An integrated passive and active control system for reducing haul truck noise

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ABSTRACT

This paper is concerned with the analysis and design of an integrated passive and active control system for reducing the haul truck noise. By using a passive enclosure with two open ends and fiberglass inner surface, a three-dimensional noise radiation problem of the engine noise is shaped to a simple problem of noise radiation from the open ends above 200 Hz. Below 200Hz where the transmission loss of the lightweight enclosure is low, a spatial global active noise control (ANC) system is constructed inside the enclosure. To improve the performance of the system, the open ends are divided into four equal diameter size ducts each followed by a louver. A single channel ANC system is set up in each duct to eliminate the noise in the frequency range from 200 Hz to 400 Hz. Experimental results demonstrate that more than 20 dB insertion loss is achieved with such a passive enclosure at frequency band above 400 Hz, and 8 dB noise reduction with global ANC systems at frequency band below 200 Hz, and 4 dB noise attenuation with single channel ANC systems between 200 Hz and 400 Hz. The results this investigation show the feasibility of hybrid noise control technique for reducing the haul truck noise.

Keywords: Insulation, Passive, Active I-INCE Classification of Subjects Number(s): 38.2

1. INTRODUCTION

Diesel engine sets are widely used as main power supplying equipment in industrial plants and facilities in commercial/residential buildings. Engine noise has been studied since the early stages of engine development. Many researchers have devoted to study and identify the characteristics of the radiated noise from diesel engine, generator, radiator fan, and engine exhaust. Various methods have been proposed to reduce engine noise [1-4].

The most conventional methods for reducing diesel engine sets noise are to use enclosures and mufflers. In order to achieve higher noise reduction level in low frequency, heavy enclosures and mufflers, and absorbing materials are required. An important requirement in the development of engine noise reduction system is the weight reduction, for the reduction of the fuel consumption. Although heavy materials have better noise reduction performance, they do not satisfy the light-weight requirement [5].

In order to attenuate the low frequency engine noise and satisfy the compact size and light-weight requirements, active noise control (ANC) method was applied. ANC uses the superposition principle of sound wave to reduce noise radiation or reduce sound pressure in a region [6-10]. In this paper the feasibility and performance of an integrated passive and active control system for engine noise are examined experimentally. One of the advantages in using the passive and active control method is the significant weight saving in passive control treatment.

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2. EXPERIMENTAL SETUP AND MEASUREMENT SYSTEM

Figure 1 shows the schematic diagram of the integrated passive and active control system, and Figure 2 is a photo of the system. For the passive control system, the designed enclosure consists of 2.5 mm steel panel, which is used to transform the three dimensional noise source to two surface sources at the open ends above 200 Hz. The inner surface of the enclosure is covered by 5 cm thick fiberglass layers which have a sound absorption coefficient larger than 0.7 above 600 Hz. For the active control systems, one set of global control sources are fixed inside the enclosure and the control sources at two open ends are mounted to each side of the ducts. The spatial ANC system focuses on the low frequency noise below 200 Hz radiating from the engine. As a result, the three dimensional residual noise is restricted to the two simple duct ends for frequency above 200 Hz. Eight single channel ANC systems at the two ends are set up to eliminate the noise from 200 Hz to 400 Hz. Moreover, the louver at the ends attenuates higher frequency noise above 400 Hz.

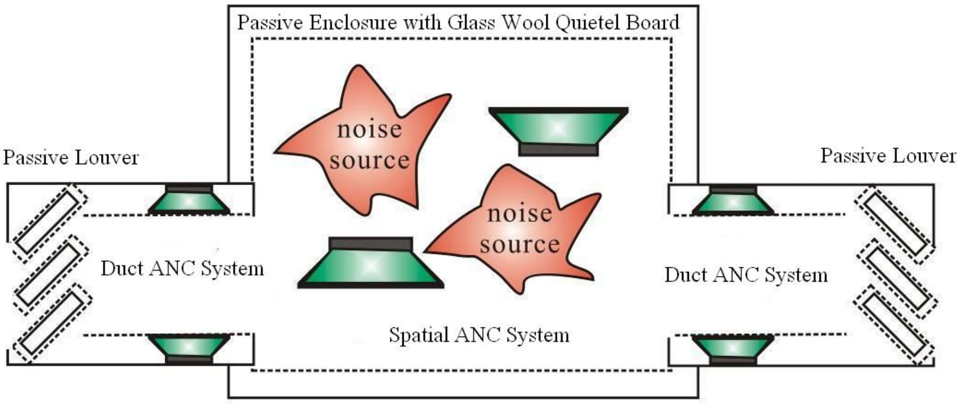

Figure 1 – Schematic diagram of the integrated passive and active enclosure.

Figure 2 – A photo of the integrated passive and active control system.

To evaluate the performance, the averaged sound pressure level $L_{\mathrm{p}}(f)$ is calculated by [12]

$$L_{\mathrm{p}}(f) = 10\log_{10}[\frac{1}{N}\sum_{i=1}^{N}10^{0.1L_{\mathrm{p}i}(f)}] \tag{1}$$

where $L_{\mathrm{p}i}(f)$ is the sound pressure level at frequency f of point i and N =11 is the number of averaged points. The measurement locations are 0.7 m away from the enclosure, 8 locations are around the enclosure with equidistance and at the same level (0.7 m in height), and the other 3 locations are equidistantly on the top central axis of the encolsure [12].

3. SYSTEM PERFORMANCE

3.1 Passive Enclosure

The insertion loss of the passive enclosure is calculated according to a classical method [12]. As shown in Figure 3, the experiment result agrees with the calculated result. The slight difference at high frequency might come from the acoustic leakage. It is clear that the passive control gives rise to good attenuation above 200 Hz but little in below. As demonstrated in Table 1, the insertion loss at 200 Hz is about 8 dB while more than 15 dB noise reduction is achieved at frequency above 400 Hz.

Table 1 - Total sound pressure level of various control modules

Frequency (Hz)	100	200	300	400	1000	2000	4000	10000
Theoretical (dB)	3.5	7.9	13.5	18.4	36.6	41.1	40.4	30.4
Experimental (dB)	7.2	9.0	15.4	18.4	27.9	33.5	35.0	29.0

In order to increase the noise reduction level, the thickness of the enclosure muffler (steel) and the absorbing material should be increased manyfold due to their negligible sound absorption (or insulation) below 200 Hz. The measured sound absorption coefficient of the fiberglass layers used in the enclosure is shown in Figure 4. The comparisons of the sound absorption coefficient between 25 mm with 50 mm thick fiberglass layers are also indicated in Figure 4. It is clear that increasing material thickness cannot improve its absorption coefficients at low frequency significantly. On the other hand, increasing the thickness of steel panel will significantly increase the weight of the passive enclosure. Therefore, this passive enclosure is not suitable for attenuating the low frequency noise of the truck.

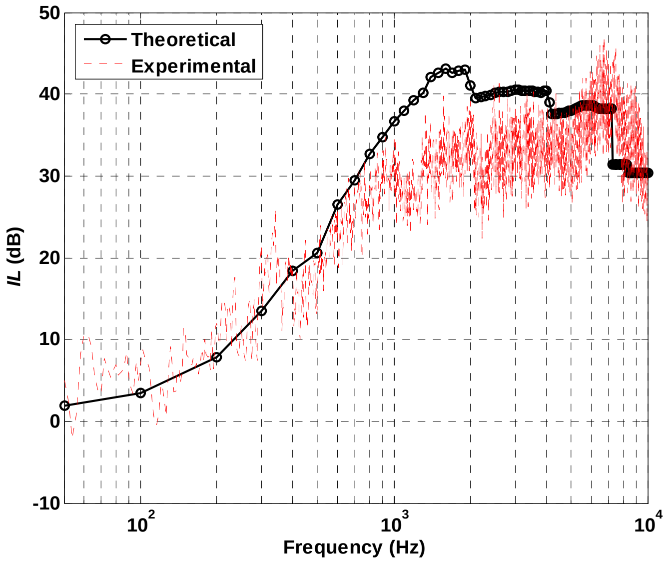

Figure 3 - Measured and predicted insertion losses of the passive enclosure.

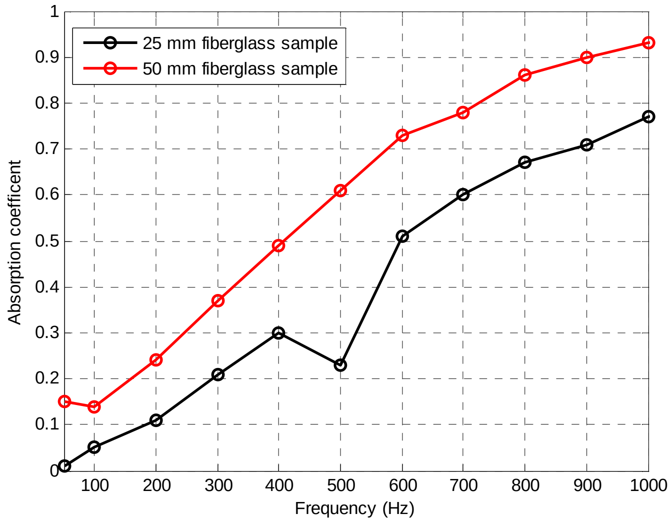

Figure 4 – Sound absorption coefficients of two fiberglass samples.

3.2 Global Active Noise Control in an Enclosure

In spatial ANC system shown in Figure 1, two primary loudspeakers are located inside the enclosure to simulate the engine noise source. A 4 - channel feed forward ANC system, which use the same signal to the primary loudspeakers as its reference signal, was implemented to model the spatial ANC system. The secondary control sources are mounted on the internal surface of the enclosure in the height 0.2 m and 0.4 m and placed around the primary sources to control the noise below 200 Hz, where the distance between the secondary sources and the center of the primary sources is 0.2 m. Four error sensors are fixed on the top of the secondary sources, whose distances are 0.7 m.

The noise spectrum of one of the error microphones of global active system is presented in Figure 5, which shows that about 20 dB noise reduction can be achieved at the error points of the ANC system. The noise reduction below 100 Hz is undesirable due to the poor frequency response of the primary loudspeakers at low frequency band (such as below 100 Hz). The performance at frequency band below 100 Hz could be improved if the signal gain measured by the error microphone is sufficient high. The spatially averaged noise spectra with and without spatial ANC system are shown in Figure 6, where the sound power reduction is about 8 dB at the band below 200 Hz white noise radiating by the primary loudspeakers, which is the key point to reduce the weight of the passive enclosure.

In general, the field incidence transmission loss of the infinite passive panel in the mass-law frequency range can be evaluated by [12]

$$TL_1 = 20\log_{10}(\pi fm / \rho c) - 5 \tag{2}$$

where c is the speed of sound in air (340 m/s), m is the surface density (kgm^{-2}), ρ is the density of the medium. At the frequency below 200 Hz, the m can benefit for 8 dB the sound power reduction from the spatial ANC system by

$$20\log_{10}(\pi fm_b / \rho c) - 20\log_{10}(\pi fm_a / \rho c) = 8 \tag{3}$$

$$\delta = m_b / m_a = 10^{0.4} \approx 2.5 \tag{4}$$

where m_b and m_a is the surface density (equal to the thickness of the same type of passive panel) while using the spatial ANC system or not. It is obvious that, in order to obtain the same transmission loss, the thickness of the steel enclosure without spatial ANC system should be about 2.5 times as that with the spatial ANC system, whose sound power reduction is about 8 dB at the band below 200 Hz. This means an increase of weight of about 2.5 times.

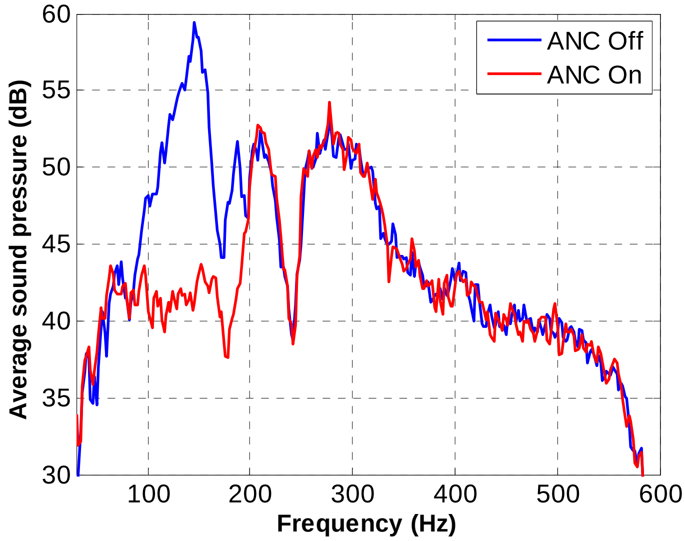

Figure 5 - Noise spectra from one of the error microphones of global active control system

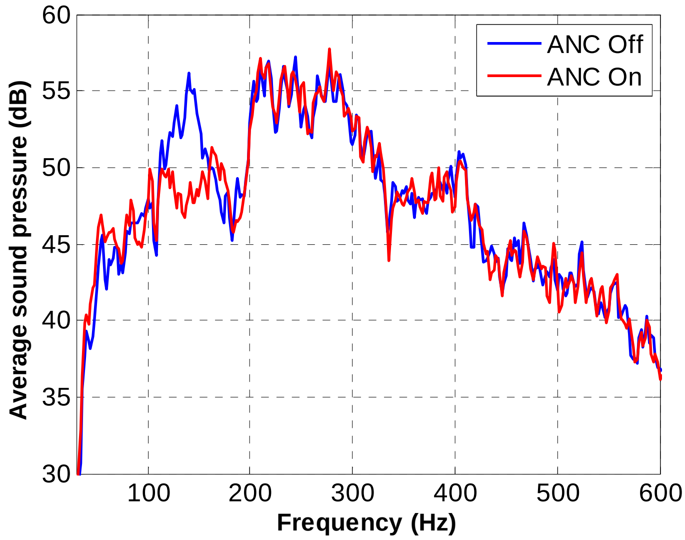

Figure 6 - Spatially averaged engine noise spectra with and without the global active noise control.

3.3 Integration of Global Active Control with Active Control at the Open Ends

To examine the integration of the two sets of active noise control systems and the performance of the active noise control at the open ends of the enclosure, global active control system and the active control systems at the two open ends were turned on separately and then simultaniously. Each open end of the enclosure was divided into four equal area size ducts each followed by a louver. A single channel feed forward active noise control system was constructed for each channel and the primary noise source (loudspeaker) is placed in the center of the enclosure driven by a band limited white noise (200 - 600 Hz) as shown in Figure 1.

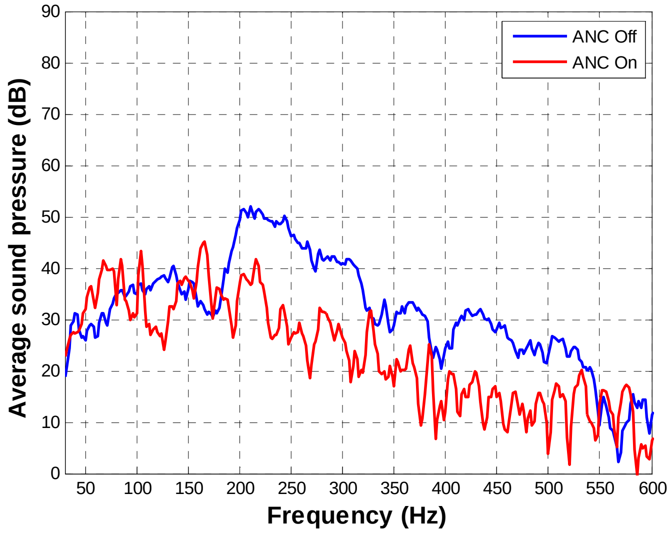

Figure 7 - Noise spectra from one of the error microphones of the duct active control system

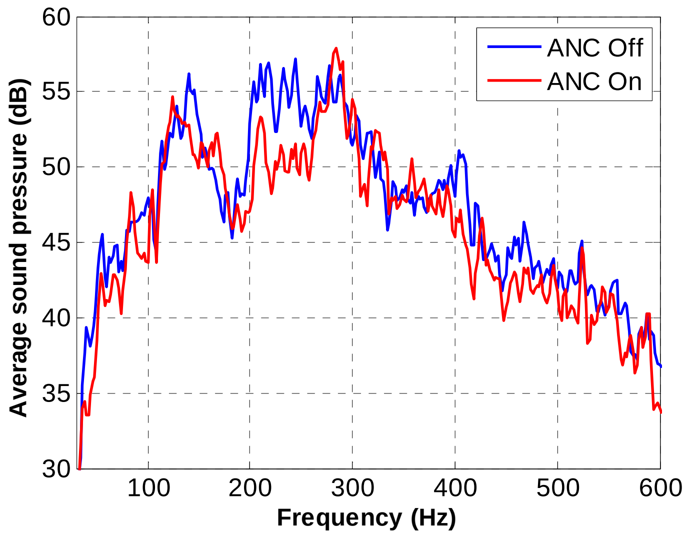

Figure 8 - Spatially averaged engine noise spectra with and without active control at the open ends.

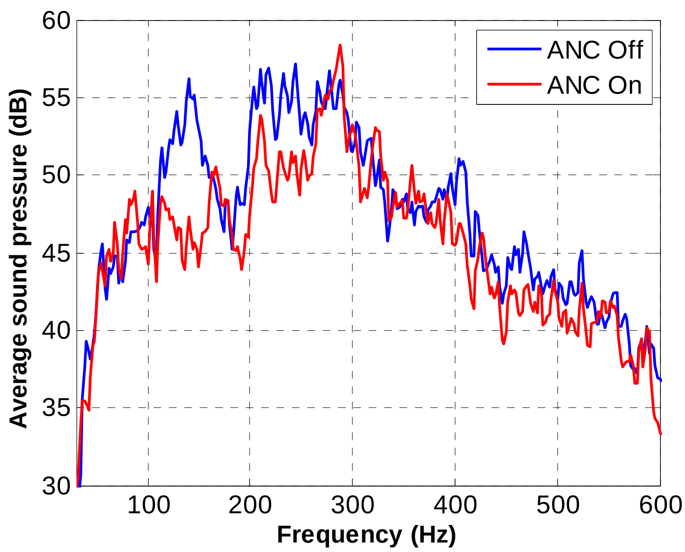

Figure 9 - Spatially averaged engine noise spectra with and without the integrated active noise control

The noise spectrum from one of the error microphones of duct active system is presented in Figure 7. The illustrational results show that more than 15 dB noise reduction has been achieved at the error points of the duct ANC system at frequency above 200 Hz, where the width of the duct is 45 cm and the length is 55 cm.

Figure 8 show the noise reduction and spatially averaged noise spectra of various control modules for the limited frequency band noise (200 – 600 Hz). As indicated by figures, power attenuation is achieved at the energy dominant low frequency noise band. The introduction of duct ANC systems acquires averaged 4 dB noise attenuation in the frequency band from 200 Hz to 600 Hz, and more than 6 dB in the band 200 – 250 Hz.

Figure 9 show the noise reduction and spatially averaged noise spectra of various control of the two sets of integrated active noise control systems. It can be seen in Figure 5, Figure 8 and Figure 9 that each active noise control system works perfectly. As illustrated by Figure 9, the noise at low frequencies (below 200 Hz) can be effectively controlled by the global active noise control system. Above the low frequency region, a generally frequency range (200 – 600 Hz) is encountered and controlled by duct ANC system as shown in Figure 8.

This experiment confirmed that the two sets of independent active noise control systems can work normally when both of them are working at the same time. The spatially averaged noise reduction achieved by the active noise control at the two open ends is about 4 dB at 200 – 600 Hz frequency band. The combination of active control at the open ends and the global active control in enclosure increases the performance of the low-frequency noise reduction in the band of 30 – 200 Hz, as shown in Figure 8. However, the noise reduction in the band from 250 – 600 Hz is undesirable, which mainly comes from the difficulty to obtain high coherent reference signal and the insufficient high signal gain in the frequency band from 250 - 600 Hz, and leads to the insufficient sound power attenuation. In additional, the noise reduction below 100 Hz is undesirable due to the poor frequency response of the primary sources at low frequency band (such as below 100 Hz).

3.4 Integration of Passive and Active Control System

As indicated above, there will be about 8 dB additional noise reduction below 200 Hz and 4 dB in the band of 200 – 600 Hz with the active noise control system, together with the passive noise reduction, the noise reduction performance of the spatially averaged noise spectra of the whole integrated system can be drawn in Figure 10 (dash blue lines). As shown in Figure 10, the engine noise recorded from a six cylinder truck engine is set as the calculated primary engine noise. More than 20 dB noise reduction can be obtained at high frequency above 400 Hz.

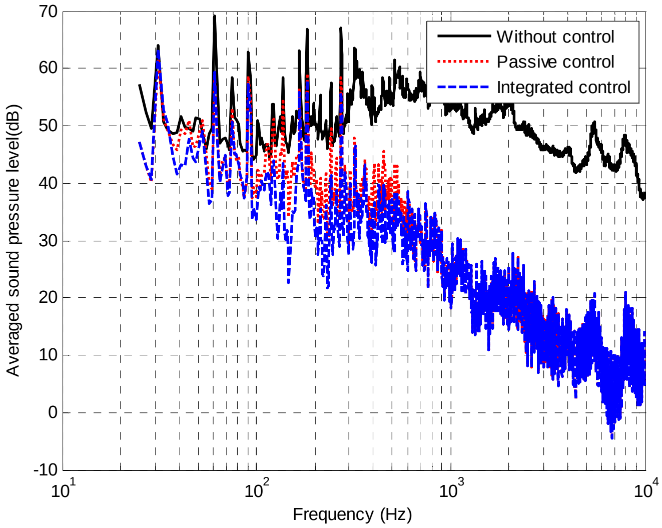

Figure 10 - Spatially averaged noise spectra of various control modules for engine noise without weighted

Total sound pressure level of various control methods is summarized in Table 2, where the passive control achieves 25.1 dBA in the whole frequency band and the active control provides 2.2 dBA in whole frequency band.

Table 2 - Total sound pressure level of various control modules

Control status	Total sound pressure level (dB) (25-10000Hz)	A-weighted total sound pressure level (dBA) (25-10000 Hz)
Without control	86.4	85.8
Passive control	69.3	60.7
Integrated control	67.2	58.5

4. CONCLUSION

An integrated passive and active passive noise control system for reducing engine noise was designed and its performance was evaluated experimentally. For the passive part, a light-weight enclosure was built with 2.5 mm thick steel panels, and 50 mm thick fiberglass layers are used to as the sound absorption mechanism inside the enclosure. Each end of the opening of the enclosure is divided into 4 equal sized ducts followed by a separate louver. This passive enclosure provided more than 25.1 dBA reduction in sound power in the entire frequency range. For the active part, 2 sets of active control systems were installed to investigate the possibility of using active noise control technique to reduce noise without increasing the weight of the enclosure. About 4 dB extra spatially averaged sound power attenuation is achieved below 600 Hz. Experimental results show the feasibility of using integrated noise control technique for reducing the engine noise. The active control provides 2.2 dBA in whole frequency band and the use of spatial ANC system will reduce the weight of the enclosure by nearly 60 percent.

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