near the spray impinging area. This distribution also corresponds to the deposit formation in the cavity. Figure 8 shows the results of elementary analysis of the deposit in Fig.6. The C/H means the mass ratio of carbon to hydrogen in the deposit. Regardless of piston type, the C/H of the deposits is highest on the undersurface of cylinder head and the second highest is on the internal surface of piston bowl.

Furthermore, at these positions, the C/H is in the order of R35>R45>Toroidal flat. While, the C/H is low on the top face of piston and there is little difference of the C/H among the piston type.

Figure 9 shows gas flow at expansion stroke obtained by CFD calculation. As seen from the gas flow of R35, R35 piston produces the reverse squish with high velocity near the squish lip. The burned gas flows to the undersurface of the cylinder head and impinges on the surface. After the impingement, the gas reaches the cylinder liner. Then the gas flows to the piston top and moves to the center. In the case of rapeseed oil combustion at low load, the burned gas contains unburned fuel, thus much unburned fuel reaches up to the cylinder liner and, after the impingement on the liner, some of the unburned fuel can contact to the piston top. Mutually effect of strong reverse squish and high swirl causes the ring-shaped high density fuel distribution on the piston top. In addition, high temperature gas in the piston bowl first impinges on the cylinder head, resulting that deposit can be easily carbonized. On this account, the C/H is high on the undersurface of cylinder head and the internal surface of piston bowl. As for Toroidal flat piston, reverse squish is weak as compared with R35, so deposit carbonization progresses slow due to weak impingement of high temperature gas on the wall. Regardless of piston type, the most of burned gas reaches the piston top with the lowering of the temperature after the impingement on the cylinder head and the cylinder liner; therefore, the C/H of the deposit is low on the piston top.

3.2. Relation between in-cylinder gas flow and fuel dilution

The preceding section suggests that the gas flow in the engine cylinder affects unburned fuel distribution and deposit formation. It is likely that the fuel dilution in lubricating oil is also affected by the gas flow. The gas flow considerably depends on piston geometry.

Figure 10 investigates the relation between the aspect ratio D_{in}/H of combustion chamber and the dilution rate shown in Fig.5. It seems to be little correlation between D_{in}/H and fuel dilution. However, focusing on R35 with swirl ratio r_s=2.2 (symbol: open circle) and Toroidal flat with r_s=2.2 (symbol: open triangle) individually, there is a tendency that lower aspect ratio offers lower dilution rate. Low aspect ratio corresponds to the deep bowl chamber. The deep bowl chamber creates a reverse squish in a vertical direction to the undersurface of cylinder head. Consequently, some amount of unburned fuel first stick on the undersurface of cylinder head and smaller amount of unburned fuel reaches the cylinder liner. The smaller adhesion amount of unburned fuel to the cylinder liner reduces the fuel dilution in lubricating oil.

Figure 11 investigates the relation between the gas velocity flowing out from the chamber and the dilution rate. Here it is assumed that the gas velocity of reverse squish from the chamber is roughly inverse proportional to the opening space of the chamber which is related to the square of the aperture ratio, $(D_{out}/D)^2$. Figure 11(b) takes swirl velocity into account. From the figures, the greater value of $(D_{out}/D)^2$ or $(D_{out}/D)^2/r_s$ reduces the dilution rate. This means the low-velocity squish flow decreases the fuel dilution.

Figure 12 is the analysis with combination of Fig.10 and Fig.11. There is a good correlation between $(D_{out}/D)^2/r_s/(D_{in}/H)$ and the dilution rate. Great value of $(D_{out}/D)^2/r_s/(D_{in}/H)$ corresponds to the condition of deep bowl chamber and low velocity outflow. The fuel dilution has an almost liner relation with $(D_{out}/D)^2/r_s/(D_{in}/H)$. The deep-bowl chamber changes the direction of reverse squish from the cylinder liner direction to the cylinder head direction. The low velocity outflow with low squish and low swirl can also carry smaller amount of unburned fuel to the cylinder liner. These reduce the adhesion of unburned fuel on the cylinder liner, resulting in smaller amount of fuel component scraped off by a piston ring. Figure 12 reveals that the change in the direction of reverse squish and the squish velocity affects the fuel dilution in lubricating oil. When biomass fuel is directly used for a DI diesel engine, re-entrant piston is necessary to improve high load emissions. However, to control the fuel dilution, the reverse squish toward the cylinder liner should be weakened and a deep bowl chamber with low swirl is desirable. Additionally in Fig.12, three conditions of R35 with D_{in}=53.4mm (r_s=2.2), R35 with D_{in}=58.4mm (r_s=3.1) and Toroidal flat with D_{in}=61.7mm (r_s=2.2) are a bit lower dilution rate than the liner correlation in the figure. This will be discussed in the next session.

3.3. Influence of excessive increase of SOF emission on fuel dilution

This section investigates the relation between the fuel dilution and the change in SOF emission during the engine operation.

Figure 13 shows the relation of SOF emission between at the start of four hours continuous operation, SOF_{0h} and at the end of the operation, SOF_{4h}. The data is obtained by Fig. 4. From the figure, long time continuous operation at low load increases SOF emission. In particular, the condition that initial SOF level SOF_{0h} is high deteriorates SOF_{4h} greatly. The increase of SOF emission during the low load operation originates from the deterioration of combustion. One possible reason of the change is the deposit formation in the nozzle hole [16]. Authors once investigated spray characteristics with the injection nozzle that contained deposit after long time operation at low load [20]. Figure 14 shows the example of the spray characteristics. The deposit formation in the hole makes the spray cone angle smaller and the spray penetration longer. The combustion with high SOF emission forms deposit easily, which deteriorates the spray characteristics. As a result, combustion further gets worse with producing higher SOF emission.

In Fig.13, the three conditions that produce high SOF_{4h} are first R35 with D_{in}=53.4mm and r_s=2.2, second R35 with D_{in}=58.4mm and r_s=3.1, and third Toroidal flat with D_{in}=61.7mm and r_s=2.2. These three conditions are low dilution rate in Fig.5 and Fig.12. Considering the data in Figs. 5, 12 and 13, SOF emission may relate to the fuel dilution. Increase of SOF emission deteriorates spray characteristics as shown in Fig.14. Long spray penetration increases the wall impingement in the chamber. This kind of excess impingement produces SOF emission due to the deterioration of combustion. This may increase fuel dilution. On the other hand, the excess wall impingement in the cavity restrains unburned fuel from flowing out of the cavity. As a result, smaller amount of fuel reaches the cylinder liner and the fuel dilution in lubricating oil is suppressed. Further, narrow cone angle inhibits the overflow of the spray from the bowl before the impingement, which also reduces the amount of fuel adhesion on the cylinder liner. Consequently, the condition with extremely SOF emission has a possibility to suppress the fuel dilution. However, the engine performance gets worse in this case.

Figure 15 takes the reduction of fuel dilution caused by the excess increase of SOF emission into account against Fig.12. The parameter ΔSOF_n is the increasing amount of SOF emission for four hours operation normalized by that of Toroidal flat with r_s=2.2 data. The data correlation in Fig.15 is improved as compared with the data in Fig.12. In this experiment, fuel dilution in lubricating oil is determined by both the strength and direction of the reverse squish flow and the increasing rate of SOF emission.

4. Conclusions

The objective of this study is to apply neat rapeseed oil to a DI diesel engine and investigate the effect of in-cylinder gas flow and combustion on the deposit formation and the fuel dilution in lubricating oil. The study focuses on the low load combustion and emissions that produce much unburned fuel. Piston configuration and swirl velocity were altered with the engine test. The test was the four hours continuous engine operation with keeping low load. Analysis of the deposit formation and the evaluation of the fuel dilution rate revealed the following things.

In-cylinder gas flow affects deposit formation. The deposit is formed in circular near the outer edge of piston top for re-entrant piston. Re-entrant piston creates strong reverse squish and high swirl. This kind of gas flow causes circular unburned fuel distribution on the piston top and forms the deposits on this location. Toroidal piston accumulates deposit on the undersurface of cylinder head. The deposit in the cavity accumulates where initial rapeseed oil spray impinges.

The C/H ratio of the deposits is highest on the undersurface of cylinder head and the second highest is on the internal surface of piston cavity. Re-entrant piston that has stronger reverse gas flow produces higher C/H ratio of the deposit than toroidal piston. The carbonization of the deposit is promoted on the wall surface where high temperature and high speed burned gas comes into contact.

The fuel dilution in lubricating oil is affected by the in-cylinder gas flow and the flow direction caused by piston configuration. The deep-bowl toroidal piston with low swirl reduces the fuel dilution. The deep-bowl chamber changes the direction of reverse squish from the cylinder liner direction to the cylinder head direction. Toroidal piston produces the lower level of fuel dilution than re-entrant piston. For toroidal piston, the low velocity outflow from the piston cavity reduces the adhesion of unburned fuel on the cylinder liner, resulting in smaller amount of unburned fuel scraped off by a piston ring. In order to control the fuel dilution, it is important to avoid extremely strong reverse squish to the cylinder liner with keeping SOF emission at low level.

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