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Simulation of the Gas Exchange Process in a Small Two-stroke Cycle Engine*

(2nd Report Comparison between Experimental
and Theoretical Treatments)

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This paper deals with a widely applicable simulation of the gas exchange process in a small two-stroke cycle gasoline engine. Theoretical result shown by the 1st report was compared with the experimental data obtained from pressure and gas analysis under various cylinder port heights and engine speeds.

In this simulation, the diffusion factor ϵ_m in the scavenging passage, the mixing factor r in the scavenging passage or the crank case, and the mixing factor ϕ_m and the short-circuit factor ϕ_d in a cylinder, are taken as follows:

For the exhaust lead of $0.026 \sim 0.184$ and the engine speed of $1000 \sim 5000$ rpm, $0 \leq \epsilon_m \leq 0.08$, $0 \leq r \leq 0.35$ (in the case of return-blow), $0 \leq r \leq 0.06$ (in the case of blow-back), $0.75 \leq \phi_m \leq 1$, $0 \leq \phi_d \leq 0.1$.

The optimum values of the scavenging efficiency and the charging efficiency are obtained at the exhaust lead of 0.096 and engine speed of 2800 rpm, $\epsilon_m = 0.01$, $r = 0.01$ (in the case of return-blow), $r = 0$ (in the case of blow-back), $\phi_m = 0.85$, $\phi_d = 0$.

Key Words: Internal Combustion Engine, Two-stroke Cycle Engine, Scavenging, Theoretical Model, Blow-back, Return-blow

1. Introduction

The 1st report⁽¹⁾ stated that the application of diffusion, mixing and other factors to the simulation of the gas exchange process in a small two-stroke cycle engine made it possible to understand the engine's scavenging performance very closely to the actual one, and that a three dimensional indication of gas components in the scavenging passage satisfactorily clarified the usefulness of such a simulation. Yet the simulation, it may well be said, is essentially one of limited nature because it is based on the assumption that the engine speed remains unchanged and the cylinder is of one single type. This report, as an attempt to demonstrate the usefulness of the simulation, compares the data obtained in 1st report with experimental results such as pressure and efficiency levels under various conditions concerning the height of scavenging and exhaust ports and the engine speed, which have a great influence on the engine's scavenging performance. Scavenging and exhaust ports in this type of engine vary greatly in shape: rectangular, diamond-shaped or round. Most of these ports have a rectangular

section and their cross-sectional area depends on their width and height. The width of this type of port to the circumference of the cylinder, however, cannot be changed significantly as it is subject to some constraints including the cylinder's thermal deformation and the failure of a ring. Experiments, therefore, were carried out by changing mainly the height of the port. From the experimental results⁽²⁾ on models and actual engines, seven typical heights of scavenging and exhaust ports are chosen so that we may be able to satisfactorily look into the blow-back and return-blow phenomena as well as the best scavenging performance. These investigations were carried out in two series of tests: In the first one the scavenging port height was kept constant and the exhaust port height was changed, and in the second vice versa. In the former the effective stroke decreases and the exhaust lead becomes greater with an increase in the height of the exhaust port, and in the latter the effective stroke is kept constant but the exhaust lead becomes smaller with an increase in the height of the scavenging port. The ports employed have an exhaust lead of $0.026 \sim 0.184$ and are essentially intended for the medium or low speed engine as the exhaust lead now commonly applied to the engine of this sort is within the range of $0.07 \sim 0.22$.⁽³⁾

Experiments were carried out under the following conditions: The stroke volume was 160 cc, and the engine speed varied within the range of $1000 \sim 5000$ rpm. The carburetor's throttle was opened fully and

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the air-fuel ratio was fixed at 13 ± 0.04 to attain practically the best performance (mean effective pressure). The inlet and exhaust pipes were made as short as possible, so that they were subject to relatively little dynamic effects⁽⁴⁾ of pulsation and inertia. The inlet and exhaust pipes were 211 and 131 mm long, respectively. The scavenging passage was 70 mm in length.

2. Experimental Equipment and Procedure

A general-purpose air-cooled, Schnürle small two-stroke cycle gasoline engine is used in the experiments. Specifications for this engine are given in Table 1. The engine is provided with seven different types of cylinders whose scavenging and exhaust ports vary in height. Fig. 1 shows a layout of experimental equipment. Fig. 2 shows how a pressure indicator and an electromagnetic valve are fitted and how the opening and closing angles of the electromagnetic valve are provided. A surge tank has a volume about 800 times as large as the stroke volume and is coated with rubber on its three sides to minimize vibration due to a pressure change. The surge tank has a laminar flow meter to measure the quantity of the air on its inlet side and a carburetor fixed on its outlet side. Fuel consumption is determined by means of a volumetric meter.

An exhaust tank has a volume about 200 times as large as the stroke volume, and it is barrel-shaped to quickly obtain a typical sample gas. Gas is taken from the rear of the tank. A semiconductor and strain gauge type pressure indicator is provided to measure the pressure including the mean effective pressure through an engine analyzer. The analyzer receives signals from the pressure indicator and gives the temperature, pressure or indicated mean effective pressure level.

The electromagnetic valve for gas sampling is fitted practically near the pressure indicator. Through the electromagnetic valve, gases before and after scavenging, as well as in the crank case and the scavenging passage, are taken out.

Table 1 Appropriate dimensions of engine

	Type	A	F	G	H	I	J	K
Open and shut angle of port	α_a (°BDC)	61	70	76	82	70	70	70
	α_e (°BDC)	56	56	56	56	66	49	37
	α_i (°TDC)	61	61	61	61	61	61	61
Height of port	σ_a	0.211	0.272	0.325	0.360	0.272	0.272	0.272
	σ_e	0.176	0.176	0.176	0.176	0.246	0.140	0.088
	σ_i	0.316	0.316	0.316	0.316	0.316	0.316	0.316
Bore $\times$ Stroke		Height of port		Suffix				
D $\times$ S: 60 $\times$ 57		$\sigma = h/s$ (h: Height)		a: Exhaust				
Appearant comp. ratio		Width of port		e: Scavenging				
E' : 6.96		$\beta = b/D$		i: Inlet				
Crankcase volume ratio		β_a : 0.159		β_e : 0.271		β_i : 0.239		
$E_k = V_k/V_h = 2.63$								

CO_2 , O_2 and CO taken out of the cylinder are measured by means of the gas chromatograph and Miyabe's gas analyzer. The direct-reading meter is also used to measure CO_2 , O_2 and CO components of an exhaust gas.

Two types of direct-reading meters are used: One employs the non-dispersive type infrared analyzing method and the other the magnetic type analyzing method. The former applies to CO_2 and CO , and the latter to O_2 . Furthermore, an exhaust gas is collected in a gas sampler (bottle containing a 20 % saline solution) and then put into the gas chromatograph and Miyabe's gas analyzer to determine the volumetric concentration of its three components, CO_2 , O_2 and CO .

The A.C. dynamometer is provided to absorb the load on the engine, and both the air-fuel ratio and the engine speed are adjusted to stay at a desired level with the carburetor's throttle fully opened; pressure measurement and gas sampling are also carried out after temperatures of the plug plate have reached a saturation point.

3. Results of Experiment

3.1 Pressure

Fig. 3 shows the measured pressure levels, P_z , P_k , P_a , P_e and P_i , in the cylinder, the crank case, the exhaust pipe, the scavenging passage and the inlet pipe,

Fig. 1 Arrangement of experimental equipment

Fig. 2 Fitting of pressure indicator and electromagnetic valve, and opening and closing angles of electromagnetic valve for gas sampling

with the scavenging and exhaust ports of A-type and the engine speed fixed at 2000 rpm. From this chart, it is clear that soon after the scavenging port opens, the pressure P_k and P_e go up drastically on the scavenging side, and then the cylinder pressures, P_g , and P_e balance. This process is characteristic of the blow-back of a combustion gas. In addition, the level of P_e becomes lower than that of P_g shortly after passing the bottom dead center, which characterizes the return-blow phenomenon - that is to say, if this happens, a fresh gas blown into the cylinder is forcibly returned to the scavenging side. Shown in this diagram are the areas of scavenging, exhaust and inlet ports (f_e , f_a and f_i); it is clear that the blow-back and return-blow phenomena result in a decrease in the effective areas of these ports. It can also be seen that P_g , P_k , P_a and P_e fluctuate with almost the same period shortly after passing the bottom dead center. This means that the exhaust pipe, cylinder, scavenging passage and crank case can internally be regarded as a single pulsation system⁽⁵⁾ and that the pressure levels, P_a and P_i , decrease to a considerable degree and may be lowered almost to the level of atmospheric pressure when the exhaust and inlet ports open in the ensuing cycle. Presumably, the period of blow-back (blow-back angle, α_r°), the period of return-blow (return-blow angle, α_{rs}°), and pressure pulsation of the pipe system vary according to cylinder type, engine speed and other parameters.

3.2 Review of experimental results

Figs. 4 and 5 make a comparison of the delivery ratio, L , the trapping efficiency, η_{tr} , the charging efficiency, η_c , and the brake mean effective pressure, P_{me} , for each type of cylinder and the engine speed in the range from 1000 to 5000 rpm at intervals of 200 rpm. Fig. 4 represents the case where the height of a scavenging port remains constant and that of an exhaust port is subject to change, while Fig. 5 does the opposite case. It can clearly be seen that η_c , as the product of L and η_{tr} , for each type of cylinder is almost proportional to P_{me} . Fig. 4 illustrates the case

Fig. 3 Pressure diagrams and angle areas of gas exchange and inlet process

where the height of a scavenging port remains unchanged and that of an exhaust port is subject to change. In this diagram it is shown that as the height of an exhaust port increases (from A to F, G and H types), the delivery ratio, L , increases while the trapping efficiency, η_{tr} , decreases. This is true despite changes in the engine speed. The delivery ratio, L , goes up and down to a greater degree with A-type cylinder whose exhaust port belongs to the lower category than those of other types. This is because A-type cylinder is affected to a larger extent by return-blow at the low speed and by blow-back at the high speed. The charging efficiency, η_c , is higher at the low speed and lower at the high speed in the

Fig. 4 Relations among delivery ratio, L , trapping efficiency, η_{tr} , charging efficiency, η_c , and brake mean effective pressure, P_{me}

Fig. 5 Relations among delivery ratio, L , trapping efficiency, η_{tr} , charging efficiency, η_c , and brake mean effective pressure, P_{me}

case of a low exhaust port (A-type cylinder). This is reversed with a high exhaust port (H-type cylinder). By contrast, no regularity can be observed for L and η_{tr} in the Fig. 5 case where the height of an exhaust port remains unchanged and that of a scavenging port is subject to change. For example, when a scavenging port falls under the highest bracket (I-type cylinder), L decreases remarkably to a significant degree at the low speed and will not go up so much at the high speed for the height of the scavenging port. In addition, when the scavenging port comes under the lowest category (K-type cylinder), L also drops to a considerable degree at the high speed. Furthermore, η_{tr} varies with the height of a scavenging port; the trapping efficiency stands at a low level when the engine speed is low and the scavenging port belongs to the lowest category (K-type cylinder), and so it does when the engine speed is high and the scavenging port comes under the highest category (I-type cylinder). The charging efficiency, η_c , meanwhile, when the scavenging port is the highest (I-type cylinder), remains at an extremely low level as it is affected greatly by the delivery ratio at the low speed. This is also the case when the scavenging port is the lowest (K-type cylinder) because it is influenced very much by the delivery ratio at the high speed. From these observations it is clear that the efficiency and performance vary greatly according to cylinder type and engine speed. It should be added here that the value of L is determined by the laminar flow meter and that of η_{tr} obtained from the concentration of oxygen in an exhaust gas.

Fig. 6 shows changes in the blow-back angle, α_r , the rate of reduction in effective area due to blow-back, $\Delta A_p/A$, the

Fig. 6 Reductions in angle area of scavenging port and gas concentrations due to blow-back and return-blow

return-blow angle, α_{rs} , the rate of reduction in effective area due to return-blow, $\Delta A_{rp}/A$, the concentration of carbonic acid gas in the scavenging passage, $\delta_e(\text{CO}_2)$, and the concentration of carbonic acid gas in the crank case, $\delta_k(\text{CO}_2)$, which are all based on the results of pressure measurement and gas analysis for each type of cylinder mentioned in Fig. 4. The concentration of a blown-back carbonic acid gas measured before the bottom dead center and that of a return-blown carbonic acid gas measured after the bottom dead center are plotted on the same line and calculated in terms of the stroke volume under atmospheric conditions.

From this diagram it is seen that the value of α_r becomes larger with an increase in the engine speed as well as with a decrease in the height of an exhaust port (from H, to G, F and A types). Thus, as α_r increases the effective area decreases; at the same time, $\Delta A_p/A$ declines with an increase in the engine speed as well as with a decrease in the height of an exhaust port.

The return-blow angle, α_{rs} , decreases with an increase of the engine speed as well as with a decrease of the height of the exhaust port. Accordingly, the effective area increases with a decrease of α_{rs} , while $\Delta A_{rp}/A$ becomes greater with an increase in the engine speed as well as with a decrease in the height of the exhaust port.

As for the concentration of carbonic acid gas, $\delta_e(\text{CO}_2)$ and $\delta_k(\text{CO}_2)$ increase as the height of the exhaust port decreases (from H to G, F and A types). From a view point of the engine speed, $\delta_e(\text{CO}_2)$ remains higher in A-type cylinder whose exhaust

Fig. 7 Scavenging efficiency, η_s , in case of changing heights of exhaust port

Fig. 8 Scavenging efficiency, η_s , in case of changing heights of scavenging port

port belongs to the lower category both at the low and high speeds. The value of $\delta_k(\text{CO}_2)$ is great in A-type cylinder at the high speed. It seems clear, therefore, that A-type cylinder is affected significantly by return-blow at the low speed and by blow-back at the high speed in view of the values of $\delta_e(\text{CO}_2)$ and $\delta_k(\text{CO}_2)$ with the delivery ratio. A great impact of blow-back on this type of cylinder at the high speed is also evident from the values of α_r and $\Delta A_r/A$. Also from the values of α_{rs} and $\Delta A_{rs}/A$, return-blow is likely to have little effects on the cylinder of A-type at the high speed. The above observations suggest that a drop in the concentration of a flow-in fresh gas due to return-blow at the low speed as well as to blow-back at the high speed causes the delivery ratio to decrease to a considerable degree.

Suppose that the height of an exhaust port remains unchanged and that of a scavenging port is subject to variation as is the case with Fig. 5 (I, F, J and K types). Then, it is clear that the type I is influenced greatly by return-blow at the low speed and by blow-back as well as return-blow, to a certain extent, at the high speed. The type K is hardly subject to the influence of blow-back and return-blow. Taking into account the loss of a fresh gas on the outflow side, or the trapping efficiency, therefore, it seems that the best charging efficiency can be obtained with a cylinder type and the engine speed that allow a moderate level of return-blow and blow-back - namely, the type F and 2800 rpm.

Figs. 7 and 8 show the relationship between the scavenging efficiency, η_s , and the modified delivery ratio, $K (= L / \text{Crel})$ for each cylinder type. Also shown in both charts are the values of η_s at the time of perfect mixing and stratification. Fig. 7 assumes the case where the height of a scavenging port remains unchanged and that of an exhaust port is subject to variation (A, F, G and H types) whereas the opposite case is given in Fig. 8 (I, F, J and K types). In both cases the engine speed is fixed at 1000, 2000, 2800, 4000 and 5000 rpm. As a consequence, the values of η_s come intermediate between perfect stratification and mixing or fail to reach the state of perfect mixing depending on the cylinder type and the engine speed. Yet the best scavenging efficiency, η_s , can also be achieved practically with the type F and at the engine speed of 2800 rpm. The value of η_s was obtained from the concentration of oxygen of gases in the cylinder before and after scavenging by the electromagnetic valve.

4. Setting the Diffusion Factor, ϵ_m , in the Scavenging Passage and the Mixing Factor, r , in the Scavenging Passage as well as the Crank Case

The factors on the scavenging side of the gas exchange process - the diffusion factor, ϵ_m , and the mixing factor, r - can be summarized as follows:

(1) The mixing factor, r , is provided to show the state of gases in the scavenging

passage as well as in the crank case immediately before the scavenging port opens. $r = 0$ means the case where a fresh gas alone exists in the scavenging passage and the crank case immediately before the scavenging port opens, and there is no residual gas due to blow-back or return-blow during the preceding cycle. $r \neq 0$ ($0 < r < 1$) represents the case in which a residual gas has to be considered to exist either in the scavenging passage alone (due to return-blow) or in both the scavenging passage and the crank case (due to blow-back).

(2) The diffusion factor, ϵ_m , is given to describe the state of gases during a period when the scavenging port is open and blow-back is taking place. $\epsilon_m = 0$ refers to the case where a combustion gas blows back in stratification to a fresh gas region during a blow-back period after the scavenging port opens. $\epsilon_m \neq 0$ denotes the case in which a diffusion occurs.

Fig. 9 compares both cases ($\epsilon_m \neq 0$, and $\epsilon_m = 0$) with cylinder types A, H and I as well as at the engine speed of 1000 ~ 5000 rpm to show changes in the initial angle when a fresh gas starts to flow into a cylinder, α_{es} , the distance of blow-back arrival, $S_{is}(\text{CO}_2)$, and the delivery ratio, L . In the types A and I with small exhaust lead, the value of α_{es} increases as blow-back grows at the high speed. The value of α_{es} is smaller in the case of $\epsilon_m \neq 0$ than in $\epsilon_m = 0$, so a fresh gas moves from a stratified region to a mixing region. The type H with large exhaust lead is subject to little blow-back only at the high speed as is evidenced by changes in the values of ϵ_m and α_{es} . These blow-back conditions can also be seen from changes

Fig. 9 Initial angle, α_{es} , of fresh gas flowing into cylinder, arrival distance, $S_{is}(\text{CO}_2)$, of blow-back gas and delivery ratio, L , in both cases of diffusion factor, $\epsilon_m \neq 0$ and $\epsilon_m = 0$

in the value of $Sis(CO_2)$ in the scavenging passage. In the types A and I, the distance of blow-back arrival is greater in the case of $\epsilon_m \neq 0$ than in $\epsilon_m = 0$, which becomes more pronounced particularly at the high speed. In the type H, meanwhile, the distance of blow-back arrival is relatively short. The delivery ratio, L , in the case of $\epsilon_m \neq 0$ is much the same as that in $\epsilon_m = 0$ if the same cylinder type and engine speed are given. Now let's take a brief look at a difference between both cases in terms of α_{es} , the flow velocity, u , and the concentration of CO_2 at the time of inflow, as all of these are supposed to have a great influence on the delivery ratio. With the engine speed is fixed at 2000 rpm, for example, in case of $\epsilon_m = 0$, we have $\alpha_{es} = 142.8^\circ$, $u = 98.5$ m/s and $CO_2 = 10.2\%$, whereas in case of $\epsilon_m \neq 0$ ($\epsilon_m = 0.05$), we have $\alpha_{es} = 139.4^\circ$, $u = 66.5$ m/s, and $CO_2 = 10.7\%$. Of these, α_{es} and the concentration of CO_2 are closely related to the effective area of the scavenging port and the fresh gas blown in respectively. Thus the diffusion factor has but a little influence on the delivery ratio because the multiplying effects of the effective area of the scavenging port, the inflow velocity, and the fresh gas blown in compensate each other.

Fig. 10 shows the effects of the mixing factor, r , applicable to a residual gas due to return-blow on the delivery ratio, L , with the type A and at the engine speed of 1000 ~ 5000 rpm. Of the above α_{es} , u and CO_2 , the effects of r on L are predominantly reflected by a drop in the concentration of CO_2 or a fresh gas blown in. In this diagram, the values of r are fixed at 0.25, 0.10, 0.02, 0.005 and 0.001 in proportion to the engine speeds of 1000, 2000, 2800, 4000 and 5000 rpm to see how the value of L might change at each one of the engine speeds if r equals zero. With the engine speed set at 1000 rpm, for instance, the value of L is smaller by about 5 % in the case of $r = 0.25$ than in $r = 0$. This is very close to measurement.

Fig. 11 provides a three dimensional indication of a gas component, C_8H_{18} , the length of the passage, and the crank angle - the state in the scavenging passage which fits very closely to the case where the engine speed is at 1000 rpm and the value of r equals zero in Fig. 10. It makes the above relations clearer. From this chart it is seen that the blow-back gas returns

in full amount to the cylinder and, hence, the effect of blow-back (α_r zone) is small in this case. Yet return-blow (α_{rs} zone) takes place in a very significant measure. The significance of the mixing factor can be grasped from this diagram as well. If return-blow occurs substantially at the low speed, therefore, there exists a large quantity of the residual gas from the preceding cycle inside the scavenging passage, thereby decreasing the delivery ratio to a remarkable degree, which coincides well with results of experiment. This is also evident in view of a difference in a cylinder type and the engine speed as is shown in Fig. 12.

Fig. 12 makes a comparison of the types A, H and I. The delivery ratio, L , decreases to a significant degree particularly with the type I and at the engine speed of 1000 rpm. This is due to a reduction in the concentration of a fresh gas resulting from return-blow and agrees closely with experiment result.

Fig. 13 shows the effects of the mixing factor, r , on the delivery ratio, L , with the type A and at the engine speed 1000 ~ 5000 rpm, taking into account a residual gas existing in the crank case due to blow-back. This diagram fixes the values of r at 0.0005, 0.001, 0.005, 0.02 and 0.05 corresponding to the engine speeds of 1000, 2000, 2800, 4000 and 5000 rpm to see how the delivery ratio, L , would change with the engine speed if r equals zero. With the engine speed fixed at 5000 rpm, for example, the value of L becomes smaller by about 7 % in the case of $r = 0.05$ than in $r = 0$. Again, this is very close to measurement.

Fig. 14 provides a three dimensional indication of gas components in the scavenging passage, the state which is almost applicable to the case where the engine speed is at 5000 rpm and the value of r equals zero in Fig. 13. From this chart it is clear that a considerable amount of the blow-back gas reaches the crank case, and that return-blow also takes place, though in a very insignificant measure. Again, it indicates the significance of the mixing factor. It seems probable, therefore, that a considerable quantity of the residual gas due to blow-back in the preceding cycle exists in the crank case at the beginning of the current cycle at the high speed and this residual gas, in

Fig. 10 Relationships between mixing factor (considered return-blow) and delivery ratio

Fig. 11 Three dimensional indication of gas component (C_8H_{18}) in scavenging passage

turn, would lower the concentration of a fresh gas blown in, thereby decreasing the delivery ratio to a substantial degree. The effects of blow-back vary according to the type of cylinder and the engine speed as indicated in Fig. 15.

Fig. 15 makes a comparison of the cylinder types A, H, I and K. From this diagram it is seen that the type I, among others, is influenced greatly by blow-back at the high speed and the type K is not. Also, from this diagram and Fig. 12 it can be clearly seen that the type I is subject to the influence of return-blow to a certain degree, along with blow-back, at the high speed.

Fig. 16 gives a three dimensional indication of gas components in the scavenging passage of the type I, making even clearer how this type is affected by blow-back and return-blow. The type K is free from influence of return-blow as well as blow-back. Both observations are very much consistent with experiment results. The fact that the delivery ratio will not go up so much may come from the above mentioned facts in the type I, in particular, which is provided with the largest scavenging port and a small exhaust lead.

Thus, it is evident that the diffusion factor, ϵ_m , enables us to understand

Fig. 12 Relationships between mixing factor(considered return-blow) and delivery ratio in case of changing cylinder type

Fig. 13 Relationships between mixing factor(considered blow-back) and delivery ratio

how a gas component or components move in the scavenging passage, while the mixing factor, r , makes it possible to recognize a drop in the delivery ratio due to the reduced concentration of a fresh gas resulting from blow-back or return-blow on the scavenging side. In the present study, the values of ϵ_m and r come within the range of: $0 \leq \epsilon_m \leq 0.08$; $0 \leq r \leq 0.35$ (for return-blow) and $0 \leq r \leq 0.06$ (for blow-back).

5. Setting of the Mixing Factor, ϕ_m , and the Short-Circuit Factor, ϕ_d , in the Cylinder

The following two approaches are taken to clarify the relationship, if any, between the factors, ϕ_m and ϕ_d , and the efficiency:

(1) With the presupposed values of ϕ_m and ϕ_d , the calculation of the relative charge, $CreI$, is made repeatedly until it settles approximately to the same level when the exhaust port opens and closes in the gas exchange process.

(2) The cylinder is assumed to be of the most simple type disregarding the length of the scavenging passage and both inlet and exhaust pipes. In this most simple type of cylinder it is assumed that conditions of gases are two extreme cases; that is, a perfectly stratified region ($K \leq 1$, $\eta_s = K$; $\phi_m = 0$, $\phi_d = 0$, $K > 1$, $\eta_s = 1$; $\phi_m = 0$, $\phi_d = 1$); and a perfectly mixing region ($\phi_m = 1$, $\phi_d = 0$). Given these assumptions, a scavenging diagram is made to show the relation between the scavenging efficiency, η_s , and the corrected delivery ratio, K ($= L/CreI$).

In the stratified region, a cubic scavenging curve ($\eta_s = -0.553K^3 + 0.665K^2 + 0.878K - 0.0498$) was drawn for the sake of convenience, supposing that the corrected delivery ratio, K , comes practically within the measured range of $0.65 \leq K \leq 1.30$ (see the curve a in Fig. 18). In addition, it was assumed that both mixing and short circuit factors, ϕ_m and ϕ_d , for the perfect stratification each take a value of 0 on this curve as well; the mixing factor for the perfect stratification ($\phi_m = 0$) and that for the perfect mixing ($\phi_m = 1$) were arranged at equal distances

Fig. 14 Three dimensional indication of gas component(C_8H_{18}) in scavenging passage

to agree with experiment results. The values of φ_m below the level of perfect mixing were also arranged at equal distances. This also applied to the values of φ_d . Note should be taken of the fact, however, that the above scavenging efficiency for the perfect mixing differs from the one based on the assumption that $Crel$ stays at a constant level ($\eta_s = 1 - e^{-k}$), because $Crel$ is normally subject to a change in the gas exchange process.

The approach (1) is commonly applied to a simulation of the gas exchange process. Fig. 17 is precisely based on the approach (1), comparing calculations with measurement in the case of the cylinder type A and the engine speed fixed at 2000 rpm. Supposing that the pressure, P_{za} , is fixed at 0.315 MPa (3.21 kgf/cm²), the temperature, T_{za} , at 1182 K, the diffusion factor, ϵ_m , at 0.05, and the mixing factor for the scavenging passage, r , at 0.1 when the exhaust port is open, and setting the values of φ_m and φ_d at, say, 0.85 and 0 respectively, then the delivery ratio, the trapping efficiency and the charging efficiency are calculated as follows: $L = 0.737$, $\eta_{tr} = 0.657$ and $\eta_c = 0.484$. The measured values are: $L = 0.753$, $\eta_{tr} = 0.681$ and $\eta_c = 0.514$. Thus the calculated values very much correspond with the measured ones.

Fig. 15 Relationships between mixing factor (considered blow-back) and delivery ratio in case of changing cylinder type

Fig. 16 Three dimensional indication of gas component (C_8H_{18}) in scavenging passage

Incidentally, the approach (2) involves more trials and errors than the approach (1), but the latter takes much more time and money as it requires the cylinder type and the engine speed to be changed very frequently. So the approach (2) is primarily employed in our study. A difference in the relative charge, $Crel$, between the approaches (1) and (2) is adjusted to come within the range of about 3 percent.

Fig. 18 is mainly based on the approach (2) to show the relationship between η_s and φ_m , and φ_d . This diagram also gives the calculations of η_s with the cylinder types A and F as well as the engine speed at 1000 ~ 5000 rpm. Close agreement between the calculated and measured values is observed (see Fig. 7). The calculated values are: $\varphi_m = 0.95 \sim 0.75$ and $\varphi_d = 0$ for the type A, while $\varphi_m = 0.95 \sim 0.80$ and $\varphi_d = 0$ for the type F. Also shown in this diagram are the values of η_s for both perfect mixing and stratification determined with the cylinder types A, F, G, H and J. Given the same corrected delivery ratio, K , a difference in the scavenging efficiency according to cylinder type and engine speed comes within the range of about $\pm 0.03\%$ for perfect stratification and about $\pm 1\%$ for perfect mixing. The values of η_s , L and $Crel$ for both perfect mixing and stratification are the ones obtained at a time when the exhaust port closes and the gas exchange process is over.

Fig. 19 indicates the relations of φ_m and φ_d with the delivery ratio, L , the trapping efficiency, η_{tr} , the charging efficiency, η_c , and the scavenging efficiency, η_s , with the engine speed fixed at 1000 ~ 5000 rpm and the cylinder types A, F, G and H - the case where the height of the scavenging port remains constant and that of the exhaust port is subject to variations. This diagram shows clearly how these efficiencies vary according to the values of φ_m and φ_d .

The delivery ratio, L , increases gradually according to cylinder types A to H,

Fig. 17 Relationships between φ_m , φ_d and efficiencies

or as the exhaust port becomes higher. In this case, it is seen that φ_m and φ_d come very close to the values for perfect mixing ($\varphi_m = 1$, $\varphi_d = 0$) at the medium speed. With the type H and the engine speed at 2800 rpm, for example, the values are: $\varphi_m = 0.95$, and $\varphi_d = 0$.

The trapping efficiency, η_{tr} , decreases also according to cylinder types A to H in the order mentioned. With the type H and the engine speed at 1000 rpm, among others, a remarkable drop in η_{tr} is observed. It is clear, in such a case, that heavy losses of a fresh gas are involved as is evidenced by the values of φ_m and φ_d ($\varphi_m = 0.9$, $\varphi_d = 0.1$).

The charging efficiency, which is expressed as the product of L and η_{tr} , varies naturally with the values of φ_m and φ_d . The best charging efficiency, η_c , can only be achieved with the type F and the engine speed at 2800 rpm when φ_m and φ_d are kept at 0.85 and 0 respectively. In other words, this is possible under conditions where gases are not perfectly mixed with each other and a direct short-circuit does not occur from the stratified region.

The scavenging efficiency, η_s , obviously changes according to the values of φ_m and φ_d as it is determined by the amount of a residual gas and a fresh gas charged. The best scavenging efficiency, as is the case with the charging efficiency, can be achieved with the type F and the engine speed at 2800 rpm when φ_m and φ_d are set at 0.85 and 0 respectively.

The above relations also apply to the case where the height of the exhaust port remains constant and that of the scavenging port are subject to variations, if only φ_m and φ_d are given. With the engine speed at 1000 ~ 5000 rpm and the cylinder types A, F, G, H, I, J and K in the present study, the values of φ_m and φ_d are brought within the range of: $0.75 \leq \varphi_m \leq 1$, and $0 \leq \varphi_d \leq 0.1$ respectively. In addition, the values of φ_m and φ_d , at which the best charging and scavenging efficiencies, η_c and η_s , are achieved through each type of cylinder, come within the range of: $0.8 \leq \varphi_m \leq 0.95$, and $\varphi_d = 0$. Also, the best charging and scavenging efficiencies, η_c and η_s , can be realized with the type F and the engine speed at 2800 rpm when φ_m and φ_d are set at 0.85 and 0 respectively.

Fig. 18 Relationships between φ_m , φ_d and scavenging efficiency

6. Summary

What has been discussed so far can be summarized as follows:

A. Pressure measurement and gas analysis reveal that the delivery ratio, L , the trapping efficiency, η_{tr} , the charging efficiency, η_c , and the scavenging efficiency, η_s , vary according to scavenging and exhaust port configurations and engine speed.

(1) The delivery ratio, L , when the exhaust lead is small, is influenced greatly by a reduction in the concentration of a fresh gas blown in due to return-blow at low speed as well as to blow-back at high speed. When the scavenging port is large and the exhaust lead is small, meanwhile, it is affected by a reduction in the concentration of a fresh gas due to blow-back to a considerable degree as well as to return-blow to a certain degree at high speed.

(2) The trapping efficiency, η_{tr} , when the effective stroke is short and the exhaust lead is large, is influenced significantly by the loss of a short-circuited fresh gas.

(3) The charging efficiency, η_c , which is expressed as the product of the delivery ratio, L , and the trapping efficiency, η_{tr} , and is almost proportional to the mean effective pressure, reaches the highest level with the proper cylinder type and engine speed - that is to say, when the heights of scavenging and exhaust ports, σ_e and σ_a , are set at 0.176 and 0.272 respectively, and the engine speed at 2800 rpm. This holds true for the best scavenging efficiency, η_s .

B. Using the diffusion factor inside the scavenging passage, ϵ_m , the mixing factor in the scavenging side, r , the mixing factor in the cylinder, φ_m , and the short-circuit factor, φ_d , it is possible to clearly relate calculated values with experimental results.

Fig. 19 Relationships between φ_m , φ_d and efficiencies

(1) The diffusion factor, ϵ_m , shows how gas components move in the scavenging passage, namely a fresh gas begins flowing into the cylinder, and a blow-back gas reaches.

(2) The mixing factor, r , indicates how the delivery ratio is affected by residual gas components in the scavenging passage or crank case due to blow-back or return-blow.

(3) The factors, ϕ_m and ϕ_d , coupled with the above factors, clearly show the relationship of the scavenging efficiency and others with the cylinder type and the engine speed.

(4) In the present study, the values of ϵ_m , r , ϕ_m and ϕ_d for each cylinder type and engine speed are brought within the range of: $0 \leq \epsilon_m \leq 0.08$; $0 \leq r \leq 0.35$ (for return-blow) or $0 \leq r \leq 0.06$ (for blow-back); $0.75 \leq \phi_m \leq 1$; and $0 \leq \phi_d \leq 0.1$. With such a cylinder type and engine speed as to allow the values of η_c and η_s to reach the highest level (the same as measurement), the values of ϵ_m , r , ϕ_m and ϕ_d are fixed for simulation purposes at the following levels: $\epsilon_m = 0.01$; $r = 0.01$ (for return-blow) and $r = 0$ (for blow-back); $\phi_m = 0.85$ and $\phi_d = 0$. This means the following: (a) When $\epsilon_m = 0.01$, a fresh gas begins flowing into the cylinder shortly after a blow-back has almost returned to the cylinder. (b) When $r = 0.01$ (for return-blow), a fresh gas on the scavenging side contains just a small amount of residual gas. (c) When $\phi_m = 0.85$ and $\phi_d = 0$, fresh and residual gases are not uniformly or perfectly mixed with each other and no loss of fresh gas due to short-circuit from the stratified region is involved in the cylinder.

C. A three dimensional indication of gas components in the scavenging passage explains even more clearly how a blow-back or return-blow occurs and how a fresh gas flows into the cylinder.

Thus it is proposed that the factors ϵ_m , r , ϕ_m and ϕ_d be applied to a simulation of the gas exchange process in a small two-stroke cycle engine for the sake of better understanding of the engine's best scavenging performance as related to the blow-back and return-blow phenomena, and also that the usefulness of the simulation be proved by a three dimensional indication of gas components in the scavenging passage.

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