Scavenging Characteristics and Supercharging Effects* of Two-stroke Hydrogen Injection Engine

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In order to improve the performance of a two-stroke hydrogen injection engine, it is necessary to clarify the scavenging characteristics with the supercharging taken into account. In the present study, the gas in the cylinder before combustion was sampled and analyzed to obtain the contents of H_2 and O_2 molecules. On the other hand, by using the chemical reaction theory, a theoretical equation with H_2 and O_2 contents as variables was derived to obtain the scavenging efficiency and the trapping efficiency. Applying this theory to the experimental results with a practical operating hydrogen fueled engine, the scavenging characteristics have been evaluated.

Also, the effects of mechanical supercharging and turbocharging on the two-stroke engines were investigated both theoretically and experimentally. As a result, it was found that supercharging was practically effective for increasing the output power.

1. Introduction

A two-stroke hydrogen injection and spark ignition engine had been developed, in which high pressure (1.0 MPa) cold hydrogen was injected into the cylinder during the first half of the compression stroke (1)(2). This engine completely eliminated abnormal combustions such as backfire or pre-ignition, and gave a higher output (20 to 25%) than the gasoline engine. It also reduced the NO_x emission because of an internal EGR and the cold hydrogen fuel. For the purpose of the improvement of the engine performance, the scavenging characteristics and the effect of supercharge on the two-stroke hydrogen injection and spark ignition engine have been investigated. In this connection, the gas in the cylinder just before combustion was sampled using an electromagnetic sampling valve, then the contents of O_2 and H_2 in the mixture were analyzed, from which the scavenging characteristics were obtained.

2. Notations

- A_1 : Mole number of fresh air supplied to the cylinder in one cycle
- A_2 : Mole number of fresh air blown off through the exhaust port, of A_1
- A : Mole number of fresh air remaining in the cylinder, $A = A_1 - A_2$ as shown in Fig. 1
- A_t : Mole number of residual air plus A
- H_2 : Mole number of hydrogen injected into the cylinder
- L_0 : Theoretical mole air-hydrogen ratio 2.38
- λ : Excess air ratio determined by A and H_2
- λ_a : Apparent excess air ratio $= A_1/L_0H_2$

- λ_r : Real excess air ratio based on A_t and H_2
 $= A_t/L_0H_2 = \lambda A_t/A$
 - λ_{O_2} : Excess air ratio calculated from analysis of oxygen content
 - λ_{H_2} : Excess air ratio calculated from analysis of hydrogen content
 - V_0 : Mole number of total gas involved in combustion $= V_f + V_R$
 - V_f : Mole number of fresh mixture gas in the cylinder $= A + H_2$
 - V_a : Mole number of the mixture after combustion of V_f
 - V_R : Mole number of residual gas in the cylinder $= rV_a$
 - r : Ratio of residual gas to the new combustion gas
 - r_m : $V_R/(V_R + V_f)$
 - r_b : $V_R/(V_R + A)$
 - η_s : Scavenging efficiency $= A/(A + V_R) = 1 - r_b$
 - η_t : Trapping efficiency $= A/A_1 = \lambda/\lambda_a$
 - η_v : Volumetric efficiency $= \lambda_0\eta_t$
- The notation in Chapter 4 is on a weight of basis instead of a mole number basis.

Fig. 1 Model of constituents of gas mixture in the cylinder before combustion

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3. Scavenging Characteristics of the Two-stroke Hydrogen Injection Engine

- (1) Method of predicting the scavenging characteristics from the constituent parts of gas in the cylinder just before combustion

The mole number of the fresh mixture can be expressed as follows ;

$$V_f = H_2 \left(1 + \frac{0.21\lambda L_0}{O_2} + \frac{0.79\lambda L_0}{N_2} \right) \quad (1)$$

After the combustion with sufficient air so that the hydrogen is perfectly burnt and it is absent in the residual gas, we have

$$V_a = H_2 \left[\frac{0.21(\lambda - 1)L_0}{O_2} + \frac{0.79\lambda L_0}{N_2} + 1 \right] \quad (2)$$

Since the constituent of the new combustion gas can be considered the same component of the residual gas, V_R can be considered equal to rV_a , and the following equation can be introduced ;

$$V_0 = V_f + rV_a \quad (3)$$

In this case, H_2O in the combustion gas was ignored because of its easy condensation during analysis. Therefore, the total mole number of the dry gas V_d just before the combustion will be expressed ;

$$V_d = V_0 - H_2O = H_2 \left[\frac{1 + 0.21L_0(\lambda + r\lambda - r)}{O_2} + \frac{0.79\lambda L_0(1 + r)}{N_2} \right] \quad (4)$$

When the molecular contents of oxygen and hydrogen in the sampled dry gas are denoted as (O_2) and (H_2) respectively, the excess air ratio λ_{O_2} is obtained based on an analysis value (O_2) as ;

$$\lambda_{O_2} = \frac{0.5r[1 - (O_2)] + (O_2)}{(1 + r)[0.5 - 2.38(O_2)]} \quad (5)$$

In the same way, λ_{H_2} is obtained based on the (H_2) from the following equation,

$$\lambda_{H_2} = \frac{1 + (0.5r - 1)(H_2)}{2.38(1 + r)(H_2)} \quad (6)$$

Assuming $\lambda = \lambda_{O_2} = \lambda_{H_2}$, where r is a ratio of residual gas to the new combustion gas, the ratio is expressed as ;

$$r = \frac{V_R}{V_a} = \frac{0.21 - 0.21(H_2) - (O_2)}{0.395(H_2)} \quad (7)$$

In addition, ratios of the residual gas to the gas just before the combustion and before H_2 injection are respectively ;

$$r_m = \frac{V_R}{V_R + V_f} = \frac{r}{1 + r + \frac{0.5}{2.38\lambda + 0.5}} \quad (8)$$

$$r_b = \frac{V_R}{V_R + A} = \frac{1}{1 + \frac{\lambda}{r(\lambda + 0.21)}} \quad (9)$$

Assuming that the air in the residual gas also participates in the combustion, the ratio of A_t to A can be obtained as follows;

$$\frac{A_t}{A} = \frac{\lambda L_0 H_2 + r\lambda L_0 H_2 - rL_0 H_2}{\lambda L_0 H_2}$$

Finally, the real excess air ratio λ_r , representing the ratio of total real air A_t to the stoichiometric air corresponding to injected hydrogen, is;

$$\lambda_r = \lambda \frac{A_t}{A} = \lambda \left[1 + \frac{r(\lambda - 1)}{\lambda} \right] \quad (10)$$

(2) Experimental apparatus

As mentioned in the foregoing, the analysis of the contents of oxygen and hydrogen of gas in the cylinder before combustion gives the excess air ratio, rate of residual gas and trapping efficiency, which are the most important factors for the performance of two-stroke engines. Taking account of the intake throttle condition, the scavenging characteristics of the conventional engine can also be obtained by the same method with hydrogen fuel. This is valid, however, only under the following conditions;

- The combustion is perfect, leaving no H_2 in the burnt gas;
- When the sampled gas represents the mean value of the gases in the cylinder; and

Fig. 2 Structure of electromagnetic sampling valve

Fig. 3 SVT relating to engine operation in a sampling cycle

- c. The sampling is done after a sufficiently long time from the finish of fuel injection to allow the uniform mixing of gas for this purpose.

The sampling valve used in the experiment was an electromagnetic type with 14 mm inside diameter and 3 mm valve lift, as shown in Fig. 2. It was water cooled to prevent preignition, and a condenser pick-up was so used that the valve lift could be measured. Fig. 3 shows the SVT (sampling valve timing) during a gas sampling cycle. SVT was controlled by the SVT system consisting of a photo-transistor type crank angle detector, a controller and an electromagnet as shown in Fig. 4. As shown in this figure, the sampled gas was led to a 150 cm³ tank first, then sent to both the gas chromatograph and the Orsat gas analyzer to measure the concentrations of oxygen and hydrogen. Once every 40 cycles, ignition was suspended and the sampling valve was opened from the TDC to 90° ATDC (3).

Fig. 5 shows a cross sectional view of the two-stroke hydrogen injection engine used in the experiment. The engine was originally a Schnürle scavenging type two-stroke engine manufactured by Suzuki Motor Co., LTD, with 76 mm cylinder bore, 80 mm stroke, three cylinders, water-cooled and 1090 cm³ total piston displacement. The cylinder head was modified to use the hydrogen injection engine, as shown in Fig. 5, and the effective compression ratio ϵ_e was increased from 5.5 : 1 to 6.8 : 1. Further, a simple air throttle valve only was installed instead of the carburetor in the intake manifold. The ignition timing was delayed, and the ignition plugs were replaced with those of a high heat range. As rocker arms were installed on the eccentric rocker cam, the hydrogen quantity could be controlled by slightly turning the rocker arm shaft with the cam by the accelerator wire and by varying the injection valve lift and its duration. The injection pressure was kept constant at 1.0 MPa, while hydrogen was injected for 60°, namely from 105° to 45° BTDC at the engine maximum torque. The temperature T_i of hydrogen at the injection valve inlet was kept at a valve between -30° and -50°C by cooling with liquid nitrogen. The flow rates of both hydrogen and air were measured respectively with laminar flow meters, while the brake output

Fig. 4 System diagram of gas analysis

was measured with an electric eddy current dynamometer.

(3) Experimental results

If a whole gas could be drawn out from the cylinder in the period from 45° BTDC to 108° ATDC in Fig. 3, it would be ideal but this is practically impossible. In this sampling device, the optimum SVT was about TDC open and 90° ATDC close. Then it was made sure what amount of sampling gas during one cycle was enough to represent the mean value of gas concentrations in the cylinder just before combustion. For this purpose, an experiment was carried out under the same λ_a , varying the engine speed and changing the sampling position by varying the protruded length l of the valve in Fig. 2. As shown in Fig. 6, it was found that one-third of the total cylinder gas at 2000 rpm was the enough quantity of the sampling gas. At a higher speed such as 4000 rpm, however, it was not enough because the sampling gas quantity became only one-fourth. On the other hand, Fig. 7 shows the experimental results about hydrogen concentration in the exhaust gas. This result indicates that a complete combustion actually takes place under a leaner mixture than $\lambda_a = 1.4$. So the above mentioned equations are valid for an extent of $\lambda_a > 1.4$.

Fig. 8 through Fig. 12 show some scavenging characteristics against the delivery ratio λ_0 which means an amount of scavenging air, that is defined as ;

Fig. 5 Cross section of two-stroke hydrogen injection spark ignition engine

Fig. 6 Measured results of λ when valve length in the combustion chamber was varied

$$\lambda_0 = \frac{\text{Fresh air supplied to the}}{\text{Weight of air with density } \rho_0 \text{ cylinder } (G_1) \text{ filled up } V_s}$$

where, ρ_0 is density of atmospheric air and V_s is a piston displacement volume. These data have been obtained by substituting the measured (H_2) and (O_2) into the questions (7) through (10) under the valid engine operating conditions mentioned above. Fig. 8 shows that more than 40% (at $\lambda_0 = 0.7$) of the total gas is residual gas, and it amounts to about 70% at $\lambda_0 = 0.25$. In Fig. 9, λ , the ratio $A (= A_1 - A_2)$ to L_0H_2 is smaller than λ_a , the ratio of A_1 to L_0H_2 . Where λ means an excess air ratio in which air in the residual gas is not included. On the other hand, air of λ_r includes the residual air, and it becomes much larger than λ_a at a small λ_0 because of a large residual gas as shown in Fig. 8. When λ is 1.0, λ_r must be equal to λ . The constituent parts of the gas just before the combustion are shown in Fig. 10. In a richer H_2 mixture (smaller λ_a value), the ratio of H_2 increases, and then the fresh air and residual gas contents decrease. However the volumetric efficiency $\eta_v = \lambda_0 A / A_1$ is independent of λ_a because H_2 is

Fig. 7 H_2 concentration in exhaust gas Fig. 8 Values of r vs. delivery ratio λ_0

injected after the closure of the scavenging and exhaust ports. And Fig. 12 shows the characteristics of the scavenging efficiency η_s and the trapping efficiency η_t . η_s is slightly higher than that of the perfect mixing process which is assumed to mix perfectly and instantaneously the fresh air with the burnt gas and these gases have a same density. This is a typical property given by the Schnürle type scavenging. Meanwhile, η_t decreases with an increase of λ_0 because the blow-off rate of the fresh air increases. But, its drop is smaller than the increment of η_s , so the amount of the effective air will increase.

4. Investigation of Supercharging Effect in the Two-stroke Hydrogen Injection Engine

(1) Theoretical calculation of scavenging air quantity in static cycle

a. In case of natural aspiration

P_2 , V_2 and T_2 denote the pressure, volume and temperature in the crankcase respectively at the time of intake port closure by the piston lower edge (2_L) during the piston descent. The air quantity G_2 at that time will be equal to $P_2 V_2 / RT_2$. This air is compressed from the position 2 to

Fig. 9 Values of λ and λ_r vs. delivery ratio λ_0

Fig. 10 Constituent parts of gas just before combustion

position 4, at which the state of G_2 becomes P_4 , V_4 and T_4 . Just after position 4 the scavenging port is opened by the piston upper edge (4_u), then the compressed air in the crankcase is sent into the cylinder until the crankcase pressure becomes the cylinder pressure which equals to the exhaust manifold pressure. In the static case, this pressure equals to the atmospheric pressure P_0 . After that, the piston goes down once to the position 5 or BDC, then goes up and closes the scavenging port again at the position 4. The fresh air quantity G_r remaining in the crankcase at that time is expressed as $G_r = P_0 V_4 / RT_4$. Therefore, the air quantity G_0 sent into the cylinder is ;

$$G_0 = G_2 - G_r$$

$$= \frac{1}{R} \left(\frac{P_2 V_2}{T_2} - \frac{P_0 V_4}{T_4} \right) \quad (11)$$

It is assumed that the compression process in the crankcase is an isothermal change because of narrow space in it, so T_4 equals to the constant air temperature T_c in the crankcase. And $P_2 = P_0$ if the engine is operated under the static aspiration. Then equation (11) becomes;

$$G_0 = \frac{P_0}{RT_c} (V_2 - V_4) = \frac{P_0 V_e}{RT_c} \quad (12)$$

Fig. 11 Volumetric efficiency η_v vs. delivery ratio λ_0

Fig. 12 Scavenging efficiency η_s and trapping efficiency η_t

b. Mechanical supercharging

In case of the mechanical supercharging, the air enters the crankcase with the same pressure as a compressor outlet pressure $P_s = P_0 + \Delta P_s$, while the exhaust gas is discharged into the atmosphere with the pressure P_0 . Through the pressure in the cylinder is not increased at that time, an increase in the brake output can be expected because of increases in λ_0 and in the concentration of the fresh air. The scavenging air quantity G_m per cycle at that time can be expressed as follows ;

$$G_m = \frac{1}{RT_c} (P_s V_2 - P_0 V_4) \quad (13)$$

c. Turbocharging

Assuming that the supercharging pressure P_s is equal to the exhaust pressure or turbine

Fig. 13 P-V diagram in scavenging process

Fig. 14 Amount of calculated scavenging air in a static simple cycle

inlet pressure P_e , the scavenging quantity G_t can be expressed as follows ;

$$G_t = \frac{P_s}{RT_c} (V_2 - V_4) \quad (14)$$

Fig. 14 shows the scavenging air quantity for a static simple cycle, which is calculated by substituting the values of the tested engine, $V_2 = 779.3 \text{ cm}^3$, $V_4 = 580.9 \text{ cm}^3$, number of cylinders = 3 and $T_c = 350^\circ\text{K}$ into the above mentioned equations. Under the same supercharging pressure ΔP_s , a much larger amount of scavenging air enters the cylinder under mechanical supercharging than under turbocharging. At the same time, however, in the former case a large amount of the scavenging air flows out because of no back pressure for the turbocharger.

(2) Increase in effective air under supercharging

The effective fresh air weight G_c corresponds to A (mole number) in the previous chapter, which can be obtained as follows ; Scavenging air quantity G_1 corresponds to G_0 , G_m and G_t in the previous chapter and it is indicated as

$$G_1 = V_s \rho_0 l_0$$

In this engine, the scavenging process can be assumed to be a perfect mixing process as shown in Fig. 12. Then the effective air quantity G_c is indicated as

$$G_c = V_s \rho_0 (1 - e^{-l_0}) \quad (15)$$

The output power N from three cylinders is

$$N = \frac{n}{60} \frac{\eta G_c H_u}{\lambda L_0} \times 3 \quad (\text{kW})$$

Fig. 15 Calculated power consumption for mechanical supercharging

Fig. 16 Experimental system diagram in supercharging

Where n is engine speed in rpm, η is thermal efficiency and H_u is a lower heat value of the fuel.

(3) Driving power for mechanical supercharger

In the turbocharging system, the pressure in the exhaust manifold is higher than in the mechanical supercharging. However it has the advantage that the exhaust energy can be utilized as a turbocharger driving power. On the other hand, the compressor in mechanical supercharging has to be driven by the engine output. In this experiment, a compressor separated from the engine was used. So, the compressor driving power was estimated as follows.

The work required to drive the compressor per cycle W_c is expressed as,

$$W_c = \frac{G_m}{\eta_c} \left[RT_0 \frac{\kappa}{\kappa - 1} \left\{ \left(\frac{P_s}{P_0} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right\} \right] \quad (16)$$

where $\kappa = C_p/C_v$: ratio of specific heats and η_c is compressor adiabatic efficiency. Then the power N_c is

$$N_c = \frac{3W_c n}{60} \quad (\text{kW}) \quad (17)$$

Fig. 15 shows N_c for the values of η_c 0.6, 0.7 and 0.8. Then the effective engine output N_e mechanical supercharger is given by

$$N_e = N_b - N_c \quad (18)$$

Where N_b is the engine output excluding N_c , and η_c in equation (16) is assumed to be 0.7.

(4) Experiments

In an experimental system as shown in Fig. 16, the compressed air for supercharging was supplied from the compressor independently of the engine, and the air flow rate was regulated to a specified supercharging pressure. In a simulating experiment of the turbocharging, the supercharge pressure P_s was made equal to the exhaust pressure P_e by means of the throttle valves of the exhaust pipe and on the intake pipe. On the other hand, no exhaust pressure was applied because the T.V. on the exhaust pipe is fully open, in the case of mechanical supercharging. The temperatures at points ① through ④ in Fig. 16 were measured by a thermocouple, while the pressures at ① through ④ were measured by piezo-type pressure pick-ups. As for the pressure pick-up ③ as the crankcase, the atmospheric pressure line was recorded by a three way cock immediately after the measurement of the crankcase pressure in order to get an absolute value of the pressure. Then the same sampling method described in Chapter 3 was applied to both the mechanical supercharging and turbocharging experiments. And the scavenging characteristics during the supercharging were obtained.

(5) Experimental results

The timely mean air temperatures in the intake system during the supercharged engine operation were firstly measured by the thermocouples. It is found from Fig. 17 that the effects of ΔP_s and λ_a on the air temperature in the crankcase at (b) and (c) are small and these temperatures can be considered constant at 80°C. And in the scavenging port (a), it is about 120°C due to flow-back of the burnt gas.

Next, Fig. 18 shows the measured P-V diagrams in the crankcase. For the mechanical supercharging, the piston position (4) at which the scavenging port is closed during the ascent of piston is shown. Near this position the crankcase pressure P_4 is about the same as the exhaust port pressure $P_e = P_0$, and it is hardly affected by ΔP_s . In the turbocharging, on the other hand, P_4 increases as ΔP_s and becomes $P_s = P_e$. The dotted lines in fig. 19 indicate the scavenging air quantities G_0 per cycle obtained by the following method ;

$$G_0 = G_2 - G_4 = \frac{1}{RT_c} (P_2 V_2 - P_4 V_4) \quad (19)$$

$T_c = 80^\circ\text{C}$, P_2 and P_4 are taken from Fig. 18. The solid lines indicated the air flow rate measured by A.M. at intake pipe. A difference between the dotted and solid lines may be caused by an incomplete isothermal change of T_c , in spite of the assumption of the isothermal change for theoretical treatment. Next, an increase in air flow rate under the mechanical supercharging amounts to 2.2 times the value under the turbocharging at $\Delta P_s = 0.03$ MPa.

To obtain the effective air quantity remaining in the cylinder and to calculate η_s and η_t , the sampling and analysis have been carried out under the supercharging operation. η_s and η_t fall on the extension lines of the non-supercharging operation as shown in Figs. 12 and 20 respectively regardless of the exhaust pressure or the type of the supercharge. And the practical

Fig. 17 Temperature at various points during turbo-charging

scavenging efficiency η_s is also a little higher than that of the perfect mixing scavenging. Fig. 21 shows the effective air quantity, in which the solid line was obtained by A.M. value multiplied by η_t value from the gas analysis, the dotted line was calculated by the equation (15) in which l_0 was also obtained by A.M.. It is found that the difference in effective air quantity between the mechanical supercharge and the turbocharge was smaller than that in Fig. 19. Fig. 22 shows another expression of Fig. 21. That is the increase in output power with ΔP_s in each case of $\lambda = 1.3$ and 1.6 when the amount of injected hydrogen is changed with no charge of air amount. In this figure, solid lines indicate the brake output measured by the dynamometer and compensated by equation (18) in the case of mechanical supercharge. And the dotted lines are obtained from the dotted lines in Fig. 21 on the assumption that the same output is produced as the measured one when the same effective air quantity and the same hydrogen fuel quantity are present in the cylinder.

Lastly, an applicable supercharging pressure in spark ignition engine is restricted by the abnormal combustions. In case of the hydrogen fuel, it is pre-ignition as shown in Fig. 23. However the pre-ignition

Fig. 18 Measured P-V diagram in crankcase

Fig. 19 Air quantity entering crankcase

can be controlled to a certain extent by the cold hydrogen injection up to $\Delta P_s < 0.03$ MPa. This method is not difficult to apply in practice when the hydrogen is pressurized by the liquid hydrogen pump.

5. Conclusions

(1) The trapping efficiency and the scavenging efficiency of a two-stroke hydrogen injection engine are obtained by sampling the gas in the cylinder just before the combustion and then analyzing hydrogen and oxygen in the sample.

(2) Since no fuel blows off during scavenging because the fuel is injected during the compression stroke, the supercharging is useful for engine power-up. Especially the turbocharging system is as effective as the four-stroke engine. Although an increase in supercharging pressure is restricted by the pre-ignition, it can be suppressed by the cold hydrogen injection up to 0.03 MPa in ΔP_s .

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Fig. 20 η_s and η_t under supercharging operation

Fig. 21 Effective air quantity in cases of mechanical- and turbo-charge

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Fig. 22 Effect of ΔP_s on engine output power

Fig. 23 Increase of brake output by turbocharging and restriction by pre-ignition