ACOUSTICS AND NOISE CONTROL

Mr. Balraj Singh[1], Dr. G.R. Selokar[2], Mrs. Priyanka Jhavar[3], Prof. Sanjay Kalariya[4]

[1]PG Scholar, Department of Mechanical Engineering, SSSIST Sehore, M.P., India
[2] Professor and Principal, Department of Mechanical Engineering, SSSIST Sehore, M.P., India
[3] Associate Prof., Department of Mechanical Engineering, SSSIST Sehore, M.P., India
[4] Professor and HOD., Department of Mechanical Engineering, SSSIST Sehore, M.P., India

ABSTRACT

Noise is one of the most common hazards we face today. It is all around us: in our homes, in our workplaces, in our cars. All too often, its negative effects are ignored. Overexposure to noise can cause immediate symptoms like irritability, stress, and inefficiency in the workplace. The long term effects of noise exposure however, are more daunting. Permanent hearing loss can occur as a result of continued exposure. Once the damage occurs, it is irreversible, but hearing loss is preventable (NIOSH 2005).

In this work we focus the boundary element method to analyze eight factors that affect enclosure design. The objective will be to understand how each factor affects the performance of partial enclosures like opening size, opening location, absorption coverage, absorption location etc. in this dissertation work we concentrate on the behavior of free-standing, partial enclosures at low frequencies (0 to 1000Hz). Structure-borne noises will not be considered. Although structure-borne noise is important, it is difficult to model and is usually case-dependent. Because of this assumption, the cases studied will not accurately reflect real conditions, but they will better demonstrate the effect of the eight factors under consideration.

Keyword: - *Acoustics and Noise Control, Mechanical work, Engine, Acoustic Waves, Acoustic Impedance etc.*

1. INTRODUCTION

The research activities for acoustic and noise control at the Herrick Laboratories include the study of the wave propagation, voice production, vibration, human perception of sounds, control and abatement of environmental noise, and fluid dynamics as well as signal processing, numerical techniques, measurements, controls, and design theory applied to making our acoustical environment safer and more pleasant. Research applications include automobiles, water craft, aircraft, domestic products and appliances, heavy equipment, design of advanced acoustical materials for noise control, health care industries, trucks, compressors, heating/ventilating equipment, computers, and fans.

The Herrick Laboratories, a graduate research facility in the School of Mechanical Engineering at Purdue University, consist of approximately 17,600 square feet (1,635 square meters) of engineering laboratory space; 10,500 square feet (975 square meters) of office space for faculty, students, and clerical staff; and 3,470 square feet (322 square meters) of machine shop and equipment space. Several facilities within the Mechanical Systems Laboratory are dedicated to research in the fields of engineering acoustics and noise control.

The main acoustics laboratory includes a reverberation room, an anechoic room and an audiometric room. The reverberation room measures 25 x 20 x 18 feet (7.6 x 6.1 x 5.5 meters) and has a practical lower limiting frequency of approximately 100 Hz. The anechoic room has a useful working volume of 12 x 12 x 12 feet (3.7 x 3.7 x 3.7 meters) and a lower limiting frequency of 100 Hz. Above this frequency, the background level is 20 dB or less. Both rooms have microphone boom systems, ventilation systems, and access doors and windows. In addition, the reverberation room has an adjacent, smaller reverberation chamber attached for transmission loss measurements of small panels. The audiometric room has interior dimensions of 8 x 8 feet (2.5 x 7.5m). It is used for jury testing and sound quality research.

There is also a large semi-anechoic room with a useful working volume of 41 x 27 x 18 feet (12.5 x 8.2 x 5.5 meters) is contained within a building addition of some 2,500 square feet (232 square meters). The building contains a dynamometer room, an instrumentation room, and mechanical equipment for ventilation and climate control of the semi-anechoic room. The room is designed for measuring the noise of large stationary and mobile machinery and vehicles under simulated operating conditions; it has an exhaust ventilation capacity of 15,000 cfm (7.1 cubic meters/second).

Two other specialized test facilities are available. A 67-inch, two-wheel chassis dynamometer for automotive noise and vibration studies allows the simulation of road speeds up to 70 mph. A quiet wind tunnel with a test section of 18 x 24 inches (46 x 61 cm) allows aeroacoustic measurements for flow velocities up to 120 mph (60 m/sec).

A very large complement of acoustics and vibration analysis equipment is available, including microphones, omni-directional sound sources, accelerometers, a scanning laser vibrometer, analog and digital filters, real time one-third octave band analyzers, Fast Fourier Transform instruments, noise monitoring stations, Binaural Heads, maximum length sequence system analyzers, state-of-the-art sound level meters, and other PC-based data acquisition and control systems with high-speed digital signal processing boards.

Acoustics is the science of sound, that is, wave motion in gases, liquids and solids, and the effects of such wave motion. Thus the scope of acoustics ranges from fundamental physical acoustics to, say, bioacoustics, psychoacoustics and music, and includes technical fields such as transducer technology, sound recording and reproduction, design of theatres and concert halls, and noise control.

1.1 Fundamentals of Acoustics

This discussion presents the basic quantities used to describe acoustical properties. For the purposes of the material contained in this document perceptible acoustical sensations can be generally classified into two broad categories, these are:

- **Sound.** A disturbance in an elastic medium resulting in an audible sensation. Noise is by definition “unwanted sound”.
- **Vibration.** A disturbance in a solid elastic medium which may produce a detectable motion.

Although this differentiation is useful in presenting acoustical concepts, in reality sound and vibration are often interrelated. That is, sound is often the result of acoustical energy radiation from vibrating structures and, sound can force structures to vibrate. Acoustical energy can be completely characterized by the simultaneous determination of three qualities. These are:

- **Level or Magnitude**. This is a measure of the intensity of the acoustical energy.
- **Frequency or Spectral Content.** This is a description of an acoustical energy with respect to frequency composition.
- **Time or Temporal Variations**. This is a description of how the acoustical energy varies with respect to time.

2. OBJECTIVE

This thesis presents design and control of active mufflers used in engine exhaust systems. Boundary element method is employed to investigate noise attenuation performances of various active muffler structures. Based on these performance evaluations, an active muffler having only one loudspeaker inclined at 45 degree with the muffler centerline is constructed and tested for active noise control (ANC). The feedback ANC with two controller tuning algorithms, the filtered-x least mean square (FXLMS) algorithm and Godart modified algorithm is applied independently for engine exhaust noise attenuation. Experimental investigation reveals that feedback ANC with FXLMS algorithm can attain an averaged 23.6 dB noise attenuation for pure tone excitation and over 6 dB noise reduction for motorcycle and automobile exhaust noises. It is also shown that feedback ANC with Godart modified algorithm has faster transient response compared to FXLMS algorithm for a pure tone noise, but has inferior noise attenuation capability than the FXLMS algorithm for the engine exhaust noise at fixed speeds (about 5 dB noise reductions) as well as during run-up tests.

3. METHODOLOGY

The primary noise of a pure tone is generated at the inlet (right end) connecting to engine exhaust tailpipe, while the control sound for anti-noise is optimized for obtaining the minimum residue sound level at the outlet (left end). Figure 2 shows amount of the maximum noise attenuation at different frequency for each active muffler. Table 1 lists the averaged noise reductions in the tested frequency range. From Fig. 2 and Table 1, it is clearly shown that it differs not much which active muffler structure has the best performance. However, the active muffler structure of Fig. 1 with only one control loudspeaker inclined at 45 degree does outperform other structures slightly, and hence is selected as the active muffler used in the following experiments.

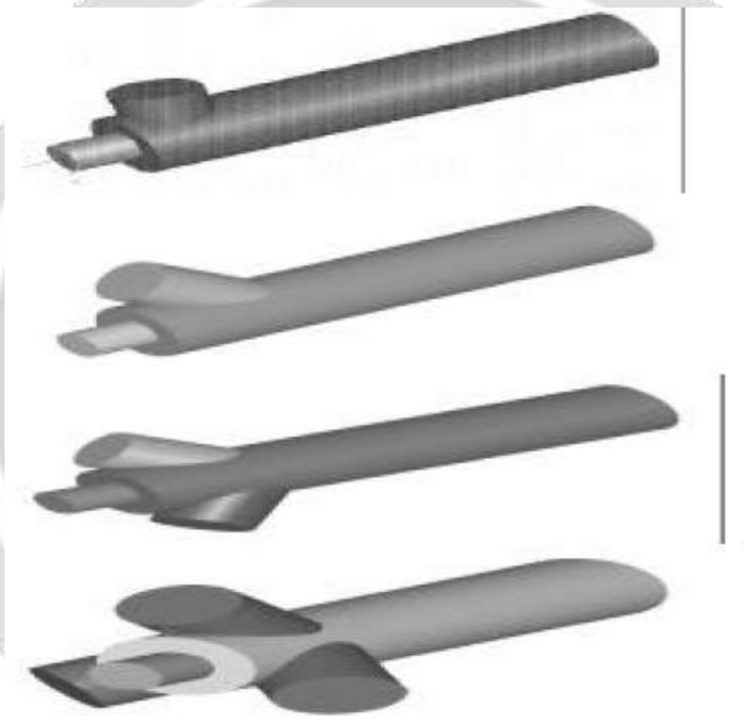

Fig-1: Active Muffler Structures

Table -1: Averaged Sound Attenuations

Structures	Noise Reduction
Figure 1(Top)	84.38 dB
Figure 2(Second Top)	86.65 dB
Figure 3(Third Top)	86.64 dB
Figure 1(Bottom)	86.45 dB

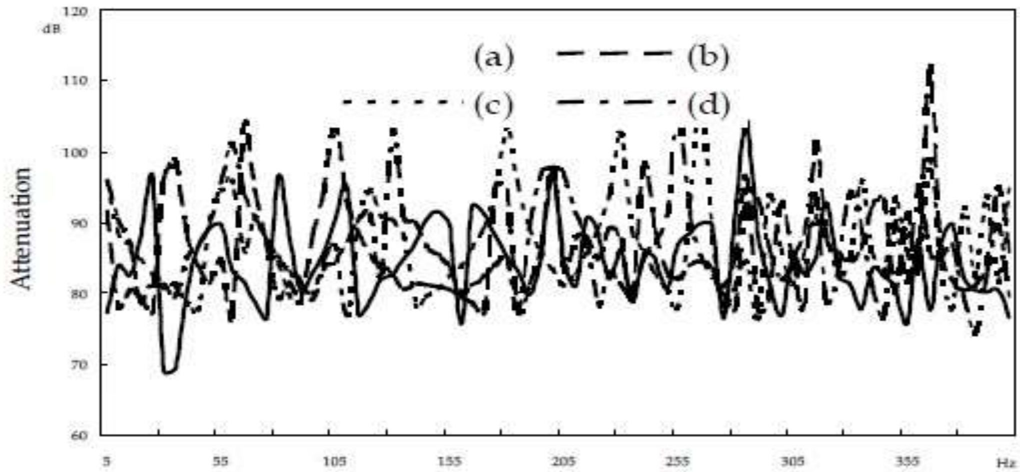

Fig- 2: Noise attenuations: (a)-(d) are for mufflers in Fig. 1from top to bottom, respectively

3.1 Performance Measures

The experimental set-up of Fig. 3 for evaluating performance of the active muffler will be run in this section. First of all, pure tone excitation is applied at the primary loudspeaker from 20 Hz to 400 Hz at every 5 Hz for testing the validity of the constructed active muffler and the FXLMS algorithm. Fig. 4 shows the resulting noise attenuations in dB at the outlet of the active muffler, thus clearly verifying the noise cancellation capability of the active muffler and the control algorithm. An averaged attenuation of 23.6 dB is obtained. Next, engine exhaust sounds from motorcycle and automobile will be applied for evaluating the performance of the active muffler.

Note that the engine exhaust sounds of motorcycle and automobile are first recorded by a recorder placed 1.5 meter behind the exhaust tailpipe with 45 inclined angles and are then replayed by the primary loudspeaker. Fig. 5 and Fig. 6 illustrate performances of the active muffler for attenuating automobile exhaust noise at idle speed of 800 rpm and a fixed speed of 1500 rpm, respectively, in both time and frequency domains. Corresponding response of the active muffler for motorcycle exhaust noise is shown in Fig. 7. Note that solid line and dashed line in the figures correspond to with and without ANC, respectively. For all these engine exhaust noise tested, over 6 dB noise attenuations are obtained.

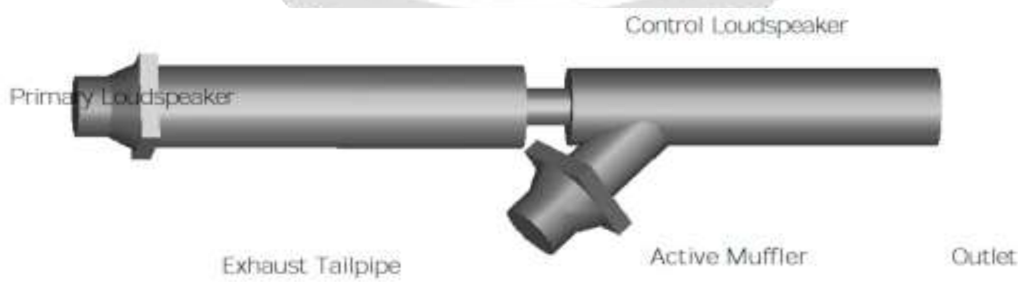

Fig-3: Experimental set-up for ANC

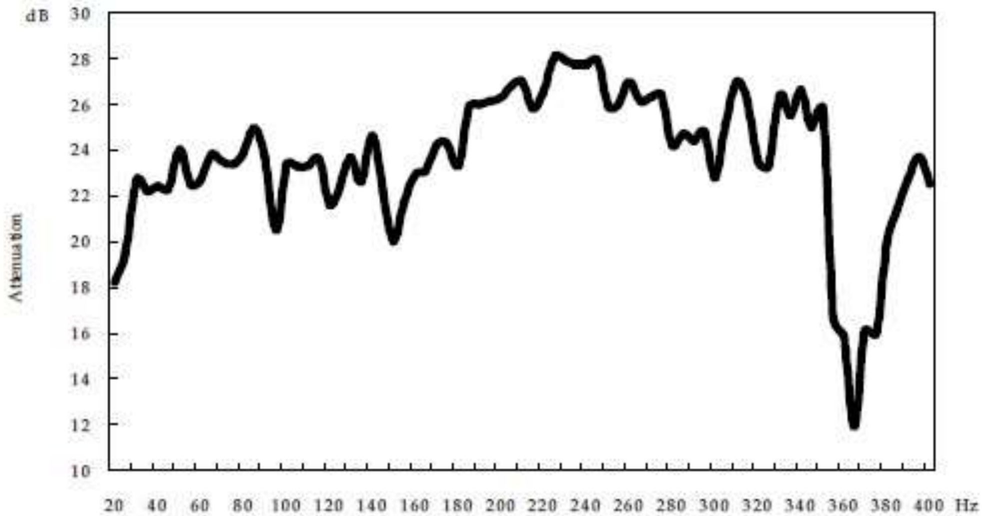

Fig-4: Noise reductions for pure tone excitation

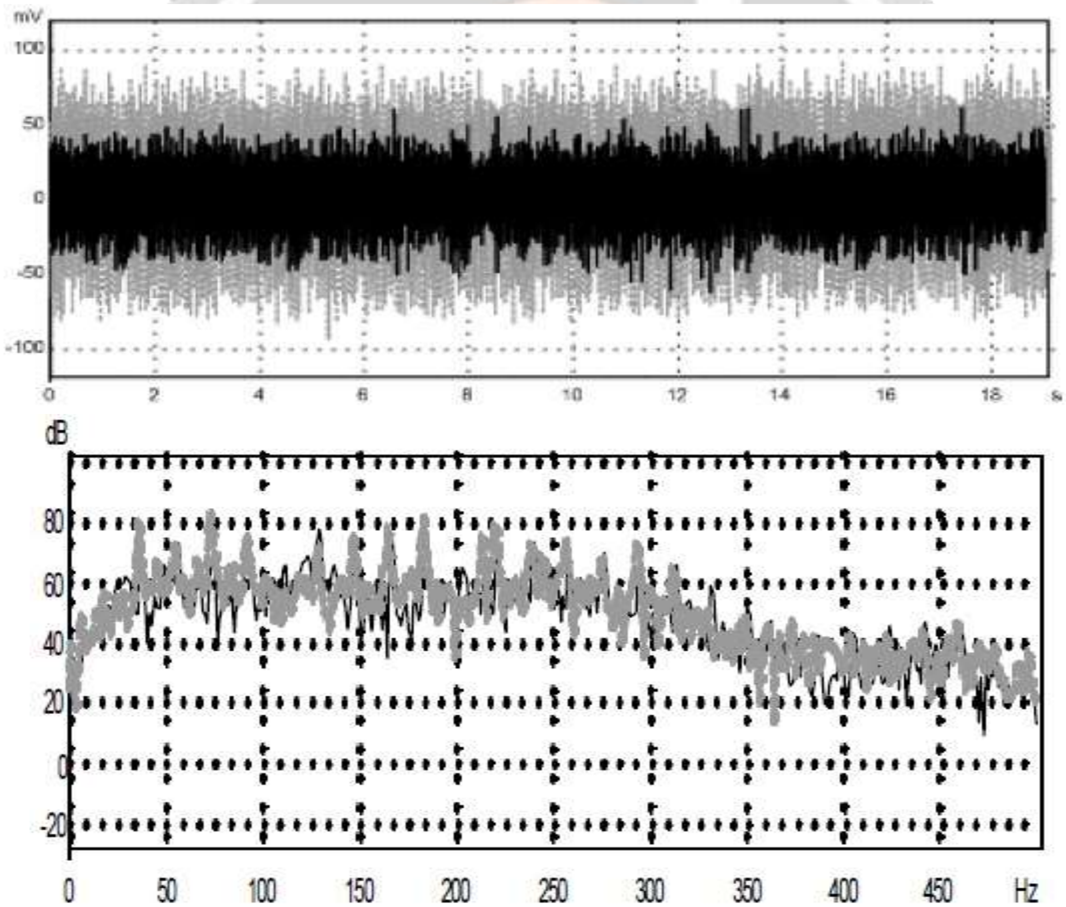

Fig-5: Time (top) and frequency (bottom) response of ANC of an automobile at 800 rpm.

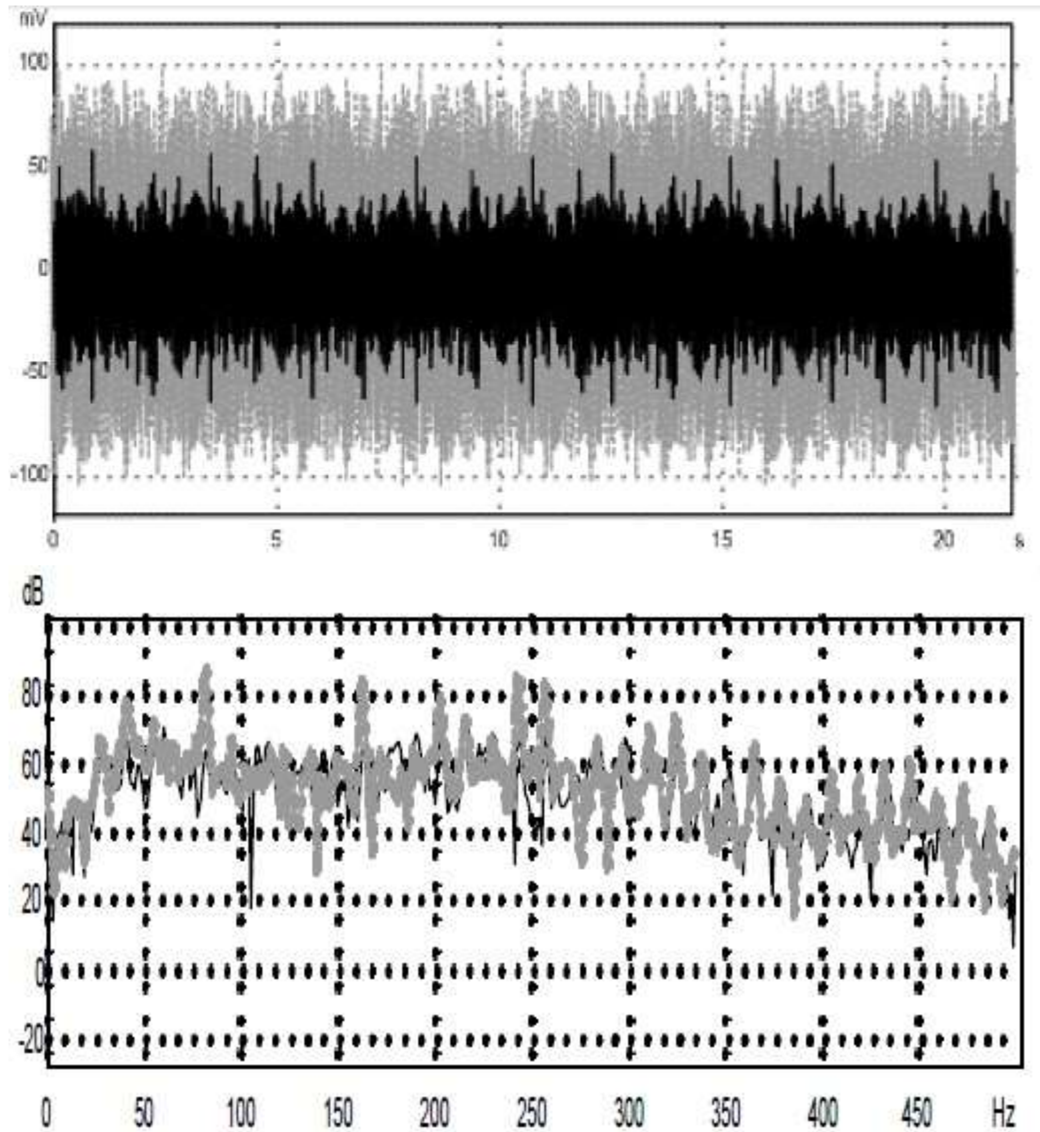

Fig-6: Time (top) and frequency (bottom) response of ANC of an automobile at 1500 rpm.

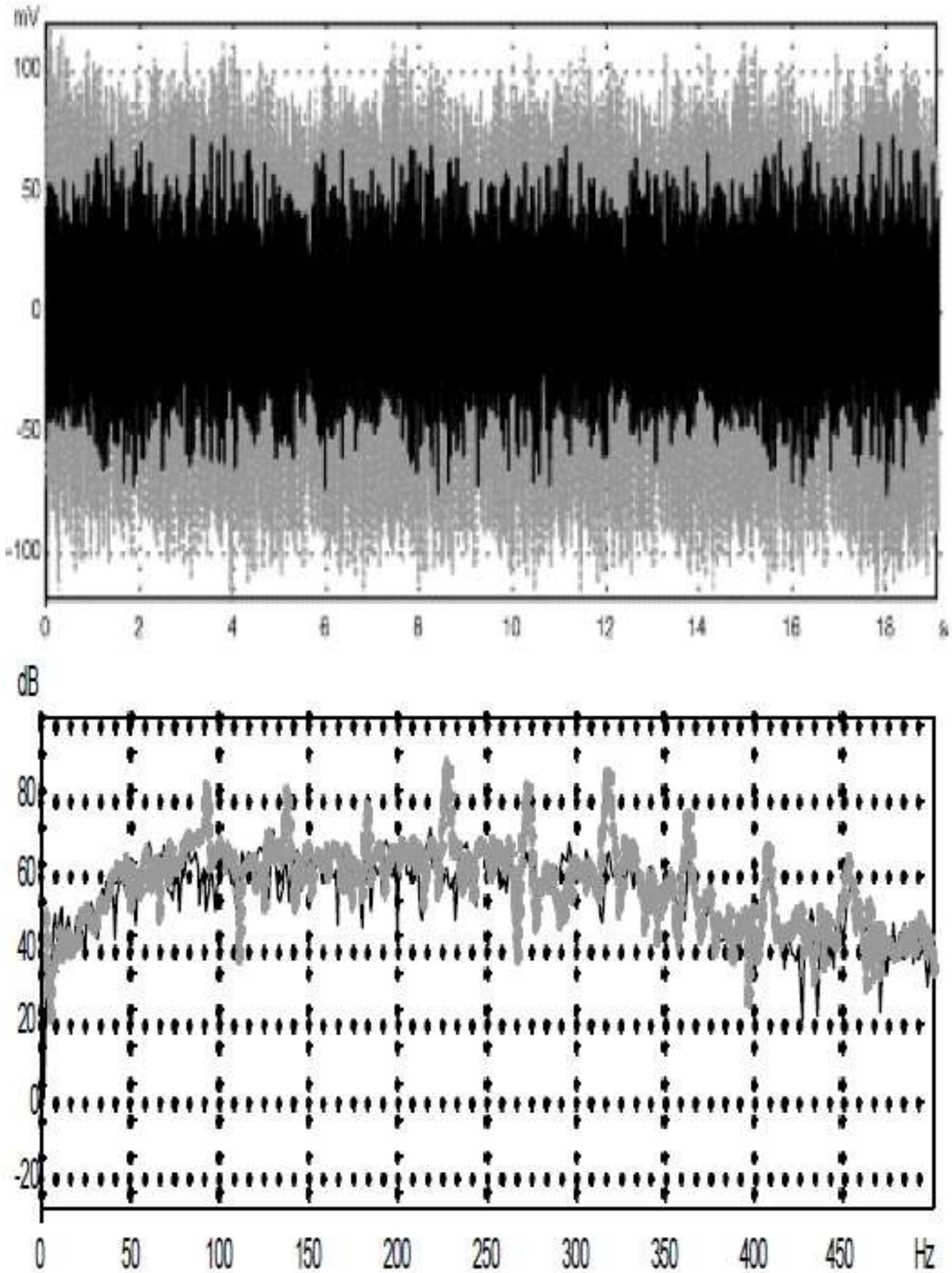

Fig-7: 8Time (top) and frequency (bottom) response of ANC of an motorcycle at 5280 rpm.

4. CONCLUSIONS

An active muffler with one control loudspeaker inclined at 45 degree with the muffler centerline is developed and tested. The active muffler is attached to engine exhaust systems for exhaust noise attenuation with a goal of attaining broadband noise suppression. Feedback ANC with FXLMS and Godart modified algorithms are independently employed for controller tuning. From experimental results it is clearly shown that although Godart algorithm has a slightly faster transient response than that of FXLMS for a pure tone noise excitation, FXLMS outperforms Godart modified algorithm in terms of noise attenuation in the overall sense for all the noise used in the experiments. Feedback ANC with FXLMS algorithm attained over 6 dB reductions of engine exhaust noise, whereas feedback ANC with Godart algorithm has only 5 dB noise attenuation.

Finally, compared to results in references [5] and [6], feedback ANC with FXLMS has a slightly better noise attenuation performance. It is, however, pointed out here that flow and temperature effects are neglected in this study. These effects will be included in the future work.

5. REFERENCES

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