726 *KSME International Journal, Vol. 11, No. 6, pp. 726~736, 1997*

Measurement of In-Cylinder Equivalence Ratio During Starting Using a Fast FID

Younggy Shin*

(*Received March 26, 1997*)

In spite of the importance of the mixture preparation process during starting in affecting HC emissions, it has been poorly understood because the process is highly complicated by the multi-component fuel, transient thermal environment, and two-phase mass transfer. For better understanding of the process, measurement of the equivalence ratio in the cylinder was attempted using a fast FID (Flame Ionization Detector). Although extensive research has been made on the measurement using the fast FID, the method is not yet straightforward and needs careful operation. In this paper, the current status of the method is described. And within the limitations of the method the in-cylinder sampling test was conducted under the engine operating condition of 900 rpm, 0. 527 bar intake pressure with gasoline injected. The test results show that 15 to 20 % of the injected fuel evaporates in the first cycle regardless of the amount of fuel injected. The lean limit of the equivalence ratio for engine firing was about 0. 6. For secure firing in the first cycle, 5 to 6 times the stoichiometric fuel amount had to be injected. The contribution of the left-over fuel in the previous cycles to the mixture formation in the present cycle was found to be substantial.

Key Words: Mixture Preparation, FID, Equivalence Ratio, Engine, Evaporation, Liquid Fuel

1. Introduction

The fast FID has drawn attentions of many engine researchers because of its capability of cycle-resolved HC measurements. When it comes to HC measurements in the exhaust port, the measurements have been proven to be reliable and very useful in understanding HC oxidation mechanisms. Encouraged by the successful application to the exhaust port, many researchers tried to extend the application to the HC measurements in the engine cylinder. Measuring the HC concentrations of unburned mixture in the cylinder is extremely important to understand mixture preparation process in a spark ignition engine from the following reason. Most of HC emissions are emitted during starting and early warm-up due to poor mixture preparation caused by cold engine condition. The mixture preparation process during the period is very complicated due to coexistence of liquid and vapor fuels and transient thermal environment. Therefore, measuring the air-to-fuel ratio of the mixture before spark ignition is the first step to understand the mixture preparation process and to propose ideas for better mixture preparation.

In spite of its promising features for fast response, the application of the fast FID to the in-cylinder HC measurements has progressed little. It is because physical phenomena such as condensation and pressure interference contaminate the in-cylinder measurement process and result in output distortion. This paper explains the mechanisms of such interference and suggests possible ways to avoid them. The knowledge is also applied to measure the in-cylinder HC concentrations in the limited engine operation conditions where the measurements are valid.

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* Department of Mechanical Engineering, Massachusetts Institute of Technology, 77 Mass. Ave., Cambridge, MA, 02139, U. S. A.

2. Physical Origin of the FID Signal

The basic physical phenomenon that is exploited in a flame ionization detector (FID) is that when a hydrocarbon is burnt, significant quantities of ions are formed. The number of ions produced is nearly proportional to the number of carbon atoms burnt in hydrocarbon (HC) form. The physical explanation for this effect is not yet fully understood. The process of ion generation is certainly to do with a non-equilibrium effect called chemi-ionization, as opposed to thermal equilibrium ionization, since the latter is responsible for negligible ionization at normal flame temperatures. The number of ion pairs generated has been established in many experiments to be about 1 ion pair per 10^6 carbon atoms (in aliphatic HC form), taking part in combustion (Lawson and Weinberg, 1969). Therefore, the ion current is simply proportional to the flow rate of carbon atoms in the hydrocarbon, CmHn, as follows:

$$i \propto m\dot{N}_{C_mH_n} \tag{1}$$

where i is the current, m is the number of atoms in the HC molecule, $\dot{N}_{C_mH_n}$ is the molar flow rate of the HC molecule. From the relation, $\dot{N}_{C_mH_n}=[HC]\dot{N}_{sample}$ where [HC] is the hydrocarbon concentration and $\dot{N}_{sample}$ is the molar flow rate of the sample gas. From the ideal gas law,

$$\dot{N}_{sample}=\frac{pQ}{R_0T} \tag{2}$$

where p and T are the pressure and temperature at the nozzle end to the flame respectively, and Q is the volumetric flow rate of the sample gas, and R_0 is a universal gas constant. Then, the ion current is finally expressed as follows:

$$i=K_im[HC]\frac{pQ}{R_0T} \tag{3}$$

Since the FID output in voltage, V_{FID}, is designed to be proportional to the current i,

$$V_{FID}=K_{FID}m[HC]\frac{pQ}{R_0T} \tag{4}$$

As far as p, T, and Q are kept constant, the FID output, V_{FID}, is proportional to the hydrocarbon concentration, [HC].

3. Schematic of the Fast FID

Figure. 1 is the schematic of the fast FID in-cylinder sampling unit. An essential difference from the conventional design is that the sample gas is mixed with the fuel gas at the nozzle exit. It is this design change which is the reason for the high frequency response, which in this case is limited only by dispersion effects within the sample tube and in the flame itself (Cambustion Ltd., 1992). When it comes to the conventional design, the flow rates are controlled by allowing each gas to pass through a capillary where the rate of flow depends upon the cross section of the capillary, the length, the pressure drop across the capillary (Patterson, 1972). The use of the capillary has been the bottle-neck to faster response. A short sample probe is desirable for good frequency response. For this reason the sample is drawn directly into the FID. Thus the flame chamber must be maintained at the pressure below that of the sample source. The design change over a conventional FID has produced important advantages, but also the significant disadvantage that the instrument cannot be merely switched on and the output interpreted as HC concentration. To get the absolute HC concentration, the device should be carefully set and calibrated.

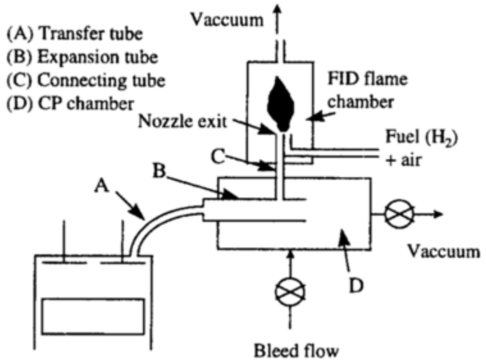


Fig. 1 Fast FID sampling unit for in-cylinder sampling.

4. Physical Phenomena Affecting the In-cylinder HC Measurements

4.1 Pressure interference

If the pressure in the CP chamber is not kept constant, the sample gas flow rate into the flame chamber, Q, will be affected. Then from Eq. (3) the current i is no longer proportional to [HC]. The phenomenon tends to happen frequently if careful caution is not taken. Thus, it should be addressed first before any measurement is made. There are three types of problems associated with the pressure interference. They are choking, sample gas mixing due to flow reversal and pressurizing of the CP chamber.

4.1.1 Choking

It is caused basically by the high cylinder pressure during the compression and /or combustion. In Fig. 1, the fluid in tubes A and B would be accelerated by the increasing pressure force. At some point, the exit to the transfer tube A becomes sonic and the flow is choked (the flow field in tube A no longer depends on the downstream conditions). The flow in tube B would adjust itself to swallow the mass flow rate and to maintain the exit pressure equal to the pressure of the CP chamber D. The mathematical relationships are detailed in Cheng et al.'s paper (Cheng, Galliot and Collings, 1989). The situation persists for some time as the cylinder pressure rises. At some later time, depending on the cylinder pressure, however, the exit to tube B also reaches sonic condition. Beyond this time, the velocity at the exit to tube B would be sonic and the pressure just inside the exit of tube B would no longer be equal to the pressure in D, but adjusts according to the inlet condition. Eventually, tube B would be pressurized and tube A would become unchoked. Finally, when the cylinder pressure goes down, tube B would become unchoked.

It is not desirable to operate the sampling unit beyond the point where the exit to tube B becomes sonic. Then, the pressure at the inlet to the connecting tube C would not be constant and the whole purpose of the constant pressure sampling system is defeated. Therefore, the CP chamber pressure should be carefully set up to avoid choking. For example, Cheng et al. estimated the relationship between the CP chamber pressure and critical cylinder pressure as shown in Fig. 2. The typical pressure just before ignition for naturally aspirated engine at WOT (wide open throttle) is ~8bar. Therefore if the CP chamber pressure is above ~0. 4 bar, choking at tube B would usually not occur until sometime into the combustion period.

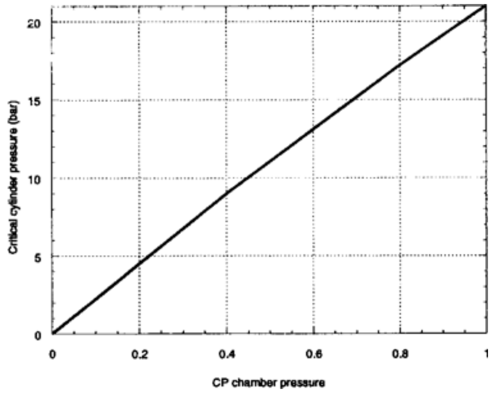


Fig. 2 The critical cylinder pressure at which choking occurs at the expansion tube exit for the sampling unit (Steady State Calculation).

4.1.2 Transient period due to flow reversal

According to Fig. 2, setting the CP chamber pressure to the atmospheric pressure would be ideal to avoid choking. However, it causes another problem related to the reversal of flow direction. For most of engine operation, the cylinder pressure during intake process is lower than the atmospheric pressure. Therefore, the fluid in transfer tube A flows from the CP chamber into the cylinder during the intake process. The fast FID starts to sample the gas in the cylinder as soon as the cylinder pressure exceeds the atmospheric pressure. However, the FID goes through a transient period before reading correct HC concentrations. Figure 3 demonstrates this case.

A single cylinder spark ignition engine was operated on the prepared air and propane mixture. Since the mixture is essentially homogeneous, the fast FID should read steady state values

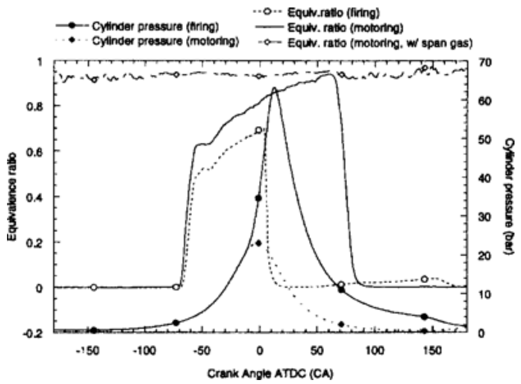


Fig. 3 Comparison of equivalence ratio converted from in-cylinder FID measurements under the propane-fueled engine operation of 900 rpm and 0. 587 bar intake pressure for the following cases of (a) firing, (b) motoring and (c) motoring with a span gas bled through the CP chamber.

when the cylinder pressure is above atmospheric pressure. In Fig. 3 shown are the following three test cases: (a) engine with ignition, (b) engine without ignition, and (c) engine running without ignition and with a span gas bled through the CP chamber. In case (c), the span gas whose HC concentration is nearly the same as that of the mixture entering the engine. A beauty of the configuration (c) is the 'in-situ' calibration of the fast FID against any possible output drift. It is possible since the FID samples the span gas while the cylinder pressure is sub-atmospheric. Figure 3 shows good plateau of the FID output for case (c). It is expected from the fact that both the mixture and the span gas are homogeneous. However, when it comes to cases (a) and (b), the outputs do not show plateau, but rise steadily right after the jumps caused by the arrival of the in-cylinder gas into the FID chamber. Case (b) shows agreement with case (a) right before the signal drops due to the lowered cylinder pressure. It means that the output reaches steady state by then. However, case (a) shows early drop of the output before reaching the steady state due to the arrival of the flame at the tip of the transfer tube. Therefore, the atmospheric CP condition can not be applied to the measurements of in-cylinder HC under engine firing.

As it is clear in Fig. 3, the transient period (the part steadily rising right after the jump) is a key obstacle to make the in-cylinder measurements possible. Its cause should be analyzed. The cause of the transient behavior has not been clarified. It seems to be related with the complex mechanism of chemi-ionization process (Sternberg, Gallaway and Jones, 1962). The energy release by the unburned sample gas is comparable to that by the FID fuel (hydrogen). It means that the energy content of the sample gas may affect the thermal environment of the flame chamber, and in turn the chemi-ionization process is affected. Under the atmospheric CP chamber condition without a span gas, the ambient air flows into the flame chamber during the intake process and the unburned mixture during the compression process. For example, the output behavior of case (b) in Fig. 3 could be explained as follows: the jump is attributed by the arrival of the unburned mixture into the flame chamber and the gradual increase of the output results from the gradual increase of the gas temperature in the flame chamber resulting from the burning of the unburned mixture.

The author proposes two ideas to improve the transient period problem. One is to measure HC concentrations under motoring as illustrated in case (c) of Fig. 3. In that case, the FID output reaches the calibrated output (case (c)) right before its drop. Since the transient period is longer than the compression period, the idea is applicable only for non-firing conditions. In this study, in-cylinder HC measurements were made based on that idea. The other idea is to keep the CP chamber pressure lower than the cylinder pressure throughout the engine cycle so that the unburned mixture can be drawn into the FID chamber from the beginning of the intake process to secure enough time to overcome the transient period. However, this idea is challenged by the possibility of pressurizing the CP chamber to be explained in the following section.

4.1.3 Pressurizing of the CP chamber under sub-atmospheric CP conditions

Under sub-atmospheric CP chamber operation, a critical issue is how to minimize the pressure fluctuations in the CP chamber. Another challenge is to avoid choking by keeping the CP chamber pressure higher than the critical value illustrated in Fig. 2. In that case the CP chamber volume is finite whereas it is infinite under the atmospheric operation. Thus, the pressure inside the CP chamber fluctuates inevitably and its magnitude depends on instantaneous sample gas flow rate and the size of the CP chamber volume. According to Summers (Summers, 1996), the magnitude of fluctuations is more sensitive to the diameter of the transfer tube than to the CP chamber volume. The minimum diameter applicable to the in-cylinder measurements is about 0. 2~0. 25 mm considering the limitations such as tube clogging by carbon deposits and slow FID response due to the reduced flow rates.

This method is most frequently used for the in-cylinder HC measurements (Collings, 1988; Galliot, Cheng, Cheng, Sztenderowicz and Heywood, 1990; Brown and Ladommatos 1991; Rose, Ladommatos, and Stone, 1994; Summers and Collings, 1995). However, it still leaves a question about its accuracy and repeatability in spite of extensive research for the reliable in-cylinder HC measurements because it is not completely free from the problems related to the pressure interference. Thus there have been few reports on its application to real engines. One meaningful example is the measurement of internal EGR (exhaust gas recirculation) reported by Cheng et al. (Galliot, Cheng, Cheng, Sztenderowicz and Heywood, 1990). For better accuracy, the effect of the burned gas on the transient period should be investigated further.

4.2 Condensation

When sampled from the cylinder of a gasoline fueled engine, the sample will condense unless the sample line is very short, or it is heated. For example, at the room temperature and pressure, in a stoichiometric gasoline mixture, only about 50% of the fuel is vaporized. At the elevated pressures the vapor fraction reduces. A full (non-isothermal) analysis shows that, when the flow is near sonic at the exit, rapid expansions cause deviations between the gas sample and tube wall of the order of 20-30K.

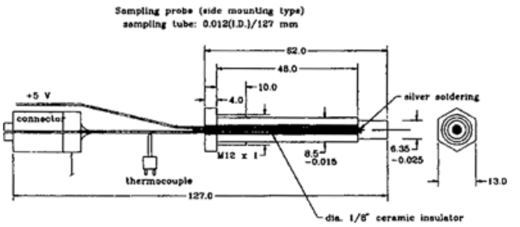


Fig. 4 Schematic of the electrically heated in-cylinder FID sampling probe.

In fact, for in-cylinder sampling, the condensation is more likely to occur in the inlet of the transfer tube, as the pressures there are high, and consequently the dew point of an unburned gasoline air mixture is elevated. To ensure that the sample does not condense either at the exit due to this expansion, or merely due to the main tube section being below the dew point of the mixture, heating is essential.

To avoid condensation, a sampling tube was electrically heated. Figure 4 shows the schematic of the heated tube. The inner diameter and length of the tube were 0. 3048 mm and 127 mm, respectively. Since the resistance of the tube was ~1.0 ohm, 5 volt DC power was applied to the both ends of the tube for electrical heating. The ground connection at the tube tip was isolated from the engine body to avoid heat loss to the body whose temperature was much lower than the control temperature. For that isolation, the tube and electrical wire were inserted into a two-hole ceramic tube (originally used for thermocouple insertion), and silver-soldered at the end. A thermocouple was silver-soldered on the tube wall to monitor the wall temperature. The whole assembly was flush-mounted on the side wall of the combustion chamber.

5. Test Condition

5.1 FID operating condition

The pressure in the CP chamber was set to atmospheric pressure. Thus, the in-cylinder measurement is valid only for engine motoring. The pressure difference between the CP chamber and the FID chamber was kept constant as 6 kPa.

5.2 Engine operating condition

A production Volvo engine (B5254 FS) was modified to fit in the Ricardo Hydra single cylinder engine mount. The engine has two intake and two exhaust valves. Its displacement volume and compression ratio are 487.0 cm3 and 10.1, respectively. Air flow rate into the engine was measured with a laminar flowmeter. The engine was motored at 900 rpm with a DC dynamometer. The equivalence ratio was calculated from the measured air flow rate and the pulse width of the injected fuel. The intake manifold pressure was set to 0. 527 bar. Indolene was used as a test fuel. Indolene is the standard fuel used for vehicle exhaust emissions certification.

5.3 Calibration of the fast FID

Although the fast FID is known to show good linearity, its output characteristic was investigated using the span gases with different HC concentrations for accurate calibration. The span gases consisting of air and propane were prepared with precision flowmeters. The accuracy of the flowmeter (Tylan Co. FC-260, FC-261) is within +/−0.1% of full scale.

Air and propane flowing through separate tubes were metered at the same time by each flowmeter. After that, the gases were merged in a single tube to form the desired span gas. The span gas was bled through the CP chamber for FID readout. The calibration result is plotted in Fig. 5. The nonlinear calibration data were curve-fitted in a second order polynomial for translation of FID output into the equivalence ratio which was proportional to HC concentration. The conversion formula from a HC concentration to the equivalence ratio of fuel C_nH_m is derived in Appendix.

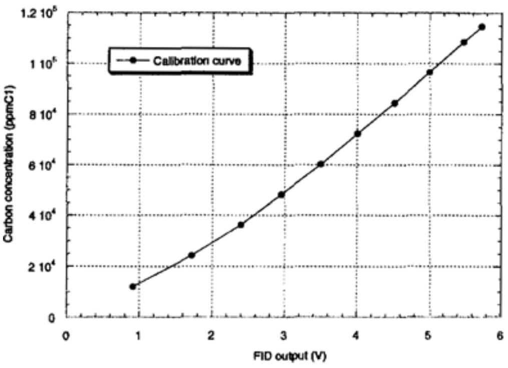
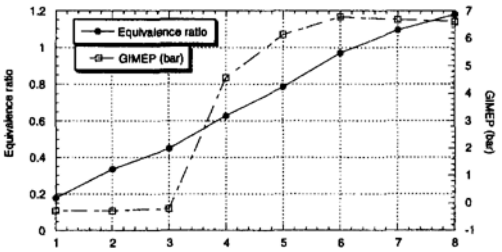


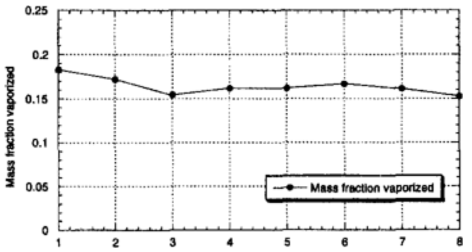
Fig. 5 Calibration of FID output with span gases.

6. Test Results

Figure 6 shows a comparison of the measured in-cylinder equivalence ratio and the corresponding GIMEP (Gross Mean Effective Pressure) with respect to the injected equivalence ratio in the first cycle. 1 unit injection corresponds to the equivalence ratio of 0.97. The corresponding GIMEP was measured in the second engine run with ignition since the current FID sampling method can not measure the correct equivalence ratio under engine firing as discussed previously. To keep good repeatability between the two sets of engine runs, coolant temperature and hydrocarbons concentrations measured by the fast FID were monitored. The engine was motored until it was restored to the same engine operating condi-



(a) Injection unit



(b) Injection unit

Fig. 6 (a) in-cylinder equivalence ratio and corresponding GIMEP and (b) mass fraction vaporized from the injected fuel for the test condition of 900 rpm, 0. 527 bar intake pressure, indolene, ambient temperature of 21 C .

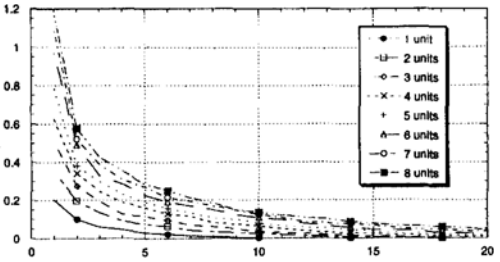
tion before the fuel injection in the first cycle. By experience, it took about less than 5 minutes and by then the hydrocarbons concentrations in the engine cylinder was within the lower detectable limit of the fast FID and the coolant temperature returned to the temperature before the fuel injection in the first cycle. To take into account cycle-to-cycle variations of the in-cylinder equivalence ratio and GIMEP, the measurements as shown in Fig. 6 were conducted several times. The test data were found to behave in repeatability good enough to be represented by the data shown in Fig. 6.

It is observed that the measured equivalence ratio is approximately proportional to the amount of fuel injected. In other words, the mass fraction vaporized is nearly constant as shown in Fig. 6 (b) where the mass fraction vaporized is about 15 to 20%. It means that fuel evaporation is not restricted at least in the first cycle either by the available fuel evaporation area or by the amount of air mass to interact with the fuel. Here the available fuel evaporation area means the area on the port wall where injected fuel spreads and convective fuel evaporation occurs. Thus, if it were not the first cycle since fuel injection, the port wall would be already wet by fuel injection from previous cycles and hence the available fuel evaporation area for the current fuel injection would be significantly limited. For construction of a data base, the test needs to be conducted in terms of intake manifold pressure, intake port temperature and engine speed.

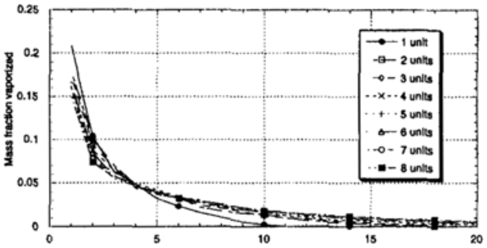
Another point to note is the quantification of the equivalence ratio in vapor phase during starting, considering the importance of the role of engine starting in view of its contribution to overall HC emissions. In conjunction with the in-cylinder equivalence measurement, the in-cylinder pressure was measured at the same time. The cylinder pressure data was converted into GIMEP (Gross Mean Effective Pressure) to infer mixture strength. Figure. 6(a) shows a good correlation between the measured in-cylinder equivalence ratio and GIMEP. According to the comparison, a firing occurred at about 0.62. Therefore, the threshold equivalence ratio for firing could be about 0.6. And for secure firing in the first cycle, the in-cylinder equivalence ratio should be at least 0. 8∼1. 0 corresponding to the injected equivalence ratio of 5 to 6. Since this test was performed only at one operating condition (900 rpm, 0. 527 bar intake pressure, and 21°C of ambient temperature), similar tests should be conducted over an extensive test matrix for commercial engine development.

Another point to investigate is the influence of previous cycles on the present cycle in terms of mixture preparation. Fuel evaporation occurs not only from the fuel injected in the present cycle, but also from the liquid film accumulated from previous cycles. Since the mixture preparation process is very complicated and essentially nonlinear to any parameters involved, it is almost impossible to predict the mixture preparation behavior using any simple theoretical model. Thus, the measurement of the equivalence ratio in consecutive cycles would be an important data base to develop a mixture preparation model. The consecutive measurement was made under two different fuel injection schedules. One is a single pulse injection in the first cycle with varying pulse widths from 1 unit to 8 units. The other is continuous injection with 1 unit pulse width.

The case of single pulse injection is shown in Fig. 7. Figure. 7(a) shows quantitatively how the amount of the fuel injected in the first cycle affects the mixture preparation process in the subsequent cycles. It is clearly seen that the subsequent cycles are affected significantly at least over 5 to 10 cycles. The same data was plotted in Fig. 7(b) in terms of the mass fraction normalized by the amount of fuel injected in the first cycle. Although they didn't collapsed into a single curve, they were quite close to each other except for the single unit injection case. It means that the mixture preparation process is not random, but behaves in fashion with respect to the injected fuel amount. Therefore it is worthwhile to think about how engine operation is related with the behavior of the mixture preparation observed in Fig. 7. An explanation of this observation is that with a larger quantity of fuel injected, there is a larger amount of liquid blown away from the intake



(a) Cycle number



(b) Cycle number

Fig. 7 (a) Equivalence ratio (b) mass fraction vaporized in the subsequent cycles after the single shot fuel injection in the first cycle.

valve vicinity during the reverse blow-down period. Because this blown-back liquid is situated further from the intake valve, the film flow component of this fuel would take a longer period to re-enter the engine. The process is a complex regenerative one, but the test results show that it is quite consistent with respect to the injected fuel amount except for the single unit injection case in which the fuel amount deposited in the intake port is too small to be blown back by the reverse blow-down flow.

As the continuation of the above consideration, a functional relationship between the injection unit and the equivalence ratio ϕ will be investigated. The measured ϕ for the continuous injection is shown in Fig. 8. It is observed that ϕ quickly builds up to its steady state value in about 10 cycles. It is attributed to the presence of the liquid fuel left over from the previous cycles. Therefore, it is important to investigate the effect of previous cycles quantitatively. One simple idea is to superimpose the response of ϕ to a single pulse injection. It would be that, for the continuous injection, the ϕ value of the current cycle is due to the contribution of the current injection, which is the first-cycle-response to the impulse excitation, plus the sum of the contributions from the previous cycles. The contribution due to the Mth previous cycle would be equal to the "unit-impulse response" delayed by M cycles. A comparison is shown in Fig. 8. The model shows relatively good agreement with the measured ϕ while the measured ϕ builds up slower than the modeled ϕ . In other words, the "unit-impulse response" model works and the transfer function represented by the mixture preparation process is linear with respect to fuel input at least for the first 20 cycle. The slower increase of the measured ϕ indicates that there exists "another" injection which tends to hinder the fuel transport into the engine.

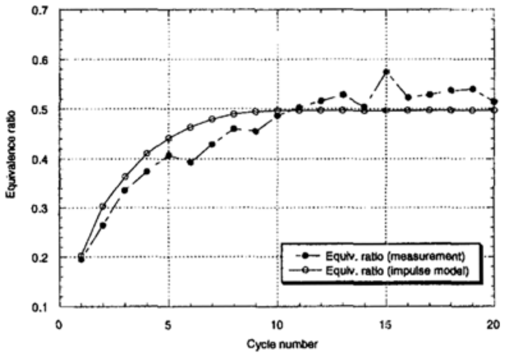


Fig. 8 Comparison of measured and modeled ϕ for the continuous injection case with 1 unit injection pulse width.

A likely explanation for the slower increase lies in the backward fuel transport process during the reverse blow down. The video observation (Shin, Min and Cheng, 1995) demonstrates that this is a significant process. The amount of the transport back, however, depends on the thickness of the fuel film present in the vicinity of the valve. In the single injection case by which the impulse function was generated, the fuel left in the port is not replenished by new injections. Therefore the blow-back effect in the subsequent cycles after injection is substantially smaller than the case of continuous injection. The result is that the "tail part" of the impulse function is higher and shorter (since the total fuel amount is conserved), and thus the model based on the impulse function

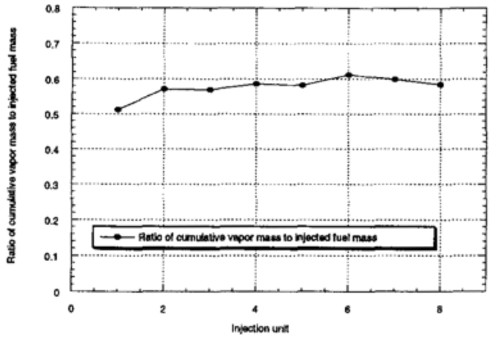


Fig. 9 The ratio of cumulative vapor mass to the mass of the injected fuel over the first 20 cycles.

overestimates the in-cylinder ϕ value in the continuous injection case.

Another feature noticeable from the in-cylinder measurement is the difference between the amount of the injected fuel and the cumulative sum of the measured vapor mass. Figure 9 shows the difference in terms of mass fraction. The ratio of mass fraction lies between 0. 5 and 0. 6. It means that only 50 to 60% of the injected fuel participates in the mixture preparation process and the rest of the fuel is unaccounted for. The path of the fuel unaccounted for and its quantification has been one of the important subjects to understand the mixture preparation process. In Fig. 9, except for the case of one unit injection, the others show about 60 % of fuel evaporation regardless of the amount of the injected fuel. When it comes to the one unit injection case, the discrepancy seems to be caused by the factors such as small amount of fuel injection, negligible blow back effect, and limited accuracy of the in-cylinder fast FID system. The fact that about 60% of the injected fuel evaporates during the first 20 cycles suggests that the fuel composition is responsible for the behavior. Heavy fuel components do not evaporate during the period and they might flow into the oil sump through the piston ring pack. That mechanism has been clearly visualized in several transparent engine tests (Shin, Min and Cheng, 1995; Saito, Sekiguchi, Imatake, Takeda and Yaegashi, 1995). Here the question was how much fuel ended up in the oil sump. The test result shown in Fig. 9 concludes that about 40% of the injected fuel would stay either within the engine cylinder or in the oil sump. The presence of large amount of liquid fuel in the engine cylinder during engine starting was clearly evidenced by Shin et al.'s visualization test results (Shin, Min and Cheng, 1995). The liquid fuel will evaporate later when engine firing starts and the coolant temperature increases.

7. Conclusions

The in-cylinder equivalence ratio during engine starting was measured using a fast FID system. Since the in-cylinder sampling system still needs careful operation and calibration, the current state-of-art of the system has been reviewed. And the in-cylinder equivalence ratio was measured under the engine test condition of 900 rpm, 0.527 bar intake pressure, and the ambient temperature of 21 °C. The following conclusions were obtained from the review and the test results.

(1) there are two types of sampling methods in terms of CP chamber pressure. The method with atmospheric CP chamber pressure is good for in-situ calibration and pressure isolation. However it is useful only for engine starting before firing since the FID output stabilization period (transient period) overlaps with combustion events. On the other hand the method with sub-atmospheric CP chamber pressure overcomes the long stabilization period, but it is still subject to the unknown intrinsic problem of the sampling system which stems from the alternating supply of unburned mixture and burned gas into the FID within one engine cycle. The problem results in poor repeatabilty of the measurements.

(2) The equivalence ratio in the engine cylinder was measured under the condition of single pulse injection in the first cycle. The result shows that about 15 to 20% of the injected fuel evaporates in the first cycle and then the equivalence ratio decays gradually, hinting that the evaporation is mainly governed by the thermal environment to which the engine is exposed.

(3) The comparison of the measured ϕ and the GIMEP measured at the same test condition reveals that a significant firing occurs at the

equivalence ratio of about 0. 6. And to build up GIMEP close to a steady state value in the first cycle, about 5 to 6 times stoichiometric fuel amount should be injected.

(4) The measurement result of the in-cylinder equivalence ratio shows that about 60% of the injected fuel evaporates during engine starting. The rest of the fuel which is unaccounted for mainly consists of heavy fuel components and stays either inside the engine cylinder or in the oil sump.

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APPENDIX

The relation to convert FID output into the equivalence ratio of a test fuel, C_nH_m, is as follows:

According to Eq. (4), the FID output, VFID, should be linear to [HC], but a real calibration result illustrated in Fig. 5. shows that the calibration curve deviates a little bit from an ideal linearity line. Therefore, for better accuracy the curve can be fitted as an second order polynomial:

$$X_{c1}=\mathrm{m}[HC]=\mathrm{fn}(V_{FID}) \tag{A1}$$

However,

From the definition of equivalence ratio,

$$\phi=\frac{F/A}{(F/A)_{st}}=\frac{1}{(F/A)_{st}}\frac{N_{CmHn}W_{CmHn}}{N_{air}W_{air}} \tag{A2}$$

The residual mass fraction is

$$x=\frac{W_bN_b}{W_{air}N_{air}+W_{CmHn}N_{CmHn}+W_bN_b} \tag{A3}$$

From the definition of [HC],

$$[HC]=\frac{N_{CmHn}}{N_{CmHn}+N_{air}+N_b} \tag{A4}$$

From Eq. (A3) and (A4),

$$\frac{N_{CmHn}}{N_{air}}=\frac{\left(1+\frac{x}{1-x}\frac{W_{air}}{W_b}\right)[HC]}{1-\left(1+\frac{x}{1-x}\frac{W_{CmHn}}{W_b}\right)[HC]} \tag{A5}$$

From (A5) and (A2)

$$\phi=\frac{1}{(F/A)_{st}}\frac{W_{CmHn}}{W_{air}}\frac{\left(1+\frac{x}{1-x}\frac{W_{air}}{W_b}\right)[HC]}{1-\left(1+\frac{x}{1-x}\frac{W_{CmHn}}{W_b}\right)[HC]} \tag{A6}$$

From (A1) and (A6)

$$\phi=\frac{1}{(F/A)_{st}}\frac{W_{CmHn}}{W_{air}}\frac{\left(1+\frac{x}{1-x}\frac{W_{air}}{W_b}\right)\frac{fn(V_{FID})}{m}}{1-\left(1+\frac{x}{1-x}\frac{W_{CmHn}}{W_b}\right)\frac{fn(V_{FID})}{m}} \tag{A7}$$

where

X_{C1} : concentration of carbons in the mixture
m : number of carbon in the fuel molecule CmHn
ϕ : equivalence ratio
F/A : mass ratio of fuel to air
$(F/A)_{st}$: stoichiometric mass ratio of fuel to air
x : residual gas fraction
N : number of moles
W : molecular weight
b : exhaust gas species
$[HC]$: molar ratio of CmHn in the mixture