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A Study on Mufflers with Air Flow*
(1st Report Generation of Noise from Expansion
Cavity Type Mufflers due to Mean Flow)

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This paper presents theoretical and experimental studies on the expansion cavity type mufflers with mean flow.

It is shown in this paper that the inlet part of the tail pipe is a point of the largest flow noise generation inside the cavity of the muffler and a very large air flow noise is generated when a jet stream from the inlet pipe directly impacts the tail pipe.

Internal Combustion Engine, Sound, Expansion Type Muffler, Air Flow, Attenuation Effect, Generation of Flow Noise, Position, Figure, Velocity

1. Introduction

Transmission loss (TL) characteristics of the expansion type muffler have been studied by Davis et al.⁽¹⁾ under the assumption that the tail pipe of a muffler is a non-reflective tube. On the other hand, assuming that the tube is completely reflective, Fukuda et al.⁽²⁾ have developed an attenuation insertion loss (IL) theory and Okuda⁽³⁾ has extended this approach. These theories essentially describe the properties of the expansion type muffler assuming plane acoustic wave propagation without air flow. In the case of a muffler with air flow, however, these theories are not always correct in estimating frequency characteristics of noise reduction. This is because these frequency characteristics change due to air flow or a secondary noise is generated by air flow inside a muffler.

Among investigations of acoustic waves in the presence of air flow, the characteristics of acoustic waves in a moving medium were analyzed by Trimmer⁽⁴⁾ in 1937, Blockhinzey⁽⁵⁾ in 1948 and Lambert⁽⁶⁾ in 1956. Morfey⁽⁷⁾ in 1969 discussed the relationship between reflected noise from the end of a tube and flow velocity. Also, Mason⁽⁸⁾ in 1969 treated the transmission of an acoustic wave in a circular tube containing air flow. However, these studies addressed only one aspect of the phenomena. In 1974, Munjal⁽⁹⁾ studied a muffler with air flow and developed a relationship between the noise reduction rate and the ratio of radiation impedances with and without air

flow. Unfortunately, this study was restricted to theoretical calculations without experimental verification and no useful muffler design information was given.

In Japan, Fujimoto⁽¹⁰⁾ in 1971 carried out an experimental study of the radiation noise from the tail pipe of a muffler associated with intermittent flow. In 1978, Obata, Hirata K. et al.⁽¹¹⁾ carried out statistical studies of the relationship between power and flow rate for the noise emanated from a muffler tail pipe. Furthermore, Hirata Y. in 1970 theoretically derived the properties of a standing wave in an acoustic tube containing air flow⁽¹²⁾ and the effect of air flow on the acoustic characteristics of a resonance type muffler⁽¹³⁾. Okuda⁽¹⁴⁾ in 1978 published a theoretical study of acoustic wave reflection at the end of an acoustic tube with air flow.

The authors have studied the effect of air flow on the relationship between the noise radiated from an expansion type muffler and the muffler design. In addition, they have investigated how to locate the position of air current noise generated inside a cavity. Furthermore, they have estimated air current noise by means of calculation and experiment. As a result, many interesting conclusions have been obtained together with important data for the muffler design. These results are outlined in this paper.

2. Effect of air flow on the noise reduction of a muffler

Referring to Fig.1, which illustrates a simple expansion type muffler without air flow, Fukuda⁽¹⁵⁾ in 1960 proposed the following approximate insertion loss equation for the frequency characteristics of improved noise reduction.

$$IL = 20 \log_{10} |m \sin(kl) \sin(kl_0)| \quad \text{dB} \quad (1)$$

This equation was also verified later from theoretical equations developed in 1969⁽²⁾. IL represents the difference of sound pressure

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level at the same relative positions with and without a muffler (cavity and tail tube), $m=S/s$ is the ratio of cross-sectional areas of the tube (S) and the cavity (s), k is the sound wave number, i.e., $k=\omega/c=2\pi f/c$ (ω =angular frequency, f =frequency, c =acoustic velocity), and l and l_0 are the lengths of the cavity and the tail pipe respectively.

When there is uniform air having a flow-rate of Mach number M , the acoustic velocity is given by Eq.(2), using the usual convention of + and - suffices respectively for the forward and backward directions of the flow relative to the plane acoustic waves.

$$c_+=c(1+M), \quad c_-=c(1-M) \quad (2)$$

The sound wave number is given by:

$$k_+=k/(1+M), \quad k_-=k/(1-M) \quad (3)$$

and the acoustic characteristic impedance is

$$Z_{0+}=Z_0(1+M), \quad Z_{0-}=Z_0(1-M) \quad (4)$$

where Z_0 corresponds to a case without air flow and is given by $Z_0=\rho c/s$ with ρ the density, and s the tube cross-section.

The energy density can be expressed by

$$E_+=|p_+|^2/\rho c_+^2, \quad E_-=|p_-|^2/\rho c_-^2 \quad (5)$$

For the expansion type muffler shown in Fig.1, we make the following assumptions that (i) plane acoustic waves propagate through the cavity with uniform air flow and (ii) the three pipes of inlet, cavity and outlet are connected in a cascade and can be analyzed as a matrix. In addition, total power transmission is assumed at each junction. Under these conditions the following approximate equation can be derived.

$$IL=20\log_{10}|m(1-M_{l0}^2)\sin(kl)\sin(kl_0)| \quad \text{dB} \quad (6)$$

where M_{l0} is the Mach number at the tail pipe

Fig.1 Expansion type muffler

Table 1 Calculated IL_{max} for different flowrates

流速 m/s	IL_{max} dB
10 ($M_{l0}=0.03$)	16
20 ($M_{l0}=0.06$)	16
30 ($M_{l0}=0.09$)	16
50 ($M_{l0}=0.14$)	16
100 ($M_{l0}=0.29$)	15
150 ($M_{l0}=0.43$)	14

Fig.2 Test setup

($M_{l0}<1$), $k_l=k/(1-M_l^2)$, and $k_{l0}=k/(1-M_{l0}^2)$. In Eq.(6), $\sin(k_l l)\sin(k_{l0} l_0)<1$, thus

$$IL_{max}=20\log_{10}|m(1-M_{l0}^2)| \quad \text{dB} \quad (7)$$

Table 1 shows examples of the calculated IL_{max} for a cavity and a tail pipe with diameter and length dimensions of 53×200 and 21×100 mm, respectively.

3. Air flow noise from a muffler

3.1 Purpose

(a) Straight tube only

(b) With muffler

Fig.3 Comparison of sound pressure level for various flowrates

When the air flowrate increases, what relationship will hold between air flow noise and muffler design? In answering this question the authors studied the effect of cavity diameter, length, tail pipe diameter, etc.

3.2 Experimental apparatus and procedures

Figure 2 indicates the test setup. A blower (turbo-centrifugal) was used for the noise source. The flowrate was controlled by a throttle valve located at the blower inlet. The velocity was measured by a 0.7 diameter Pitot tube at 166 mm upstream position from the cavity inlet and the mean flowrate was obtained. The noise was measured at a position 300 mm away from the tube at a 45 degree angle. The improvement in noise reduction was determined from the difference (IL) of the sound pressure levels with and without a muffler.

3.3 Generation of air flow noise by a muffler

Figure 3(a) indicates the frequency characteristics of a radiated noise (SPL) from a straight tube with no muffler attached. Essentially no change in SPL is observed when the flowrate is changed from 10 m/s to 30 m/s, except in the high frequency range. A large noise component is observed at 800 Hz. This component corresponds to the 835 Hz fundamental frequency of the blower impeller of the sound source.

When the test setup is replaced with a tube system containing a muffler as shown in Fig.3(b), a significant noise increase is observed at a flowrate of 30 m/s in both the low frequency range around 250 Hz and the high frequency range. This may be due to air flow noise generated by the muffler.

3.4 Influence of cavity diameter

In Fig.4, improved noise reduction is observed between 200 and 600 Hz for a 10 m/s air flow. Also, the sound pressure level is smaller for $m=25$ than for $m=6.4$. In the case of a 30 m/s flowrate, however, we see that much larger SPL values are found for $m=25$ than in the case of a 10 m/s flowrate.

Figure 5 denotes a comparison between calculated and measured values for improved

Fig.4 Sound pressure level for different cavity diameters

noise reduction using Eq.(1). The required frequency condition for plane wave propagation is $f=1.22c/d$ Hz⁽¹⁶⁾ (c =acoustic velocity, d =diameter of pipe) or less. Therefore, the calculation is made only below this frequency (4kHz). Equation (1) does not take into account the effect of air flow; nevertheless, the calculated values agree fairly well with the measurements for a 10 m/s flowrate. On the other hand, at 30 m/s, there is a significant difference. Also, the larger the cavity diameter, the larger is the noise increase due to air flow.

3.5 Influence of cavity length

Figure 6 is an example of the noise increase when the flowrate changes from 10 m/s to 30 m/s in cavities with lengths between 200 to 600 mm. It is noted that the noise increase

Fig.5 Noise reduction efficiency and noise increases for different diameters

Fig.6 Noise increases for different cavity lengths

due to air flow is extremely large with a cavity length of 200 mm. In this case there are two significant noise peaks, one is the low frequency range around 250 Hz and the other in the high frequency range of 3–4 kHz.

3.6 Influence of tail pipe diameter

Figure 7 shows a variation of the sound pressure level, improved noise reduction and noise increase of the emitted noise, of various tail pipe diameters. The improved noise reduction decreases as the tail pipe diameter increases. Also, the noise is smaller for high flowrates and high frequencies. This may be because, for smaller tail pipe diameters, the velocity of the air flow entering the tail pipe increases and causes more noise at high frequencies. The resonance frequency f_c is also a function of the tail pipe length. This resonant frequency is given by

$$f_c = c(1 - M_0^2)/2(l_0 + \Delta l) \quad \text{Hz} \quad (8)$$

where Δl_0 is an aperture compensation term. For each figure, it is estimated that the high frequency components between 1.2–1.7 kHz are affected by the resonance of the tail pipe length.

4. Effect of adapters on air flow noise

A number of adapters have been devised to

Fig. 7 Improved reduction and noise increases for different diameters

prevent air flow noise, assuming that this noise is generated mainly by the air turbulence caused by a sudden change in slope of the cavity inlet/outlet. Table 2 presents the most effective adapters.

First of all, we need to investigate whether the acoustic characteristics of these mufflers change depending on the type of adapter used. Figure 8 is a comparison between the muffler A and a muffler with B+D+E

Table 2 Mufflers with various adapters

Mufflers	Sketch	Remarks
A		No adapter
B		Bellmouth on inlet end
C		Perforated pipe on inlet end
D		Bellmouth on tail pipe inlet
E		Perforated conical plate on cavity outlet end

Fig. 8 Comparison between simple muffler and combined one with composite adapter (speaker test)

Fig. 9 Comparison between simple muffler and combined one with composite adapter (continuous air current test)

adapters in terms of improved noise reduction, using a loud speaker source. It can be seen that there is hardly any improvement in the noise reduction. As shown in Fig. 9, employing a blower as a sound source leads to less radiated noise from the tube end when adapters are used. This may be the result of a noise suppressive effect of the adapter on air flow.

These phenomena have also been investigated in more detail, as shown in Fig. 10. Types B and C adapters are intended to prevent turbulent air due to sudden changes in slope by installing the adapter at the inlet end of the muffler. With these types B and C, there is a large improvement in noise reduction at low frequencies. However, in the high frequency range, type B adapter does not give any improvement and type C adapter causes an increase of noise due to the high frequency noise generated from the perforated tube. In the cases of adapter types D and E, there is a slight improvement in noise reduction in the low frequency range. In the high frequency range, type D adapter has a significant effect on air flow noise suppression, while the type E adapter is less effective. The following describes the authors' explanation of these results.

5. Locating air flow noise inside the muffler cavity

5.1 Purpose

It is important to locate where a secondary noise is generated by air flow within the muffler cavity. The authors proceeded to investigate where the largest noise generating location was inside the cavity, on the basis of the previous study on the comparative effectiveness of a number of adapters.

5.2 Theoretical analysis

Let us assume that the power level of the emanated noise, $L_{w'}$, from the tube end without the muffler, as shown in Fig. 11, is defined by

$$L_{w'} = 10 \log_{10} W' / 10^{-12} \quad (9)$$

From Eq. (9), the emanated power W' is calculated as follows:

$$W' = 10^{(L_{w'} - 10)/10} \quad (10)$$

When a muffler is present, the power level, L_w , of the emanated noise from the muffler tail pipe and emanated power, W , are given by:

$$L_w = 10 \log_{10} W / 10^{-12} \quad (11)$$

$$W = 10^{(L_w - 10)/10} \quad (12)$$

Next, let us assume that no muffler is attached and the air flow noise is negligible at low flowrates. In that case, the power level and emanated power of the noise generated at the tube end will be represented by $L_{w'l}$ and W_l' respectively. Furthermore, let us define the noise parameters L_{wh} and W_h' for the case where a significant air flow noise is generated because of high flowrates (all other conditions remaining the same). In addition, with a muffler attached, the foregoing parameters are expressed by L_{wl} , W_l , L_{wh} and W_h , respectively. Also, the power ratio λ is defined as

$$W_i = \lambda W_l' \quad (13)$$

Using these definitions, the insertion loss

IL (dB) of the muffler is obtained from $L_{wl}' - L_{wl}$, i.e.:

$$IL = 10 \log_{10} 1/\lambda \quad (14)$$

Therefore, λ is given by:

$$\therefore \lambda = 10^{-1/10 IL} \quad (15)$$

The emanated power W_h at high flowrates is calculated using:

$$W_h = \lambda W_{h'} + W_f \quad (16)$$

where W_f is the power of the initial air flow. From these equations, the secondary noise power level L_{wf} (dB) which emanates from the muffler due to the air flow is given by

$$L_{wf} = 10 \log_{10} W_f / 10^{-12} = 10 \log_{10} \frac{W_h - \lambda W_{h'}}{10^{-12}} \quad (17)$$

$$= 10 \log_{10} [10^{L_{wh}/10} - 10^{(L_{wh'} - (L_{wl}' - L_{wl}))/10}] / 10 \text{ dB}$$

Consequently, the air flow sound pressure level L_f at a distance 'r' from the tube end is expressed by

$$L_f = L_{wf} - 10 \log A \quad (18)$$

where the propagating area A (m²) is given by $4\pi r^2$ in a free space and $2\pi r^2$ in a semi-free space.

Furthermore, by assuming sound pressure levels of L_h' , L_l' , L_h and L_l corresponding respectively to the power level L_{wh}' , L_{wl}' , L_{wh} and L_{wl} , at the same relative position from the pipe end, the air flow sound pressure level can be calculated as follows

$$L_f = 10 \log [10^{L_h/10} - 10^{(L_h' - (L_l' - L_l))/10}] \text{ dB} \quad (19)$$

In other words, the frequency characteristics of the air flow sound pressure level L_f generated in the muffler can be calculated by obtaining the frequency characteristics of

Fig. 10 Noise generation for each adapter ($w = 28$ m/s)

Fig. 11 Powers W , W' and power levels L_w , L_w' of emanating noise

muffler and air flow.

5.3 Experiment and calculation

Figure 12 shows air flow noise calculations assuming a muffler of 300mm cavity length and a flowrate of 28m/s, while neglecting air

Fig.12 Calculation of the air flow noise from the total measured noise

Fig.13 Calculation of the air flow noise at various positions inside the cavity

flow noise at flowrates less than 10 m/s. Each adapter is the same as shown in Table 2. It is seen that for both the low and high frequency ranges, the air flow noise becomes a significant fraction of the total.

Figure 13 shows the generated noise at the cavity inlets 1 and 2, obtained from the difference of air flow noise for the mufflers D and B+D. The same figure shows the generated noise at the cavity outlet corner 3 also calculated in the same way as for the mufflers D and D+E. Similar calculations are also shown for the tailpipe inlet 4 and the mufflers A and D. Referring to these results, it is clear that, of all the positions generating noise, position 4 generates the largest flow noise in both the low and high frequency ranges. In the former experiments shown in Fig.10, low frequency noise was reduced most effectively by means of adapters that do not have steep slopes at the cavity inlet, e.g. types B, C, etc. Referring to the result shown in Fig.14, however, the noise reduction was most effectively achieved by preventing impact noise generated when a jet stream from the inlet tube end collides against the tail

Fig.14 Calculation of the air flow noise from the total measured noise

Fig.15 Calculation of the air flow noise at various positions inside the cavity

Fig.16 Wall surface pressure distribution inside the cavity

tube inlet end. The actual generating position of the noise might be 4 according to the result of the experiment.

Another experiment was also carried out with a muffler of 200 mm cavity length. The results obtained as shown in Figures 14 and 15 were essentially the same as those shown in Fig.13. To verify these results, we measured the pressure distribution at the cavity wall as shown in Fig.16. With a cavity length of 400mm, the jet stream occurs close to the wall surfaces at $x/l = 0.7$. With a cavity length of 300mm, it is estimated that the total jetting energy might actually impact the tail pipe inlet. Referring to the results shown in Figures 5 and 6, the fact that the generated air flow noise increases with an increasing cavity diameter and a decreasing cavity length can be understood assuming that the tail pipe acts like a reed instrument.

6. Conclusions

The result of the present research is summarized as follows:

- (1) The air flow noise generated from the muffler is negligible for flowrates less than 10 m/s in the tube. For flowrates higher than 30 m/s, the noise becomes significant.
- (2) The inlet of the tail pipe is a point of the largest air flow noise generation inside the cavity. Very large air flow noise is generated when a jet stream from the inlet pipe directly impacts the tail pipe. The generated noise increases as cavity diameter increases or cavity length decreases.

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