On the Effects of Intake or Exhaust Pipe System in a Crankcase-Compressed Two-Stroke Cycle Engine

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On the Effects of Intake or Exhaust Pipe System in a Crankcase-Compressed Two-Stroke Cycle Engine

Norihiro Sawa

Abstract

To examine the influences of an intake or exhaust pipe system on the delivery ratio in a small crankcase-compressed two-stroke cycle engine, the author has performed the theoretical analysis on the matching condition for the intake or exhaust pipe effect and then has measured the amount of the delivery ratio and analyzed some pressure indicator diagrams for the intake or exhaust pipe.

Some conclusions reached are summarized as follows:

a) In the extreme cases of a long intake pipe or high speed, the residual pulsation waves in the intake pipe have great influence on the delivery ratio and such a pulsation effect is generally governed by $q_i=15\ a_i/NL_i$.

b) The maximum delivery ratio occurs by the inertia effect due to the intake pipe and the matching condition is expressed by $1/Z_{iM}^2=(180/\theta_i^*)^2+U^2$.

c) The pressure wave remaining in a long exhaust pipe contributes to the scavenging action of the next cycle and such a pulsation effect is defined as $Q_e=(1+\theta_{es}^*)\cdot q_e$.

d) The maximum increase in the delivery ratio $(k-k_0)_M$ due to the exhaust pipe is obtained by exhaust blow-down waves and such a blow-down effect is given by $Z_e=\dfrac{4}{3}\cdot(\theta_{es}^*/360)$.

1. Introduction

It has well been known long since the fact that the breathing capacities of engine are largely governed by an intake or exhaust pipe system, and the author has also presented already a report[1] concerning the effects of intake or exhaust pipe length on the delivery ratio in a small crankcase-compressed two-stroke cycle engine.

Successively, in order to investigate systematically the influences of various engine factors on the delivery ratio, he has made a few theoretical analyses and then measured the delivery ratio in 2-stroke cycle one changing the various dimensions of the test engine over a wide range of engine operating condition, and some pressure diagrams in the intake or exhaust pipe were obtained mainly by a magnetic oscillograph to analyze the air flow in the suction or exhaust process.

2. Fxperimental apparatus and method

The test engine shown in Fig. 1 is a crankcase-compressed two-stroke cycle engine for a moter bicycle, the dimensions

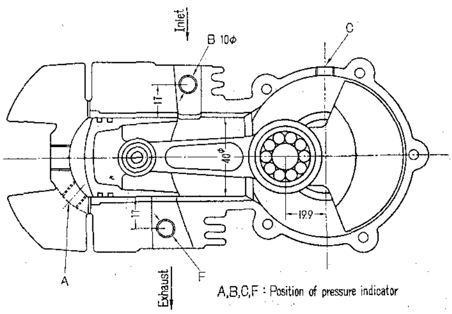

Fig. 1. Test engine.

Table 1. Dimension of test engine

			Engine-50	Engine-125
Cylinder bore×Stroke		(mm)	40 ϕ×39.8	55 ϕ×52.5
Stroke volume		(V_h cc)	50	125
Crankcase volume (at T.D.C.)		(V_k cc)	161	452
Mean volume of crankcase during inlet opening period		(V_{km} cc)	150	435
Compression ratio		(ε)	7 : 1	7 : 1
Sectional area of intake pipe		(A_s cm^2)	1.49	3.14
Port timing (symmetrical type)	Inlet		±60° (T.D.C.)	±70°
	Exhaust		±67° (B.D.C.)	±69.7°
	Scavenging		±55° (B.D.C.)	±57°
Scavenging type			Schnüle	Schnüle

of which are described in Table 1. In Fig. 2 which shows the general layout of experimental apparatus. a surge tank with a flow-meter of round nozzle type is connected to the intake pipe directly and then a carburetor is installed on the outside wall of the tank. Straight pipes with so many different kinds of length and diameter are prepared for intake or exhaust pipe system that we are able to get many pipe systems by combinating them.

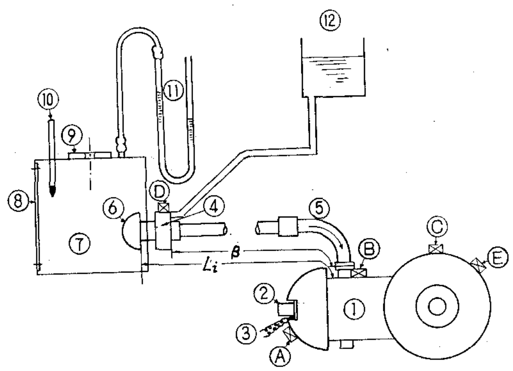

Fig. 2. General layout of experimental apparrtus.

① Test engine ② Spark plug ③ Thermocouple
④ Carburetor ⑤ Intake pipe ⑥ Air-cleaner
⑦ Surge tank ⑧ Rubber sheet ⑨ Flow-meter
⑩ Thermometer ⑪ Manometer ⑫ Fuel tank
Pressure indicator :
Ⓐ Cylinder Ⓑ Inlet port Ⓒ Crankcase
Ⓓ Carburetor Ⓔ Crank-mark

Considering the results of the previous experiments[1], all the tests regarding the intake pipe system are made in the motoring state and without the exhaust pipe to eliminate the influences of exhaust pipe on the delirery ratio. On the other hand, effects of the exhaust pipe system are examined in the firing state using the same test engine with intake pipe of shorter length.

All these tests are carried out with the engine speed changing from 1500 to 4500 rpm, i. e. the air amount to the engine is always measured by a flow-meter after the temperature of spark plug seat having reached a steady value, and to analyze the inatke or exhaust process in a two-stroke cycle engine, the pressure variations at such positions an inlet port, crankcase and exhaust port are picked up by means of each pressure indicator of the electric-capacity type as shown in

Fig. 3 and then are recorded on a magnetic oscillograph through several amplifiers of direct or alternating current type.

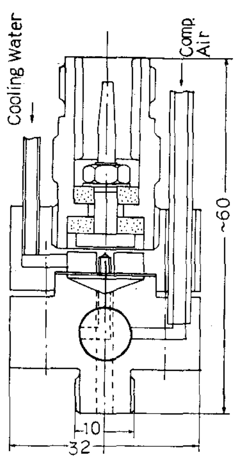

Fig. 3. Pressure indicator

3. Experimental results and considerations

3-1 On the Pulsation Effect due to the Intake Pipe

The pulsation effect due to the intake pipe in an engine are shown in Fig. 4 and 5, which illustrate a delivery ratio curve and pressure fluctuations at the inlet port respectively. From both the figures, it is seen that the delivery ratio has shown a tendency to incresse when

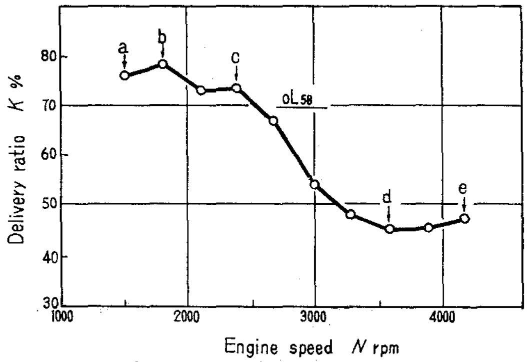

Fig. 4. Delivery ratio for pipe length ${}_0L_{58}$ (E-50)

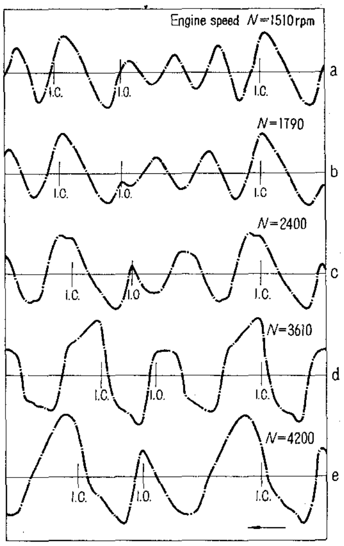

Fig. 5. Pressure dirgrams at the inlet port (E-50, ${}_0L_{58}$)

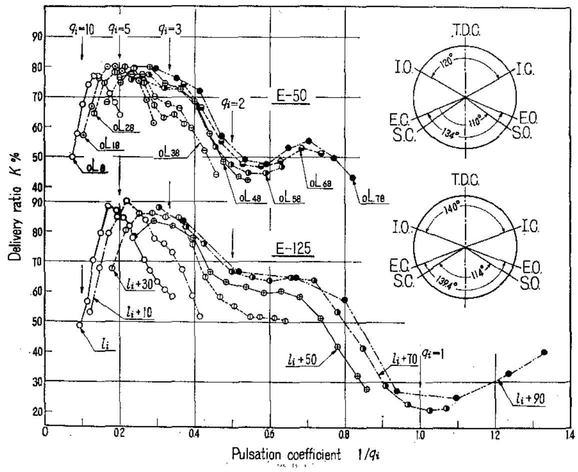

Fig. 6. Pulsation effect due to the intake pipe length (E-50, E-125)

the positive part of pulsation waves is superposed on the intake period and vice versa.

Such pulsation effect due to the intake pipe system is governed by the following expression[3].

$$q_i = (a_i/4L_i)/(N/60) = 15a_i/NL_i \tag{1}$$

Where q_i is pulsation coefficient, a_i : sonic velocity, L_i : length of intake pipe, N : engine speed.

As shown in Fig. 6, the delivery ratiocurves (K) drop down at at several positions of q_i having integer value, because in the case, a negative wave just arrives in the next intake period. Changing the diameter of intake pipe with a given constant length, it is evident from Fig. 7 that the increase of delivery ratio at

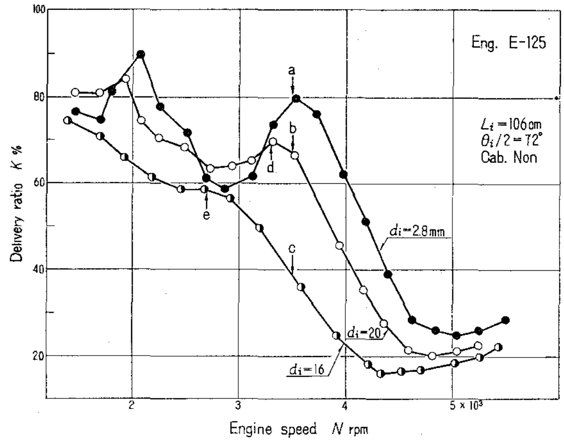

Fig. 7. Delivery ratio for each intake pipe diameter (E–125)

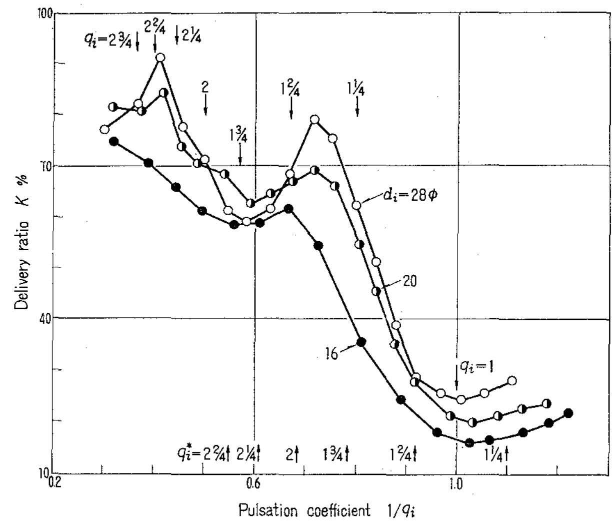

Fig. 8. Pulsation effect due to each intake pipe diameter (E–125)

nearly $N = 3000$ rpm is affected by the pipe diameter ; particularly, larger the pipe diameter, the higher the engine speed where the peak values of delivery ratio curve are obtained by the pulsation effect.

Rearranging these experimental results with the pulsation coefficient q_i instead of engine speed (N), all the peaks in the curves coincides with each other nearly at a constant value of $q_i\left(=1\frac{1}{2}\sim 1\frac{1}{4}\right)$ as shown in Fig. 8. Then the pulsation coefficient (q_i) seems to be an important parameter as a criterion showing whether the pulsation effect due to intake pipe system is positive or negative. Strictly speaking, however, a matching condition of the pulsation effect varies in accordance with the change of intake period (θ_i) as shown in Fig. 9. Considering the relation between the pulsation wave and the inlet timing as illustrated semantically in Fig. 10, these facts are understood as follows :

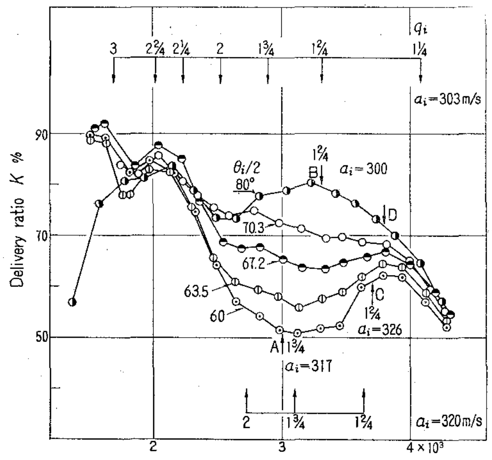

Fig. 9. Delivery ratio for each intake period (θ_i), (E–50)

a) In the case of $q_i=2$, the negative part of pulsation wave arrives at the timing of inlet opening and also its positive part enters successively during intake period, so that the pulsation wave effect is very small.

Since the delivery ratio is mainly governed by the first positive wave which just reachs the inlet port at the closure, the delivery ratio for a short intake period $\theta_i/2=60°$ is small and it can not be increased by the pulsation effect.

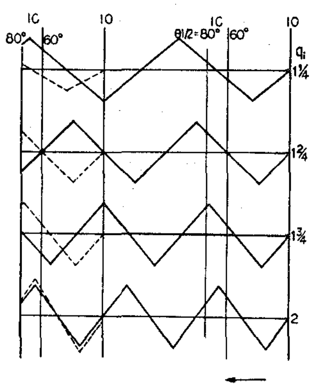

Fig. 10. Matching state of the pulsation wave

b) In the case of $q_i=1\frac{3}{4}$, a part of the positive waves reachs the cylinder at the inlet opening, but the negative wave also enters the cylinder during the latter half period of the intake process. In the case of short intake period ($\theta_i/2=60°$), negative pulsation wave affects remarkable on the delivery ratio because the inlet port closes entirely before the first positive wave is coming. On the other hand, when a long intake period ($\theta_i/2=80°$) the delivery ratio increase considerably by the first positive wave, which has sufficiently higher value at the inlet closure.

c) When $q_i=1\frac{1}{2}$, the delivery ratio for $\theta_i/2=60°$ is also small but can be increased due to the complet superposing of the residual positive wave upon the whole period of intake process. For $\theta_i/2=80°$, the delivery ratio is augmented by the positive wave superposed on the first half period of the intake process as the first positive wave is high.

d) In the case of $q_i=1\frac{1}{4}$, the delivery ratio for $\theta_i/2=60°$ is not so low, because of only a part of the positive pulsation wave superposing on the intake period in spite of the first positive wave; then it is seen that the delivery ratio increases, in general, at $q_i=1\frac{1}{2}\sim1\frac{1}{4}$.

From the preceding considerations, the matching conditions for the pulsation wave remained in the intake pipe system are also dominated by the so-called inertia effect due to the inlet pipe.

3–2 On the Inertia Effect due to the Intake Pipe

The fact that the maximum delivery ratio occurs mainly by the inertia effect due to the intake pipe system is well known and has already reported by the author[4]. Now he applys the approximate inertia theory proposed by Professor T. Asanuma[5] to the intake pipe system of the crankcase-scavenged two-cycle engine as shown in Fig. 11 and exmines the conditions occurring the maximum delivery ratio through the inertia effect and obtaines the following relation;

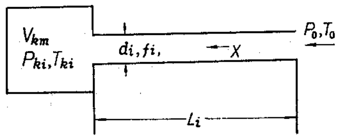

Fig. 11. Intake pipe system

$$1/Z_{iM}^2=(180/\theta_i^*)^2+U^2 \tag{2}$$

Where $Z_{iM}(\equiv \omega L_i^*/a_i)_M$ is an inertia coefficient where the maximum delivery ratio is given, θ_i^* is the effective intake period and L_i^* is the equivalent length of inlet pipe. U is coefficient of flow reistance per air column in intake pipe, which is calculated from the following equation[1]

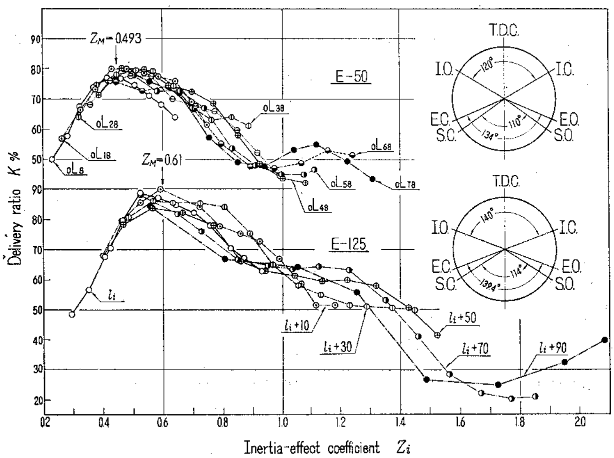

Fig. 12. Inertia effect due to the intake pipe (E- 50, E-125)

$$U = \frac{V_h}{4f_iL_i}\cdot\frac{dK/d\theta}{\beta^2(\theta)}\cdot(\lambda_1+\lambda_2+xL_i/d_i+\lambda_3) \tag{3}$$

Where V_h : stroke volume, f_i, d_i, L_i : sectional area, diameter, length of intake pipe respectively, K: delivery ratio, θ : crank angle, λ_1, λ_2, x, λ_3 : coffiicient of steady flow resistance in entrance of intake pipe, carburetor, wall of intake pipe and inlet port respectively, $\beta(\theta)$: ratio of flow velocity.

To strengthen the relation given by Eq. (2), all results obtained by changing the intake pipe length in the test Engine–125 as well as Engine–50 are plotted against the inertia coefficient (Z_i) as shown in Fig. 12. It is shown in the Figure that the maximum delivery ratio for every pipe length in Engine–50 are obtained in the range of values $Z_i=0.45\sim0.55$, which shows a good agreement with the value of $Z_{iM}(=0.493)$ calculated by Eq. (2) assuming that the value of U is 1.03 obtaining from Eq. (3) and the effective intake period (θ_i^*) is 110°.

Besides the experimental values of Z_i measured in Engine–125 agree with the value of $Z_{iM}(=0.61)$ computed from the same equation putting $U=0.9$, $\theta_i^*=130°$.

Similarly the delivery ratio is largely affected by the crankcase volume as shown in Fig. 13, particularly the larger the crankcase volume the lower the engine speed where the maximum delivery ratio occurs. However, if rearranging the experimental results with Z_i as shown in Fig. 14, the maximum delivery ratio for each crankcase volume is also occuring nearly at the value of $Z_i=0.493$, which computed by Eq. (2) like the former.

Increasing the opening period of inlet port (θ_i^*), the engine speed for the maximum delivery ratio also in

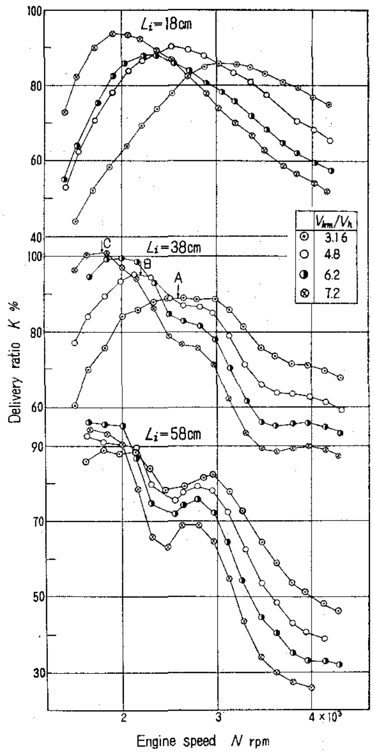

Fig. 13. Delivery ratio for each crankcase volume (E–50)

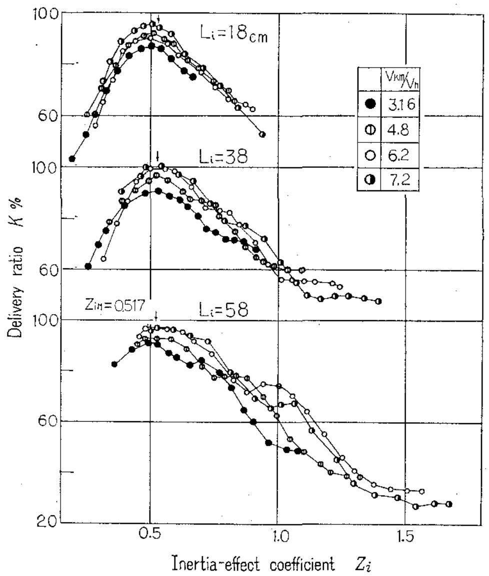

Fig. 14. Inertia effect due to the crankcase volume (E–50)

increases as shown in Fig. 15, but Fig. 16, shows that the maximum delivery ratio has the same value of the inertia coefficient (Z_{iM}) caluculated for each opening period from the same equation. To examine the effects of flow resistance, the experiments are carried out changing the throttle valve of carburetor and the diameter of intake pipe. These results are shown in Fig. 17 & 18 respectively. Using the inertia coefficient (Z_i) calculated from the engine speed for the maximum delivery ratio, the flow resistant coefficient (U) can be counted backward from Eq. (2). On the other hand, the flow resistant coefficient (U) is also determined from Eq. (3) When the coefficient of the steady flow resistance is used. Then both the values are compared with each other in Table 2. It seems that, since both the values agree frirly well, the influence of carburetor or pipe diameter on the matching condition of the inertia effect should be considered roughtly as the influence of flow resistance coefficient.

Through the above examinations, it seems to be confirmed that the matching conditions for the maximum delivery ratio, that is, the optimum values inertia coefficient (Z_i), the resistance coefficient (U) and the effective intake period (θ_i^*) can be

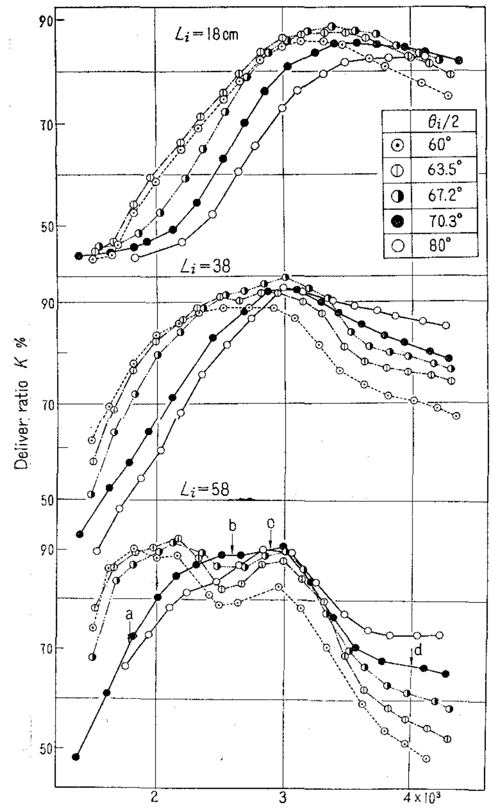

Fig. 15. Delivery ratio for each opening period of inlet port (E-50)

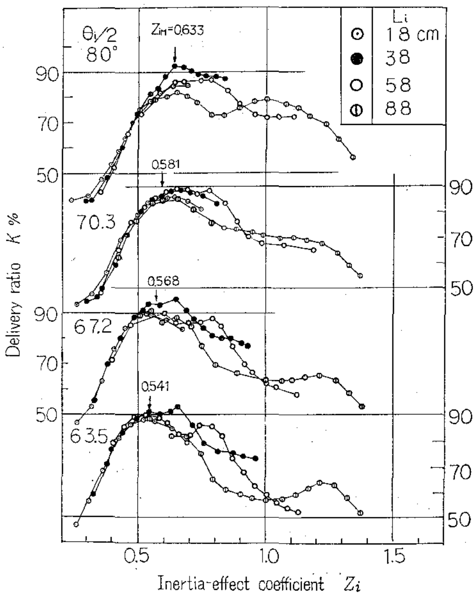

Fig. 16. Inertia effect due to the opening period of inlt port (E-50)

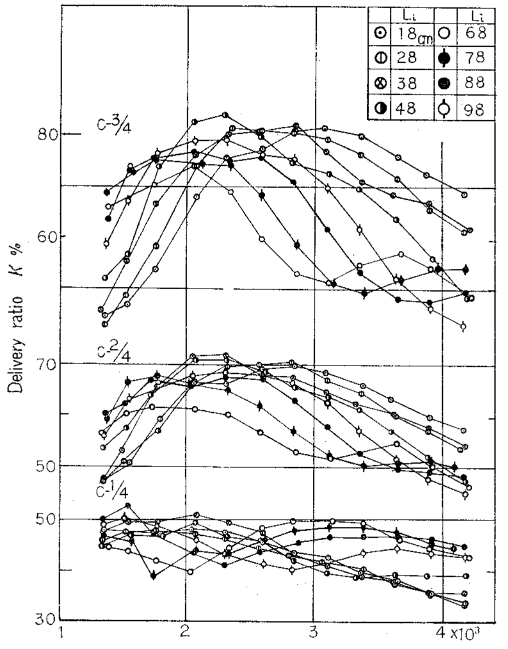

Fig. 17. Delivery ratio for each throttle of Carburetor (E-50)

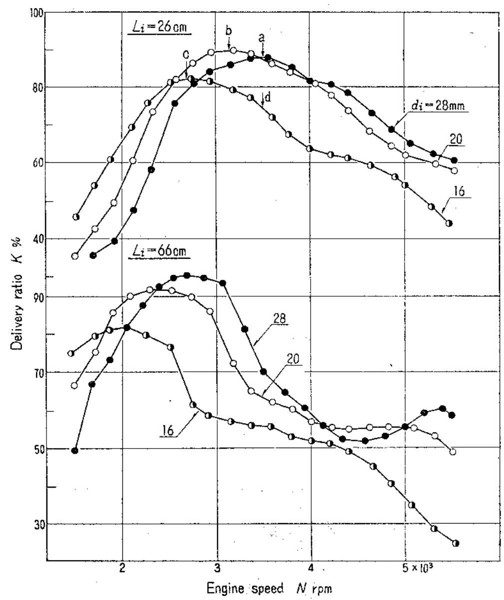

Fig. 18. Delivery ratio for each intake pipe diameter (E-125)

Table 2. Flow resistant coefficient (U)

Test engine	Opening of Carburetor or Diameter of intake pipe	Z_{iM} (Mean value) Experimental value	U is counted backward from Eq. (2)	U is determined from Eq. (3) (Mean value)
E-50	C-4/4	0.512	1.07	1.045
	C-3/4	0.497	1.17	1.125
	C-2/4	0.465	1.40	1.36
	C-1/4	0.330	2.55	2.32
E-125	C-4/4	0.60	0.93	0.90
	16 mmϕ	0.556	1.15	0.98
	20 mmϕ	0.635	0.82	0.828
	28 mmϕ	0.683	0.58	0.63

calculated by Eq. (2) and such computed values are shown in Fig. 19. Accordingly the effect of such engine factors as L_i, V_k, f_i, N, U etc. on the maximum delivery ratio can be also determined, so that Eq. (2) seems to be a useful formula for the engine design.

3-3 On the Pulsation Effect due to the Exhaust Pipe

The pulsation wave remaining in the exhaust pipe after the scavenging process

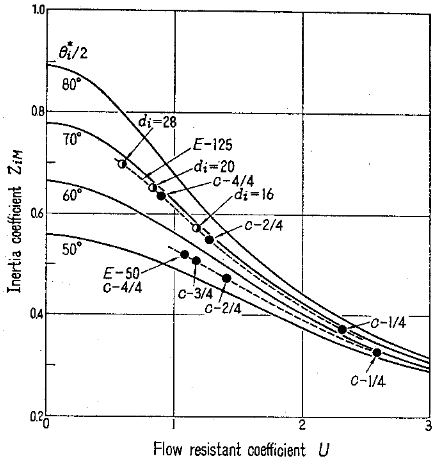

Fig. 19. Matching condition for the inertia effect.

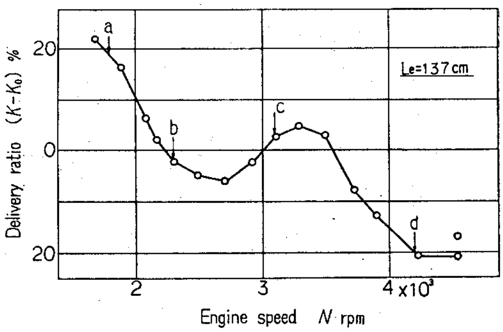

Fig. 20. Delivery ratio curve for pipe length L_e=137 (E-50)

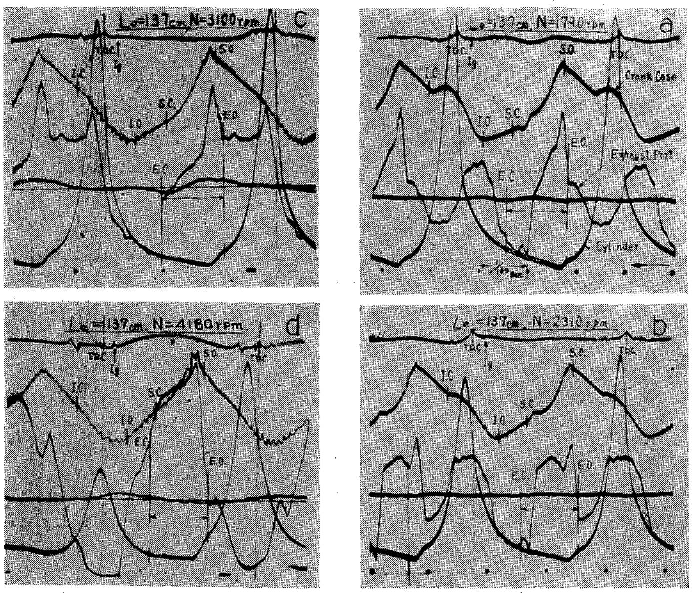

Fig. 21. Pressure diagrams for pipe length L_e=137 (E-50)

contributes to the next scavenging one, and then the delivery ratio $(K-K_0)$ is also affceted by such pulsation waves as shown in Fig. 20.

In Fig. 21 (a), N=1790 rpm; the number (q_e) of pulsation wave entering the cylider during a cycle is equal to $2\frac{1}{4}$, while the pulsation wave number (Δq_e) for the effective scavenging period is 3/4. Then the sum of both numbers Q_e $(=q_e+\Delta q_e)$ is nearly equal to 3, so that the delivery ratio shows very high value because of the negative wave superposes on the latter half period of the exhaust-scavenging process (θ_{es}^{*}). The delivery ratio at $N=2310$ rpm decreases extremly as show by a point b in Fig. 20, as the positive pulsation wave superposing on the latter half period of θ_{es}^{*}.

On the other hand, the delivery ratio at $N=3100$ rpm inceases slightly on account of the relatively small positive wave superposed on the whole period of θ_{es}^{*}, although the θ_{es}^{*} period is almost covered by the positive wave as shown in Fig. 21 (c). Fig. 21 (d) $N=4180$ rpm, shows that the θ_{es}^{*} period is completely covered by the positive wave of exhaust blow-down. Then the delivery ratio decreases extremely.

From these considerations, it is found that the difference of delivery ratio $(K-K_0)$ has an inclivation to increase with superposing of the negtive part of pulsation waves upon the latter half period of θ_{es}^{*} and vice versa.

Consequently, such a dynamic effect due to the exhaust pipe system on the delivery ratio $(K-K_0)$ is usually called the exhaust pulsation effect. This considered as follows ; in the first half period of the exhaust-scavenging process in a two stroke cylce engine, whose inlet port is controlled by the piston, the cylinder and scavenging pressures are both high and besides a forced scavenging action due to the piston motion follows, so that the influence of pulsation wave on the delivery ratio is comparatively as little as was expected.

On the contraty, in the latter half period the scavenging pressure drops down so steeply that the delivery ratio depends mainly upon whether or not the negative wave is superposing on the period.

Then the pulsation coefficient (Q_e) is defined as a number of the pulsation waves involed between exhaust openning (E. O.) and effective scavenging closure (S. C.*) and it is an important parameter showing whether the pulsation wave placed during the latter half period of scavenging process is positive, or negative, and is given

$$Q_e = (a_e/4L_e) \div \{(N/60) \times 360/(360+\theta_{es}^{*})\} = (1+\theta_{es}^{*}/360) \cdot q_e \qquad (4)^{6)}$$

Where a_e : sonic velocity in exhaust pipe system, L_e : length of exhaust pipe, N: engine speed, θ_{es}^{*}: effective period of exhaust-scavenging process, and $q_e=15a_e/NL_e$ is similar to Eq. (1).

When Q_e is eqaual nearly to $\frac{3}{4}\sim 1$, $1\frac{3}{4}\sim 2$ and $2\frac{3}{4}\sim 3$, the delivery ratio $(K-K_0)$ increases because of superposing the negative wave on the latter period of the scavenging action.

To certify such a consideration, some experimental results for each exhaust pipe of various length are plotted against the pulsation coefficient (Q_e) in Fig. 22, in which some peaks on the delivery ratio curve occurs aproximately at the value of $Q_e=1\frac{3}{4}\sim 2$, $2\frac{3}{4}\sim 3$ and so on. Accordingly, the pulsation coefficient (Q_e) seems to be important parameter giving the mathing criterion of the pulsation effect due to exhaust pipe system, and then it would be applicable to the other

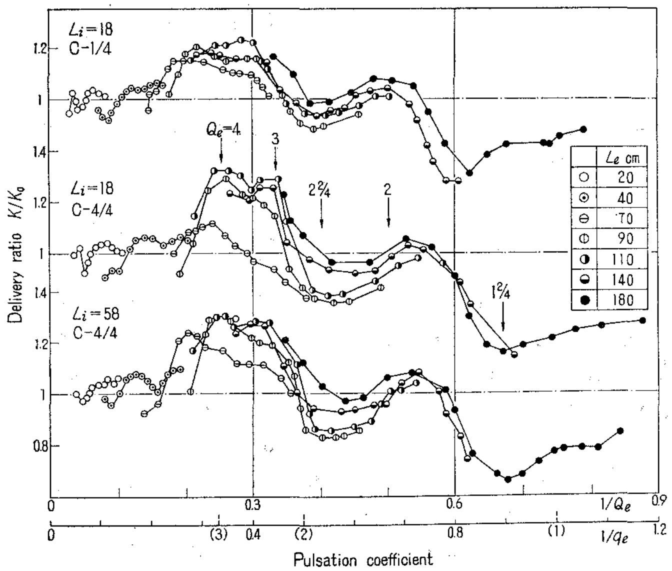

Fig. 22. Pulsation effect due to the exhaust pipe

exhaust pipe system such as a cone type as proved in Fig. 23.

3-4 On the Blow-Down Effect due to the Exhaust Pipe

According to some experiments on the effect of exhaust pipe system, it seems certain that the increasing in the delivery ratio $(K-K_0)$ due to exhaust pipe effect depends considerably on the matching state between the negative wave following after the positive wave of blow-down and the effective closure of scavenging port. Then the maximum increase of delivery ratio $(K-K_0)$ is also obtained just when three quarters of the blow-down period is nearly equal to the effective exhaust-scavenging period (θ_{es}^*).

These matching conditions are expressed simply by

$$Z_{eM}(\equiv \omega L_e^*/a_e)_M = (4/3)\cdot(\theta_{es}^*/360) \qquad (5)^{7)}$$

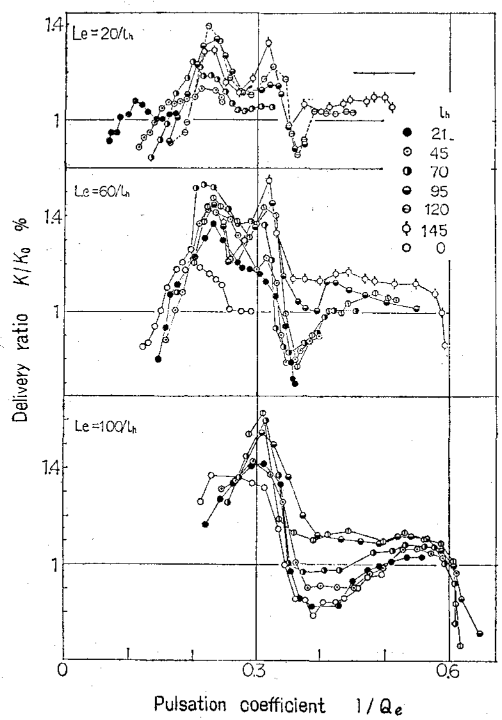

Fig. 23. Pulsation effectdue to the conical pipe $(\theta_h=2°)$

Where Z_{eM}: blow-down coefficient where the maximum delivery ratio $(K-K_0)_M$ is given, ω: angular velocity, a_e: sonic velocity, L_e^*: equivalent length of the scavenging and exaust pipe system, θ_{es}^*: effective exhaust-scavenging period, which counts in the crank angle from the exhaust opening (E.O.) to the effective scavenging closure (S.C.*).

In this equation, θ_{es}^* is given as the engine design and a_e are measured respectively. Therefore it is only necessary to obtain the value of L_e^*, which is calculated from the period of pressure wave during the exhaust blow-down and affected largely by the various factors in the exhaust pipe system.

Both theoretical values of L_e^* calculated on the inertia theory and the impedance theory are compared with the experimental values obtained in an air model engine similar to the exhaust pipe system of the actual engine. In practice, the theoretical period (T_M), from the exhaust openning to the time for the cylinder pressure taking the maximum negative value, instead of L_e^* is used for comparing with the experimental values obtained from the oscillograms, as shown in Fig. 24.

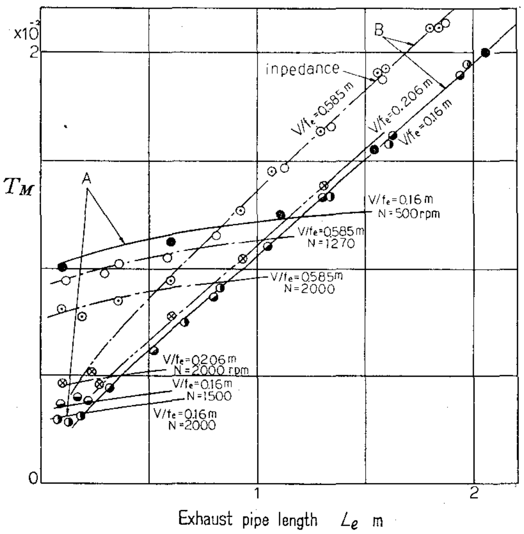

Fig. 24. Theoretical period (T_M) calculated by means of the inertia theory and the impedance theory.

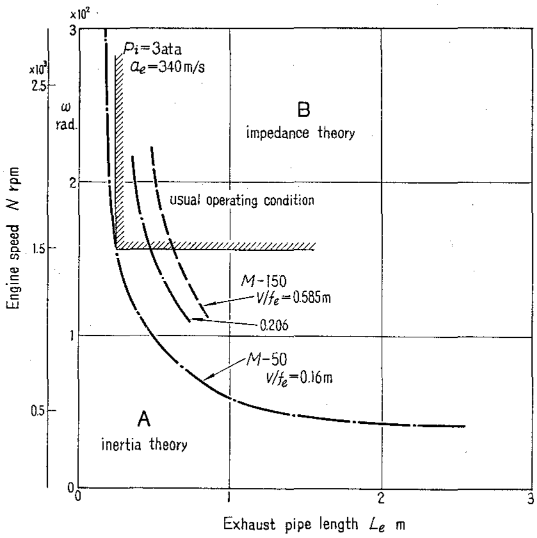

Fig. 25. Applicable range of both theories.

From a good agreement in the figure, it is seen that each theory has an applicable range respectively, and further the impedance theory can be applied to usual operating conditions in the actual engine, where the exhaust pipe effect should be expected as shown in Fig. 25.

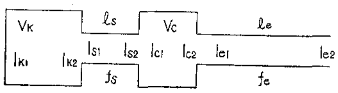

Fig. 26. Exhaust-Scavenging pipe system.

If the impedance theory is to be used to a exhaust-scavenging pipe system of an actual engine, which is composed of crankcase ($\overline{V}_k$), scavenging passage (l_s, f_s), cylinder ($\overline{V}_c$) and exhaust pipe (l_e, f_e) as shown in Fig. 26 then the equivalent length (L_e^*) is also to be calculated by the approximate equation (6)[7].

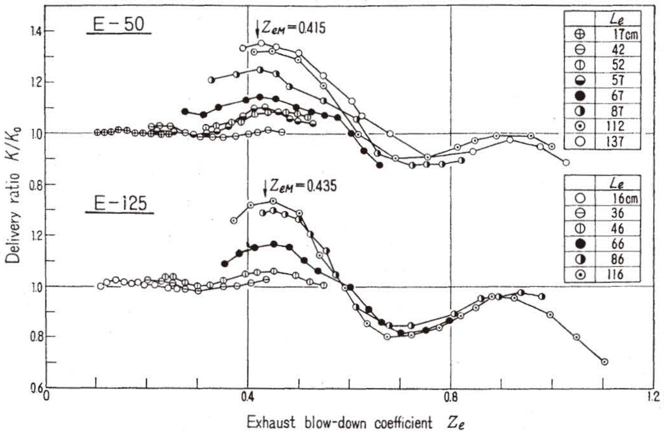

Fig. 27. Exhaust blow-down effect due to the exhaust pipe length.

$$\mathrm{Cot}\,(L_e/L_e^*) = \left(\overline{V}_c + \frac{\theta_{sc}^* - \theta_{s0}^*}{\theta_{sc}^* - \theta_{E0}} \cdot \overline{V}_k\right) \Big/ f_e L_e^* = \overline{V}_m / f_e L_e^* \tag{6}$$

To certify the relation given by Eqs. (5) and (6), all experimental results for Engine–125 as well as Engine–50 are plotted against the exhaust blow-down coefficient $Z_e\,(=\omega L_e^*/a_e)$ in Fig. 27. It is shown in Figure that the maximum increase of delivery ratio $(K-K_0)_M$ for every pipe length are obtained in the range of values $Z_e=0.41\sim0.43$, which agree with the value $Z_{eM}=0.415$ for Engine–50 computed

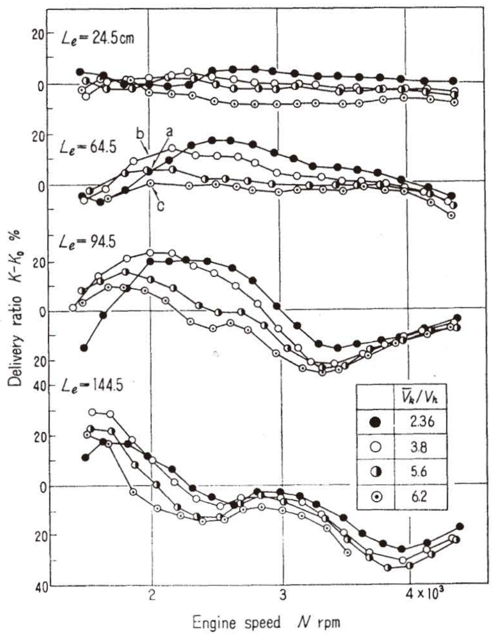

Fig. 28. Delivery ratio $(K-K_0)$ for each crankcase volume (E-50)

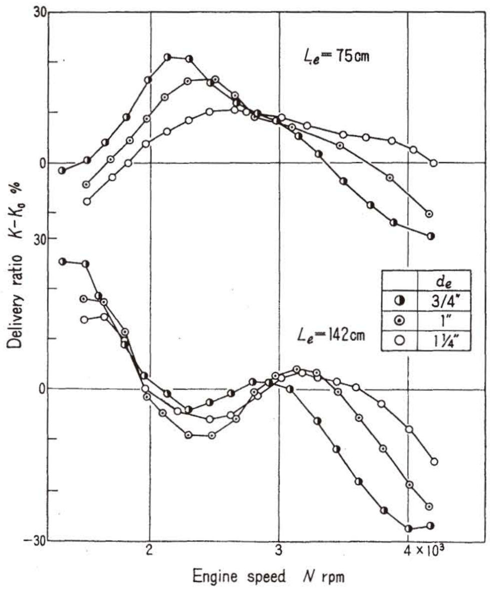

Fig. 29. Delivery ratio $(K-K_0)$ for each exhaust pipe diameter (E-50)

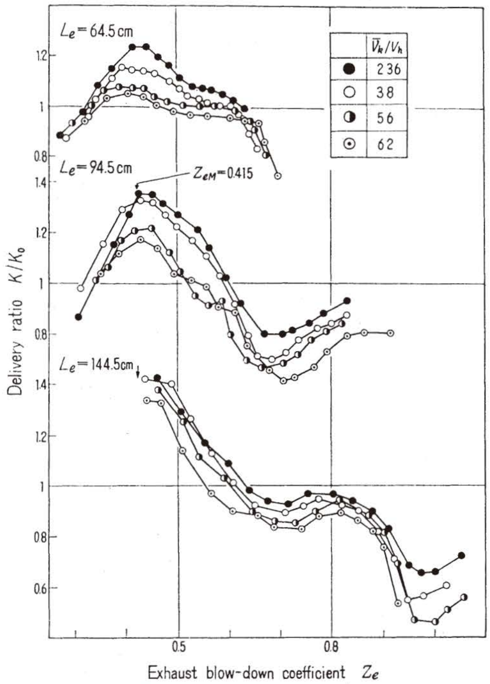

Fig. 30. Exhaust blow-down effect due to the crankcase volume (E-50)

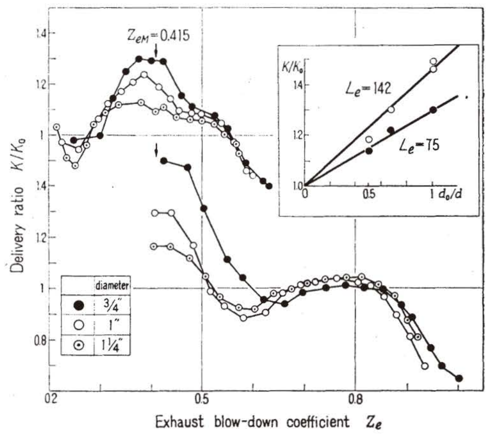

Fig. 31. Exhaust blow-down effect due to the exhaust pipe diameter (E-50)

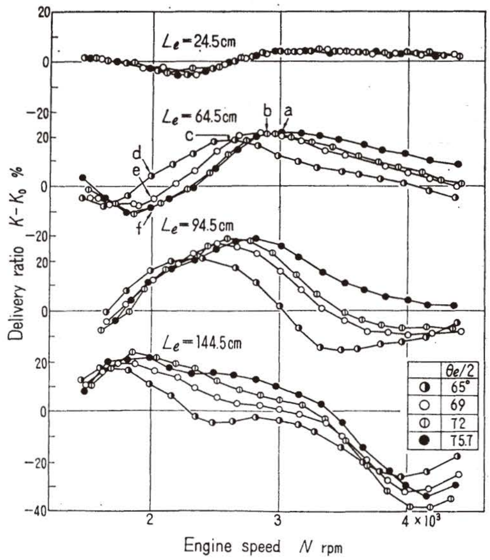

Fig. 32. Delivery ratio $(K-K_0)$ for each opening period of exhaust port (E-50)

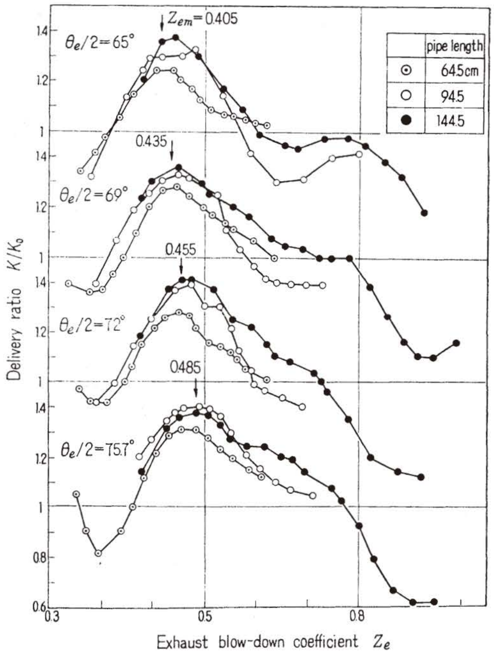

Fig. 33. Exhaust blow-down effect due to the opening period of exhaust port (E-50)

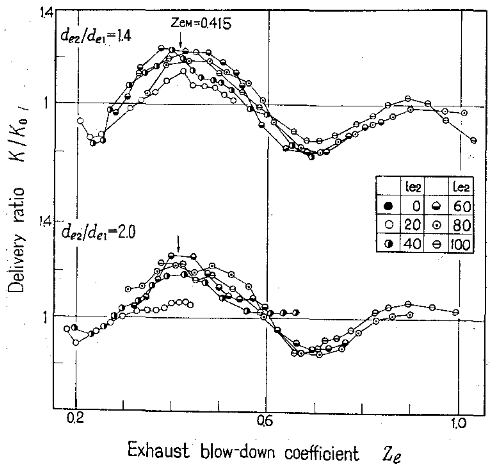

Fig. 34. Exhaust blow-down effect due to the exhaust system with stepped pipe (E–50, $l_{e1}=44$ cm)

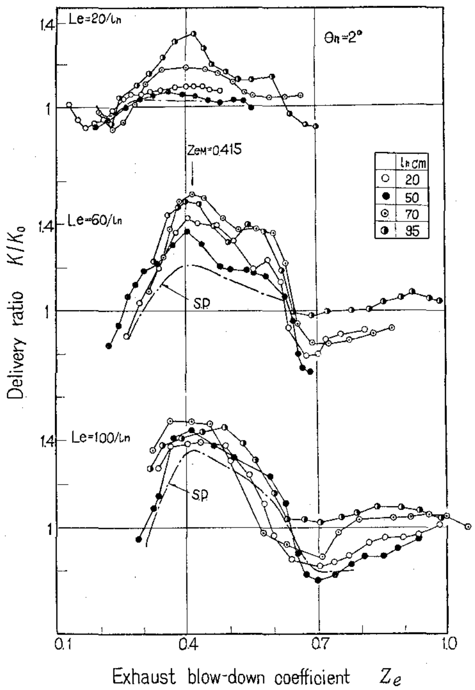

Fig. 35. Exhaust blow-down effect due to the exhaust system with conical pipe (E–50, $\theta_h=2°$)

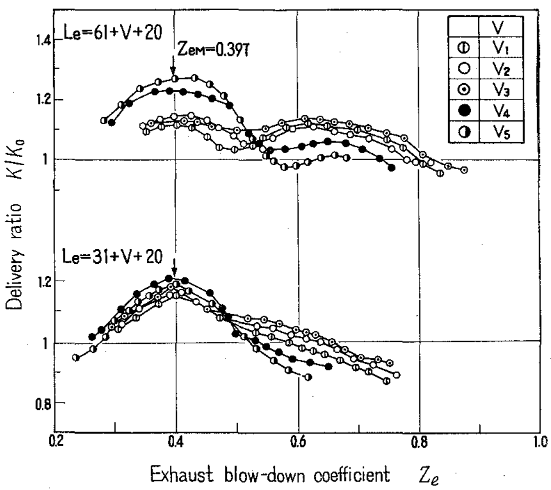

Fig. 36. Exhaust blow-down effect due to the exhaust system with expansion chamber (E–50)

by Eqs. (5) and (6), and in the case of Engine–125, while the experimental values (Z_e) also shows a good agreement with the value of $Z_{eM}=0.435$ calculated from the same equation.

Similarly, the delivery ratio ($K-K_0$) is largely affected by the crankcase volume or the sectional area of exhaust, pipe as shown in Figs. 28 & 29. However, if rearranging the expermental results with Z_e as shown in Figs. 30 & 31, the maximum increase of delivery ratio for each case is also occuring nearly at the value of $Z_e=0.415$, which computed by Eq. (5).

Increasing the exhaust-scavenging period (θ_{es}^*), the engine speed for the maximum increase of delivery ratio also increases as shown in Fig. 32, but Fig. 33 shows that the maximum increase of delivery ratio has the same value of exhaust blow-down coefficient (Z_{eM}) calculated for each opening period from the same equation.

It seems, therefore, to be seen that the optimum values of engine speed (ω) and effective exhaust-scavenging period (θ_{es}^*) can be determined by Eq. (1). if the optimum dimensions of the exaust-scavenging pipe system (L_e^*) would be obtained by Eq. (6). and then it will be applicable to the other exhaust pipe system as shown in Figs. 34, 35 and 36.

4. Conclusions

The results reached are summarized as follows:

a) In the extreme cases of a long intake pipe or high speed, the residual pulsation waves in the intake pipe have great influence on the delivery ratio and such a pulsation effect is generally governed by the expression

$$q = 15a_i/NL_i \tag{1}$$

Strictly speaking, however, these matching conditions must be discussed considering the pulsation effect and the inertia effect due to the intake pipe system. In a word; (i) if the inertia effect were utilized largely, the delivery ratio would increase at $q_i=1\frac{1}{2}$, while it would decrease at $q_i=2$. (ii) Even if the inertia effect is not large, the delivery ratio augments at $q_i=1\frac{1}{2}\sim1\frac{1}{4}$, where the positive wave superposse on the latter half period of intake process, but it diminishs at $q_i=1\frac{3}{4}$ because of the negative wave during intake period. (iii) In general, the matching condition is the same as in the case of (1), because the first positive wave coming into the cylinder at the inlet closure has a relative higher value when q_i is larger than 2.

b) The maximum delivery ratio occurs by the inertia effect due to the intake pipe and the matching condition of the effect in a crankcase-compressed two-stroke cycle engine is expressed approximately by the following equation.

$$1/Z_{iM}^2 = (180/\theta_i^*)^2 + U^2 \tag{2}$$

c) The pressure wave remaining in a long exhaust pipe contributes to the scavenging action of the next cycle and such a pulsation effect is important for the engine performance ane is defined as the following pulsation coefficient (Q_e).

$$Q_e = (1+\theta_{es}^*/360)\cdot q_e \tag{4}$$

d) The maximum increase in the delivery ratio ($K-K_0$) due to the exhaust pipe is obtained by the exhaust blow-down waves and such a blow-down effect is given by the expression

$$Z_{eM}(\equiv \omega L_e^*/a_e)_M = (4/3)\cdot(\theta_{es}^*/360) \tag{5}$$

where L_e^* is also calculated by the approximate equation (6).

5. Acknowledgement

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