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Letter to the Editor

New active muffler system utilizing destructive interference by difference of transmission paths

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1. Introduction

The exhaust noise of an internal combustion engine is usually composed of the most dominant frequency components directly related to engine revolution and wideband noise. For example, the four-cylinder gasoline engine used in experiments has dominant C2 and C4 components, which are double and quadruple of the rpm frequency, and mid-to-high frequency wideband noise.

Muffler is the most popular method to reduce exhaust noise. Generally the muffler has to work over a broad engine rpm range with a fixed internal structure. It makes the muffler design very difficult and results in a compromise, which makes the internal structure very complex and increases backpressure that reduces engine power and lowers fuel efficiency. The engine power loss due to conventional passive exhaust muffler system is known to be at least 10% and it is reported that 5% improvement of fuel efficiency results in about 2000 US dollar savings for heavy trucks [1].

To reduce the problems caused by high backpressure of conventional mufflers, various attempts like dual mode muffler, which has a valve operated by either gas pressure or an actuator and can change the gas stream flow under various levels of engine load, have been tried with only limited success [2]. Active muffler systems, which usually use a speaker to cancel the exhaust noise, have been very actively researched in recent years. In these methods, the speaker makes canceling noise which has the same magnitude and 180° phase difference [3–6]. The success in duct noise cancellation has proved the usability of this approach which uses the same principle [7]. However, this method has failed to be commercialized despite good results both in the laboratory and in actual car-installation test. The speaker was one of the main reasons for the failure. The engine noise has very high sound pressure level and dominant very low-frequency components. The low cost production of a suitable speaker, which has to be small but powerful enough to produce a very high sound pressure level and has very good low-frequency characteristics, is almost

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impossible to achieve. So along with other practical problems related to manufacturing, durability, and maintenance, this method has not yet made the transition into successful commercial products. Various other attempts like using a volume variable resonator or multiple valve system have also failed due to numerous problems inherent in their design [8,9].

The authors have proposed a new method for exhaust noise control of an internal combustion engine [10]. In this method, a U-shaped bypass pipe is attached to the original exhaust pipe. The bypass pipe has a length variable mechanism so that when two divided noises meet, the pipe length difference is adjusted to make the major component of the noise have a 180° phase difference (half of a wavelength) thereby canceling the noise by destructive interference. This method uses the noise itself to cancel noise and does not require a secondary noise source like a speaker; hence it can overcome many problems of other active noise control methods.

A simple experiment at the laboratory with pipes and a speaker as a noise source has proved more than 20 dB decrease of the major noise component. So, in theory, this technique combined with a very low-backpressure muffler to remove the remaining wideband noise could be a very good candidate for active noise control. However, the necessary length difference required in the low rpm range is still too big to be used in a car.

In this paper, the authors propose a much-improved concept and prove its potential with experiments. The major improvement is the combined use of a current passive muffler with a new system. In the low engine rpm range, only a passive muffler is employed and in the high rpm range, where only a short length difference is required, the proposed system with a very low-backpressure muffler is employed. In this case, the whole system size can be significantly reduced and the system becomes much more practical. This method still needs some more space than the passive muffler system, so it is better suited for larger vehicles like buses, SUVs and commercial trucks rather than passenger cars. As an example, large trucks hauling containers on highways in United States and Australia spend most of the operation time on highway where the proposed system's benefit of increased power and lower fuel consumption become very critical.

In a previous research, a system with one bypass pipe to remove C2 and a simple muffler was used for noise measurement and compared to the current passive muffler [11,12]. A second bypass pipe was added to remove C4 [13]. In this research, the system's total performance was measured and analyzed. The noise reduction performance was carefully measured and torque increase was confirmed by an engine dynamometer experiment.

2. New active exhaust noise control method

2.1. Basic concept

The basic concept of the destructive interference is shown in Fig. 1. The system is composed of a straight pipe and a U-shaped bypass pipe attached to the straight pipe. Let us assume a sinusoidal wave is traveling from the left to right direction inside the pipe. The wave is divided into two parts at point A and the divided waves are made to meet again at point B. If the transmission length difference of the two paths, $2L_2$, is equal to or an odd multiple of the half-wavelength, $\lambda/2$, of the sinusoidal wave, w_0, the two waves will have 180° phase difference and the amplitude of the combined wave, $w_1 + w_2$, will be drastically reduced as illustrated in Fig. 1(b)–(d) by the destructive interference between two waves.

This can be explained with an equation as follows. The input wave can be expressed as

$$w_0(t) = B_0 \sin(2\pi f t). \tag{1}$$

Then, the combined wave, $w_1 + w_2$, will be

$$\begin{aligned} w_{1+2}(t) &= w_1(t) + w_2(t) \\ &= B_1 \sin\left(2\pi f\left(t + \frac{L_1}{v_c}\right)\right) + B_2 \sin\left(2\pi f\left(t + \frac{L_1 + 2L_2}{v_c}\right)\right) \\ &= B_1 \sin\left(2\pi f\left(t + \frac{L_1}{v_c}\right)\right) + B_2 \sin\left(2\pi f\left(t + \frac{L_1 + (2n-1)\lambda/2}{v_c}\right)\right) \\ &= B_1 \sin\left(2\pi f\left(t + \frac{L_1}{v_c}\right)\right) + B_2 \sin\left(2\pi f\left(t + \frac{L_1}{v_c}\right) + (2n-1)\pi\right) \\ &= (B_1 - B_2)\sin\left(2\pi f\left(t + \frac{L_1}{v_c}\right)\right), \end{aligned} \tag{2}$$

where B_0, B_1, B_2 are the magnitude coefficients and v_c is the speed of the sound.

2.2. Proposed system

The proposed system's concept is shown in Fig. 2. With the control of the valve at the left, the exhaust gas and noise are fed to a conventional muffler at the bottom in a low rpm range. In the higher rpm range, where active control is engaged, the valve is switched to drive the exhaust gas and noise to the active muffler system at the top. The bypass pipes are composed of inner and outer tubes and actuators are used to change the length. The first and second bypass pipes control C2 and C4, respectively. A simple muffler, which has the minimum backpressure, is used to remove the remaining noise.

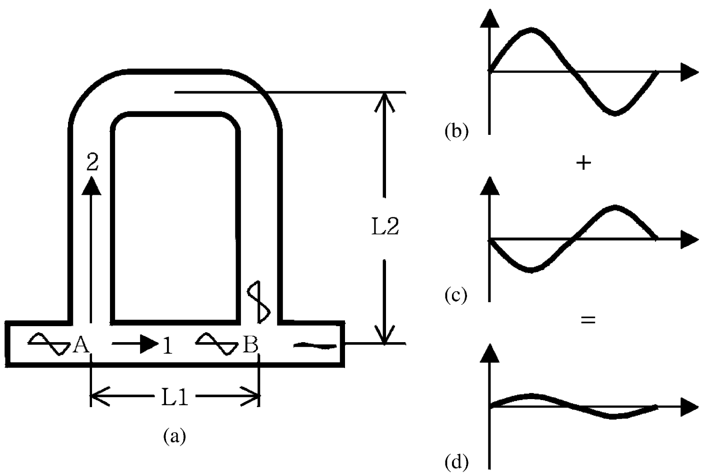

Fig. 1. Principle of the muffler using difference of transmission paths.

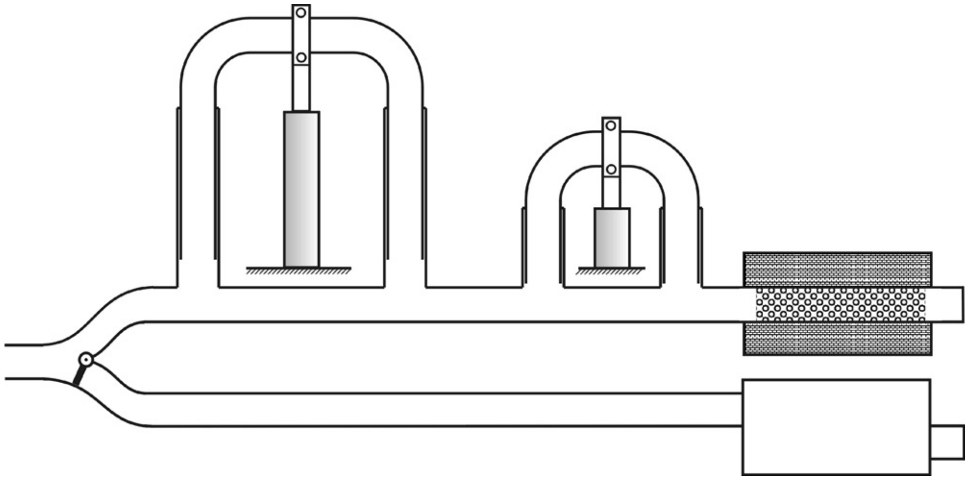

Fig. 2. Conceptual drawing of the proposed muffler system.

2.3. Transmission loss of a bypass pipe

Transmission loss (TL) is one of the most popular measures of muffler's noise reduction performance. One bypass pipe system should have the maximum TL at the frequency determined by its length difference and the peak frequency should change following the bypass pipe's length change. Such change was measured and analyzed with experiments [11–13].

2.3.1. Definition of TL

TL is defined as the difference between the power incident on the muffler proper and that transmitted downstream into an anechoic termination [14]. If we assume the same cross-sectional area and tail pipe terminating anechoically, TL equals 20 times the logarithm (to the base 10) of the ratio of the acoustic pressure associated with the incident wave, A_n, and that of the transmitted wave, A_1, as in

$$TL = 20\log\left|\frac{A_n}{A_1}\right|. \tag{3}$$

2.3.2. TL simulation of bypass pipe system

The proposed bypass pipe system's TL characteristics were estimated by simulation using SYSNOISE. Figs. 3 and 4 show the simulated TL characteristics of a single and a double bypass pipe system, respectively. In the simulation, the cross-section of the pipe is rectangular shape, $50 \times 50\,\text{mm}^2$, and each side length of the first and second bypass pipe is 1 and 0.5 m, respectively. The measurement of the setup is only for simulation purpose to check the characteristics before experiment and is not necessarily related to engine experimental setup size. As expected, Fig. 3 shows the peak attenuation at target frequency and its odd harmonic frequencies. Fig. 4 shows additional attenuation at double the target frequency, which is caused by the second bypass pipe. The result suggests that the proposed system will work as expected.

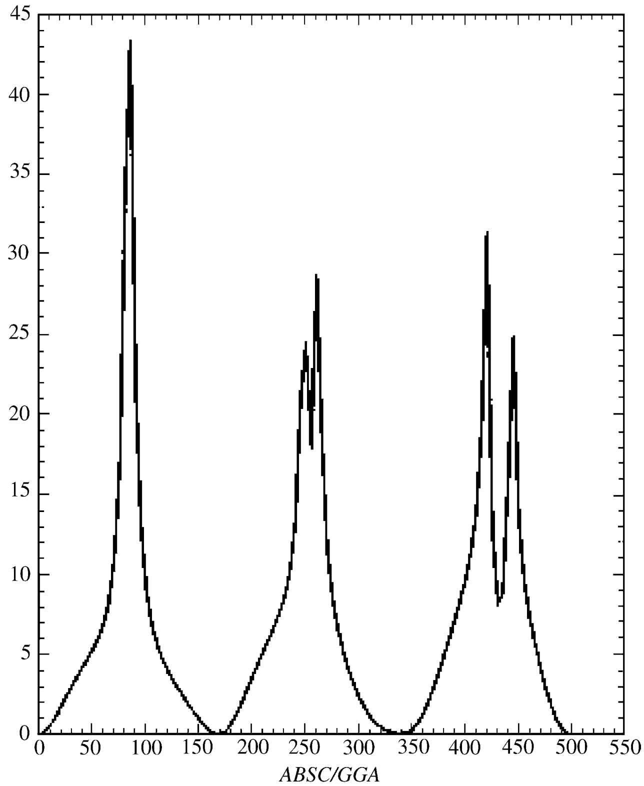

Fig. 3. Simulated transmission loss of a single bypass system.

2.3.3. TL measurement of a bypass pipe

The length-varying mechanism makes it impossible to correctly measure the TL of the proposed system because the length has to be fixed when the noise is measured to get TL. So TLs at various lengths were measured and overlapped to show the changing TL characteristics with length change. One bypass pipe system was used to analyze the TL characteristics. A speaker was attached at one end as a wideband noise source and a two-microphone method was used to measure the TL [15]. The pipe length was changed from the maximum to minimum length and fixed during measurement. The TL measured at various points is shown in Fig. 5. Among the peaks at the left half, the left-most peak is the TL at the maximum length and the right-most peak is the TL at the minimum length. The peak points move to higher frequency with decrease of length as expected. Although Fig. 5 shows frequencies only up to 250 Hz, experiments also have shown peak attenuation at odd harmonics of target frequency. Comparison to Fig. 3 shows the major TL characteristics to be very similar. The peaks at the right side of Fig. 5 were not observed in simulation. These peaks' frequencies are also dependent on length change; however, their magnitudes are always changing and sometimes it does not happen. The frequency of the peak turns out to be the frequency of a standing wave whose wavelength is equal to total bypass pipe length. The result strongly suggests that we can either remove or reduce the major noise component C2 by varying bypass pipe length according to rpm change.

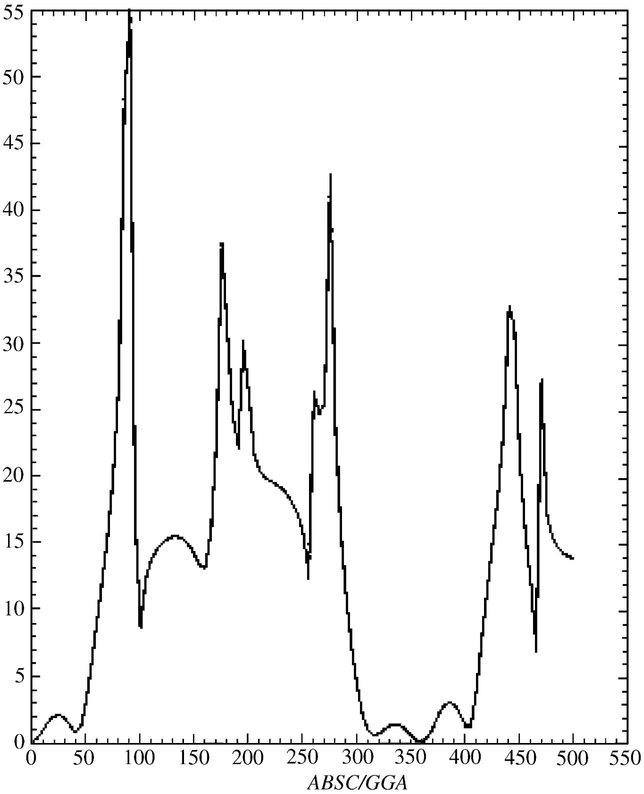

Fig. 4. Simulated transmission loss of a double bypass system.

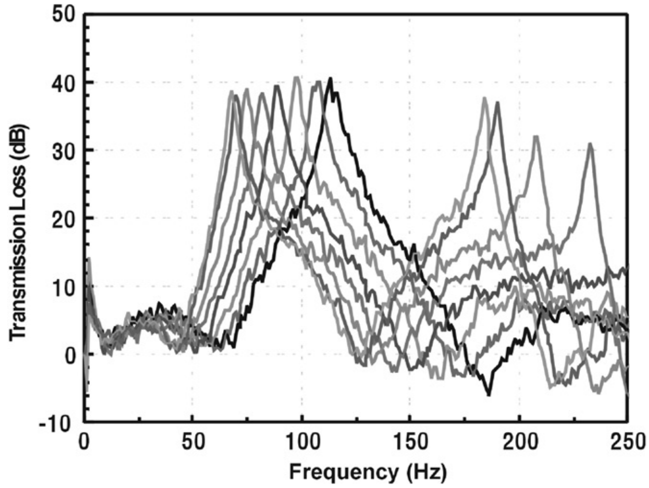

Fig. 5. Transmission loss peak change due to pipe length change.

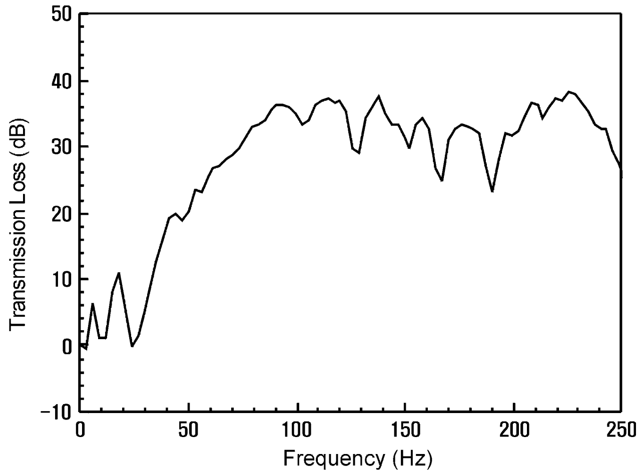

Fig. 6. Transmission loss of the original passive muffler.

Fig. 6 shows the TL of the original passive muffler. Comparison to Fig. 5 shows that the proposed system's TL in the designed rpm range is almost 40 dB and even better than the passive system. Considering that the proposed system has a minimal backpressure, this proves our goal to make a new muffler system, which has the same or better noise reduction characteristics compared to the passive system and very low backpressure.

3. Exhaust noise measurement experiment

The noise attenuation characteristic of the proposed system was measured with engine experiments and compared to the passive muffler's performance. A four-cylinder 2.0 l gasoline engine was used. The original muffler was used as a passive muffler for performance comparison. An aftermarket-tuning muffler with a simple internal structure of strait perforated pipe wrapped with glass wool was used as a simple muffler to remove wideband noise. Bypass pipes were machined from steel pipes and stepping motors and linear motion guides were used for the sliding mechanism. The active control range was designed as 1821–2856 rpm at room temperature. However, the actual range should be adjusted as the exhaust gas temperature changes due to operation time and load. A thermocouple was installed to monitor the temperature change. Two bypass pipes were installed to remove C2 and C4 simultaneously. The experimental setup is shown in Fig. 7. An accelerometer was installed at the engine block. The acceleration signal was low-pass filtered and connected to an $F-V$ converter. The controller calculates control signals using the output of the $F-V$ converter and sends it to motor drivers to change the bypass pipe length.

3.1. Exhaust noise measurement condition

While the active control was engaged, engine rpm was changed from 800 to 4000. A microphone was installed at 1 m from the end of the muffler system to measure the exhaust noise. For the rpm range out of control limit, the bypass pipe was stopped either at the minimum

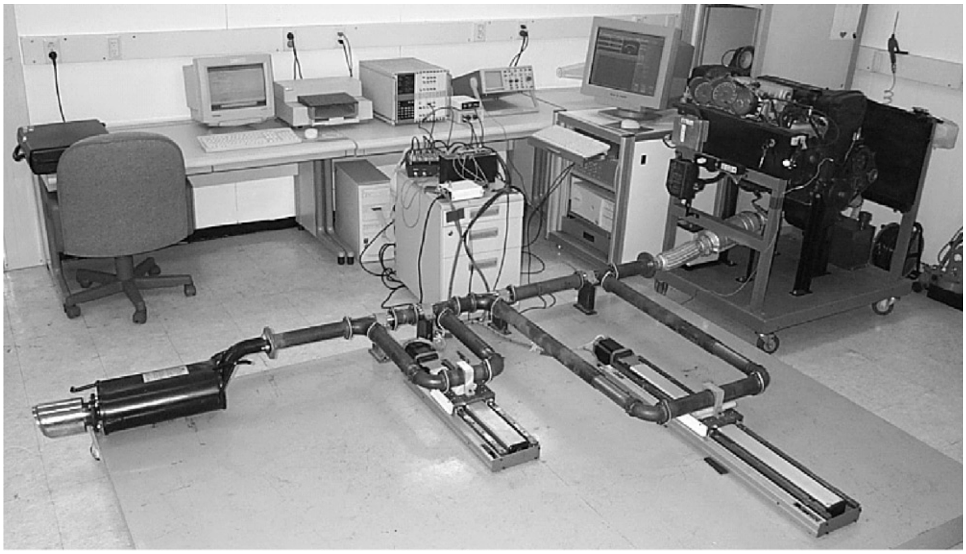

Fig. 7. Exhaust noise measurement experimental setup.

or maximum position. The muffler noise picked up by the microphone was sent to the LMS CADA-X system. The acceleration signal from the accelerometer on engine block was low-pass filtered using 300 Hz cut-off frequency and used as a trigger signal for the CADA-X. The engine exhaust noise with 200 rpm increment was measured and added to waterfall graph for analysis.

In order to get the correct measurement of exhaust noise, only the exhaust noise has to be measured. However, due to limitations of the laboratory, the exhaust noise was measured while the engine sound is present. So the measurement provides only qualitative result for comparison purpose. So the unit for y-axis in Fig. 8 is taken as voltage instead of Pa.

3.2. Performance comparison

Exhaust noise was measured under four different conditions and the results are shown in Fig. 8. In the figure, (a) is the case of measured exhaust noise without muffler, (b) is the exhaust noise with the original passive muffler, (c) is the exhaust noise with one bypass pipe for C2 control and the simple muffler, and (d) is the exhaust noise with two bypass pipes for simultaneous C2 and C4 control and the simple muffler. In (a), it is shown that the original engine noise has a wideband noise mainly between 250 and 450 Hz and it becomes larger when the rpm is over 2000. The most dominant C2 is directly related to the engine rpm and exists between 30 and 120 Hz. C4, which is a second order component, has peaks smaller than C2 and exists between 60 and 240 Hz. (b) shows the passive muffler's performance. The muffler works very well as expected. All noise components are well suppressed over all rpm range. In (c), C2 is well controlled proving that the bypass pipe for C2 control is working very well. There are large peaks in a higher rpm range. However, they are out of the designed control range. In (d), C4 is also very well suppressed. Compared to the performance of the passive muffler in (b), the proposed muffler system shows very similar noise reduction performance. Considering the proposed system has minimal backpressure, it strongly

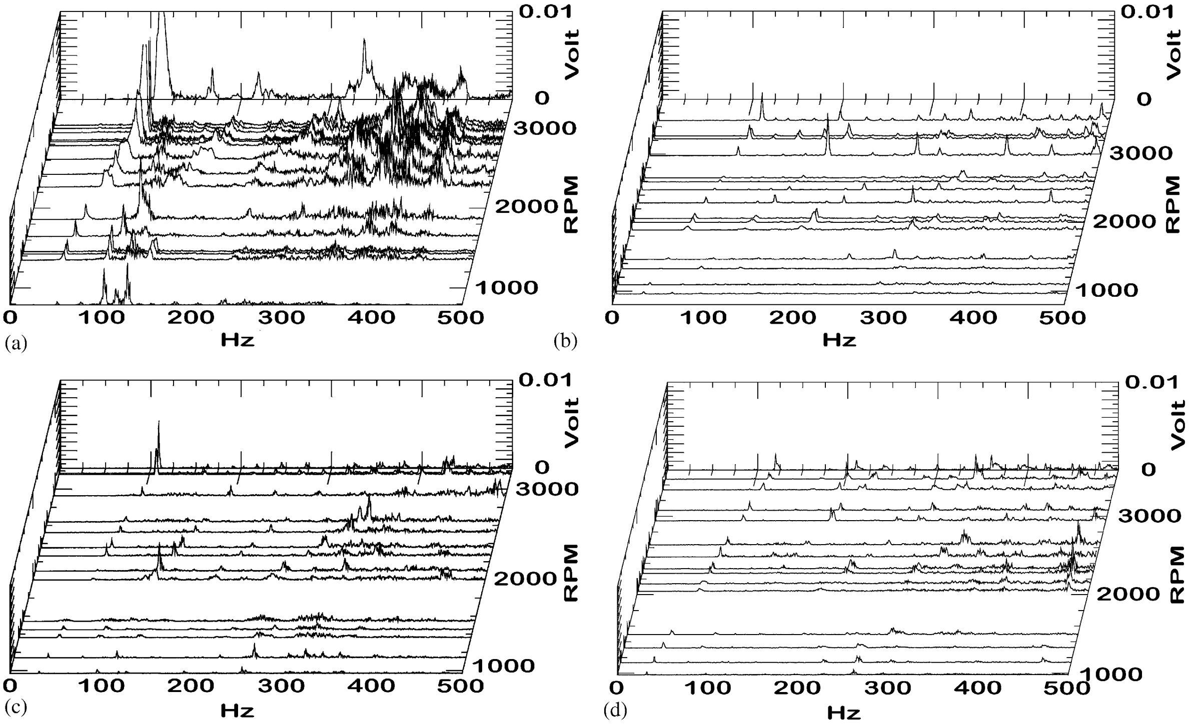

Fig. 8. Measured engine noises with various configurations: (a) without muffler, (b) with original passive muffler, (c) with one bypass pipe and simple muffler and (d) with two bypass pipes and simple muffler.

suggests that this system can achieve higher engine power and increased fuel efficiency without sacrificing noise reduction performance.

4. Torque measurement using engine dynamometer

To prove the proposed system's potential for higher engine power, the torque was measured using an engine dynamometer. The same type of engine as in noise measurement experiment was used and another muffler system was designed and installed.

4.1. Experimental condition

Unlike the noise measurement experiment, the exhaust gas temperature changes a lot in engine dynamometer experiment. The temperature change affects noise propagation speed and changes wavelength. So the controller has to compensate the temperature change. In this experiment, the control range was designed between 1356 and 2924 rpm in room temperature. The experiment was conducted using a power train-testing engine dynamometer with the help of a motor company. The experiment was performed in three steps under maximum torque condition. First, the engine torque curve without muffler was measured. This will give maximum torque of the engine. Second, the passive muffler was installed and torque was measured. This experiment will show the

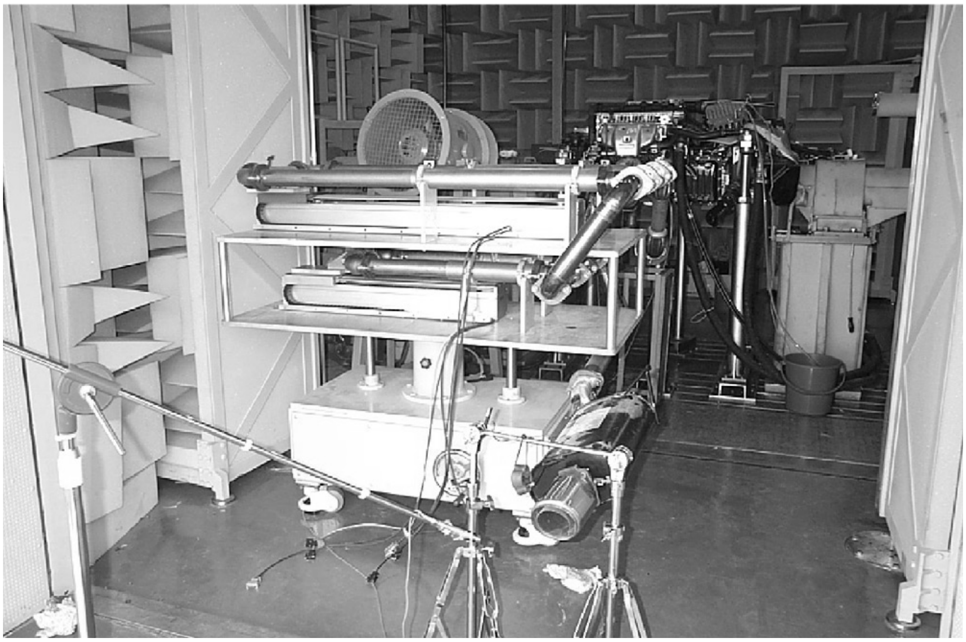

Fig. 9. Torque measurement experimental setup.

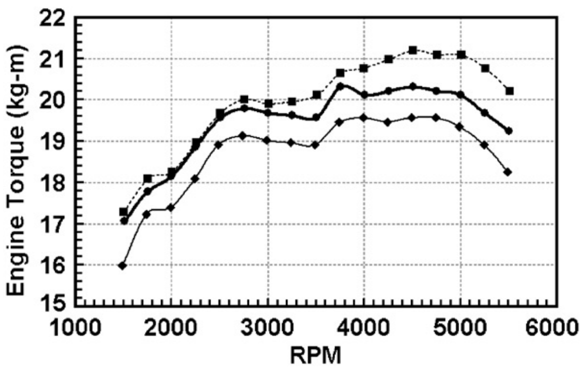

Fig. 10. Engine torque curves.

decreased torque curve due to the muffler's backpressure. Third, the proposed system with one bypass pipe for C2 control and a simple muffler is installed and the torque curve was measured. This will give an increased torque curve due to drastically reduced backpressure.

4.2. Experimental result

The experimental setup for torque measurement is shown in Fig. 9 and the measured three torque curves are shown in Fig. 10. The torque was measured with a 250 rpm increment. In Fig. 10, the top line is a torque curve without muffler, bottom line is a torque curve with passive muffler, and middle line is a torque curve with the proposed system. Between 2000 and 2500 rpm, the proposed system recovers more than 90% of torque lost by the passive muffler and the torque is well recovered even outside the control range. This result proves the proposed system's big potential for increased power due to its minimum backpressure structure. However, the result in Fig. 10 should be used very cautiously and only qualitatively. It is because the proposed system makes a big change to boundary conditions so that the engine power increase should be measured after careful provisions for many constraints including catalytic converter, optimum design of simple muffler, engine ECU tuning, etc.

5. Conclusions

A new concept of active muffler system, which can dramatically reduce the conventional passive muffler's backpressure and overcome problems of other active muffler methods, is proposed and its superior performance for noise reduction and engine power boost is qualitatively confirmed by engine experiment. This method has a U-shaped length-variable bypass pipe attached to an exhaust pipe and uses destructive interference occurring when two divided exhaust noises are made to meet. To overcome the large bypass pipe length required in a low rpm range, an improved design is proposed in which a passive muffler is also installed and used in the low rpm range. The bypass pipe system is only used in the high rpm range thereby reducing the total system size. The length-variable bypass pipe's good transmission loss characteristic is confirmed by experiment and the proposed system of two bypass pipes and a simple muffler have shown a noise reduction performance comparable to the original passive muffler. The proposed system's minimum backpressure design provides a great potential for boosting engine power and improving fuel efficiency. The engine dynamometer test has proved that the proposed system recovers almost all torque lost by the passive muffler. The result is not final in any sense and has to be used only qualitatively because many factors have to be considered for the exact evaluation. Although a high price actuator was used in the experiment, the system can be made relatively cheap once it is designed for real production because of simple structure and low calculation requirement for controller. The expected improvement of fuel efficiency was not confirmed in this research due to limitation on experiment.

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