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The Effect of Crankcase Volume and the Inlet System on the Delivery Ratio of Two-Stroke Cycle Engines

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IN ORDER TO IMPROVE ENGINE PERFORMANCE, increasing the air charge is most important. The air charge of the crankcase scavenged two-stroke cycle engine is particularly sensitive to pressure fluctuations in the exhaust and inlet systems because of the small pressure differences available for scavenging the cylinder. To date, the results of many fluctuation studies have been published (1-9)* and the optimum tuning conditions of the inlet and exhaust pipes have been solved both theoretically and experimentally. It is a well-known fact that the air charge can easily be made larger than the piston displacement by pipe tuning. However, the dimensions of the exhaust and inlet systems, which are designed without regard to utilization of the pressure fluctuations, are not always suitable for full utilization. Since the air charge is increased by the pressure difference, it is necessary to consider not only matching the frequency of fluctuation to the engine speed, but also to the time-areas** of the ports and the flow resistance of the inlet and exhaust systems.

By changing the crankcase volume and lengths of the exhaust and inlet pipes, the authors have experimentally investigated the effect of the crankcase volume on the delivery ratio, and the effect of exhaust and inlet systems to compensate for the drop in delivery ratio caused by increasing the crankcase volume. Further experiments have been carried out to find the conditions under which both the tuned exhaust and inlet systems are most effectively utilized for increasing the delivery ratio.

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*Numbers in parentheses designate References at end of paper.

$$\text{**Time-Area (in.}^2\text{-sec)} = \frac{\text{Angle-Area (in.}^2\text{-deg)}}{6 \times \text{Engine Rpm}}$$

ABSTRACT

Increasing the air charge of crankcase scavenged two-stroke cycle engines is essential for improved performance. Accordingly, the crankcase volume and dimensions of the inlet system were experimentally investigated in order to fully utilize the dynamic effect of both the exhaust and inlet systems with a view towards increasing the delivery ratio. To achieve full utilization of the effect of the inlet system, it was found that the time-area of the inlet port should be far larger than that considered for the usual engine. The drop in delivery ratio caused by increasing the crankcase volume can be fairly well compensated for by tuning the exhaust and inlet systems.

TEST PROCEDURE

Two crankcase scavenged two-stroke cycle gasoline engines were used; their main dimensions are shown in Table 1. A schematic diagram of the experimental arrangement is shown in Fig. 1. The discharge end of the exhaust pipe was inserted into the expansion chamber (about 1.4 cu ft) through which the exhaust gas was discharged. The carburetor was located between the inlet pipe and the surge tank. Three faces of the rectangular surge tank (about 1.7 cu ft) were made of rubber membrane to damp the pressure pulsations in the tank. Air was supplied from the sinking tank through the control valve to keep the pressure in the surge tank at atmospheric level. The airflow rate of engine A was measured by the displacement volume of the sinking tank. Since measurements with the sinking tank took a long time, the airflow rate of engine B was measured with a sharp-edge orifice. In this case, the sinking tank was used only to supply air to the surge tank. By comparing the flow rate measured with the orifice with that of the sinking tank, it was confirmed that the pressure pulsations were too well damped in the surge tank to disturb the measurements with the orifice.

Inasmuch as the throttle valve of the carburetor was held fully open and the area of the fuel metering orifice was fixed, the fuel-air ratio was affected by the operating conditions of the engine. However, the cylinder pressure at the beginning of blowdown is scarcely affected by the variation in the fuel-air ratio which is caused by both changes in the engine speed and the length of the inlet pipe. It is affected far more by the delivery ratio than the fuel-air ratio. In this experiment, the length of the exhaust pipe was represented by the distance between the cylinder wall and the discharge end of the exhaust pipe, and the length of the inlet pipe by that between the cylinder wall and the inlet of the carburetor. The carburetor was removed when the engine was driven by an electric dynamometer in order to investigate only the effect of the inlet system on the delivery ratio.

CRANKCASE VOLUME

THEORETICAL TREATMENT - For the arrangement of experimental data, it is convenient to determine the relationships between the factors affecting delivery ratio. These relationships were investigated without consideration of the dynamic effect of the exhaust and inlet systems.

The state of the air in the crankcase can be expressed by

$$\frac{1}{p}\frac{dp}{dt} + \frac{1}{V}\frac{dV}{dt} - \frac{1}{G}\frac{dG}{dt} - \frac{1}{T}\frac{dT}{dt} = 0 \quad (1)$$

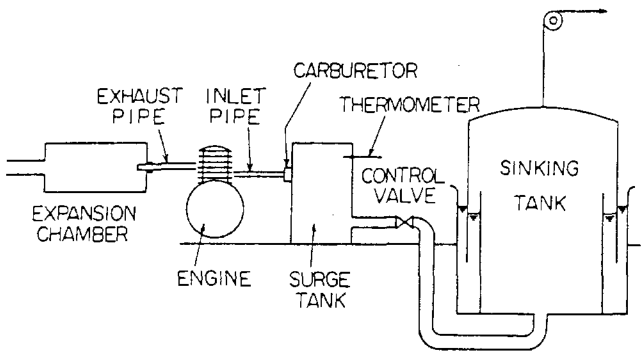


Fig. 1 - Schematic diagram of experimental arrangement

Table 1 - Test Engine Specifications

	Engine A (Tohatsu TEA-65)	Engine B (Kawasaki KF-3-G)
Type	Schnürle Scavenged, Air-Cooled	
Cylinder Bore, Stroke and Displacement	2.56 in. x 2.36 in. 12.15 in.^{3} (199 cc)	2.28 in. x 2.20 in. 9.03 in.^{3} (148 cc)
Normal Output	5.5 PS/3600 rpm	3.5 PS/3600 rpm
Crankcase Compression Ratio	1.41	1.51
Inlet Pipe Diameter, in.	1.14	1.02
Exhaust Pipe Diameter, in.	1.57	1.10

	Engine A: Period of Port Opening, deg	Engine A: Angle-Area, in.^{2}-deg	Engine B: Period of Port Opening, deg	Engine B: Angle-Area, in.^{2}-deg
	(Symmetrical Timing)		(Symmetrical Timing)	
Inlet Port	132	87.4	125	47.4
Scavenge Port	111.4	50.4	112.8	55.3
Exhaust Port	145.2	143.2	134	53.6

In order to simplify the calculation, the change in state is assumed to be adiabatic. As a result, the above equation can be transformed as follows:

$$\frac{1}{p}\frac{dp}{dt} + \frac{\kappa}{V}\frac{dV}{dt} - \frac{\kappa}{G}\frac{dG}{dt} = 0 \quad (2)$$

The flow state through both the inlet port and the carburetor is replaced by the equivalent port of area F_i defined as follows:

$$\frac{1}{\mu^2 F_i^2} = \frac{1}{\mu_i^2 f_i^2} + \frac{1}{\mu_c^2 f_c^2} \quad (3)$$

The flow state through both the exhaust and scavenge ports is similarly replaced with that of the equivalent port of area F_s given by

$$\frac{1}{\mu^2 F_s^2} = \frac{1}{\mu_e^2 f_e^2} + \frac{1}{\mu_s^2 f_s^2} \quad (4)$$

The change in the weight of air in the crankcase is expressed as follows:

For flow from the crankcase,

$$\frac{dG}{dt} = -\mu F\sqrt{\frac{2\kappa g p_o \gamma_o}{\kappa - 1}}\sqrt{\left(\frac{p}{p_o}\right)^{(\kappa-1)/\kappa} - 1} \quad (5)$$

For flow into the crankcase,

$$\frac{dG}{dt} = \mu F\sqrt{\frac{2\kappa g p_o \gamma_o}{\kappa - 1}}\left(\frac{p}{p_o}\right)^{1/\kappa}\sqrt{1 - \left(\frac{p}{p_o}\right)^{(\kappa-1)/\kappa}} \quad (6)$$

where F becomes F_i given by Eq. 3 during the inlet period, and F_s given by Eq. 4 during the scavenge period.

On the other hand, the weight of air in the crankcase can be expressed as follows:

$$G = V\gamma_o\,(p/p_o)^{1/\kappa} \quad (7)$$

By introducing Eqs. 5-7 and $d\theta/dt = 6n$ into Eq. 2, the following equations are obtained:

For flow from the crankcase,

$$\frac{1}{p}\frac{dp}{d\theta} + \frac{\kappa}{V}\frac{dV}{d\theta} + \frac{\kappa}{6n}\sqrt{\frac{2\kappa g p_o}{(\kappa-1)\gamma_o}}\frac{\mu F}{V} \cdot \left(\frac{p}{p_o}\right)^{-1/\kappa}\sqrt{\left(\frac{p}{p_o}\right)^{(\kappa-1)/\kappa} - 1} = 0 \quad (8)$$

For flow into the crankcase,

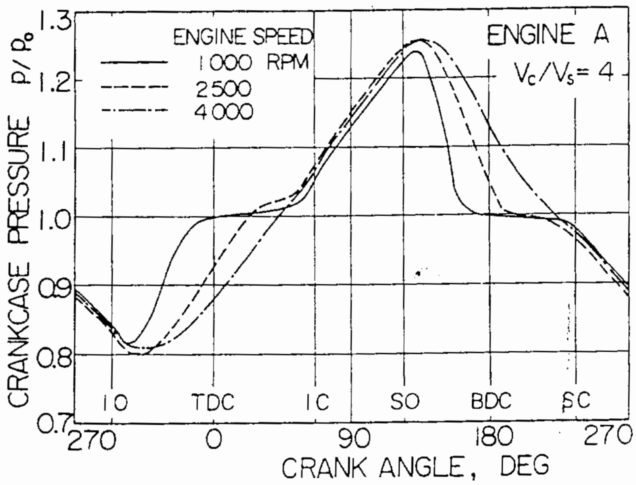
$$\frac{1}{p}\frac{dp}{d\theta} + \frac{\kappa}{V}\frac{dV}{d\theta} - \frac{\kappa}{6n}\sqrt{\frac{2\kappa g p_o}{(\kappa-1)\gamma_o}}\frac{\mu F}{V}\sqrt{1 - \left(\frac{p}{p_o}\right)^{(\kappa-1)/\kappa}} = 0 \quad (9)$$

Assuming a certain pressure for the starting point at inlet port opening, the pressure variation can be calculated stepwise with either Eq. 8 or Eq. 9. This procedure is reiterated until the pressure value obtained agrees with that assumed in the preceding cycle. The delivery ratio 1_o is calculated from the pressures at inlet port opening and closing as follows:

$$1_o = \left\{\left(\frac{p_1}{p_o}\right)^{1/\kappa} - \left(\frac{p_2}{p_o}\right)^{1/\kappa}\right\}\frac{V_1}{V_s} \quad (10)$$

By taking $\mu_e = \mu_i = \mu_s = 0.8$ and $\mu_c = 1$, and using the port dimensions of engine A, the pressure variation was calculated for various crankcase volumes and engine speeds. An example is shown in Fig. 2. At 1000 rpm engine speed, the induction and discharge of the air are accomplished early in the inlet and scavenge periods, respectively, due to the excessively large port areas. The pressure in the crankcase is approximately atmospheric at the dead centers. After top dead center, the air in the crankcase is discharged by the movement of the piston. The pressure at inlet port closing cannot become high. During the later half of the scavenge period, the negative pressure produced by the motion of the piston is decreased by the reverse flow through the scavenge port. Therefore, the crankcase vacuum at inlet port opening (which draws in fresh air from the outside) is similarly decreased.

At 2500 rpm engine speed, the duration required to draw the fresh air into the crankcase is prolonged. However, the



IO: Inlet port opens. SO: Scavenge port opens.
IC: Inlet port closes. SC: Scavenge port closes.

Fig. 2 - Calculated variation of crankcase pressure

pressure at inlet port closing becomes higher than at 1000 rpm because of the decrease in the reverse flow from the crankcase. At 4000 rpm, it becomes low again due to the poor time-area of the inlet port. At 2500 rpm, the duration of discharge of the scavenge air is longer than at 1000 rpm. However, due to the decrease in the reverse flow through the scavenge port, the pressure at scavenge port closing is the lowest. At 4000 rpm, the pressure at scavenge port closing becomes high again due to prolongation of the scavenge air discharge period. Fig. 3 shows the calculated delivery ratio.

The engine speed n_o, at which maximum delivery ratio is produced, decreases with an increase in the ratio of the crankcase clearance volume to the piston displacement. However, the maximum value of the delivery ratio is scarcely affected by the clearance volume. Taking n_o/n as the abscissa, the relationship between the delivery ratio and engine speed is practically independent of the crankcase clearance volume (Fig. 4). The ratio n_o/n is used for arranging the experimental data. As the time-area of the port changes inversely with engine speed, the abscissa n_o/n may be considered proportional to the time-areas. At $n_o/n = 1$, maximum delivery ratio is attained and the time-areas are the most favorable. At all other values of n_o/n, the corresponding time-areas are n_o/n-multiplied by the most favorable time-areas, when no change is made in the port timing. At $n_o/n > 1$, the delivery ratio is low due to the increase in the reverse flow caused by the excessively large port areas. At $n_o/n < 1$, it is low because of the poor time-areas of the ports.

DELIVERY RATIO AT VARIOUS CRANKCASE VOLUMES - The crankcase volume of engine A was increased to the desired amount by connecting a receiver to the side wall of the crankcase through a 1.52 sq in. hole. Fig. 5 shows the delivery ratio of engine A measured with the shortest possible exhaust and inlet pipes. Since the pipe lengths are represented by the distance from the cylinder wall to their open ends, the shortest possible exhaust and inlet pipe lengths, which were determined by constructional restrictions, were 6.9 and 5.2 in., respectively. The engine speed n_o, at which maximum delivery ratio is obtained, decreases with an increase in clearance volume. This engine speed n_o changes almost in inverse proportion to the square root of the clearance volume. Taking n_o/n as the abscissa, the delivery ratio can be represented (curve D in Fig. 6). These experimental results agree quantitatively with calculated results. From any one of the known relationships between

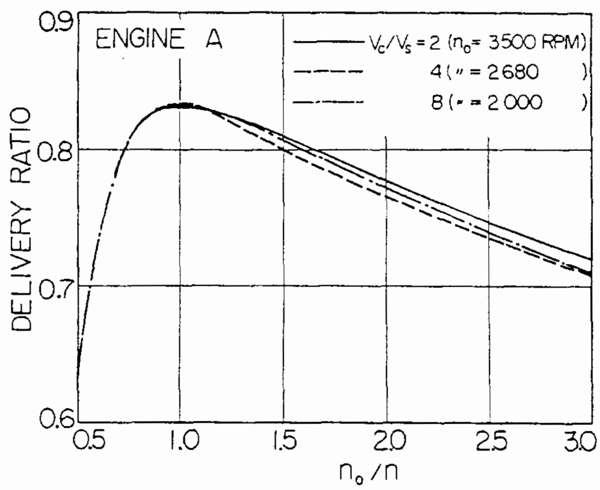


Fig. 4 - Relationship between delivery ratio and ratio of engine speed n_o, giving a maximum delivery ratio, to engine speed n (calculated)

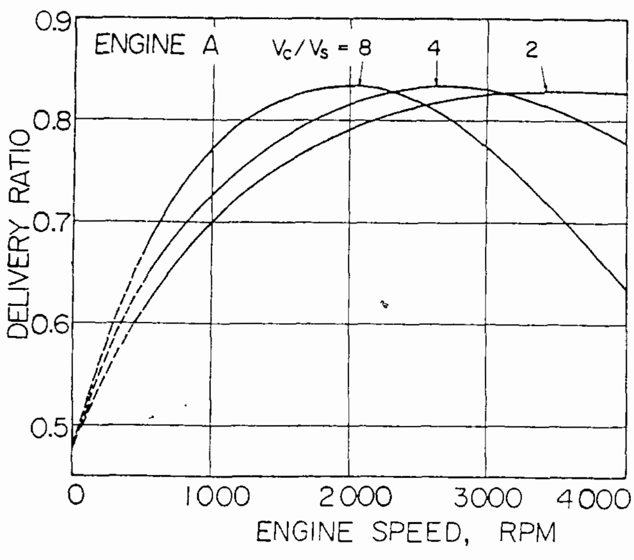


Fig. 3 - Relationship between delivery ratio and engine speed (calculated)

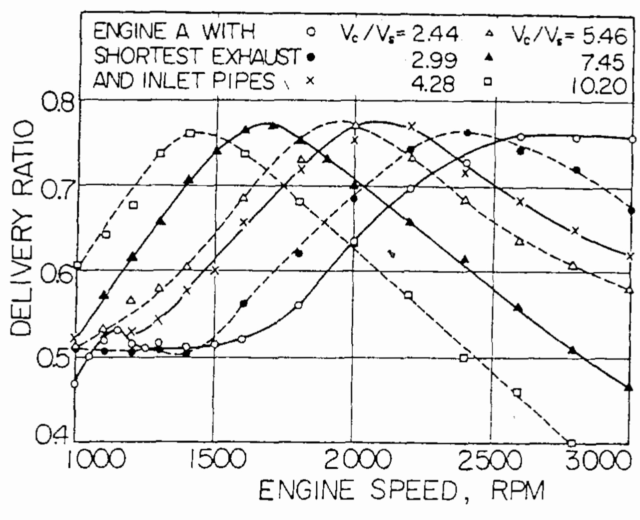


Fig. 5 - Relationship between delivery ratio and engine speed

the delivery ratio and the engine speed obtained experimentally, it is possible to estimate the relationship for an arbitrary value of the clearance volume.

The delivery ratio at a given engine speed was investigated under various conditions. When the exhaust or inlet pipe was tuned, each length was selected to give a maximum delivery ratio at the shortest length of the other pipe. As shown in Fig. 7, the degree of improvement by increasing the delivery ratio above that obtainable with the shortest exhaust and inlet pipes by means of the properly tuned exhaust pipe, is scarcely affected by changing the crankcase clearance volume, because the exhaust pipe effect depends on the blowdown. However, the degree of the improvement in delivery ratio produced by a properly tuned inlet pipe is markedly less with an increase in clearance volume.

The decrease in delivery ratio seems to be caused both by the extinction of the ramming effect of the inlet pipe, whose length becomes shorter with an increase in crankcase volume, and by the lack of inlet port time-area. The delivery ratio, obtainable by combining exhaust and inlet pipes that are independently matched to the engine speed, decreases with an increase in clearance volume. The effect of the diffuser (cone angle 8 deg, area ratio of both ends 9), which is fitted to the end of the exhaust pipe, is remarkable in the case where only the exhaust pipe is properly tuned, but is slight in the case where the properly tuned inlet pipe is used. Figs. 8 and 9 show the results measured at engine speeds of 1800 and 3000 rpm, respectively. At 3000 rpm, the delivery ratio, measured with properly tuned exhaust and inlet pipes, is smaller at some values of V_c/V_s than the delivery ratio measured with the properly tuned exhaust pipe and the shortest inlet pipe. Therefore, it is evident that ex-

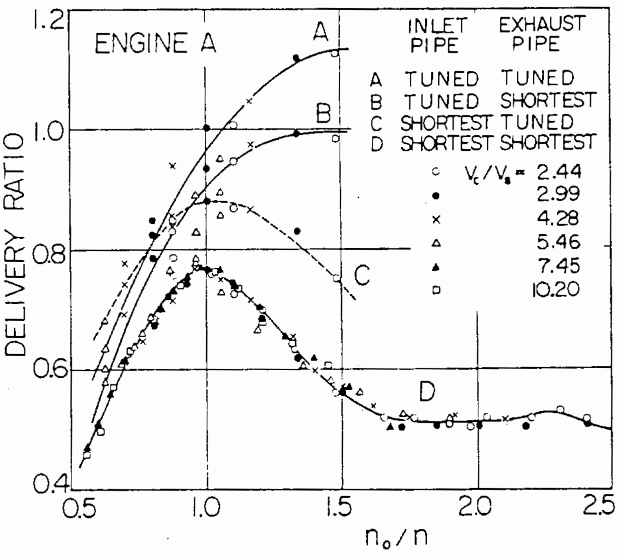


Fig. 6 - Effect of properly tuned exhaust and inlet pipes

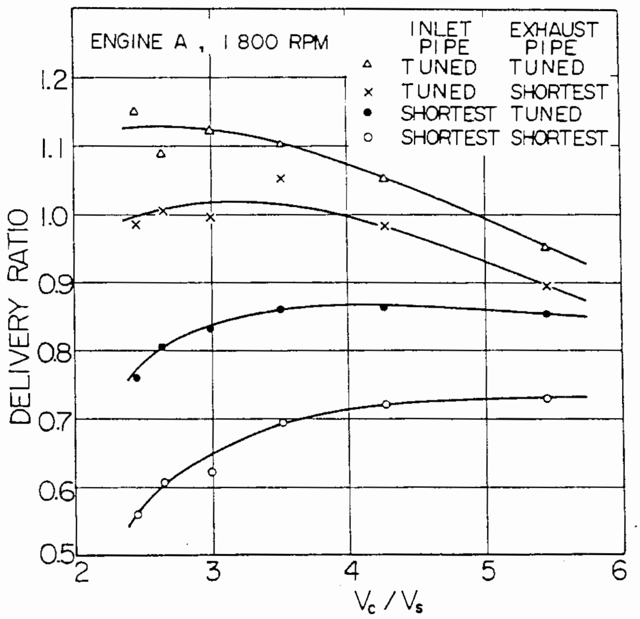


Fig. 8 - Effect of crankcase clearance volume V_c

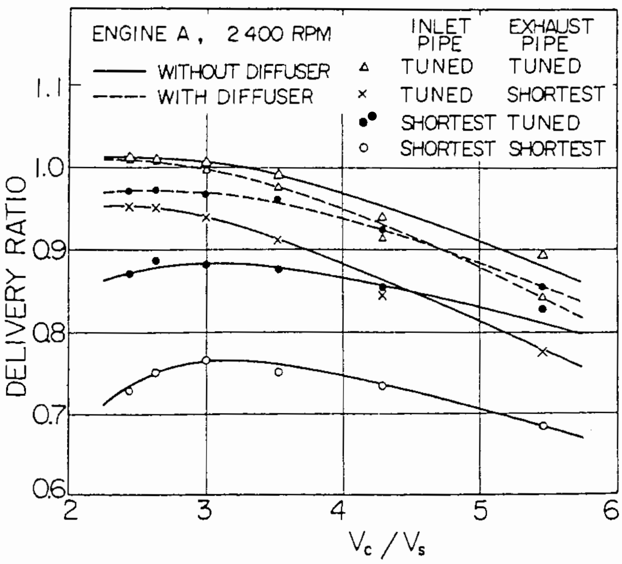


Fig. 7 - Effect of crankcase clearance volume V_c

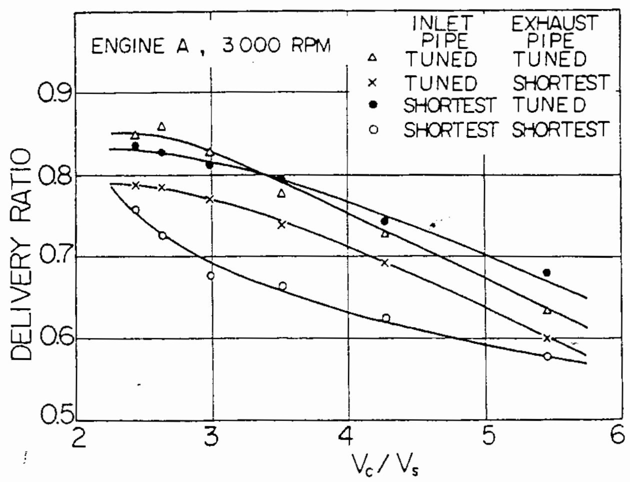


Fig. 9 - Effect of crankcase clearance volume V_c

haust and inlet pipes which are independently matched do not always give the most favorable results. The maximum delivery ratio obtained by the use of the tuned pipes increases with a decrease in clearance volume. However, the rate of its increase falls gradually. No advantage seems to accrue from making the clearance volume excessively small.

Using the engine speed n_o, at which the delivery ratio obtainable with the shortest pipes becomes maximum, the delivery ratios obtained by first tuning only the exhaust pipe, then only the inlet pipe, and, last, both pipes, are plotted in Fig. 6. The delivery ratio under each condition can be roughly represented by a curve, though some difference appears at the higher engine speeds. It is possible to determine the condition under which the effect of the exhaust and inlet pipes becomes most pronounced from the figure. The degree of increase in delivery ratio obtained by the tuned exhaust pipe (curve C) over that obtained by the shortest pipes (curve D) is scarcely affected by n_o/n (that is, the time-areas of the ports). As shown by curve B in Fig. 6, the degree of improvement in delivery ratio produced by the tuned inlet pipe becomes more marked with an increase in n_o/n. In order to increase the delivery ratio by tuning the exhaust and inlet pipes, the time-areas of the ports must be much larger than those for the shortest pipes.

Fig. 10 shows the crankcase pressures at inlet port opening and closing. These measurements were made with an indicator of the photoelectric cell type. With the tuned inlet pipe, the closing pressure is high but the opening pressure is hardly changed. With the tuned exhaust pipe, only the opening pressure is affected. Employing both tuned pipes, the closing and opening pressures are high and low, respectively; both of these help to increase the delivery ratio. In the same figure, the drop in pressure at closing, which is caused by an increase in engine speed under the condition of the tuned inlet pipe, is remarkable. Therefore, the insufficiency of the time-area of the inlet port seems to be chiefly responsible for the extinction of the inlet pipe effect at high engine speeds.

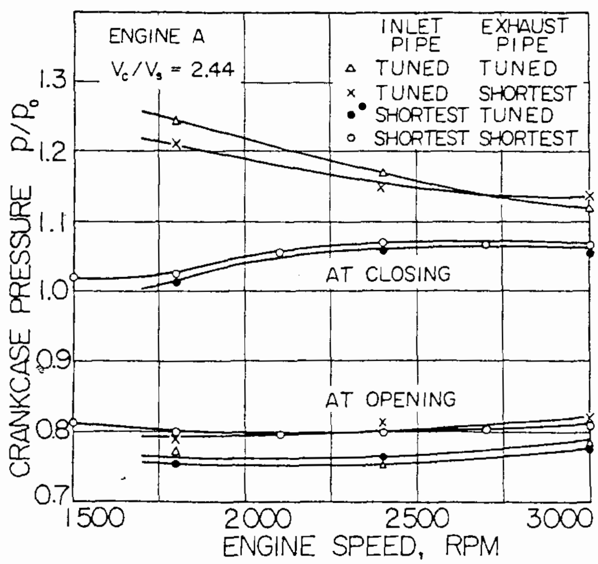


Fig. 10 - Relationship between engine speed and crankcase pressures at opening and closing of inlet port

TUNING OF EXHAUST SYSTEM

The delivery ratio can be increased by fitting an exhaust pipe of suitable length (Fig. 11). Since the pressure fluctuation due to the blowdown is quite large and continues in the exhaust system after exhaust port closing, it affects the blowdown process of the next engine cycle. Therefore, the tuning frequency of pressure fluctuation relative to the engine speed should be considered for a whole engine cycle, that is, 360 deg of crankangle. Assuming that the exhaust system is a pipe with an open end and a closed end, the number of pressure fluctuations during an engine revolution is approximately given by

$$\frac{60/n}{4\, l_e/a_e} = \frac{15\, a_e}{n\, l_e} \tag{11}$$

where:

a_e = Mean sonic velocity of gas in exhaust system

l_e = Length of exhaust pipe plus correction value of open end (0.82 X radius of pipe)

Changing the length of the exhaust pipe under constant engine speeds, the relationship between the delivery ratio and number of pressure fluctuations $15\, a_e/nl_e$ is investigated. The delivery ratio becomes maximum at $15\, a_e/nl_e = 3$ (Fig. 12).

TUNING OF INLET SYSTEM

The delivery ratio can be improved by selecting the inlet pipe length so as to increase the pressure at inlet port clos-

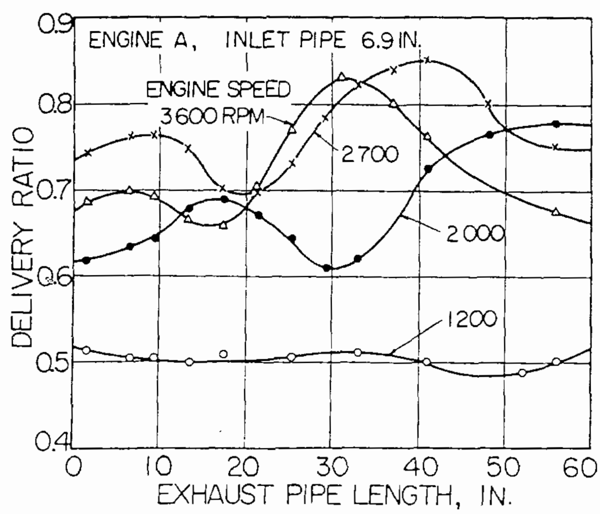


Fig. 11 - Effect of exhaust pipe length

ing. The pressure fluctuations, which continue in the inlet system after inlet port closing, little affect the suction of the next cycle. Ignoring the pressure fluctuation produced by the suction of the previous cycle, only the frequency of fluctuation relative to the engine speed has to be considered for the inlet port opening period. The acoustic frequency ν of the inlet system, which consists of the crankcase and the inlet pipe, is given by

$$\frac{2\pi\nu l_i}{a_i}\tan\frac{2\pi\nu l_i}{a_i} = \frac{f_{pi}\, l_i}{V_m} \tag{12}$$

where:

a_i = Mean sonic velocity of air in inlet system
f_{pi} = Cross-sectional area of inlet pipe
l_i = Length of inlet pipe plus correction value of open end
V_m = Mean crankcase volume for inlet port opening period

The number of pressure fluctuations during the inlet port opening is given by

$$\frac{60/n \quad \theta_i/360}{1/\nu} = \frac{\nu\theta_i}{6n} \tag{13}$$

where:

θ_i = Inlet port opening period, deg

The delivery ratio is shown in Fig. 13 plotted as a function of the number of pressure fluctuations calculated from Eq. 13. The delivery ratio becomes maximum at $\nu\theta_i/6n$ = 0.7 - 0.8. That is, the maximum delivery ratio is obtained by selecting the length of inlet pipe to make three-quarters of the period of pressure fluctuation coincident with the inlet port opening period.

INLET PIPE LENGTH

The delivery ratio of the engine in the firing state is lower than that in the motoring state because of both the rise of temperature and the increase in crankcase pressure caused by the reverse flow of exhaust gas. The delivery ratio of engine B with a 14.2 in. long inlet pipe was measured in the motoring and firing states. The relationships of the delivery ratio to the engine speed in both states are quite similar (Fig. 14). In order to investigate the fundamental factors of the inlet system, the motoring test with a simplified inlet system may be better than the firing test. The experiment was carried out in the motoring state without the carburetor.

As shown in Fig. 15, the engine speed giving a maximum

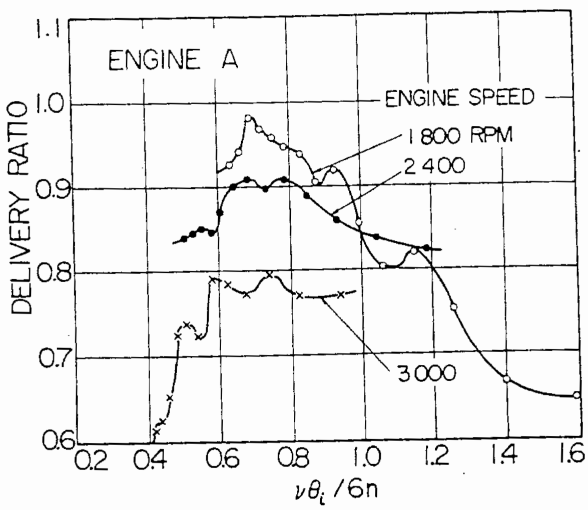


Fig. 13 - Number of pressure fluctuations in inlet pipe during inlet port opening period

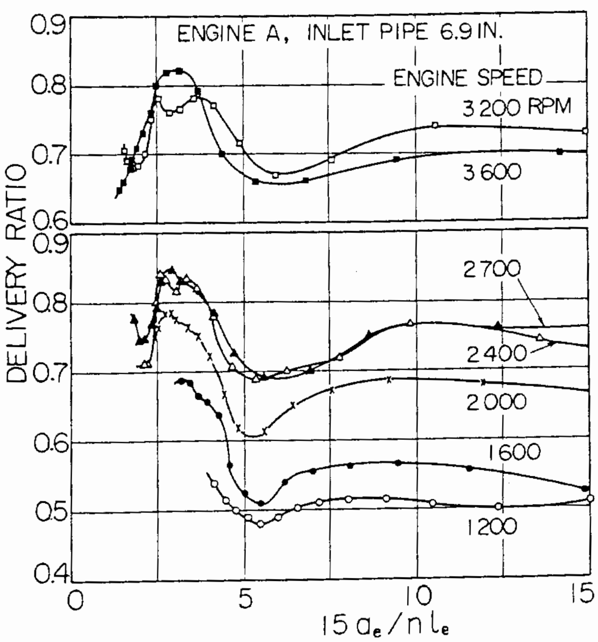


Fig. 12 - Number of pressure fluctuations in exhaust pipe during one engine cycle

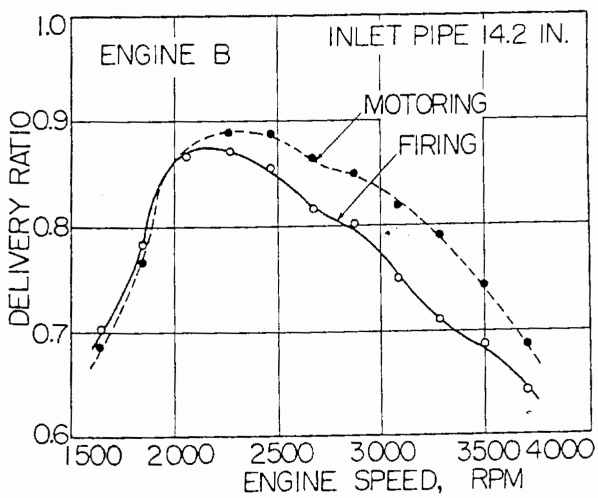


Fig. 14 - Comparison of motoring and firing delivery rati

delivery ratio decreases and the maximum delivery ratio becomes gradually higher with an increase in length. This tendency indicates that the effect of the inlet pipe is more profitably utilized with an increase in the inlet port time-area. Fig. 16 shows the range of inlet pipe lengths making the delivery ratio higher than that for the shortest length of 4.3 in. Note that with the shorter lengths, the delivery ratio is improved by the inlet pipe ramming effect over a wide range of engine speeds, and the range in which the delivery ratio decreases is quite small. The two fine lines in Fig. 16 show the lengths calculated from Eq. 13, which produce the pressure fluctuations of 0.5 and 0.75, respectively, during the open period of the inlet port. The inlet pipe which gives a value of $\nu\theta_i/6n$ larger than 0.5 can improve the delivery ratio.

INLET PORT OPENING PERIOD

REMODELING INLET PORT OF ENGINE B - The dimensions of the inlet port of engine B are shown in Fig. 17. The inlet port was designed so as to be open 125 deg of crank-angle and to be fully open at top dead center, where piston skirt edge reached the upper edge of the port. To increase the inlet port opening period, the piston skirt was cut off at the proper distance from its edge with the breadth of the cut portion 0.08 in. larger than that of the port. To decrease the port opening period, the distance from the inlet port to the center of the crankshaft was increased by inserting a packing of proper thickness between the cylinder and the crankcase. In order to avoid changing the timing of the scavenge and exhaust ports, the spacers of the same thickness as the packing were bolted to the piston crown facing the ports. Since it was not large, the change in the compression ratio did not seem to affect the motoring delivery ratio of the engine. The angle-areas for the various inlet port opening periods used in the experiment are shown in Table 2.

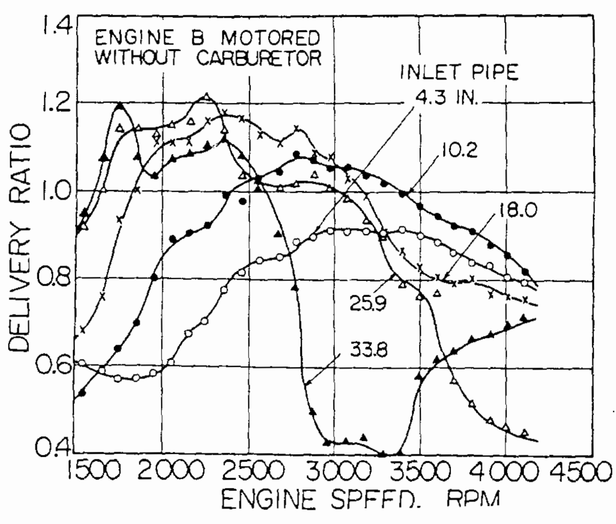


Fig. 15 - Effect of inlet pipe length

PRELIMINARY CONSIDERATIONS THROUGH THEORET-

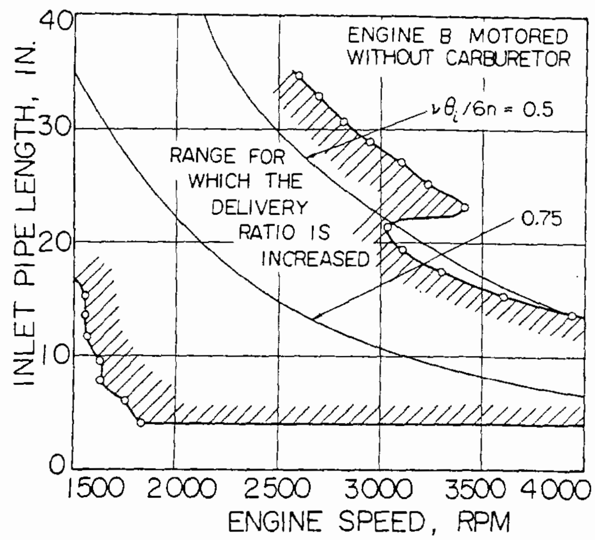


Fig. 16 - Range of inlet pipe lengths for which delivery ratio is higher than that for shortest length (4.3 in.)

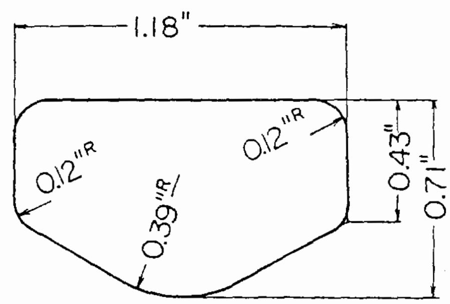


Fig. 17 - Engine B inlet port dimensions

Table 2 - Angle-Areas of Inlet Port of Remodeled Engine B

Period of Port Opening, deg	Angle-Area, $in.^2$-deg	Period of Port Opening, deg	Angle-Area, $in.^2$-deg
118	38.8	142	62.5
121	42.5	150	68.2
125	47.4	158	73.9
134	55.5	166	79.8

ICAL CALCULATIONS - Assuming the change of state in the crankcase to be adiabatic, replacing the flow state through the port with that through an equivalent port, and ignoring the dynamic effects of the exhaust and inlet pipes, the pressure change in the crankcase is given by Eqs. 8 and 9. Then, the delivery ratio is calculated by introducing the pressures at both inlet port closing and opening to Eq. 10. Taking $\mu_e = \mu_i = \mu_s = 0.8$, the pressure variation in engine B without the carburetor is calculated. A result is shown in Fig. 18 for 3000 rpm. In the case of a 100 deg inlet port opening period, the pressure in the crankcase at inlet port closing is lower than atmospheric due to the poor time-area. Consequently, the delivery ratio is low as shown in Fig. 19. In the case of a 110 deg period, the pressure reaches atmospheric before inlet port closing. After the pressure reaches atmospheric, air is discharged from the crankcase through the inlet port. Since the amount of discharged air is not large, the pressure at inlet port closing is the highest and the delivery ratio is also high.

For the 130 and 150 deg periods, the pressure is approximately atmospheric prior to inlet port closing. However, because a large amount of air is discharged from the crankcase due to the reverse flow through the inlet port, the delivery ratio again becomes low. Therefore, the most suitable period of inlet port opening, when the utilization of the inlet system effect is taken into consideration, seems to be closely related to the extent to which the reverse flow through the inlet port is prevented by the inlet system effect.

RELATIONSHIP BETWEEN EFFECT OF INLET PIPE LENGTH AND INLET PORT OPENING PERIOD - The motoring delivery ratio of remodeled engine B was measured without the carburetor. The results for the shortest possible inlet pipe (4.3 in.) are shown in Fig. 20. The delivery ratio at the lowest engine speed is low, because the time-areas of the inlet, scavenge, and exhaust ports are excessively large, and consequently both the outflow of the charge air from the crankcase through the inlet port and the reverse flow through the scavenge port are also large. The delivery ratio

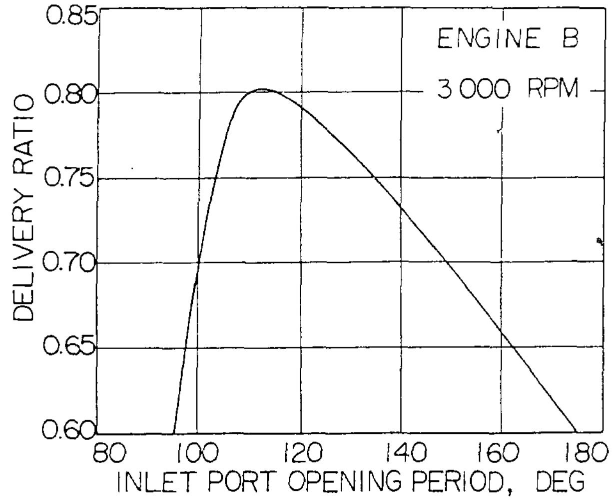


Fig. 19 - Effect of inlet port opening period (calculated)

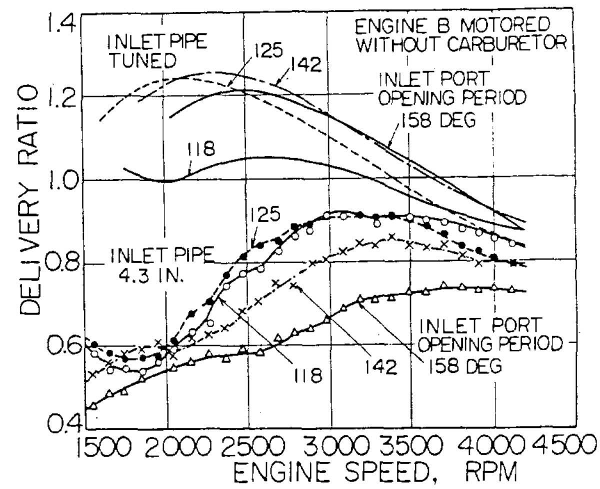
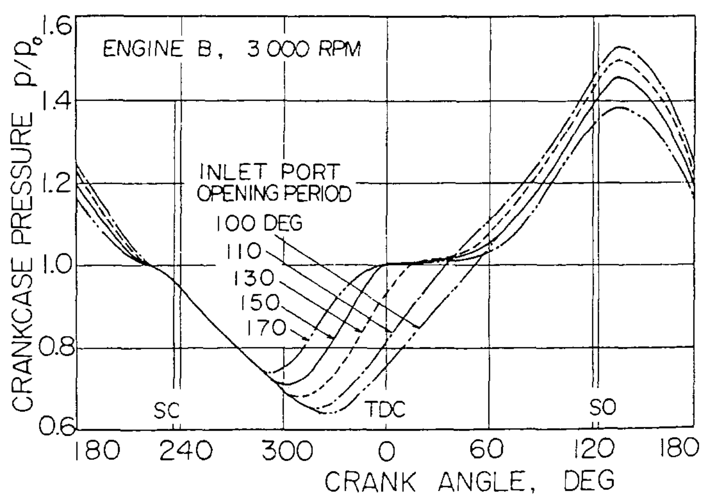


Fig. 20 - Effect of inlet port opening period



SC : Scavenge port closes. SO : Scavenge port opens.

Fig. 18 - Calculated variation of crankcase pressure

becomes higher with an increase in engine speed, reaching maximum. After attaining this maximum, it decreases with a further increase in engine speed due to the poor time-areas of the ports. The engine speed at which the delivery ratio becomes maximum increases with the prolongation of the inlet port opening period. However, this maximum delivery ratio does not always represent the highest at that engine speed, because the amount of air discharged from the crankcase before the inlet port closes increases with prolongation of the inlet port opening period.

In order to determine the relationship between the inlet port opening period and the effect of the inlet pipe, delivery ratios are shown in Fig. 21 for the shortest length (4.3 in.) and the most suitable length of the inlet pipe. For the 4.3 in. length, the delivery ratios at 3000 and 4000 rpm engine speeds become maximum at port opening periods of 120-125 deg or below. On the other hand, the delivery ratio at 2000 rpm becomes maximum near a 130 deg period. Since the delivery ratio in Fig. 20 shows a tendency to increase with a decrease in engine speed below 2000 rpm, the pressure fluctuation in the scavenge system seems to affect the delivery ratio in this range of engine speeds. If the effect of this fluctuation can be eliminated, the period at which the delivery ratio at 2000 rpm reaches maximum may become shorter than that shown in Fig. 21.

The most suitable period of port opening for the tuned inlet pipe is longer than that for the shortest one. Therefore, it is evident that the effect of the inlet pipe can be fully utilized with an opening period longer than the usual one. The highest delivery ratio at each engine speed, which is obtained by changing the length of the inlet pipe from 4.3 to 35.7 in., is also shown in Fig. 20. There is no remarkable difference among the delivery ratios at the various periods of port opening except for the 118 deg period. For the 4.3 in. length, the delivery ratio for the 118 deg period is nearly equal to that for the 125 deg period. However, for the most suitable length, there is a considerable difference between the two delivery ratios. Fig. 22 shows the delivery ratio for the length selected so that it will be highest at 3000 rpm. The delivery ratio for the 125 deg period is higher than that for the 118 deg period.

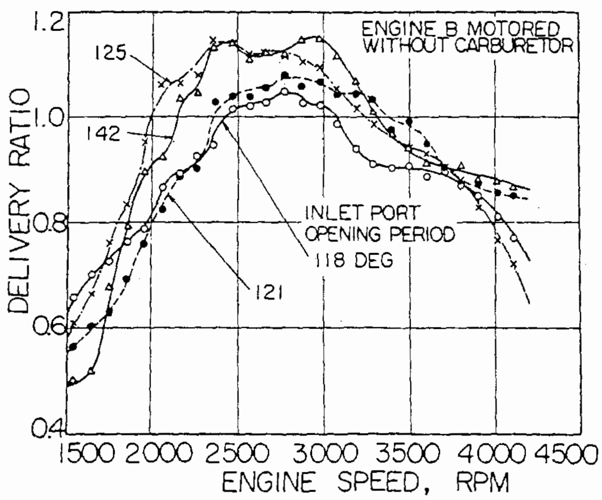


Fig. 22 - Delivery ratio for inlet pipe selected to produce highest possible delivery ratio at 3000 rpm

FLOW RESISTANCE OF CARBURETOR AND AIR CLEANER

The flow resistance of the carburetor, which is fixed to the external end of the inlet pipe, diminishes the effect of the inlet pipe. The delivery ratios in Figs. 23 and 24 were measured by first installing only the carburetor, second only the air cleaner, and, last, both the carburetor and air cleaner. The throttle valve of the carburetor was fully opened, and the length of the inlet pipe was represented by the distance from the cylinder wall to the inlet of the carburetor or the air cleaner. Comparing the delivery ratio in Fig. 23 with that in Fig. 15, it is seen that only the engine speed giving a maximum delivery ratio is varied by the effect of the inlet pipe equipped with the carburetor, the maximum de-

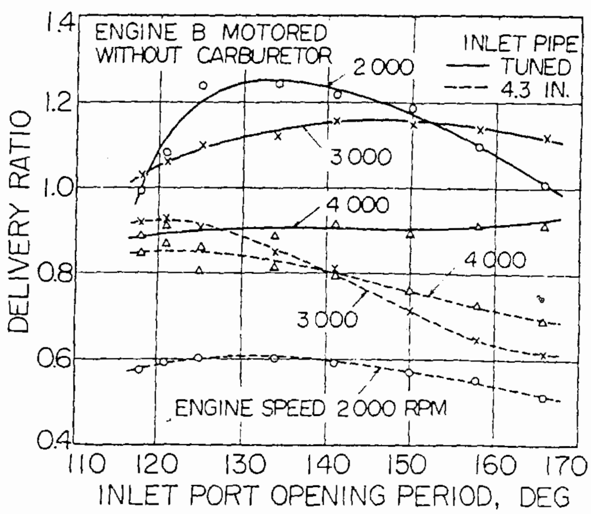


Fig. 21 - Effect of inlet port opening period

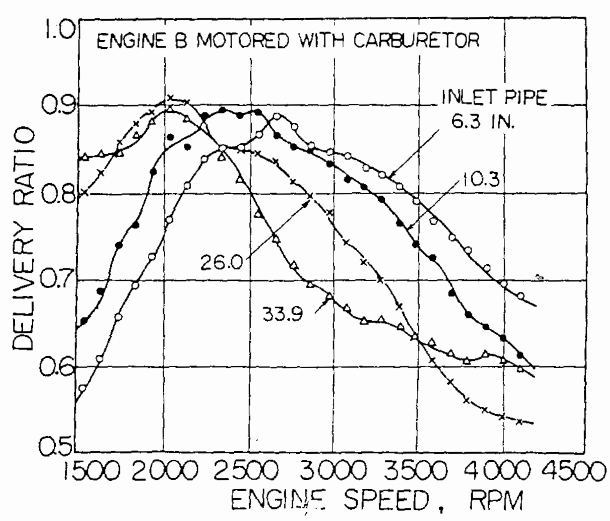


Fig. 23 - Effect of inlet pipe length with carburetor installed

livery ratio, however, being scarcely improved. Fig. 24 shows the delivery ratio by taking the pipe length as the abscissa. The air cleaner was an oil bath type and its pressure loss was about 2 in. H_2O at 3600 rpm.

Replacing the flow state in the carburetor and the air cleaner with that of an equivalent orifice, the influence of their flow resistance may be easily investigated. The orifice equivalent to the carburetor is fixed to the external end of the inlet pipe, and its area is made equal to the throat area of the carburetor. The orifice equivalent to the air cleaner is fixed to the external end of the inlet pipe, or to the inlet of the carburetor, through a chamber which has a volume equal to that of the air cleaner between its outlet and the oil surface. The pressure loss in the orifice should be equal to that in the air cleaner.

The rarefaction wave produced at the inlet port propagates in the inlet pipe and is reflected at the pipe end, that is, the orifice. The influence of the orifice on the ramming effect of the inlet pipe depends on the pressure of the reflected wave propagating towards the inlet port. With an increase in the orifice area, the pressure of the reflected compression wave becomes higher and, consequently, the effect of the inlet pipe becomes more remarkable. Ignoring the friction loss and the change of entropy in the inlet pipe, the pressure p_5 of the reflected wave produced by the rarefaction wave of the pressure p_3 is given by

$$\phi\sqrt{1-(P_3+P_5-1)^2}\left\{1-\frac{2}{\kappa-1}(P_5-P_3)^2\right\} - \sqrt{\frac{2}{\kappa-1}}\,(P_3+P_5-1)^2\,(P_5-P_3) = 0 \qquad (14)$$

where:

$P = (p/p_0)^{(\kappa-1)/2\kappa}$

ϕ = Area ratio of orifice to inlet pipe (throttle ratio)

Subscripts 3, 5 = States of rarefaction wave and its reflected wave, respectively

The derivation of the above equation is described in the Appendix. Fig. 25 gives an example of the calculation for $\kappa = 1.4$, showing the relation between the throttle ratio and the pressure of the reflected wave. There is no improvement in the delivery ratio under the condition $p_5/p_0 < 1$, where the pressure of the reflected wave is below atmospheric.

The delivery ratio was measured by fixing an orifice to the external end of the inlet pipe. The results are shown in Fig. 26. The degree of improvement on the delivery ratio by the tuned inlet pipe becomes smaller with a decrease in throttle ratio. The range of throttle ratios, in which an improvement in delivery ratio is not found, coincides with the range giving in Fig. 25 the pressure p_5/p_0 of the reflected wave less than 1. The area ratio of the carburetor throat to the inlet pipe of engine B is 0.38. It is evident from Fig. 26 that the effect of the inlet pipe is nullified due to the excessively small area ratio. The delivery ratio of

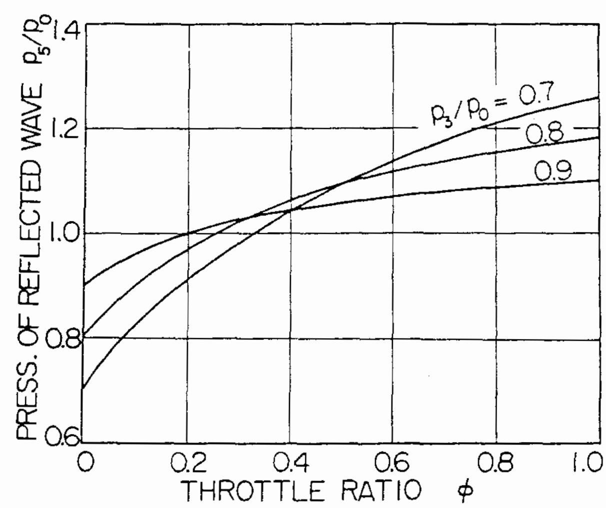


Fig. 25 - Pressure of wave reflected from orifice (calculated)

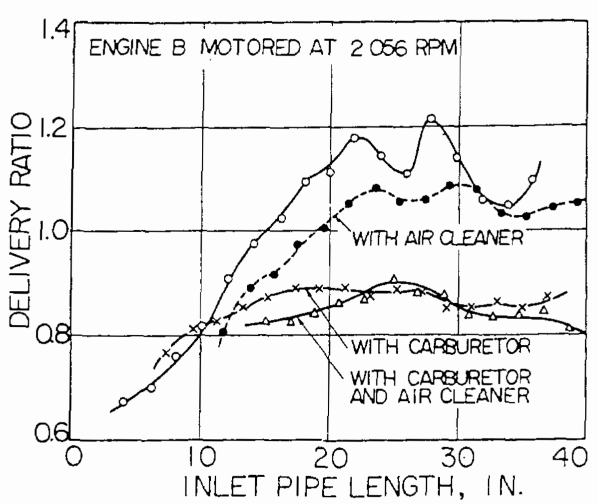


Fig. 24 - Effect of carburetor and air cleaner

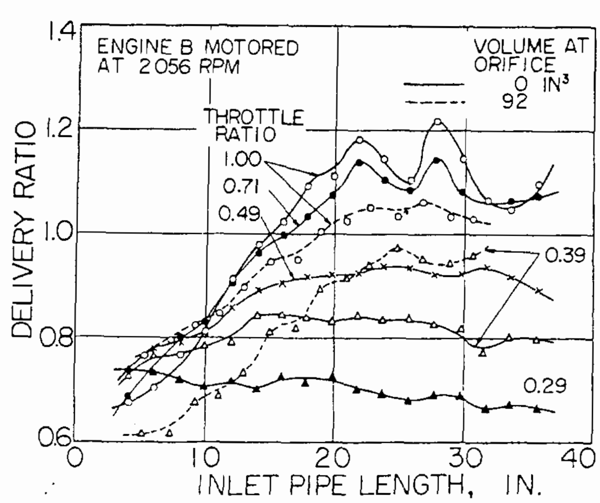


Fig. 26 - Effect of orifice fixed to external end of inlet pip

engine A described in terms of crankcase volume is fairly well improved by the inlet pipe with a carburetor, because the area ratio of the throat to the inlet pipe is 0.62.

In the case of a small throttle ratio, the delivery ratio is improved by putting a chamber between the orifice and the inlet pipe. The dotted lines in Fig. 26 show the results obtained by the use of a chamber having a volume that is ten times the piston displacement. The area of the equivalent orifice of the air cleaner, determined by measuring the pressure loss under steady-state conditions, is 0.59 sq in. and consequently the throttle ratio is 0.72. Thus, the air cleaner is hardly related to the effect of inlet pipe. The influence of the flow resistance of the carburetor varies with its position.

Fig. 27 shows the delivery ratio at various positions of the orifice in the inlet pipe. The position is shown by the ratio x/1 of the distance x between the cylinder wall and the orifice to the length of inlet pipe 1. The decrease in delivery ratio can be kept small by installing the orifice near the inlet port. If the carburetor is fixed near the inlet port, the fuel-air ratio is greatly affected by the pressure fluctuation in the inlet pipe. It is then necessary to use a carburetor that is insensitive to the pressure fluctuation. The change in fuel-air ratio due to the pressure fluctuation can be avoided by fitting the carburetor to the open end of the inlet pipe. However, there is then a large decrease in the delivery ratio due to its flow resistance.

CONCLUSIONS

The effect of the crankcase volume and the dimensions of inlet system on the delivery ratio of crankcase scavenged two-stroke cycle engines was experimentally investigated with consideration given to utilizing the dynamic effect of the exhaust and inlet systems. The results obtained may be summarized as follows:

1. For given port areas, there is an engine speed which results in maximum delivery ratio. This engine speed decreases with an increase in crankcase volume; however, the maximum value of the delivery ratio is scarcely affected by the crankcase volume.

2. The drop in delivery ratio caused by an increase in crankcase volume is fairly well compensated for by properly tuned exhaust and inlet pipes. It seems that little advantage is obtained by making the crankcase volume excessively small.

3. The effect of the exhaust pipe is hardly disturbed by the time-areas of the ports. The effect of the inlet pipe becomes more remarkable with increases in the time-areas, expecially the time-area of the inlet port. Therefore, to improve the delivery ratio by tuning the exhaust and inlet pipes, the time-areas of the ports should be made far larger than those which would be proper under the condition of the shortest exhaust and inlet pipes.

4. When a short inlet pipe is fixed to the engine, the delivery ratio can be improved over a wide range of engine speeds. By selecting the pipe length so as to make three-quarters of the period of pressure fluctuation coincident with the period of inlet port opening, the delivery ratio is most remarkably improved by the utilization of the pressure fluctuation. When the number of pressure fluctuations during the inlet port opening is below 0.5, the delivery ratio decreases with the installation of an inlet pipe.

5. The influence of the carburetor and the air cleaner, which are fixed to the end of the inlet pipe, can be investigated by replacing them with an equivalent orifice. The effect of the inlet pipe is nullified at throttle ratios below 0.3-0.4.

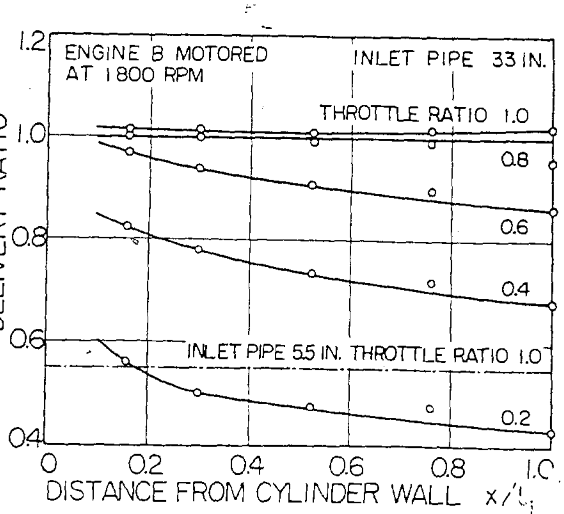


Fig. 27 - Effect of orifice position

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SYMBOLS

f_c = Throat area of carburetor

f_e, f_i, f_s = Areas of exhaust, inlet and scavenge ports, respectively

F_i, F_s = Areas of equivalent orifices for inlet and scavenge ports, respectively

g = Gravitational acceleration

G = Weight of air in crankcase

l_o = Delivery ratio

n = Engine speed, rpm

p = Pressure in crankcase

p_o = Atmospheric pressure

p_1, p_2 = Pressures in crankcase at inlet port closing and opening, respectively

t = Time, sec

T = Absolute temperature of air in crankcase

V = Crankcase volume

V_1 = Crankcase volume at inlet port opening or closing

V_c = Clearance volume of crankcase

V_s = Piston displacement

γ_o = Specific weight of air at atmospheric pressure

μ = Flow coefficient

μ_c, u_e, μ_i, μ_s = Flow coefficients of carburetor, and exhaust, inlet, and scavenge ports, respectively

κ = Ratio of specific heats

θ = Crankangle, deg

APPENDIX

When the rarefaction wave produced at the inlet port propagates in the inlet pipe and is reflected at the orifice fitted to the pipe end, the pressure of the reflected wave can be theoretically calculated as follows.

The friction loss and the change of entropy in the inlet pipes are ignored to simplify the calculation. Assuming that the equations of the flow state in the orifice are represented by those of steady flow, and that the pressure in the orifice is equal to that at the end of the inlet pipe, the velocity of air in the orifice is given by

$$u_t = \sqrt{\frac{2}{\kappa - 1} \left\{ 1 - \left(\frac{p_4}{p_o}\right)^{(\kappa - 1)/\kappa} \right\}}\, a_o \qquad \text{(A-1)}$$

where:

a_o = Sonic velocity of air at atmospheric pressure

u = Velocity

Subscript 4 = State at end of inlet pipe

Subscript t = State in orifice

The specific weight of air in the orifice is given by

$$\gamma_t = \gamma_o \, (p_4/p_o)^{1/\kappa} \qquad \text{(A-2)}$$

where:

γ = Specific weight of air

The equations of energy and continuity between the orifice and the end of the inlet pipe are given by

$$\frac{\kappa}{\kappa - 1} \frac{p_o}{\gamma_o} = \frac{\kappa}{\kappa - 1} \frac{p_4}{\gamma_4} + \frac{1}{2g} u_4^2 \qquad \text{(A-3)}$$

$$\phi \, \gamma_t u_t = \gamma_4 u_4 \qquad \text{(A-4)}$$

where:

ϕ = Area ratio of orifice to inlet pipe (throttle ratio)

Eliminating u_t, γ_t, γ_o, and γ_4 from both $\kappa g p_o/\gamma_o = a_o^2$ and Eqs. A-1 through A-4, the boundary condition at the orifice is given by

$$\phi \sqrt{\frac{2}{\kappa - 1} (1 - P_4^2)} \quad P_4^{-2} = U_4 \Big/ \left(1 - \frac{\kappa - 1}{2} U_4^2\right) \qquad \text{(A-5)}$$

where:

$$P = (p/p_o)^{(\kappa - 1)/2\kappa}$$

$$U = u/a_o$$

The following relations induced from the characteristic equations (9) exist for the pressure p_3 of the rarefaction wave propagating towards the orifice, the pressure p_5 of the reflected wave, and the pressure p_4 and the velocity u_4 at the end of the inlet pipe:

$$P_4 = P_3 + P_5 - 1 \qquad (A-6)$$

$$U_4 = \frac{2}{\kappa - 1} (P_5 - P_3) \qquad (A-7)$$

Substituting the above two equations in Eq. A-5, we obtain Eq. 14:

$$\phi\sqrt{1 - (P_3 + P_5 - 1)^2 \left\{1 - \frac{2}{\kappa - 1}(P_5 - P_3)^2\right\}} - \sqrt{\frac{2}{\kappa - 1}}(P_3 + P_5 - 1)^2 (P_5 - P_3) = 0 \qquad (14)$$

DISCUSSION

P. H. SCHWEITZER
Schweitzer and Hussmann Engineering

THE VALUABLE RESEARCH of Profs. Nagao and Shimamoto has taught us that the air charge of a crankcase-scavenged engine depends on its porting, its crankcase volume, and on the tuning of its inlet and exhaust system.

To recapitulate the main points:

For a given set of ports, small crankcase volume is best for high engine speed and large crankcase volume for low speed. Explanation: With small volume and low speed, the crankcase discharges too fast into the cylinder and during the rest of the transfer-port-open period there is reverse flow.

Even in a nontuned engine (short inlet and exhaust pipes) there is an optimum engine speed for a given set of porting. When the speed is higher than optimum, the port-time-areas do not allow sufficient fillup of the cylinder; if the speed is lower than optimum, there is a reverse flow by the now excessively large port-time-area.

The optimum engine speed n_o changes nearly in inverse proportion to the square root of the clearance volume. If the speed is so selected as to best suit the clearance volume, the clearance volume does not affect the delivery ratio. Tuning either the inlet or the exhaust pipe or both increases the delivery ratio.

If the exhaust pipe alone is tuned correctly, the delivery ratio increases some 20% both at n_o and also at other speeds. If the inlet pipe alone is tuned, the gain in delivery ratio increases with decreasing speed ($n_o/n > 1$) and reaches maximum at $n_o/n = 1.5$, where the engine speed is 2/3 of the optimum for a nontuned engine. This means that after the clearance volume has been selected to suit a given engine speed, and we then tune the intake, the optimum speed will go down to 2/3 of the original speed. When both intake and exhaust are tuned, the same is true, but the gain is further increased and the delivery ratio reaches 1.14 at $n_o/n = 1.5$.

At a fixed speed intake tuning brings more profit at small crankcase volume; exhaust tuning at large crankcase volume. Obviously a large crankcase volume neutralizes (dissipates) the ramming effect, but does not affect exhaust tuning, because the cylinder is in-between.

A diffuser helps the tuned exhaust in the case of shortest intake but not in the case of the tuned intake.

The pressure records show that inlet tuning helps build up pressure in the crankcase through ramming, but does no good to crankcase vacuum; exhaust tuning enhances the crankcase vacuum by the blow-down effect on the cylinder, but leaves the crankcase pressure alone. All effects diminish with increasing speed. A tuned engine is better at all speeds.

The designer wants to make the most of this wealth of information. How shall he proceed?

After selecting bore, stroke, and rpm, he may use the curves in Fig. 3 and select the proper crankcase volume, which gives the highest delivery ratio. After this he wants to lay out the porting with the intention of taking advantage of inlet tuning, or exhaust tuning, or both.

With nontuned inlet and exhaust the specific inlet, exhaust, and blow-down time-areas (defined as: time-area/displacement) have certain optimum values for engines of various speeds, sizes, and of a given general classification. To use the examples of engines A and B of Table 1, they have specific inlet time-areas of 373 and 243×10^{-6} sec/in., respectively.

Do you regard specific time flow areas a useful index in porting layout? If so, how much should the specific inlet time-area be increased if inlet tuning is most profitably to be utilized?

Fig. 20 indicates that for 3600 rpm 142-158 deg inlet port period was best for engine B. A 150 deg port opening corresponds to 20% mean port height and the original 125 deg to 14.5% mean port height. This would indicate 38% increase is needed for tuning. Is this about right? Seeing the amazing increase in delivery ratios with proper tuning, no designer will want to forego such a performance gain.

A well-tuned exhaust, I understand, calls for adequate blowdown time-area. For untuned engines I have found 8 to 10×10^{-6} sec/in. gives good results. What should the figure be for tuned exhaust systems?

Since playing with pipe lengths is relatively inexpensive, but with porting less so, the designer would like to start out with a porting which has to be changed little or none during the experimental development. Your guidance in porting layout, therefore, is of great value.

I note that your recommendation for best exhaust pulse frequency agrees with my own if the duration of the port opening is 120 deg.

What is your opinion on the merit of rotary valves and reed valves compared to port-controlled inlet ports? What is your opinion of the so-called fluid diodes?

Noting the authors' advanced work, and the spectacular recent success of the Japanese motorcycle industry, it is obvious we can learn much from the Japanese.

DANIEL S. SANBORN
McCulloch Corp.

THE AUTHORS ARE TO BE commended for the thoroughness with which they have investigated such an extensive range of parameters, and for the consistency of the results that they have achieved.

It is obvious that the full story of this very comprehensive analytical and experimental study is too long for convenient presentation in a single paper of this type. The authors' speculations on the reasons for, and factors controlling, the trends that have been so completely documented could well be the subject of a supplementary paper.

Good correlation of airflow measurements as obtained from a sharp-edge orifice, with those obtained from the sinking tank, indicate that the surge tank in the inlet system is effective in minimizing pressure fluctuations at the entrance to the engine inlet pipe. Were any comparisons made (with open exhaust, for instance) to verify that the expansion chamber, shown in Fig. 1, was large enough so that the exhaust pipe was discharging into an essentially constant pressure?

A theoretical study by this discusser of a simplified system, neglecting duct friction, utilizing the "Method of Characteristics," indicated that there was a distinct inlet duct area for optimum augmentation of the delivery ratio. Even if extrapolation was made for comparison at optimum engine speed, it appeared that duct area might be as important as length in realizing the full potentials of inlet ramming. Optimum duct area is undoubtedly related to the inlet port timing and area that can be utilized and may be restricted by venturi size requirements, if a conventional carburetor is used. It would be interesting, however, to know if the authors considered varying this parameter in their investigation.

The authors state that a significant portion of the pressure fluctuation generated by one blow-down pulse continues in the exhaust system until the next cycle. They imply that the successive pulses must be in phase with one another, so that the "carry-over" from one will reinforce the next for optimum effect. This condition would require an integral number of pressure fluctuations during one engine cycle (corresponding to integral values of the parameter $15\ a_e/nl_e$ of Fig. 12). The data presented in Fig. 12, however, appear to show little effect of this parameter; in fact, the results for engine speeds of 3200 and 2400 rpm indicate highest values of delivery ratio at $15\ a_e/nl_e$ equal to 2.5 and 3.5 where any "carry-over" pressure fluctuation would be expected to diminish the effect of the following blow-down. It is submitted, for the authors' consideration, that the phasing of the reflected blow-down pulse with the exhaust flow and port opening area of the same cycle is the only consideration of great importance. If the reflected pulse aids the scavenging of one cycle, it would be expected to give up energy and be attenuated by this and subsequent friction losses by the time of the next engine cycle. Nothing will be gained if the pipe is so short that the negative reflection gets back to the exhaust port while its flow is still supercritical. Also, the lowest pressure should appear at the port when it has its maximum area (bdc) so that the greatest benefit to the exhaust flow will result. It is considered just a coincidence that, because of normal port timings, this results in approximately three pressure fluctuations per engine cycle. The gain in delivery ratio fortunately occurs over quite a wide speed range, with fixed geometry, as can be inferred from Fig. 12. An additional bonus results from this arrangement: If the negative pressure peak reaches the port near bdc, what remains of the following positive reflection will reach the exhaust port about the time the transfer port closes and can be used to oppose the subsequent loss of charge from the exhaust, or even to return short-circuited charge to the cylinder as supercharge. This will not increase the delivery ratio, but can favorably affect the power output and specific fuel consumption.

It would be interesting to know if any attempts were made to take low pressure indicator diagrams of the transient pressure in the inlet duct at the port. This would substantiate the use of mean crankcase volume and neglect of variable inlet port area in determining the acoustic frequency of the inlet system in the authors' Eq. (12). Comparison of this transient inlet pressure with the resultant delivery ratio would give a clearer insight into the test results presented by Fig. 13.

It is not clear whether the carburetor was installed during the motoring tests which are compared with the firing tests in Fig. 14 to show the effect on delivery ratio. Also, the exhaust pipe length used is not specified. This comparison appears to show a favorable effect of firing at low speed where sufficient exhaust lead is available for the blow-down to be sufficiently complete before there is much transfer port opening. Back-flow into the crankcase is thus minimized, and a subsequent negative reflection from the end of the exhaust pipe can assist the scavenging. Increase in exhaust lead would be expected to raise the cross-over point above the 2000 rpm speed shown in Fig. 14.

The data plotted in the lower left area of Fig. 16 are quite confusing. This entire boundary appears to result from the upturn of the delivery ratio for the 4.3 in. pipe length, as shown in Fig. 15. It would be very illuminating to know if this length is truly unique, or if the results are influenced by other factors of geometry resulting from limitations of the test installation. Do the longer pipe lengths exhibit a corresponding upturn in delivery ratio at speeds below 1700 rpm, where the 4.3 in. length begins to show improvement? Also, if the pipe area were optimized, is it anticipated that these trends would change significantly? The discontinuity in the upper boundary shown in Fig. 16 appears to result from characteristics of intermediate pipe lengths which are not included in Fig. 15. Even more interesting, however, is the increase in delivery ratio of the long (33.8 in.) pipe which appears at high (3500) rpm in Fig. 15. Do the authors have any speculations on the cause of this unexpected upward trend and on its limitations?

Fig. 25 shows the theoretical effect of the carburetor throat area in diminishing the favorable reflected positive pressure wave from the end of the inlet pipe. Fig. 24 shows how seriously the carburetor negates the improvement that can be achieved by tuning the inlet system. Fig. 27 indicates that this adverse effect can be minimized by positioning the carburetor close to the inlet port. This position would have the added advantage of reducing "spit-back" of fuel that may take place at off-optimum operating conditions. The authors issue a warning, however, regarding the problem of severe pressure fluctuations influencing control of fuel-air ratio. Presumably, the carburetor throat area is dictated by the venturi depression required to achieve good carburetor control. It would be interesting, with the carburetor at the open end of the pipe, to reduce the pipe area to approach that of the carburetor throat. With compensation of pipe length for reduced area (to maintain the same pressure phase relations at the port) it is anticipated that the improvement in the reflection coefficient, according to Fig. 25, would much more than offset the additional friction losses incurred by the smaller pipe. Alternate solutions are represented by injection carburetors, variable venturi throats, or direct fuel injection, with its other significant advantages.

The basic attraction of the small two-stroke cycle engine is its high ratio of output-to-mechanical complication, leading to a low cost per horsepower. The possibilities of performance improvement (by minor modification of existing hardware) that have been demonstrated by this very comprehensive experimental program cannot long be ignored by the engine manufacturer; nor can the two-stroke engineer avoid being inspired to further investigation and development along these lines.

C. FAYETTE TAYLOR
Massachusetts Institute of Technology

THIS PAPER IS the most complete and scholarly piece of work I have seen on the subject of air capacity of crankcase-scavenged two-cycle engines. It is most interesting, and of great practical value to the designer.

The use of dimensionless ratios is greatly to be commended, as far as it goes. The paper would be even more valuable and easier to apply if pipe lengths, pipe and port areas, and engine speeds had also been presented in dimensionless terms. For these I would suggest:

1. Area/piston area.
2. Length/piston stroke.
3. Piston speed/acoustic velocity.

Use of these ratios would relieve the results of their dependency on particular cylinder dimensions (see presentation of similar work on four-cycle inlet pipes (1, 2)).* For those who still prefer to see actual lengths, areas, and rpm, there is no harm in presenting these values as well.

I could not find the authors' definition of delivery ratio in the paper. I presume it to be the SAE definition, based on piston displacement.

With regard to air capacity, since the crankcase of the two-cycle engine used has many points of similarity with the cylinder of a four-cycle engine, it should be interesting to compare the authors' study of inlet pipe tuning with a similar study made at M.I.T. on a four-cycle cylinder (Ref. 1, pp. 260-261, and Ref. 2, pp. 196-199). Their observation that the pressure fluctuations left in the inlet pipe after inlet closing have little effect on the delivery ratio agrees with our findings. Another agreement is that with long inlet pipes, delivery ratios in excess of one are possible.

On the other hand, we found that the peak of the delivery ratio curve occurred at different speeds with inlet pipes of the same length but of different diameters. If a similar relation holds for the two-cycle engine, the curves shown in Fig. 13 would peak at a different number of fluctuations with another pipe diameter.

The four-cycle engine is much less sensitive to exhaust pipe tuning than the two-cycle engine, because the exhaust process is much less involved with the intake process. I have not seen a careful study of exhaust-pipe effects on four-cycle engines, although I am aware that racing cars have found tuned exhaust pipes helpful.

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*Numbers in parentheses designate References at end of discussion.

A change in crankcase volume ratio for the two-cycle engine would correspond with a change in compression ratio for a four-cycle engine, as far as the inlet process is concerned. In the case of unsupercharged four-cycle engines, differences in compression ratio have very small effects on delivery ratio. However, it must be remembered that the ratio of clearance volume to piston displacement is one to two orders of magnitude smaller in the case of the four-cycle engine.

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AUTHORS' CLOSURE TO DISCUSSION

THE AUTHORS wish to thank the discussers for their fine comments and their interest in this paper. We would like to express our special recognition to Prof. Schweitzer for his relevant recapitulation of the paper.

As to the questions raised by Prof. Schweitzer, the specific time flow areas as suggested are useful index in porting layout. Since flow conditions in crankcase-scavenged engines are greatly affected by the crankcase volume, the optimum values of specific time-areas for a given engine speed may have to be determined from consideration of the crankcase volume. The specific inlet time areas of engines A and B are 333 and 243 x 10^{-6} in.^{2}-sec/in.^{3} at 3600 rpm, respectively, while the ratios of the crankcase clearance volume to the displacement are 2.44 and 1.96, respectively. Supposedly, the larger crankcase volume needs the larger specific time-area. If the inlet port timing is kept constant, the optimum value of specific inlet time-area for inlet tuning is probably one and a half times larger than that for a nontuned engine, because the delivery ratio on the condition of inlet tuning becomes maximum at $n_o/n = 1.5$. Even if the inlet port opening period is increased to make the time area large, the optimum value of inlet time area may be little changed from the above optimum value by the estimation with Fig. 20.

The mean inlet port heights of engine B are 14.5% at the original opening period of 125 deg and 17.5% at 150 deg opening period, respectively. Then, 20% increase in the mean port height is needed for the best tuning at 3600 rpm. In this experiment the inlet time area was increased by cutting the piston skirt, leaving port dimensions intact. As shown in Fig. A, the maximum area of inlet port was kept constant in spite of the increase in the port opening period. When the inlet port height is also made larger, the necessary time-area will be given at the opening period shorter than that in the experiment.

The adequate valve of blow-down time area for exhaust tuning cannot be estimated from our experiment. As shown in Fig. 6, however, the increase in the delivery ratio by exhaust tuning is little affected by n_o/n, that is, the port time areas. The adequate blow-down time area may not be much different from that for nontuned engines.

Mr. Schweitzer's notes on the best exhaust pulse frequency are appreciated. The authors wish to discuss this subject together with the questions of Mr. Sanborn.

The authors have not carried out investigation to compare the merit of rotary valves with those of piston-controlled inlet ports. The ramming effect of the inlet system may be most effectively utilized with rotary valves, the timing of which can be favorably selected.

By fitting a reed valve and at the same time closing the inlet port, the delivery ratio for the reed valve was compared with that for the piston-controlled port of the same engine. The dimensions of the reed valve were so selected that the delivery ratios for both types were equal at the highest engine speed in the nontuned condition. As shown in Fig. B, in the nontuned condition, the delivery ratio for the reed valve was higher than for the piston-ported valve at all range of engine speeds. Therefore, reed valves are advisable for engines with nontuned inlet systems. On the other hand, in the case of inlet tuning, the delivery ratio

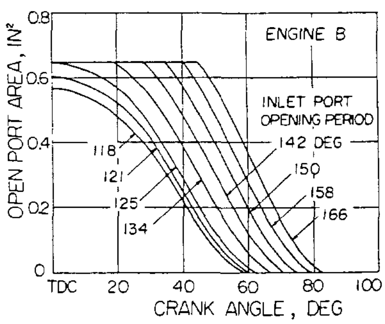

Fig. A - Open inlet port area of engine B

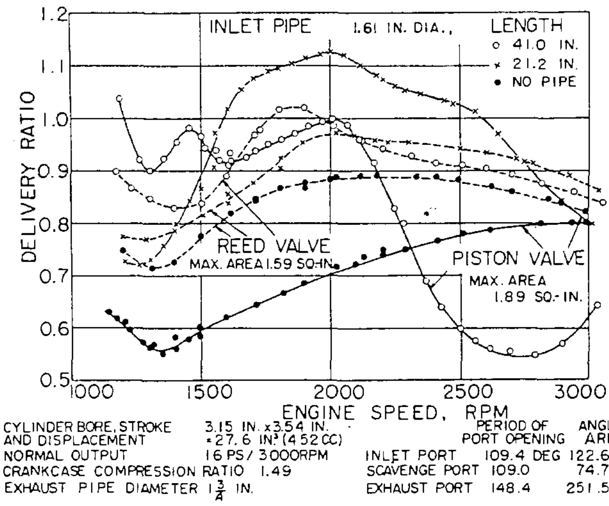

Fig. B - Comparison between reed valve and piston valve for the piston-ported valve was higher than that for the reed valve at all engine speeds. Reed valves are not suitable for utilizing the ramming effect of the inlet system because they cannot produce large pressure pulse in inlet systems.

Applying fluid diodes to engines is a very interesting and fine subject. Fluid diodes may be useful for preventing the reverse flow through the ports.

In the questions of Mr. Sanborn, the volume of the expansion chamber in Fig. 1 is two hundred times larger than the piston displacement of engine A. By discharging the exhaust gas into the expansion chamber and also directly into the atmosphere (open exhaust) the delivery ratios of engine A with a 6.9 in. long exhaust pipe were measured at various engine speeds. The difference between both delivery r' ɔs was less than 2%. Though the difference betweer h delivery ratios in the case of long exhaust pipe is u' wn, small pressure variation in the expansion chamber s not seem to affect the delivery ratio seriously.

ɛ inlet duct area could be an important parameter but v iot varied in the investigation, because rebuilding the ‡.. age near the inlet port was not easy. The effect of the inlet pipe diameter had been tested with another engine which had a reed valve or piston-controlled inlet port. As shown in Fig. C, the delivery ratio for the reed valve is scarcely affected by the pipe diameter, because the pressure pulse in the inlet pipe is too small to utilize the ramming effect. The delivery ratio for the piston valve shows different values for three different pipe diameters, that is, the delivery ratio for the 1.34 in. diameter pipe is far smaller than that for the 1.61 in., while the 1.89 in. shows in its maximum value nearly equal to that of the 1.61 in. If a pipe diameter larger than 1.89 in. is used, the maximum delivery ratio can be expected to be smaller. This is because the pressure fluctuation in the pipe will become presumably smaller by increasing the pipe diameter, even if the pipe length is made longer to keep the frequency of the inlet system constant. Therefore, the ramming effect will be diminished; then the optimum pipe diameter may exist as stated by the discusser.

In Fig. 12, the delivery ratios for engine speeds of 3200 and 2400 rpm indicate peaks at $15\,a_e/nl_e$ equal to 2.5 and

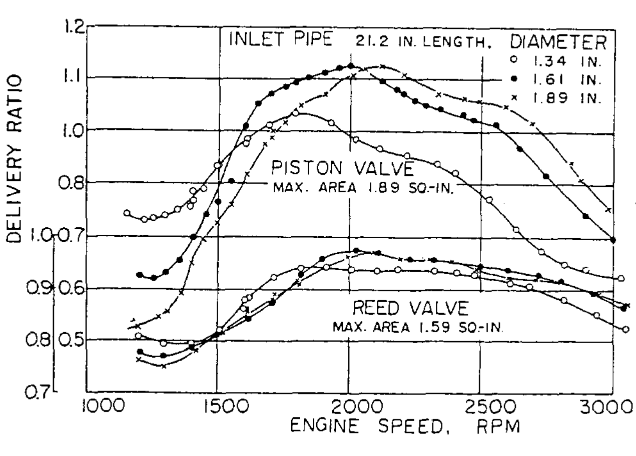

Fig. C - Effect of pipe diameter

3.5 as pointed out by the discusser. The occurrence of two peaks is supposedly due to the irregularity of the pattern of pressure fluctuation in the exhaust pipe. If small variations of delivery ratio are ignored, it can be considered that the delivery ratio becomes maximum at $15\,a_e/nl_e = 3$, that is, three pressure fluctuations per engine cycle. It is desirable for exhaust tuning to phase the reflected blow-down pulse with the exhaust period of the same cycle and, at the same time, to phase the remaining pressure fluctuation after exhaust port closing with the blow-down of the next cycle. However, both conditions cannot always be met satisfactorily at the same time. If the remaining pressure fluctuation after exhaust port closing is negligible, it should only be considered to phase the blow-down pulse with the exhaust period. In our experiment, the remaining pressure fluctuation did not seem to be small enough to affect the scavenging of the next cycle. Further, in engines of normal exhaust port timing, the exhaust pipe lengths which satisfy the two conditions as mentioned above, are not much different. Therefore, the authors preferred to arrange the experimental data with the pressure fluctuations in one engine cycle. At $15\,a_e/nl_e = 3$, the blow-down pulse is well utilized to the scavenging of the same cycle and the remaining pressure fluctuation is phased with the blow-down of the next cycle.

It is understood from the records of pressure in the inlet pipe at the port that the delivery ratio becomes maximum at the inlet pipe length which sets the phase of the peak of positive pressure at the time of inlet port closing. The calculated period of acoustic oscillation of the inlet system with the above-mentioned pipe length is about three quarters of the inlet period. Since the pattern of pressure fluctuation during the inlet period is not a sine wave, the accurate measurement of its frequency cannot be expected with the pressure records at the inlet port. Therefore, it is difficult from the pressure records to substantiate the use of mean crankcase volume and the neglect of variable inlet port area in calculating the acoustic frequency.

In Fig. 14, the motoring delivery ratio was compared with the firing one for the same inlet system using a carburetor. The exhaust pipe length used was 5.9 in. The discusser's opinions on the reason of the difference between both delivery ratios agrees with authors'.

In Fig. 15, the delivery ratio for the 4.3 in. pipe length shows an upturn trend at engine speeds below 2000 rpm. The longer pipe length exhibits the same trend at the lower engine speed. Also, the same trend appeared with another engine with a different test installation. The cause of the upturn trend could not be made clear.

The hatched area in Fig. 16 shows the range of inlet pipe lengths which give the delivery ratio higher than that for the 4.3 in. long pipe. The inlet pipes of various lengths were investigated. A few examples of many results obtained for them is shown in Fig. 15. The curves in Fig. 15, which give the relation between the delivery ratio and the engine speed, have small variations. The discontinuity in the upper boundary in Fig. 16 is due to those variations. The increase in delivery ratio for the 33.8 in. long pipe, which appears at 3500 rpm in Fig. 15, is produced by the pressure fluctuation which remains in the inlet pipe after port closing.

The carburetor throat at the open end of the pipe diminishes the ramming effect of the inlet pipe due to throttling. It may be possible to increase the delivery ratio by reducing the inlet pipe area, even if the friction loss is much increased. It is expected from Fig. 26 that the ramming effect is well utilized at the area ratio of the throat to the inlet pipe larger than 0.7. Inlet pipes of different diameter, incidentally, with an orifice near the port, were tested on another engine. As shown in Fig. D, reduction in diameter brings a gain of delivery ratio at low engine speeds. If the inlet pipe length is adjusted to keep the frequency of the inlet system constant, the decrease in the delivery ratio at high engine speeds may disappear or become less. If the effect of pressure fluctuation on the carburetor is left out of consideration, reduction of the pipe diameter by placing the carburetor near the port may be better for the delivery ratio.

Prof. Taylor's suggestion on the use of dimensionless ratios is greatly appreciated. We will attempt to rearrange our experimental data by the use of dimensionless ratios, as referred to in the papers concerning four-cycle engines.

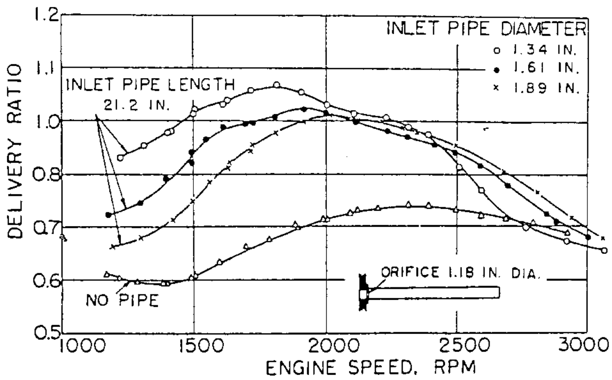

Fig. D - Effect of throttling

As to the delivery ratio, we followed the SAE definition as presumed. The delivery ratio is the ratio of delivered volume in atmospheric condition to piston displacement.

Prof. Taylor's comparison between four-cycle and two-cycle engines regarding inlet tuning is appreciated and very interesting. He has indicated many similar points, predicting some important characteristics which are not included in our experiment.