On the Effects of the Pressure Wave on the Fuel Flow Amount in a Small Two-Stroke Cycle Gasoline Engine

On the Effects of the Pressure Wave on the Fuel Flow Amount in a Small Two-Stroke Cycle Gasoline Engine

Norihiro Sawa* and Masayosi Sawa**

Abstract

The author carried out systematically some experiments to examine the influences of the intake pipe system on the characteristics of carburetor and the fuel blow-out from an intake pipe in a small crankcase-compressed two-stroke cycle gasoline engine. Some conclusions reached are summarized as follows:

a) The air-fuel ratio in motoring differs scarcely from that in firing.

b) When the carburetor is installed at the engine side, the air-fuel ratio is affected by the intake pipe length.

c) With an intake pipe of constant length, the air-fuel ratio varies extensively by the position of the carburetor in the intake pipe system.

d) The variation in air-fuel ratio is largely governed by the matching condition q_s and depends partly on the amplitude of pulsation wave in the intake pipe system.

e) The blow-out ratio of fuel is remarkably influenced as is done in the case of delivery ratio by the pipe length, the port timing and the opening of the carburetor, and then the fluctuation of that is governed by the matching condition of an inertia effect.

1. Introduction

It has long since been known the fact that the engine performance or the breathing capacity is largely governed by an intake or exhaust pipe system in the engine and the author has also presented already some reports concerning the theoretical and experimental analysis on the matching condition for the intake or exhaust pipe effect in a small crankcase-compressed two-stroke cycle gasoline engine.

In those experiments, the author found two phenomena that the blow-out of fuel increase with the pipe length and with the carburetor put near the inlet port side in the pipe with a given constant length, the engine combustion unstable and poor.

Nevertheless, it is necessary to prevent the fuel blow-out for the fuel economy and there are many practical demands for putting a carburetor in the intake pipe in place of at the entry of it but very little is known about such phenomena, that is, the influence of an intake-pipe system on the combustion state of engine, the air-fuel ratio characteristics of carburetor and the blow-out of fuel.

Thereupon, to examine in detail the influences of that, he has measured the

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breathing capacity, the fuel consumption, the amount of blow-out fuel and the static pressure in venturi-part of carburetor changing the length of intake-pipe and the position of carburetor in the pipe with a given constant length, and recorded some pressure diagrams in the venturi-part of carburetor, the entrance of intake pipe and observed the combustion state indicated by the cylinder pressure.

2. Experimental apparatus and method

The author used two crankcase-compressed two-stroke cycle gasoline engines of the same type for motor bicycle i. e. an Engine–50 and an Engine–125, whose dimensions are described in Table 1.

Table 1. Dimension of test engine

Engine			E-50	E-125
Cylinder bore×stroke		(mm)	$40\,\phi\times39.8$	$55\,\phi\times52.5$
Stroke volume		(cc)	50	125
Compression ratio			7	7
Mean volume of crankcase during inlet opening period		(cc)	150	435
Sectional area of intake pipe		(cm^2)	1.49	3.14
Port timing	Inlet		±60° (T.D.C.)	±70°
	Exhaust		±67° (B.D.C.)	±69.7°
	Scavenging		±55° (B.D.C.)	±57°

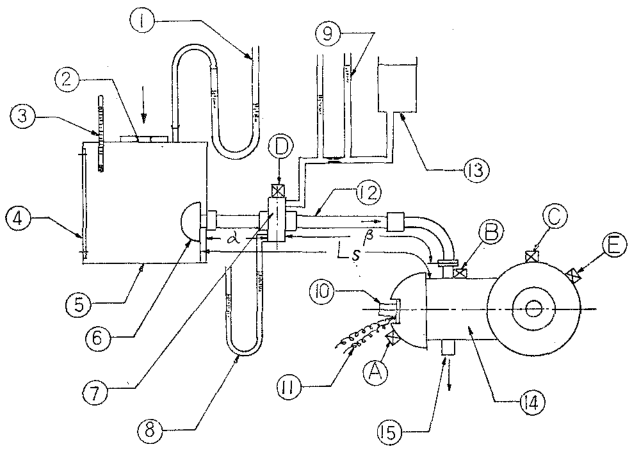

Fig. 1. General layout of experimental apparatus.

① Manometer ② Air flow-meter ③ Thermometer ④ Rubber sheet
⑤ Surge tank ⑥ Air cleaner ⑦ Carburetor ⑧ Manometer
⑨ Flow-meter of venturi type ⑩ Spark plug ⑪ Thermo couple
⑫ Intake pipe ⑬ Fuel tank ⑭ Test engine ⑮ Exhust pipe

Pressure indicator:

Ⓐ Cylinder Ⓑ Inled port Ⓒ Crankcase Ⓓ Carburetor Ⓔ Crank-mark

In Fig. 1 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 system directly, and in this case especially a carburetor and an intake pipe are connected with a short vinyle pipe and also the carburetor is fixed firmly by an iron stand to aroid the mechanical vibration from the engine body. Moreover ten straight pipes (10 cm in length, 13.8 mm in inner diameter) are prepared for intake pipe system so that the position of carburetor as well as the length of intake pipe can be changed in the many variations by their combination.

To observe the fuel level in a float-chamber and examine the vibrating state of it, the author prepared a small window in the float chamber wall. Then as shown Fig. 2, the venturi part of carburetor is equipped with a manometer and a pressure indicator of capacity type to measure a static and dynamic pressures in it and the measuring utensil of the blow-out fuel are set at the entry of the intake pipe.

On the other hand, to eliminate the influences of an exhaust pipe on the delivery ratio, a short exhaust pipe ($L_e=3$ cm) has been used throughout experiments based on the results of previous paper. All tests carried out with the engine speed change from 1400 to 4200 rpm and the amount of air flow, the fuel consumption and the blow-out fuel are measured by the flow-meter, the static pressure in venturi part of carburetor by the manometer after the temperature of spark plug seat having reached a steady value.

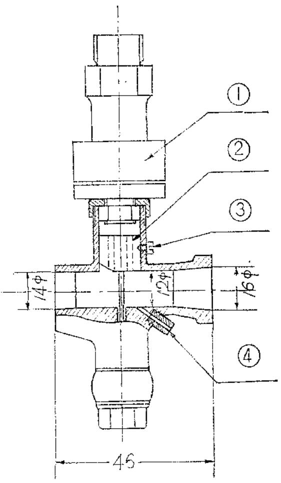

Fig. 2. Detail of carburetor (AMAL M 12 C_2 type)
① Pressure indicator
② Trottle valve
③ Set screw
④ Connector to manometer

In such tests, the throttle valve in the carburetor is always adjusted to a full open state and even in motoring operation the fuel is supplied in the some manner as firing. Furthermore to analyze the influences of

Table 2. Experimental items

No.	Experimental items	Intake pipe	Instrument
1	Combustion state	$l_s=78$; ${}_0L_{78}$, ${}_{30}L_{48}$, ${}_{50}L_{28}$, ${}_{70}L_8$	Pressure indicator. Cathode-ray oscillograph
2	Delivery ratio	$l_s=18$; ${}_0L_{18}$, ${}_{10}L_8$ $l_s=38$; ${}_0L_{38}$, ${}_{10}L_{28}$, ${}_{30}L_8$ $l_s=78$; ditto (1)	Flow-meter of round nozzle type
3	Fuel consumption	$l_s=78$; ${}_0L_{78}$, ${}_{10}L_{68}$, ${}_{20}L_{58}$, ${}_{30}L_{48}$ ${}_{50}L_{28}$, ${}_{60}L_{18}$, ${}_{70}L_8$	Flow-meter of venturi type
4	Negative pressure in venturi-part of carburetor	$l_s=78$; ditto (3)	Manometer
5	Rising and vibration of fuel level in float chamber	$l_s=78$; ditto (1)	Eye measurement
6	Pressure variation in carburetor	$l_s=18$; ${}_0L_{18}$, ${}_{10}L_8$ $l_s=78$; ${}_0L_{78}$, ${}_{30}L_{48}$, ${}_{50}L_{28}$	Pressure indicator, Magnetic oscillograph

pulsation wave on the characteristic of the carburetor, the combustion state and the blow-out phenomena, the pressures in the cylinder (A), the intake port (B), the crank-case (C) and the venturi part of carburetor (D) and the entry of an intake pipe Ⓔ are picked up by means of each pressure indicator of electric capacity type as shown in Fig. 1 and the crank-marks are indicated by another pick-up (F) of the same type, and then those electric variations are all recorded on a magnetic or a cathode oscillograph set through several amplifiers of direct current type respectively.

3. Experimental results and considerations

3.1 The effects of intake pipe system on the carburetor performance

3.1.1 The characteristic of air-fuel ratio

To compare both cases of motoring and firing, the delivery ratio and the mixture ratio for the test engine E–125 with the intake-pipe length $l_s = 15$ cm are plotted against the engine speed in Fig. 3.

It is shown in the figure that it is not necessary at all to consider whether the engine operates in motoring or firing as well as in the case of the four-stroke cycle engine. However, when the carburetor are installed on the engine side, the mixture ratio as well as the delivery ratio are largely affected by the intake-pipe length as shown in Fig. 4 in contrast with the case of exhaust-pipe length in Fig. 5, particularly the longer the pipe length, the worse the characteristics of mixture ratio becomes.

There seems to exist the influences of pulsation wave in the intake-pipe on the carburetor performance.

To examine in detail these influences, changing the position of carburetor in

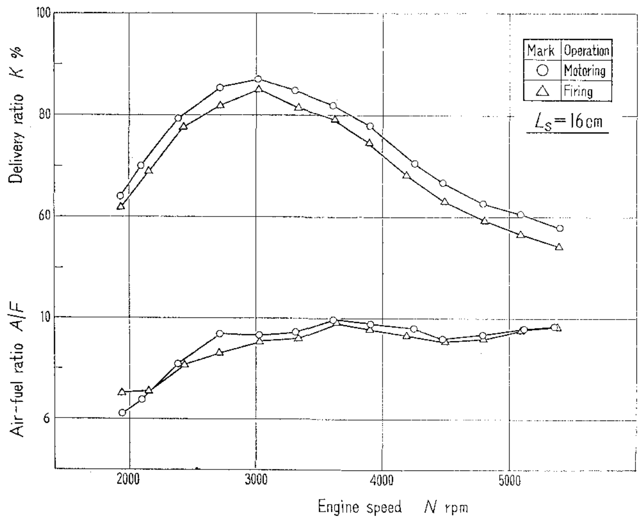

Fig. 3. Comparison of delivery ratio in motoring and firing (E–125)

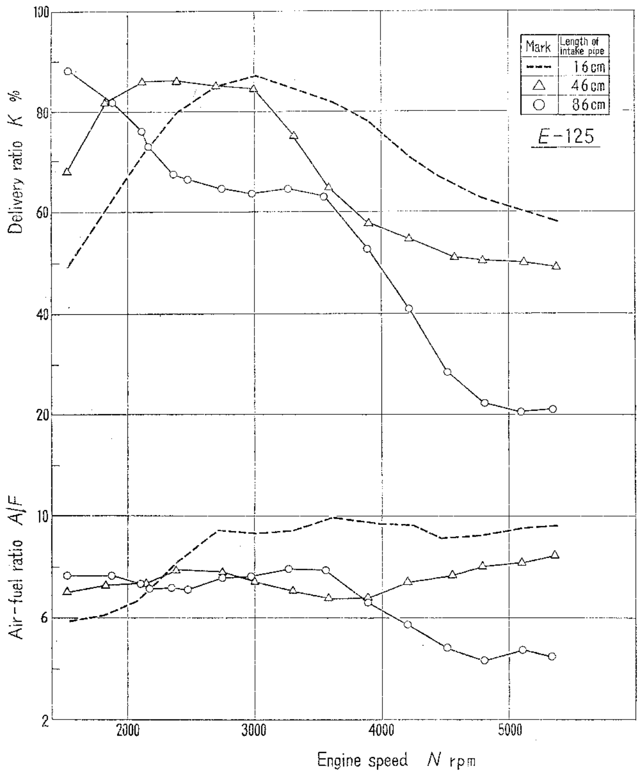

Fig. 4. Air-fuel ratio and length of intake pipe (Test engine E-125)

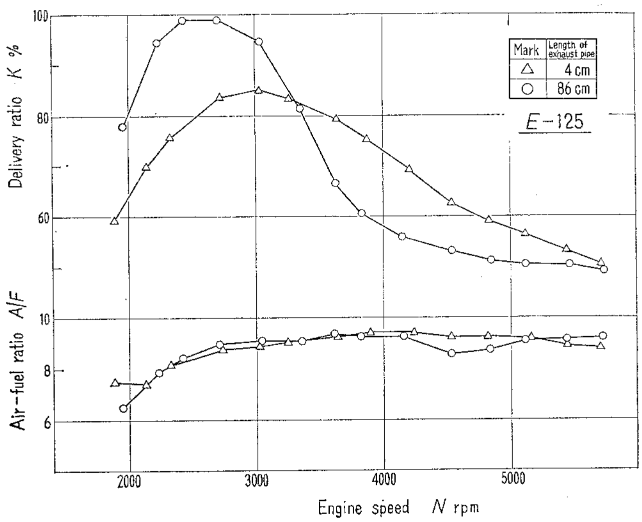

Fig. 5. Air-fuel ratio and length of exhaust pipe (E-125)

the intake pipe with a given constant length, the air amount is measured by an air flow-meter on a surge tank, the fuel consumption by a flow-meter of venturi type and the static pressure in venturi-part is examined by means of a manometer as shown in Fig. 1 and the combustion state indicated by the cylinder pressure are observed on a cathode oscillograph set and the results are shown in Fig. 6 and Table 3, and in the notation of intake pipe $_{\alpha}L_{\beta}$, α means the pipe length between

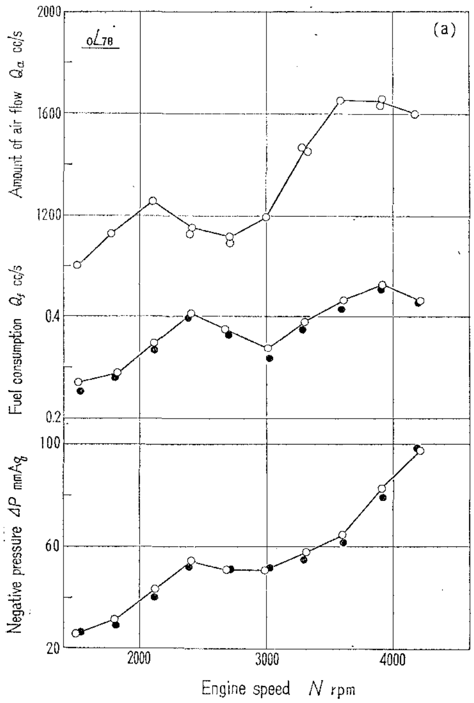

Fig. 6 (a). Experimental result.

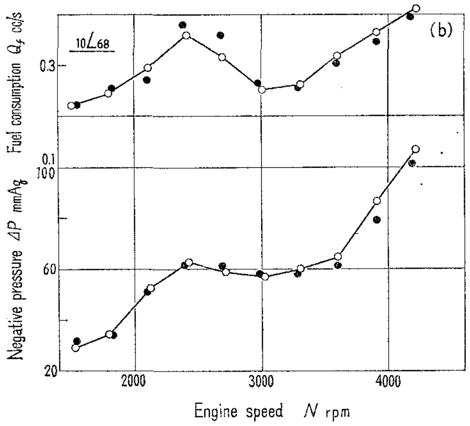

Fig. 6 (b). Experimental result.

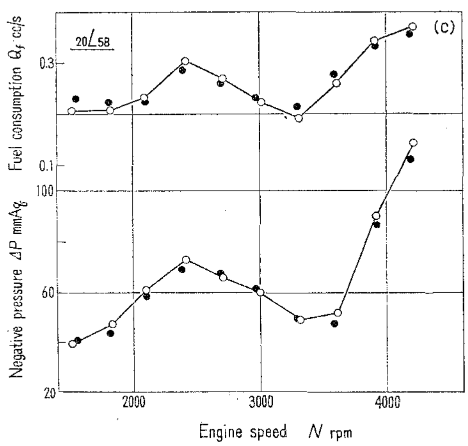

Fig. 6 (c). Experimental result.

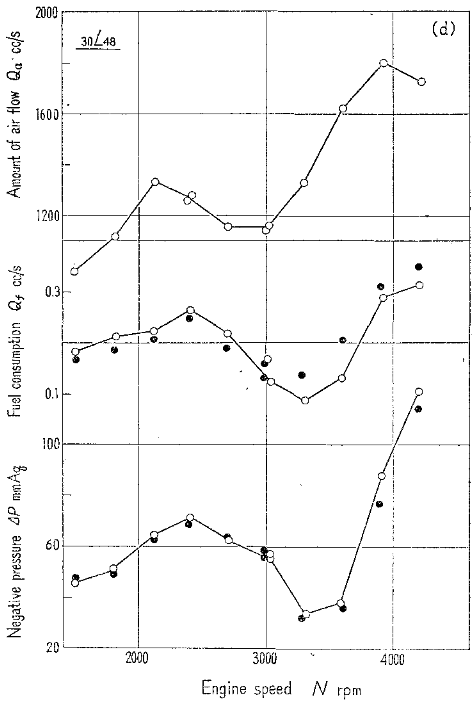

Fig. 6 (d). Experimental result.

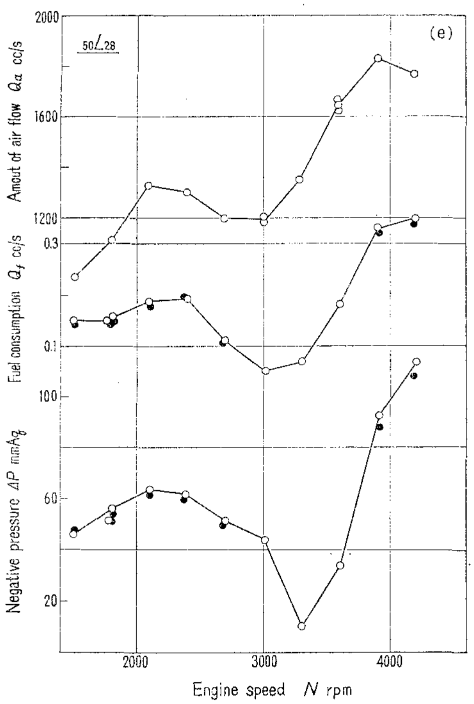

Fig. 6 (e). Experimental result.

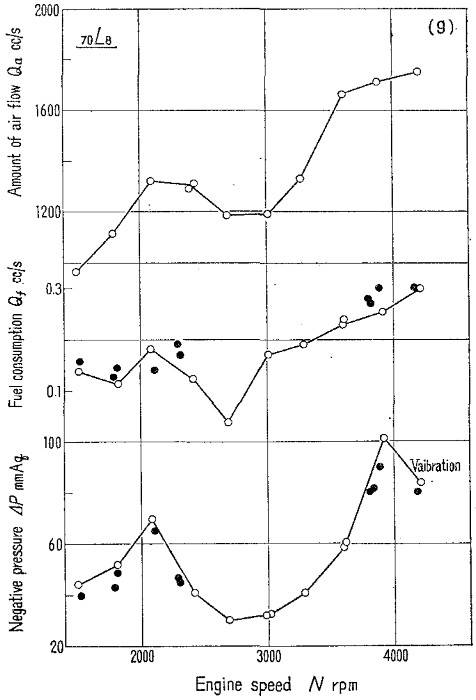

Fig. 6 (g). Experimental result.

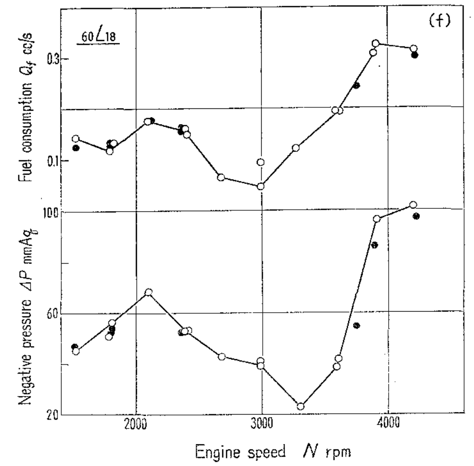

Fig. 6 (f). Experimental result.

the air cleaner and the carburetor and β the distance from the carbutetor to the inlet port and then the nominal length of intake pipe (l_s) equals to $\alpha+\beta$, so that the effective total length of it $L_s=l_s+8$ cm except the case of $_0L_{78}$.

In those figure and table, when the carburetor is put near the air cleaner, the breathing capacity and the static pressure in venturi-part indicated a slightly similar tendency to that of curves of the fuel consumption, in consequence the combustion state is very satisfactory in all range of engine speed; on the contrary the nearer the carburetor approaches to the inlet port i.e. from $_{30}L_{48}$ to $_{70}L_8$, the worse the engine combustion becomes.

Especially in the case of $_{70}L_8$, the combustion state is so poor that the engine is mis-fired completely except at the low speed ($N=1500$ and 2100 rpm) and the high speed ($N=3900$, 4200 rpm).

In those cases, the curves of breathing capacity in a long intake pipe $l_s=78$ cm drop down at nearly $N=3000$ rpm because of the pulsation wave being superposed

Table 3. Combustion state at each position of carburetor

	$_0L_{78}$ (L_s=78+10.8 cm)			$_{30}L_{48}$ (L_s=78+8 cm)			$_{50}L_{28}$ (L_s=78+8 cm)			$_{70}L_8$ (L_s=78+8 cm)		
Engine speed r.p.m.	Combustion state	Rising of fuel level mm	Vibration of fuel level	Combustion state	Rising of fuel level mm	Vibration of fuel level	Combustion state	Rising of fuel level mm	Vibration of fuel level	Combustion state	Rising of fuel level mm	Veritcal vibration of float
1500	◎	0	nothing	◎	0.5	nothing	◎	10	small	○	15	small
1800	◎	”	”	◎	”	”	◎	”	”	×	17	”
2100	◎	”	”	◎	”	”	○	13	”	○	”	”
2400	◎	”	”	◎	”	a little	○	17	float vibrates horizontally	×	”	”
2700	◎	”	”	×	1	”	×	”		×	”	”
3000	◎	”	”	×	”	”	×	”		×	”	”
3300	◎	0.5	a little	×	17	large	×	”	float vibrates vertically	×	”	large
3600	◎	”	”	×	15	”	×	”		×	”	”
3900	◎	”	”	○	16	”	○	”		○	”	”
4200	◎	”	”	○	10	”	○	”		○	”	”

◎ good combustion ○ poor combustion × mis-firing

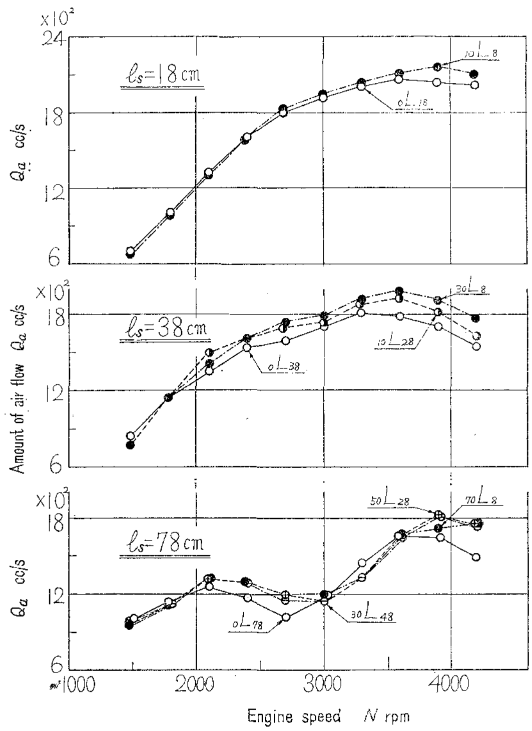

Fig. 7. Position of carburetor and amount of air flow (E–50).

on the intake period as well as in the case of previous paper, but the breathing capacity for every pipe length (l_s) is not affected at all by the position of carburetor as shown in Fig. 7.

On the other hand, the fuel consumption curves indicate a slightly similar tendency to the curves of breathing capacity as shown in Fig. 8, however the nearer the carburetor to the engine side, the lower the value of that, in consequent the larger becomes the air-fuel ratio as shown in Fig. 9 and then the peak of curve occurs at nearly $N=3000$ rpm.

Acoordingly it can be concluded that the combustion state due to the position

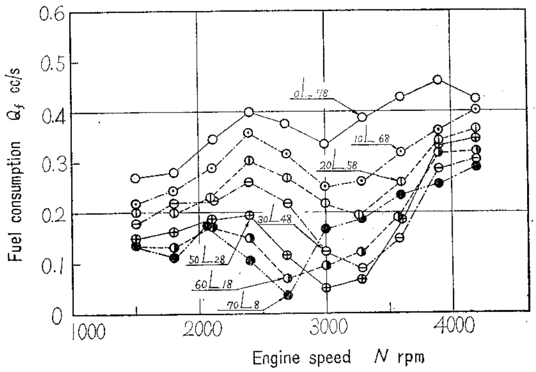

Fig. 8. Position of carburetor and fuel consumption (E-50).

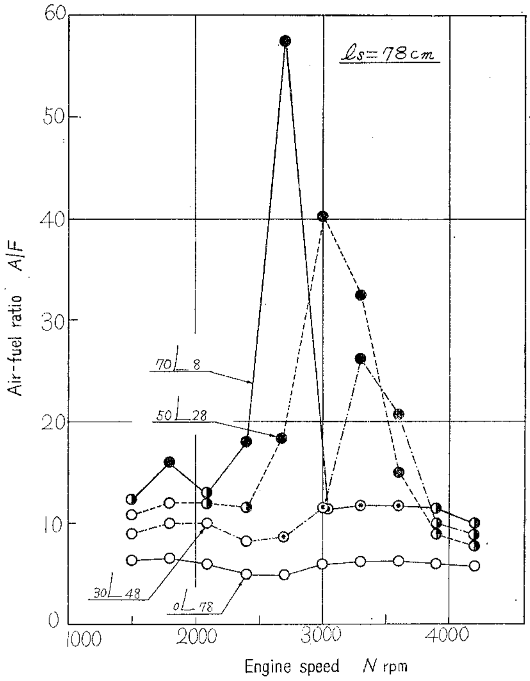

Fig. 9. Position of carburetor and air-fuel ratio (E-50).

of carburetor, namely the poor combustion or the mis-firing, is caused essentially through the lack of fuel.

Comparing the curves of fuel consumption in Fig. 8 with the static pressures in venturi-part of carburetor shown in Fig. 10 in order to examine in detail the variation in fuel consumption caused by the position of carburetor, for example, the static pressures for ${}_{20}L_{58}$ show the higher negative value than that for ${}_{0}L_{78}$, notwithstanding the fact that the fuel amount for the former should be fairly smaller than that for the latter.

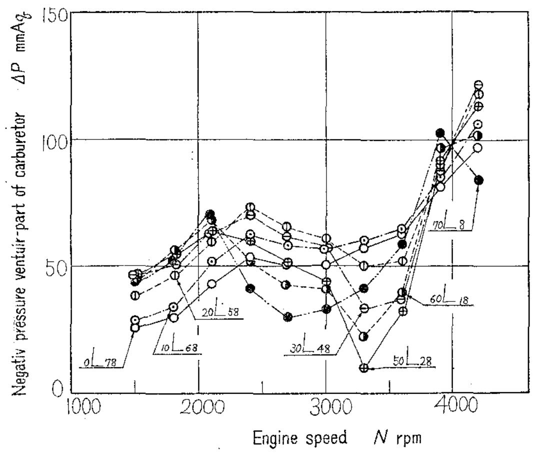

Fig. 10. Nagative pressure in venturi-part of carburetor (by manometer).

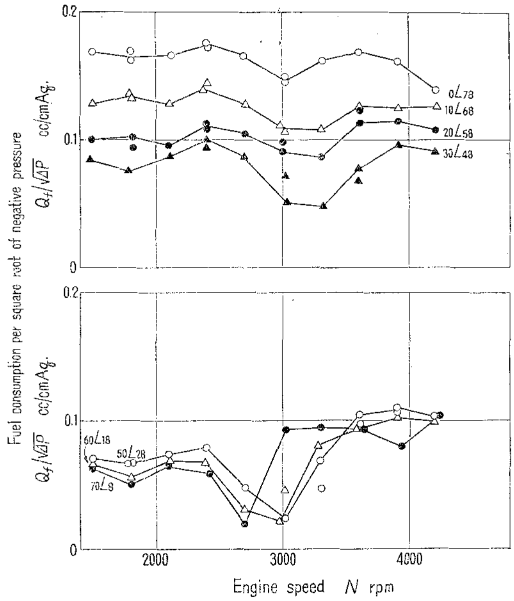

Fig. 11. Fuel consumption per root of megative pressure (E-50).

Now, rearranging these experimental results with the ratio ($Q_f\sqrt{\Delta p}$) of the fuel consumption (Q_f cc/s) to a square root of the venturi pressure (Δp cmAq.) as shown in Fig. 11, it seems that there is clearly no correlation the fuel amount and the venturi pressure, because the static pressure measured by a manometer indicates the mean value of pressures caused by the air velocity and pulsation waves. Therefore it appears to be a matter of course that such mean and static pressure should not be proportional to the fuel consumption.

As shown in Table 3, the near the carburetor to the inlet port, the higher becomes the fuel lever, which also vibrates violently.

It is seen obviously in the same table that such phenomena in the fuel level correspond closely to the combustion state. The irrational fact that the fuel flow decreases with the rising of the fuel level in the float chamber can be explained as follows.

If a large positive pressure of the pulsation wave in the intake pipe acts the fuel level in the main jet of carburetor and consequently the level is gone down, the time required for the restoration of it equals the half period of proper vibration of the fuel column containing between the main jet and the float chamber, which is estimated roughly by the following equation.

$$T=\pi\sqrt{\frac{l_1+(f_1/f_2)l_2+(f_1/f_3)l_3+(f_1/f_4)l_4+(f_1/f_5)l_5+(f_1/f_6)l_6}{g(1+f_1/f_6)}} \qquad (1)$$

where f is the sectional area; l is the length in the model of carburetor as shown in Fig. 12.

The value of half period ($T=0.277$ sec) calculated by Eq. (1) using the dimentions of carburetor is so large that the fuel is restrained by the positive wave to flow out from the jet. To confirm this presumption, it is necessary to record directly the pressure variation in the venturi-part using a high speed typed pressure indicator instead of the manometer.

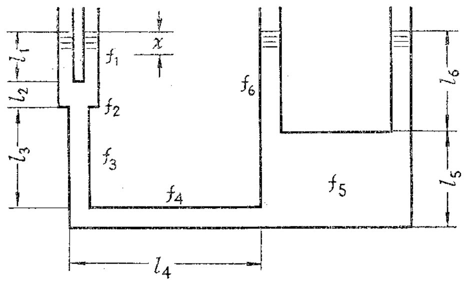

Fig. 12. Model of vaibrating system in carburetor.

$f_1=0.038$ cm^2 $f_2=0.0805$ cm^2 $f_3=0.0019$ cm^2
$f_4=0.0598$ cm^2 $f_5=4.9$ cm^2 $f_6=1.1$ cm^2
$l_1=0.7$ cm $l_2=1.34$ cm $l_3=0.3$ cm
$l_4=0.75$ cm $l_5=1.0$ cm $l_6=1.37$ cm

3. 1. 2 Pressure variation in verturi-part of carburetor

The pressure variations in the venturi-part, the inlet port, the cylinder and the crankcase are shown in Figs. 13, 14 and 15.

In comparison between these pressure variations and the fuel consumption per a square root of the static negative pressure ($Q/\sqrt{\Delta p}$) shown in Fig. 11 or combustion states in Table 3 (or Fig. 6), the amplitude and the sign of pulsation wave arrived at the venturi-part at the inlet port opening seem to be most important to the variation of the $Q/\sqrt{\Delta p}$ curves or combustion states.

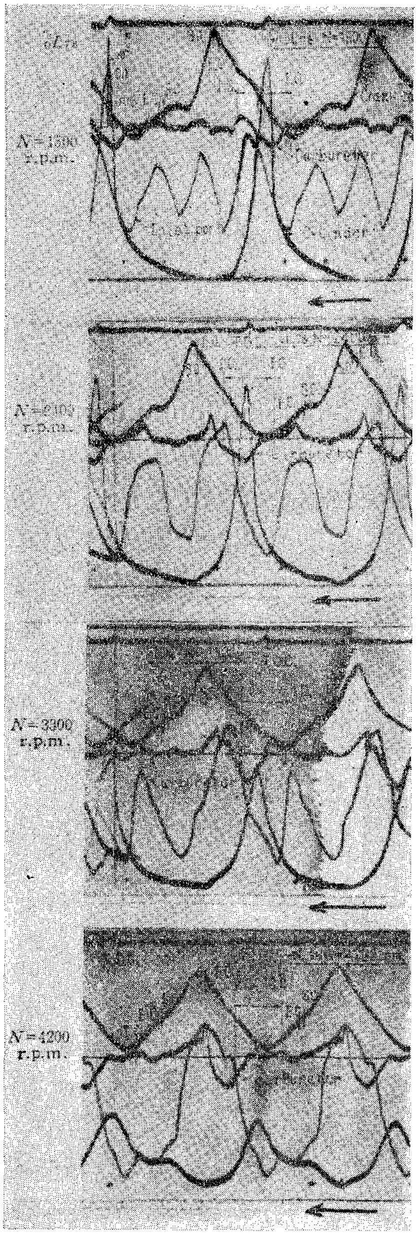

Fig. 13. Pressure variations in venturi-part of carburetor ($_0L_{78}$).

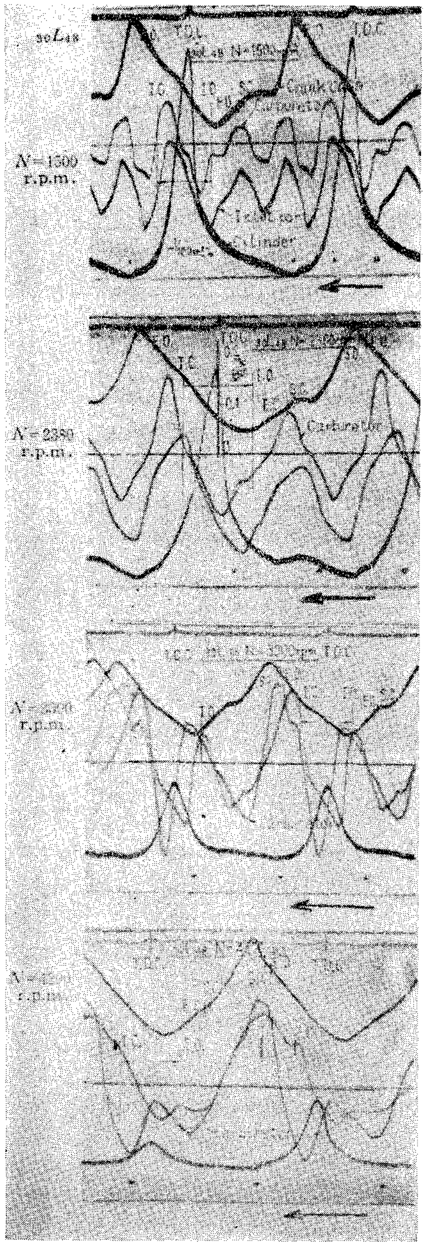

Fig. 14. Pressure variation in venturi-part of corburetor ($_{30}L_{48}$).

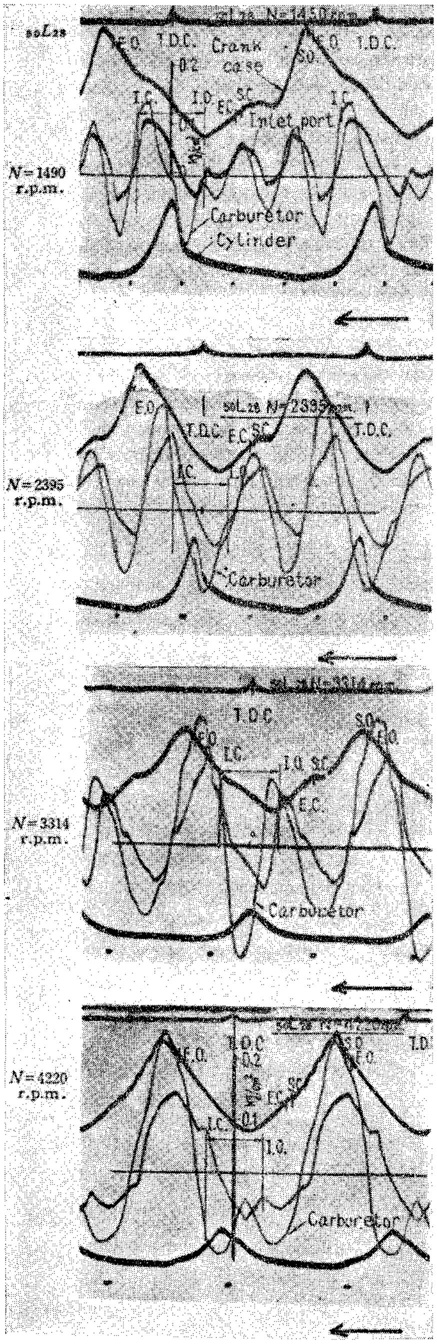

Fig. 15. Pressure variation in venturi-part of carburetor ($_{50}L_{28}$).

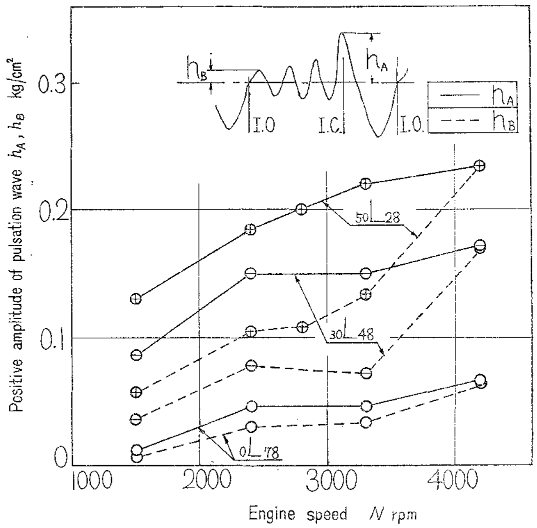

Fig. 16. Positive amplitude of pulsation wave at I.O. and I.C. (E-50).

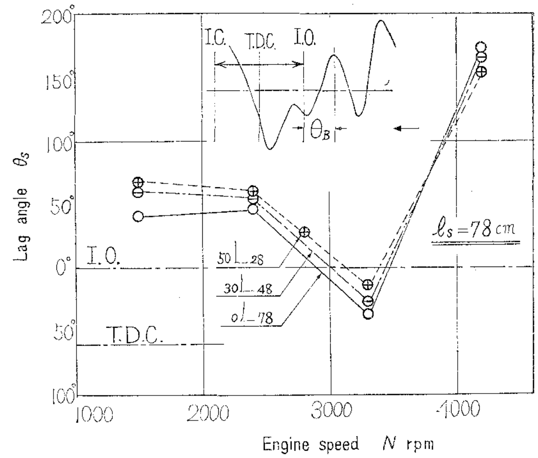

Fig. 17. Lag angle θ_s between I.O. and positive wave at near I.O. (E-50).

In order to certify such consideration, the maximum amplitude of positive wave at the inlet port closure h_A and at the inlet port opening h_B obtained from the oscillograms is plotted in Fig. 16.

It is seen therefrom that the latter amplitude h_B affects mainly the flow amount of fuel during the intake process, and tends to become larger as the carburetor approaches the inlet port and the engine runs faster.

On the other hand, the lag angle θ_B between the inlet port opening (I.O.) and the positive wave near I.O. is plotted against the engine speed in Fig. 17, because the angle seems to be a criterion indicating the sign of pulsation wave.

As shown in the Figure, such angles remain constant in spite of the position of carburetor, but change considerably with the engine speed. It is concluded easily that the lag angles for the engine speed $N=2700\sim3600$ rpm equal approximately to zero; in other words the positive waves occur just before or after the inlet opening (I.O.), so that the fuel is obstructed from flowing out from the main jet and accordingly the ratio of the fuel consumption (Q_f) to a square root of the venturi pressure ($\sqrt{\Delta p}$) for every position of carburetor decreases in this speed range and the combustion becomes worse.

In the opposite direction, owing to the increase of $Q_f/\sqrt{\Delta p}$, the combustion state for ${}_{70}L_8$ at higher speed ($\sim$4200 rpm) becomes slightly good as shown in Table 3, because the negative part of pulsation wave with a large amplitude arrives at the inlet opening (I.O.).

Then the pulsation coefficient q_s expressed by the following equation seems now to be an important parameter as the criterion showing whether the pulsation wave at the inlet opening is positive or negative.

$$q_s=(a_s/4\cdot L_s)/(N/60)=15\cdot a_s/(NL_s) \qquad (2)$$

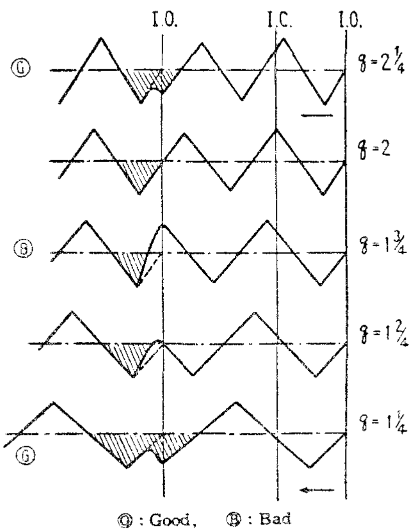

Fig. 18. Matchng condition of pulsation wave.

where a_s is the sonic velocity in the intake pipe system; L_s is the effective total length; N is the engine revolution per minute.

In Fig. 18, when the positive wave reaches the venturi-part just before I.O.; that is $q=1+3/4$, the flow of fuel decreases remarkably or ceases entirely as shown in Fig. 10 because of the shortening of the period of the negative pressure in venturi-part after I.O..

On the contrary, at $q=1+1/4$ or $2+1/4$, that is $n+1/4$, the negative wave arrives at the venturi just before I.O., and then the pressure in venturi chamber during the first half of the intake process is so low and so long that the fuel is supplied sufficiently to the engine. However, the amplitudes of pulsation wave for $n\geqq3$ are so small that the combustion state is scarcely affected by the pulsation wave.

3.2 The effects of intake pipe system on the fuel blow-out

To examine in detail the blow-out phenomena, the fuel consumption (θ_f) and the blow-out amount of fuel (q_f) are mainly measured by the flow-meter and the blow-out ratio (q_f/θ_f) are plotted against the engine speed in the Fig. 19. In the figure, it is clear that the blow-out ratio (q_f/θ_f) is largely affected by the dimensions

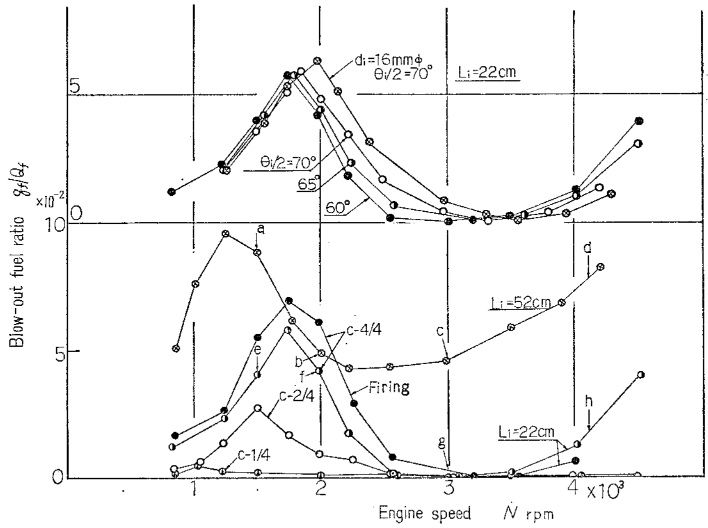

Fig. 19. Blow-out fuel ratio and engine speed.

of engine as well as in the case of the delivery ratio, particularly the longer the pipe length (L_i), the lower the engine speed (N_{max}) where the maximum blow-out ratio $(q_f/\theta_f)_{max}$ is given.

In order to understand such variation of the blow-out ratio curve, it is necessary to analyze the pressure indicator diagrams. Some typical oscillograms for $_0L_{44}(L_s=52\text{ cm})$ and $_0L_{14}(L_i=22\text{ cm})$ are presented in Figs. 20 and 21 respectively, where the numbers of oscillograms (a, b, ···, etc.) correspond the notations (a, b, ···, etc.) in Fig. 19.

In these oscillograms, when the inlet port opens, the pressure at the inlet port shows at first a negative pressure due to the depression in the crankcase caused by the motion of piston, and then it changes the sign to positive on which results

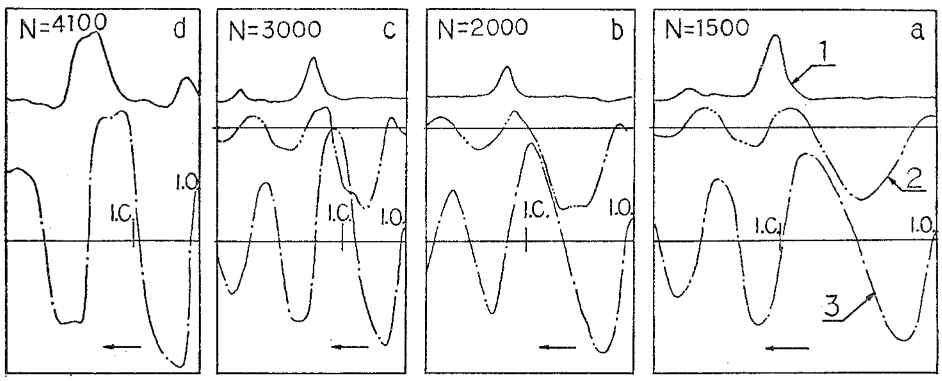

Fig. 20. Osillogram for $_0L_{44}(L_s=52\text{ cm})$; E-50.

1: Blow-out force 2: Pressure in venturi part of carburetor
3: Pressure in inlet port

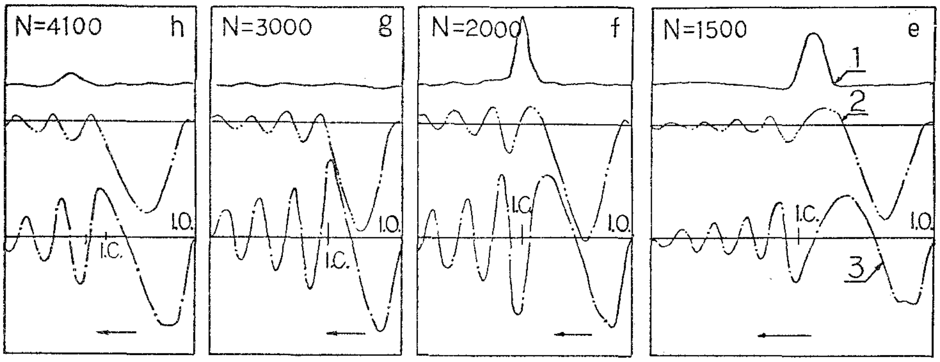

Fig. 21. Osillogram for ${}_0L_{14}(L_s=22\,\text{cm})$; E-50.
1: Blow-out force 2: Pressure in venturi part of carburetor
3: Pressure in inlet port

mainly from the inertia of air column in the intake pipe.

At lower engine speed, such positive pressure wave enters into the crankcase completely, but owing to too late closure of the inlet port it flows back again into the intake pipe as shown in Fig. 20 (a) and Fig. 21 (e), (f) and consequently the blow-out ratio reachs nearly a maximum value at the point (a), (e), (f) in Fig. 19. However, if the engine speed increases slightly and the inlet port closes just when the positive wave enters into the crankcase, in other words when the entering velocity of mixture is equal to zero as shown in Fig. 20 (b), (c) and Fig. 21 (g), the blow-out ratio attains a minimum value as shown in Fig. 19.

When the engine runs so fast as in the case of Fig. 20 (d) and Fig. 21 (h), the blow-out ratio increase again, since the inlet port closes too fast before the positive wave flows into the crankcase.

Accordingly, it appears to be certain that the blow-out ratio depends considerably on the matching condition between the pressure wave and the closure timing of inlet port as well as in the case of the delivery ratio. Then the number m in the matching condition of inertia effect expressed by the following equation seems now to be an important parameter as the criterion showing whether the blow-out amount due to inertia effect is maximum or minimum.

$$1/Z_{iM}^2=\left(m\cdot\frac{180}{\theta_i^*}\right)^2+U^2 \tag{3}$$

where $Z_{iM}=(\omega L_i^*/a_i)_M$ is inertia coefficient where the maximum inertia-effect is given, θ_i^* is the effective intake period, U is the coefficient of flow resistance per air column in intake pipe and m is the numbers of air column vibration.

If the number m equals to an odd number, then the blow-out ratio (q_f/Q_f) decreases, because the inlet port closes just when the entering velocity of mixture is equal to zero.

Conversely, when the number m equals to an even number, the blow-out ratio attains a maximum value in order to the issuing velocity of mixture is equal to zero.

Some experimental results are plotted against Z_i in place of N in Fig. 22, where all curves for each pipe length become hallow at the value of $m=1$, 3 and peak at $m=2$, which shows a good agreement with the value calculated by Eq. (3) as shown same figure.

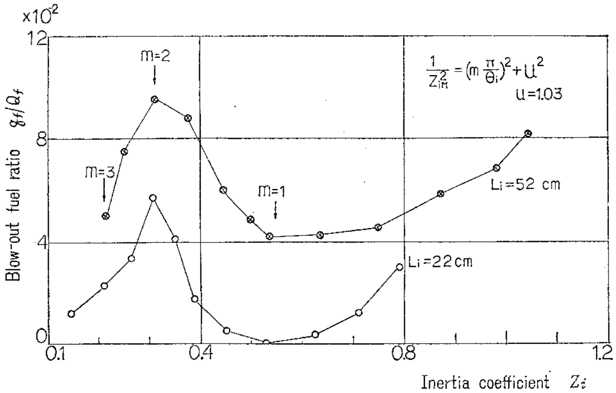

Fig. 22. Blow-out fuel ratio and inertia coefficient.

4. Conclusions

The results reached are summarized as follows:

a) The air-fuel ratio in motoring operation differs scarcely from that in firing.

b) In the case of the carburetor is installed at the engine side, the air-fuel ratio as well as the delivery ratio are affected by the intake-pipe length.

c) With an intake pipe of constant length, the delivery ratio is scarcely affected by the position of carburetor in the pipe system, but the fuel consumption varies extensively by the position of it.

d) When the carburetor is inserted fairly close to the inlet port, the fuel level in the float chamber vibrates and rises up to the ceiling of chamber, and then the fuel consumption decreases more than in the case of the normal runnings.

e) In general, the nearer the carburetor to the inlet port, the worse the combustion state in cylinder becomes.

f) The poor combustion or mis-firing is caused by the lack of fuel, which depends largely on the amplitude of pulsation wave in the intake pipe and partly governed by the matching condition q_s, that is, whether the positive or the negative of pulsation wave arrives at the venturi-part when the inlet port opens.

g) The blow-out ratio of fuel is largely governed by the various factors-pipe length, pipe diameter, the timing of inlet port and the opening of throttle valve of carburetor—as well as in the case of delivery ratio, and then the fluctuation of that is governed by the matching condition of inertia effect. $1/Z_{im}^2=\left(m\cdot\frac{180}{\theta_i}\right)^2+u^2$.

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