MEMOIRS OF THE SCHOOL OF ENGINEERING OKAYAMA UNIVERSITY Vol. 9, No. 2, DECEMBER 1974

Reduction of Engine Exhaust Noise by Throttling in an Exhaust Manifold

Kazuie NISHIWAKI* and Yuzuru SHIMAMOTO*

(Received December, 10, 1974)

Synopsis

This paper covers the work done in an attempt to reduce exhaust noise level without the decay of brake mean effective pressure of a two-cylinder two-stroke cycle engine, by means of an exhaust manifold having a throttle plate in its junction. The data are shown for various dimensions and configurations of the manifold and the discussion is given on the effect of exhaust throttling on exhaust noise level and engine performance.

The principal results obtained by this study are as follows:

(1) The lower limit of the throttle area was about one-third of a cross-sectional area of the manifold from the view point of brake mean effective pressure.

(2) Within this limit brake mean effective pressure was scarcely influenced by throttling under the condition where a number of pressure oscillation related to the pulsation effect during an interval between discharges was less than 1.7.

(3) A reduction in exhaust noise level attained was 8 dB(A) at higher engine speeds.

(4) The junction angle of the manifold gave little significance to exhaust noise level and brake mean effective pressure.

* Department of Mechanical Engineering

1. Introduction

It may be inevitable in most cases that flow resistance due to silencers and throttled sections in an exhaust system reduces engine output. Therefore, designers endevour to make a compromice between engine output and noise reduction through silencers. The utilization of dynamic effects arising in an exhaust system may be one possible way to make better noise suppression without the decay of brake mean effective pressure, in particular for a two-stroke cycle engine, the performance of which is sensitive to pressure oscillation in the exhaust system.

From this view point, an attempt was made on a two-cylinder two-stroke cycle engine to set an throttle plate at the junction of the exhaust manifold so that it might reduce the amplitude of pressure oscillation transmitted to silencers. As to the influence of the throttle plate on engine performance, there is a possibility that the oscillating component of flow velocity is absorbed through the junction into the adjacent pipe and thus the throttle gives little influence on engine performance for appropriate dimensions of the manifold in respect to the dynamic effects. This principle is deduced from the experimental work by the authors(1) which indicates that a throttle plate at the end of an exhaust pipe having a resonance pipe gives little influence on inlet charge.

2. Experimental Apparatus

The engine used for the investigation was a two-cylinder crankcase-scavenged otto engine with lead valves (Mitsubishi 2G-10), having a displacement volume of 180 cm^3 and an exhaust port opening period of 142 deg.(symmetrical timing). Fig.1 shows the configurations and dimensions of the manifold used in the experiment. As is indicated in the figure, the junction angles of the manifolds tested are 180 deg., 90 deg., and 45 deg. The lengths of the manifolds l are shown on

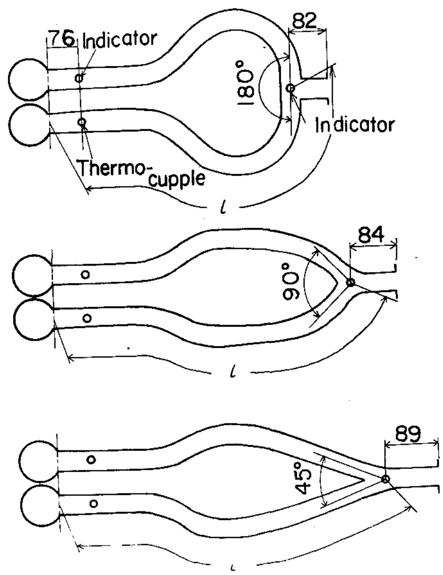

Fig.1 Configurations and dimensions of the manifolds

Table 1 Length of the manifolds

Junction angle α	Length l		
180 deg.	588 mm	700	1400
90		746	1446
45		798	

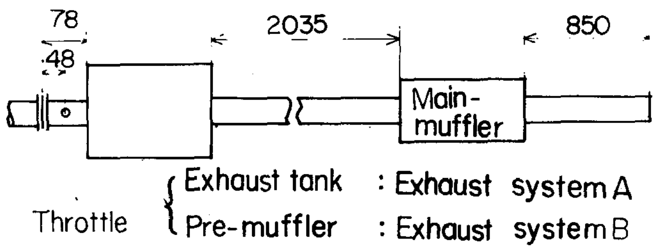

Fig.2 Exhaust systems behind the manifold

Table 1. The throttle plate was located just behind the junction. The experiment was done for three different area ratios of the opening area of the throttle to the cross-sectional area of the manifold, including 1/1, 1/3, and 1/6. In Fig.2, two exhaust systems to be connected behind the manifold are shown. The exhaust system A was used for the investigation of basic characteristics of each manifold concerning engine performance, and the exhaust system B for the measurement of exhaust noise level. The tests were carried out at full engine load.

3. Engine Performance Influenced by Throttling

3.1 Delivery Ratio

Delivery ratio,based on atmospheric conditions and the displacement volume, was obtained from flow rate through orifices in pipes. The pipes were connected to an inlet surging tank the volume of which was 1320 times as large as the displacement volume.

Fig.3 presents delivery ratio versus engine speed for the various throttle ratios and junction angles of the manifolds followed by the exhaust system A. To clarify the dynamic behaviour, the manifolds are reduced to an acoustic model as shown in Fig.4. For simplicity,

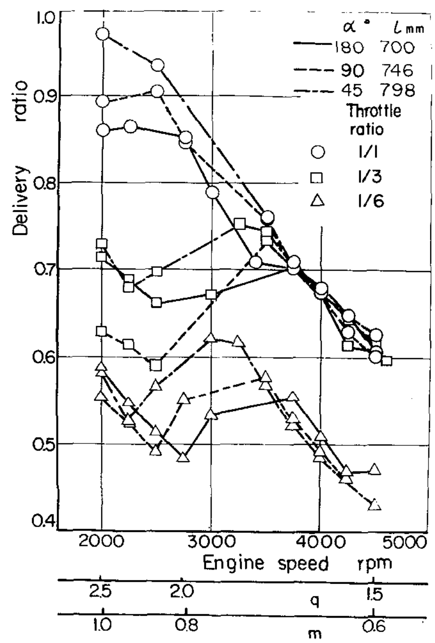

Fig.3 Delivery ratio versus engine speed, q, and m for various throttle ratios and junction angles

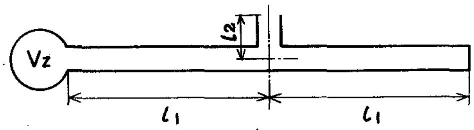

Fig.4 Acoustic model

the exhaust tank is discarded, for the volume of the tank was so large that a pressure amplitude produced there may be negligible compared with the one in the manifold. The model shown in the figure is related to the dynamic behaviour of the inertia effect. For the model related to the pulsation effect, the boundary between the cylinder and the pipe is replaced by a closed end boundary. Using those symbols which are indicated in the figure, and applying the acoustic impedance theory (2) to this model, a natural frequency ν_m (Hz) for the inertia effect may be obtained from the following relation.

$$-\frac{1}{\tan kl_2} = \frac{f/V_z k \ \tan kl_1 + 1}{-f/V_z k + \tan kl_1} - \tan kl_1$$

where

$k = 2\pi\nu_m / a$, a : sound velocity

The sound velocity was given by the indicator diagram measured just behind the exhaust port. A number of pressure oscillation m during the exhaust port opening period is then given by

$$m = \nu_m \theta_0 / 6n$$

where

n : engine speed (rpm), θ_0 : exhaust port opening period (degree)

A natural frequency ν_q (Hz) for the pulsation effect may be given by

$$\nu_q = a / 4 l_1$$

For the analysis of the pulsation effect in an exhaust system of a two-cylinder engine, any characteristic value has not been shown yet. For present purposes, a number of pressure oscillation related to the pulsation effect for an interval of the discharges (180 deg. in this case) is then taken as this characteristic value, considering that the pressure wave transmitted from the other cylinder might be a predominant factor in exhaust and scavenging processes rather than the from preceding cycle. Thus let q be this number, we have

$$q = 180 \nu_q / 6n$$

The characteristic values m and q calculated for the manifold without throttling the length of which was 700 mm, are shown below the abcissa in Fig.3. Examination of these values reveals that the inertia effect predominates over the pulsation effect in the lower speed range extended from 2000 rpm to around 3700 rpm, where delivery ratio is greatly influenced by throttling. One important relation is observable from the figure, namely, that delivery ratio for a throttle ratio of

1/3 is scarecely different from that for the manifold without throttling in the speed range higher than around 3700 rpm, where the pulsation effect is influential judging from the value of q. These trends are the same as those which were observed on the experimental work by the authors(1) on a sigle-cylinder two-stroke cycle engine with an exhaust pipe having a resonance pipe related to the pulsation effect. Based on the analysis of the previous work, it is deduced that the trend observed at the higher speeds is owing to the resonance in the manifold for the pressure oscillation of the pulsation effect. A considerable decrease in delivery ratio for a throttle ratio of 1/6 is seen in the figure within the speed range by this test. This may be attributed to the fact that the mean back pressure increases with a decreasing throttle area. Thus, the lower limit of the throttle ratio may be around 1/3, as far as delivery ratio is concerned.

Experimental results for different lengths of the manifold with a junction angle of 180 deg. are shown in Figs.5 and 6. The former is the data for the speed range where the inertia effect is predominant, and the latter is the result for the speed range where delivery ratio is chiefly influenced by the pulsation effect. It is seen from Fig.5

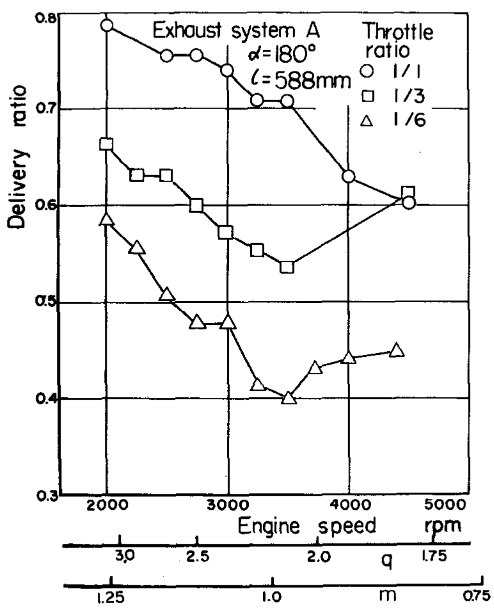

Fig.5 Delivery ratio versus engine speed, q, and m in the range where the inertia effect is predominant

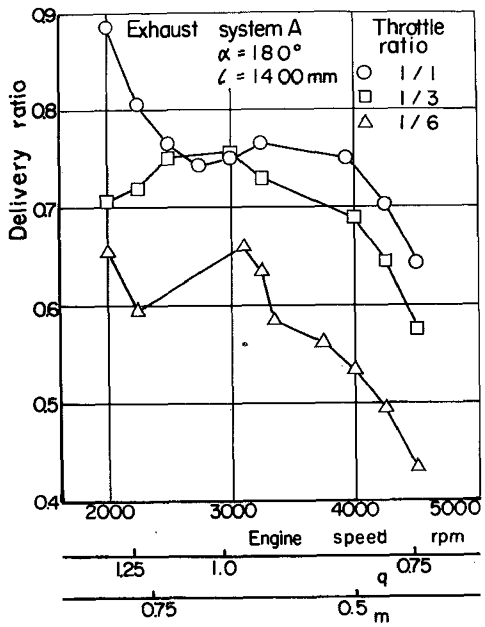

Fig.6 Delivery ratio versus engine speed, q, and m in the range where the pulsation effect is predominant

that delivery ratio is greatly influenced by throttling. On the other hand, it appears in Fig.6 that a difference between delivery ratios at throttle ratios of 1/1 and 1/3 is comparatively small. As stated above, this trend is due to the resonance in the manifold which causes little difference to delivery ratio for the varying throttle ratio. Fig.7 shows the data for a junction angle of 90 deg. and a manifold length of 1440 mm. Comparing the results indicated in Figs.6 and 7, little significance is seen concerning the junction angle.

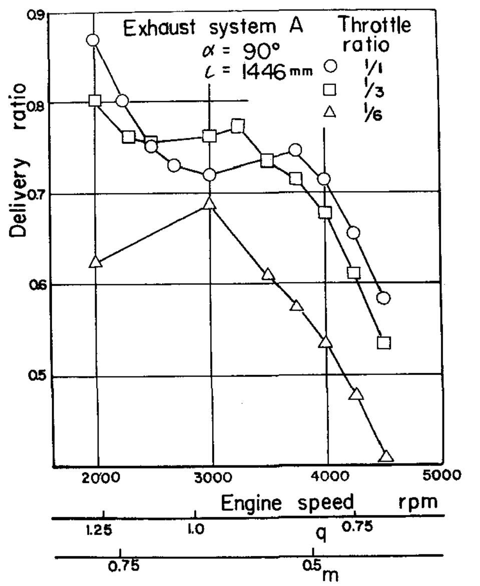

Fig.7 Delivery ratio versus engine speed, q, and m for a junction angle of 90 deg.

3.2 Brake Mean Effective Pressure

A series of tests were carried out for the manifolds with a junction angle of 180 deg. and throttle ratios of 1/1, 1/3, and 1/6. Figs. 8, 9, and 10 show the brake mean

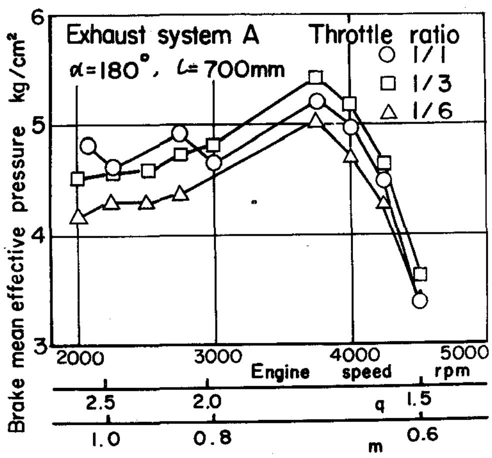

Fig.8 Brake mean effective pressure versus engine speed, q, and m in the range where both the inertia and pulsation effects are influential

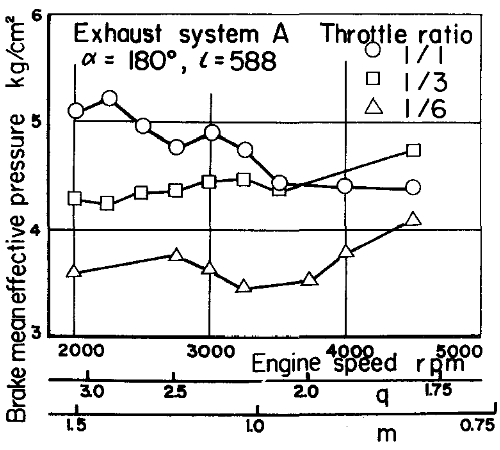

Fig.9 Brake mean effective pressure versus engine speed, q, and m in the range where the inertia effect is predominant

effective pressure related to engine speed, q, and m. Here the brake mean effective pressure is seen to decrease with decreasing throttle ratio in the range where the inertia effect is predominant. The amount of the decrease, however, is not so large compared with the decrease in delivery ratio as previously shown. Presumably this is owing to a short-circuited fresh charge during the scavenging period. One interesting trend is observable from these figures, namely, that brake mean effective pressure for a throttle ratio of 1/3 increases slightly comparing with that for a throttle ratio of 1/1 in part of the range where the pulsation effect is influential. The tendency may not be well explained by the resonance in the manifold. To clarify this point, the indicator diagrams were taken just behind the exhaust port. The results are shown in Fig.11. The high pressure pulse observed at 3750 rpm just before the time of exhaust port closing is the reflected pressure pulse from the other cylinder whose exhaust port is closed. This pulse may be effective to prevent fresh charge from flowing out after scavenging port closing. It will be noted on this figure that an amplitude of the reflected

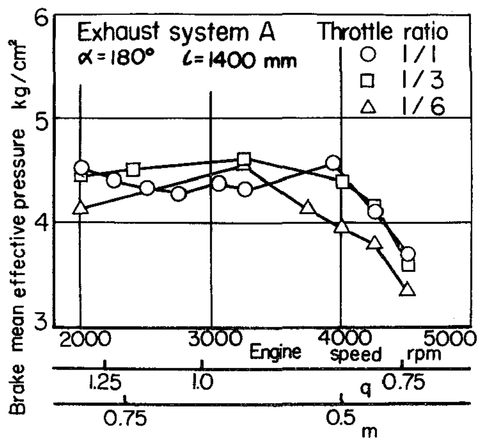

Fig. 10 Brake mean effective pressure versus engine speed, q, and m in the range where the pulsation effect is predominant

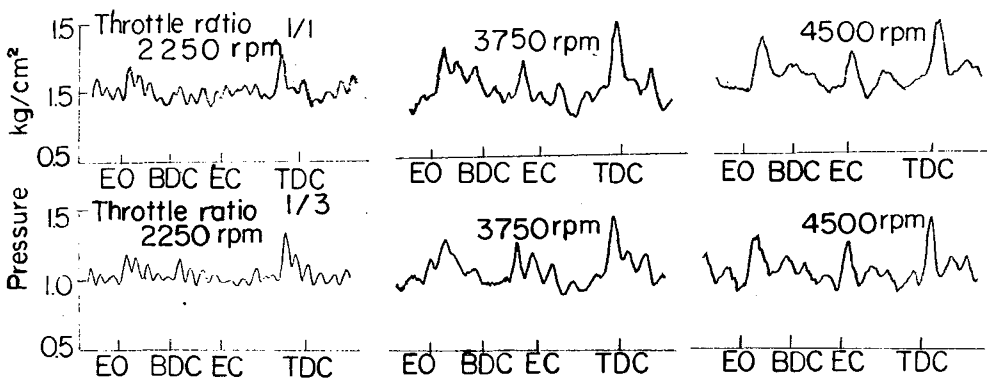

Fig.11 Indicator diagram just behind the exhaust port

pulse is greater at a 1/3 throttle ratio than that at a 1/1 throttle ratio. As will be expected, higher reflected pulse results in improved charging, and therefore in increased brake mean effective pressure.
It is seen from the figure that reflected pulse at 2250 rpm reaches too early and that at 4500 rpm comes back too late to improve charging. Thus the favourable effect of throttling on brake mean effective pressure may be mainly due to the reflected pulse which increases in amplitude with a decreasing throttle ratio, and small influence on delivery ratio on account of resonance in the manifold. As a considerable decrease in brake mean effective pressure is seen at a throttle ratio of 1/6, the lower limit of the throttle ratio may be about 1/3 also in respect to brake mean effective pressure.

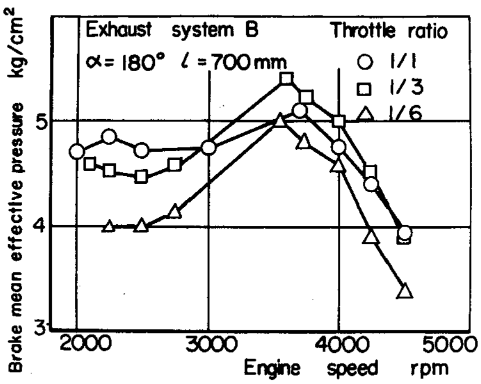

Fig. 12 Brake mean effective pressure versus engine speed for the exhaust system with a silencer

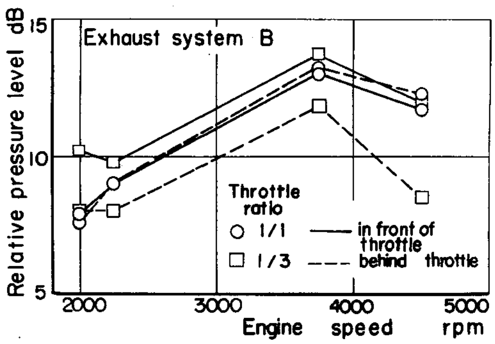

Fig. 13 Relative pressure levels in front of and behind the throttle

4. Exhaust Noise

For the study of exhaust noise emitted from the exhaust system with the manifold under consideration, the exhaust system B shown in Fig.1 was attached behind the throttle plate.

The measuring and recording equipment consists of a condenser microphone (Rion UC-11), which is used in conjunction with a precision sound level meter (Rion NA-51), a spectrum analyzer (SA-35), and a level recorder (Rion LR-01E). For spectrum analyze, the output of the precision sound level meter was recorded on a data recorder (TEAC R-410), and supplied to the spectrum analyzer thereafter.

As noise leve is affected by cylinder pressure at the beginning of blow-down, brake mean effective pressure, which is approximately proportional to this pressure, was obtained for the exhaust system B. The result is shown in Fig. 12. Comparison of brake mean effective pressures for the exhaust system B and A reveals that little difference arises between them, in particular at throttle ratios of 1/3 and 1/6. One explanation for this might be the fact that reflected pressure wave from the exhaust system following the throttle is damped in large volume of the pre-muffler and through the throttle, and thus gives little influence to the pressure variation at the exhaust port, which was experimentally observed on indicator diagrams. From the above discussion, the exhaust system B and A in conjunction with the manifold are considered to be under approximately the same condition in respect to brake mean effective pressure.

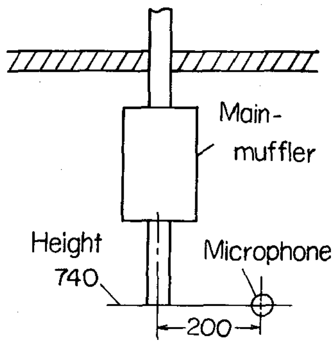

Fig. 14 Location of the microphone

In Fig. 13, relative pressure levels measured just in front of and behind the throttle are shown for various throttle ratios. As would be expected, the pressure level is seen to decrease with a decreasing throttle ratio.

Noise level and spectrum were measured outside of the test room as shown in Fig. 14. No significant influence of reflected sound was observable by the preliminary test. Figs. 15 and 16 show noise level in dB(C) and dB(A), respectively, obtained for the manifold with junction angle of 180 deg. and a length of 700 mm. It is seen from

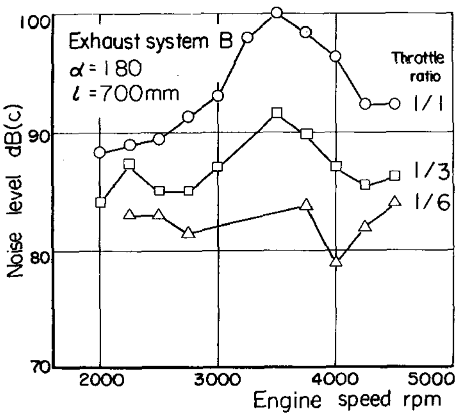

Fig. 15 Noise level in dB(C) versus engine speed for various throttle ratios

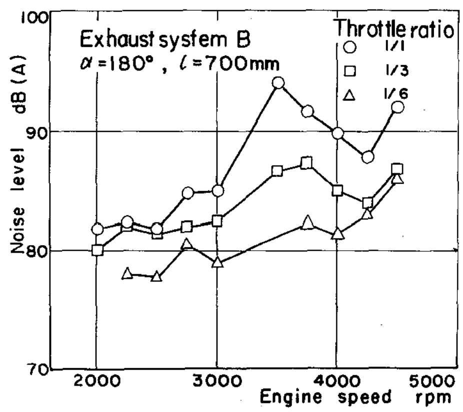

Fig. 16 Noise level in dB(A) versus engine speed for various throttle ratios

these figures that a considerable reduction in noise level is attained by throttling, in particular at higher speeds, for instance noise levle at a throttle ratio of 1/3 decreases by 8dB(A) at 3500 rpm as compared with that at a throttle ratio of 1/1. Though a decrease in noise level at a throttle ratio of 1/6 is seen to be greater, a decay of engine output is significant as previously shown. In Fig. 17, noise level in dB(C) at a throttle ratio of 1/3 is shown for various junction angles. It appears in Fig.17 that a difference in noise level for different junction angles is less than 3 dB. Therefore the result indicates that a junction angle of the manifold has little influence on noise level.

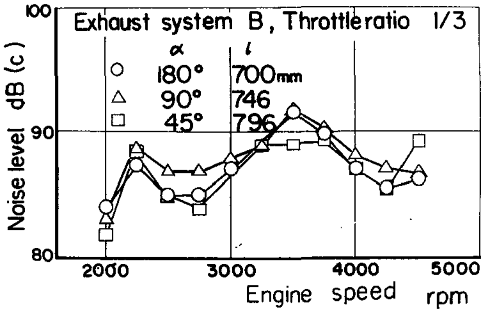

Fig. 17 Noise level in dB(C) for various junction angles

Fig. 18 shows the typical noise spectrums for the manifold with a junction angle of 180 deg. and a length of 700 mm for various throttle ratios. It is noticeable in Fig. 18 that a reduction in spectrum level by throttling is more pronounced at lower frequencies. This change in noise spectrum results in smaller difference in noise levels in dB(C) and dB(A) at throttle ratio of 1/3 and 1/6 as shown in Figs. 15 and 16.

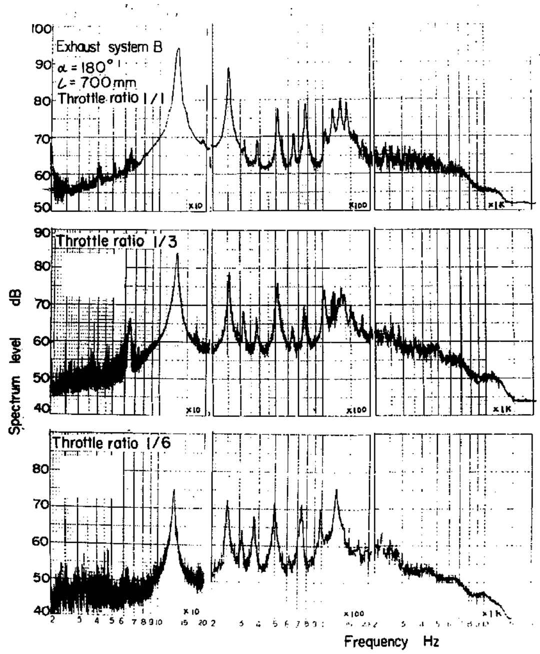

Fig. 18 Typical noise spectrums as influenced by throttling

5. Conclusion

By the use of the manifold having a throttle at its junction, a considerable reduction in noise level was attained without a significant decay in engine output. The experiment suggests that the manifold acts as a resonance pipe for pressure oscillation related to the pulsation effect and thus absorbs the variation of flow velocity to avoid excessive back pressure.

The lower limit of the throttle ratio was around 1/3. Within this limit, brake mean effective pressure was scarecely influenced by throttling under the condition where q is less than 1.7.

A reduction in noise level attained at a throttle ratio of 1/3 was more pronounced at higher speeds and the maximum reduction obtained was 8 dB(A).

The junction angle of the manifold gave little influence on engine output and noise level within the speed range in the test.

References

(1) K.Nishiwaki and Y.Shimamoto : Preprint of 50th Meeting, Japan Soc. Mech. Engrs., No.720-14(1972), 5.

(2) M.Maegawa : Preprint of 54th Lecture Meeting, Japan Soc. Mech. Engrs.,(1953),25.