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A Hybrid Active/Passive Exhaust Noise Control System (APECS) for Locomotives

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13. ABSTRACT (Maximum 200 words)
This report documents the analysis and development of an Active/Passive Exhaust Noise Control System (APECS) for railroad passenger locomotives that could provide at least 10 dBA noise reduction.

Based on preliminary analysis and tests on actual locomotives in 1995 and 1996, a combination active roof-mounted system and a passive exhaust silencer were designed. The roof-mounted system consisted of ten enclosures, each with 2 12-inch speakers as actuators; eight microphones mounted on the roof as control sensors; a cabinet containing the controller and other electronics; a passive exhaust silencer specially designed for this system; and other components. An active liner system was also considered instead of the roof-mounted system; however, the latter was chosen since it could be built with commercially available and less costly components.

This design was installed and tested on a locomotive in 1999. The tests included measurements of noise reduction performance and an inspection for any physical degradation of the system.

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PREFACE

This project has been carried out as part of the Federal Railroad Administration (FRA) "Next Generation High-Speed Rail Technology Program."

The authors are grateful for the support and cooperation of the following commuter rail systems: MBTA in Boston, SCRRA in Los Angeles, and Metra in Chicago, who provided the test locomotive, test site, personnel, and equipment to help in the development of the Active/Passive Exhaust Noise Control System (APECS). The authors also acknowledge the support of the Electro-Motive Division (EMD) of General Motors who were responsible for all locomotive interface issues and who provided personnel on site during the installation and test of the APECS. In particular, the authors thank Greg Prentice of EMD without whose diligence, hard work and organization the final installation and test of the system would never have occurred.

METRIC/ENGLISH CONVERSION FACTORS

ENGLISH TO METRIC

LENGTH (APPROXIMATE)

1 inch (in)	= 2.5 centimeters (cm)
1 foot (ft)	= 30 centimeters (cm)
1 yard (yd)	= 0.9 meter (m)
1 mile (mi)	= 1.6 kilometers (km)

AREA (APPROXIMATE)

1 square inch (sq in, in^2)	= 6.5 square centimeters (cm^2)
1 square foot (sq ft, ft^2)	= 0.09 square meter (m^2)
1 square yard (sq yd, yd^2)	= 0.8 square meter (m^2)
1 square mile (sq mi, mi^2)	= 2.6 square kilometers (km^2)
1 acre = 0.4 hectare (he)	= 4,000 square meters (m^2)

MASS - WEIGHT (APPROXIMATE)

1 ounce (oz)	= 28 grams (gm)
1 pound (lb)	= 0.45 kilogram (kg)
1 short ton = 2,000 pounds (lb)	= 0.9 tonne (t)

VOLUME (APPROXIMATE)

1 teaspoon (tsp)	= 5 milliliters (ml)
1 tablespoon (tbsp)	= 15 milliliters (ml)
1 fluid ounce (fl oz)	= 30 milliliters (ml)
1 cup (c)	= 0.24 liter (l)
1 pint (pt)	= 0.47 liter (l)
1 quart (qt)	= 0.96 liter (l)
1 gallon (gal)	= 3.8 liters (l)
1 cubic foot (cu ft, ft^3)	= 0.03 cubic meter (m^3)
1 cubic yard (cu yd, yd^3)	= 0.76 cubic meter (m^3)

TEMPERATURE (EXACT)

[(x-32)(5/9)] °F = y °C

METRIC TO ENGLISH

LENGTH (APPROXIMATE)

1 millimeter (mm)	= 0.04 inch (in)
1 centimeter (cm)	= 0.4 inch (in)
1 meter (m)	= 3.3 feet (ft)
1 meter (m)	= 1.1 yards (yd)
1 kilometer (km)	= 0.6 mile (mi)

AREA (APPROXIMATE)

1 square centimeter (cm^2)	= 0.16 square inch (sq in, in^2)
1 square meter (m^2)	= 1.2 square yards (sq yd, yd^2)
1 square kilometer (km^2)	= 0.4 square mile (sq mi, mi^2)
10,000 square meters (m^2)	= 1 hectare (ha) = 2.5 acres

MASS - WEIGHT (APPROXIMATE)

1 gram (gm)	= 0.036 ounce (oz)
1 kilogram (kg)	= 2.2 pounds (lb)
1 tonne (t)	= 1,000 kilograms (kg)
	= 1.1 short tons

VOLUME (APPROXIMATE)

1 milliliter (ml)	= 0.03 fluid ounce (fl oz)
1 liter (l)	= 2.1 pints (pt)
1 liter (l)	= 1.06 quarts (qt)
1 liter (l)	= 0.26 gallon (gal)
1 cubic meter (m^3)	= 36 cubic feet (cu ft, ft^3)
1 cubic meter (m^3)	= 1.3 cubic yards (cu yd, yd^3)

TEMPERATURE (EXACT)

[(9/5) y + 32] °C = x °F

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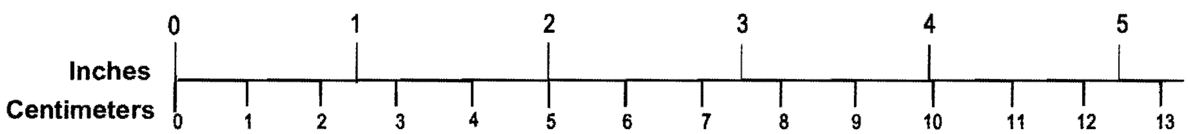

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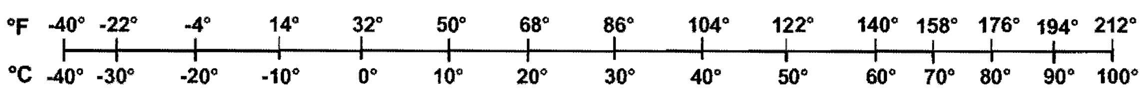

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EXECUTIVE SUMMARY

Trains operated in the United States are powered primarily by diesel-electric locomotives, which are generally noisier than comparable electric locomotives. Based on measurements made on an SD40-2 freight locomotive, which uses a similar power plant to an FP40PH passenger locomotive, the primary sources of diesel-electric locomotive noise are from the engine exhaust and the cooling fans. This report focuses on controlling the engine exhaust noise through a combination of passive and active systems. Two active system concepts were considered: the active liner, and the roof mounted feed-forward system. Resistive exhaust silencers were used as the passive system.

The goal of this program is to design, build and test an exhaust noise reduction system that will reduce the A-weighted exhaust noise signature of a passenger locomotive by 10 dBA. This system should be built compatible with existing passenger locomotives and durable to run in normal service for a period of six months. This project has been carried out as part of the Federal Railroad Administration (FRA) "Next Generation High-Speed Rail Technology Demonstration Program."

Preliminary analysis indicated that the most effective and cost efficient system was a combination active roof-mounted feedforward system that could effectively control tonal peaks up to 200 Hz, and passive exhaust silencers that could effectively control broadband noise above 200 Hz. Though an active liner system had potentially better performance compared to the active roof-mounted system, the latter was chosen since it could be designed with commercially available and less costly components.

Preliminary tests were conducted to aid in the development of a locomotive exhaust reduction system. The first series of tests was performed on an F40PH diesel-electric locomotive on December 3, 1995, at the Massachusetts Bay Transportation Authority's (MBTA) commuter rail system in Boston, Massachusetts. In these tests, measurements were made of the temperature and sound pressure in and around the exhaust duct.

Another series of tests was implemented on an F59 PHI diesel-electric locomotive on July 15, 1996, at the Southern California Regional Rail Authority's (SCRRA) commuter rail system in Los Angeles, California. These tests were to: measure the reduction achievable in locomotive exhaust tones below 250 Hz with an experimental roof mounted active noise control system; acquire transfer function data necessary to evaluate analytically alternate system configurations; and acquire acoustic, temperature and flow velocity data to facilitate the design of an integrated active/passive silencer.

The results of these tests demonstrated that an active noise control system could provide substantial reduction in low frequency exhaust noise between 0 and 150 Hz at fast idle (12.3 dB), throttle 4 (9.0 dB) and throttle 8 (5.5 dB) with lesser reductions at idle (1.9 dB) and throttle 6 (1.9 dB).

From the results obtained, a prototype Active/Passive Exhaust Noise Control System (APECS) was designed, using ten speaker enclosures with two 12-inch high fidelity speakers, and arranged

around the exhaust port; eight control microphones mounted to the roof; a cool box to thermally protect the speakers from the exhaust; an exhaust silencer specifically designed for this test; a cabinet that housed the controller and other electronics; and other components.

On July 26, 1999, another series of tests was implemented on an F40PH-2 passenger locomotive at the Chicago Metra's 51st St. Yard in Chicago, Illinois. In these tests, the APECS installed and tested was to demonstrate the acoustic performance of the system, and to evaluate the ability of the system to perform properly in a railroad environment. For most of the operating conditions, the noise reduction system provided audible reductions in locomotive exhaust noise, by reducing the A-weighted exhaust noise to within a few decibels of the goal of 10-dBA. The installation of the cool box/control speaker assembly and the silencer went smoothly with few fitting problems. The control speakers in their protective cool box proved to be very durable even in the hostile environment of the locomotive engine compartment. The passive silencer performed well acoustically, and except for the protective cover on the thermal insulation which began to melt at high temperature, also performed well mechanically, and the thermal insulation provided the needed thermal protection to the cool box. Backpressure was somewhat higher than specified for the engine, though it is not known if this is a problem. The control microphones performed well after a problem with a DC offset was corrected. All the other components performed well including the optical tachometer, controller, signal input/output boards, and RTD boards. Problems were encountered cooling the electronics enclosure due primarily to the power amplifiers. Proper temperature control (both hot and cold) needs to be addressed before the system can be placed in service for any length of time.

The test of the APECS was performed with the locomotive under stationary conditions. Originally, the plan was to put the test locomotive with the APECS into service for 6 months, but that was not done due to funding constraints. Consequently, the APECS was installed in the locomotive without making all of the modifications necessary for in-service operation.

1. INTRODUCTION

1.1 BACKGROUND

This project has been carried out as part of the Federal Railroad Administration (FRA) "Next Generation High-Speed Rail Technology Demonstration Program."

Diesel-electric locomotives are generally noisier than comparable electric locomotives. Since the U.S. railroad industry is powered primarily by diesel electric locomotives, noise from these units is a substantial barrier to the introduction of high-speed passenger rail service in the U.S.

As indicated in Figure 1, the primary sources of diesel electric locomotive noise are the engine exhaust and the cooling fans. The data in the figure are based on measurements made in the frequency range from 40 Hz to 10 kHz at 100 feet to the side of an SD40-2, a 3000 HP freight locomotive, on the Burlington Northern Railroad. Although the SD40-2 is a freight locomotive, it has the same power plant as an F40PH, a very common passenger locomotive. Rolling noise data are based on measurements at 100 feet from track centerline with the locomotive coasting by.

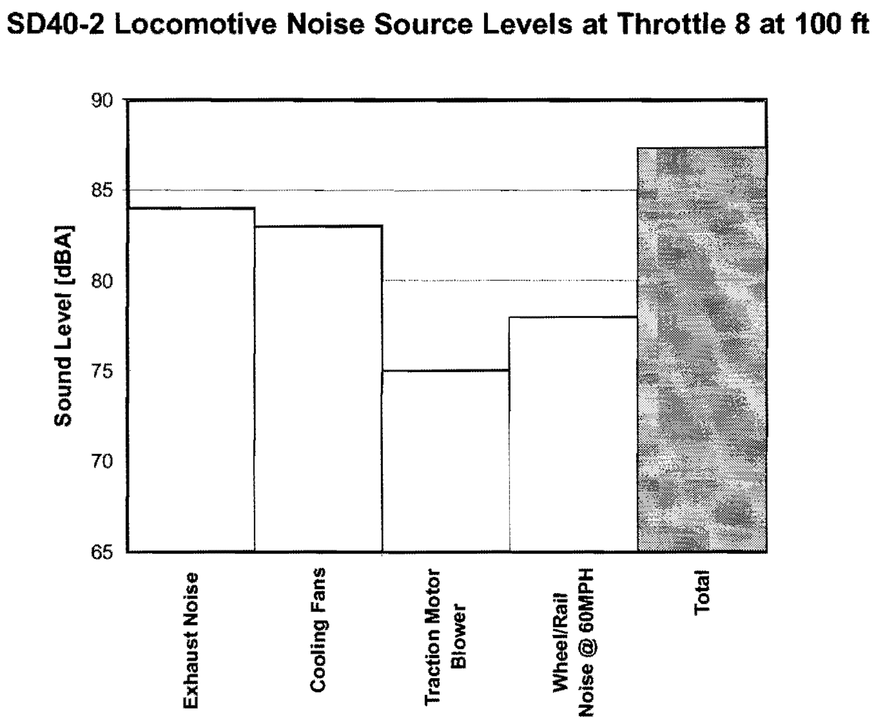


Figure 1. Noise Sources on an SD40-2 Diesel Electric Locomotive Measured at 100 ft with the Locomotive Running at Throttle 8 at Full Load

It is clear from the figure that both engine exhaust and the cooling fans must both be reduced before significant reductions in diesel electric locomotive noise can be achieved.

This program focuses on controlling the engine exhaust noise through a combination of passive and active systems. Because of the limited space inside the locomotive engine compartment, passive silencers alone cannot provide sufficient noise reduction, especially at low frequency. A purely active system on the other hand would become quite costly and complex at high frequency where significant broad band noise attenuation is needed. Resistive exhaust silencers were considered as passive systems in these tests. These silencers were selected, because such silencers typically have low backpressure, a critical requirement in locomotive silencers. High backpressure can have a significant negative impact on engine efficiency, an overriding economic consideration with locomotives.

Two concepts were considered for active systems: the active liner and the roof mounted feedforward system. The active liner concept is illustrated in Figure 2. On opposite sides of an exhaust pipe are two cavities separated from the pipe by a flow resistive liner. At the back of each cavity is an actuator (a high temperature speaker). A control pressure sensor (a high temperature microphone) is placed just behind the liner in each of the cavities. A feedback

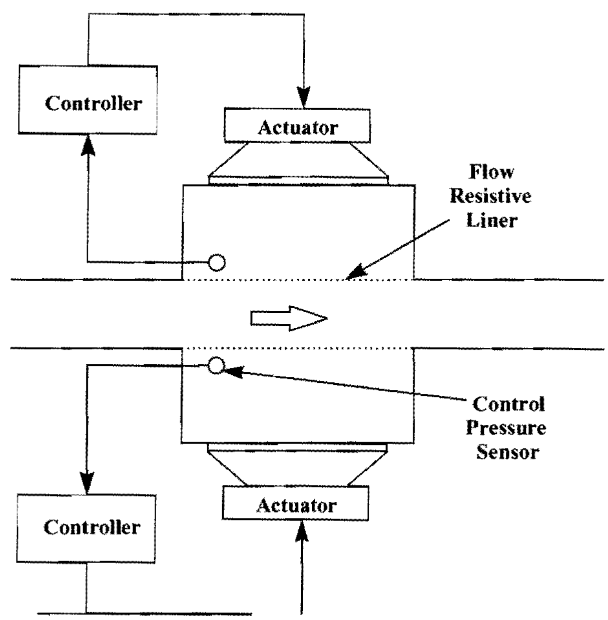


Figure 2. Active Liner Concept

loop connects the sensor through a control filter and power amplifier to the actuator. When activated, the feedback control system will attempt to drive the dynamic pressure behind the liner to zero. Doing so will increase the particle velocity through the liner, enhancing the attenuation of an acoustic wave that attempts to propagate down the pipe. A similar effect will occur passively when the depth of the cavity is one quarter, three-quarters, five quarters, etc. of an acoustic wavelength. A comparison of the performance of the active and passive systems is shown in Figure 3 for a 4-inch deep cavity at 650°F that extends 24 inches along the length of the pipe. The passive silencer has a substantial peak in insertion loss (or IL, defined as the difference between the uncontrolled and the controlled sound pressures, measured in dB) at 1.2 and 3.6 kHz corresponding to the cavity being ¼ and ¾ of an acoustic wavelength. However, below 1 kHz and at 2.4 kHz the insertion loss drops precipitously. The active liner improves the performance substantially both below 1 kHz and in the vicinity of 2.4 kHz.

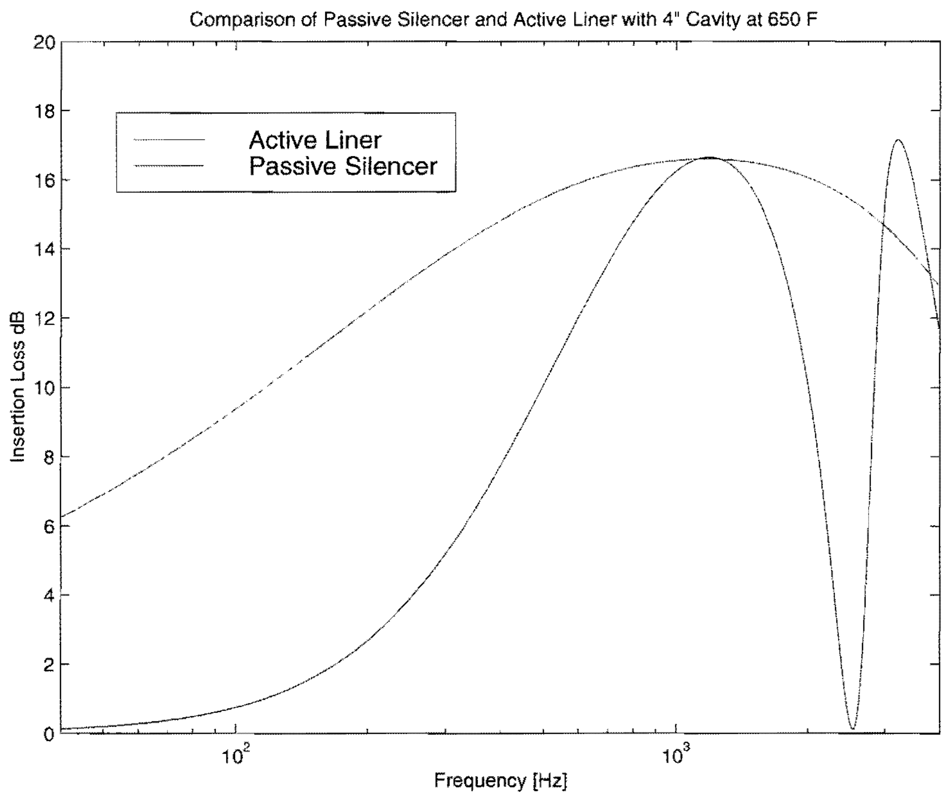


Figure 3. Comparison of Active and Passive Liner Performance

The roof mounted feedforward system is illustrated in Figure 4. The figure shows a section through the locomotive hood in the vicinity of the exhaust outlet, usually called the exhaust stack. Mounted close to and surrounding the exhaust stack are a number of actuators (speakers). Mounted further away near the edge of the hood of the locomotive are a number of control pressure sensors (microphones). Figure 5 shows a block diagram of the control system. A tachometer reference signal is fed to the control filter that in turn drives the actuators through a number of power amplifiers. The resulting sound, engine exhaust plus control signal, is monitored by the control microphones on the hood of the locomotive. The signals from these microphones are fed to the controller where they, in combination with the tachometer reference signal, are used in a Least Mean Square (LMS) algorithm to adapt the control filter to minimize the noise at the control microphones. Figure 6 shows a typical estimate of the noise reduction

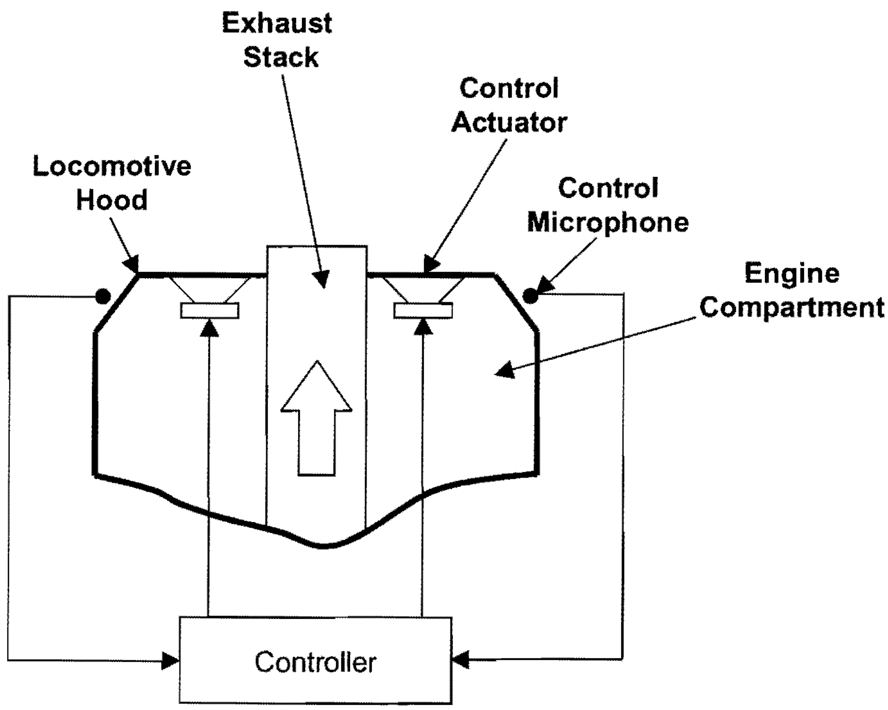


Figure 4. Simplified Roof Mounted Feedforward System for Exhaust Noise Control

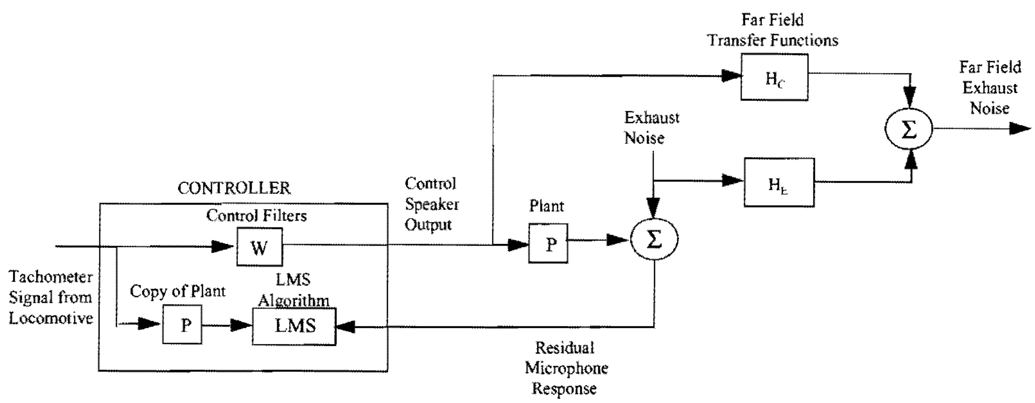


Figure 5. Feedforward System Block Diagram

that would be expected from this arrangement. The calculation was done for eight control speakers and eight control microphones and shows the average far field noise reduction in the horizontal plane around the locomotive. For dimensions typical of the locomotive and the exhaust stack the figure shows that the system is effective only up to about 200 Hz. This upper frequency limit can be extended slightly by moving the control speakers closer to the exhaust outlet. In addition some improvements in noise reduction can be obtained by moving the control microphones farther away. Further performance improvements and extensions of the effective frequency range of the treatment require the addition of passive systems.

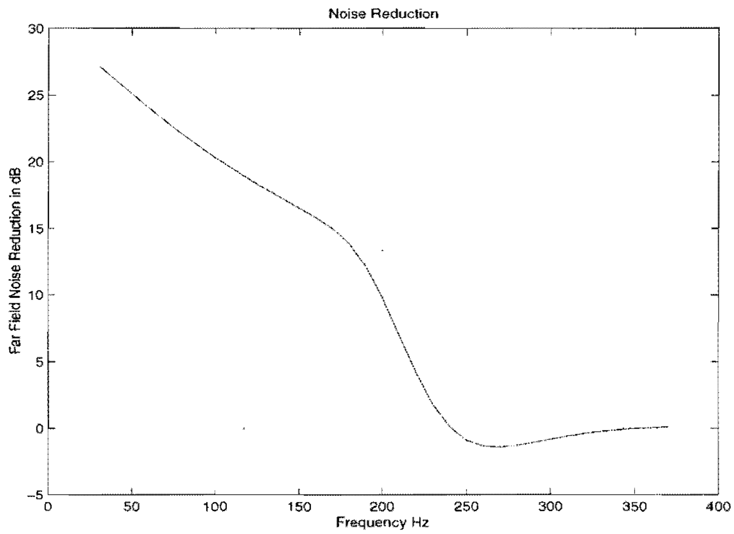
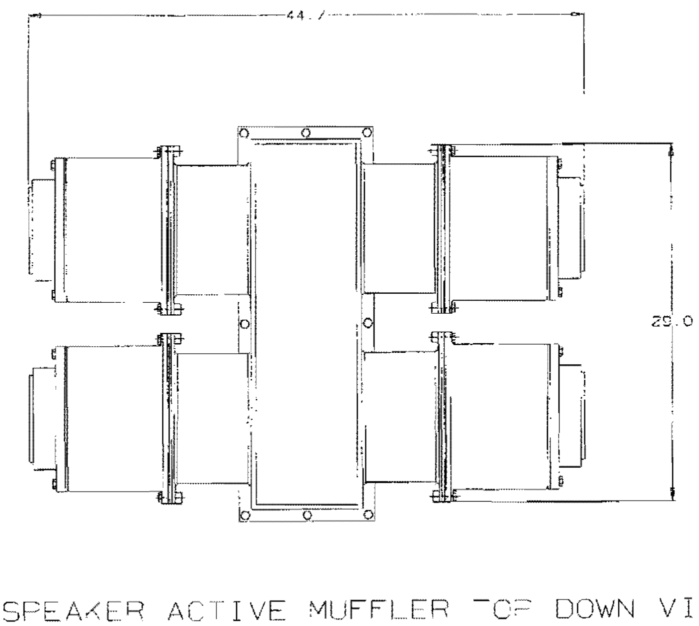


Figure 6. Simulated Performance of the Feedforward Roof Mounted System

The active liner system was initially favored as the active system, since it was relatively compact and could substantially reduce noise throughout the frequency range desired. However, no system could be designed that both could fit in the space available and survive the exhaust stream temperatures of up to 950°F without becoming extremely expensive. The design examined employed eight drivers, four on each side of the exhaust duct (see Figure 7). Each driver consisted of an electro-dynamic actuator capable of 100-lb force vector driving a metal speaker cone with a metal bellows surround. External cooling air was required to ensure the reliability of the actuators. However, even with this robust arrangement, the estimated system performance and reliability was not enough to proceed further with this concept. Instead, two system designs were selected, each focused on a particular frequency range. The result is the roof mounted feedforward system with an integrated passive silencer.

While the roof mounted feedforward system is limited in the frequency range over which it can be employed, it does offer a number of advantages over the active liner concept. First since the speakers do not need to vent into the exhaust duct, they can be placed in a much more benign and protected environment. This allows the use of commercial grade loudspeakers significantly improving reliability. Finally the control microphones are also in a more benign environment allowing the use of commercial grade microphones rather than expensive high temperature devices.



Dimensions in Inches

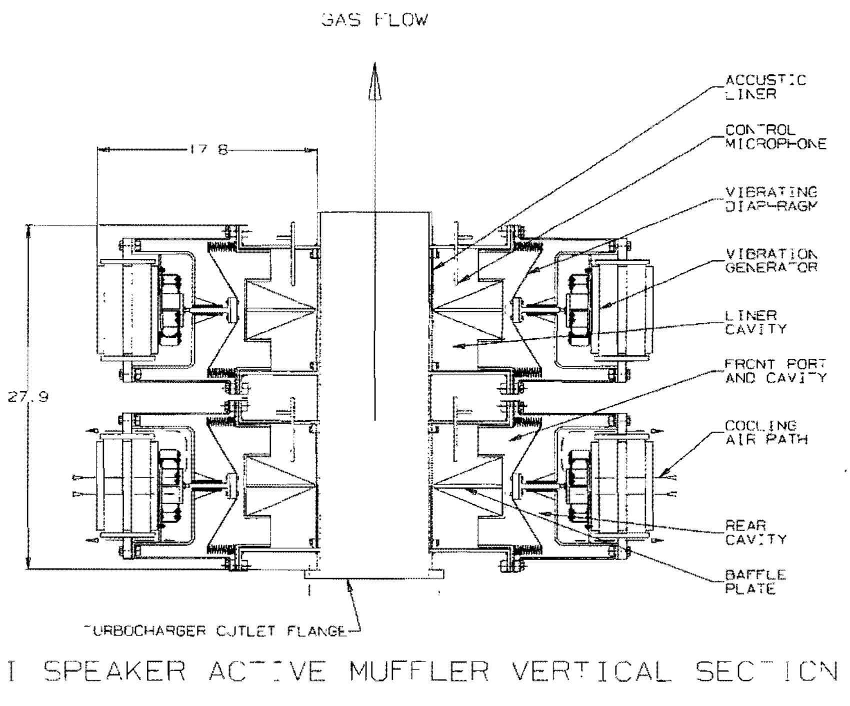


Figure 7. Typical Active Liner Actuator Design

1.2 OBJECTIVE

The goal of this program is to design, build and test an exhaust noise reduction system that will reduce the exhaust noise signature of a passenger locomotive by 10 dBA. The hardware should be sufficiently durable and compatible with the locomotive that the system could be run in normal service for a period of six months. The 10-dBA goal is based on the relative magnitude of the source strengths on the locomotive as shown in Figure 1. A 10-dBA reduction will bring exhaust noise down to the level of the traction motor blowers the next highest source after the cooling fans. Of course, all of this is done in anticipation of the cooling fans ultimately receiving similar treatment.

1.3 ORGANIZATION OF THE REPORT

This report is organized into seven sections. Section 2 describes many of the system design issues and the rationale for arriving at the particular system configuration ultimately selected. Section 3 describes the preliminary testing that provided the information to support the system design. Section 4 describes the prototype system that was fabricated for testing on a locomotive. Section 5 presents the testing carried out on the prototype. Section 6 outlines the expected modifications to the prototype design necessary before the system can be tested in-service, and Section 7 presents conclusions and recommendations.

1.4 OTHER DOCUMENTATION

Other documents written provide detail that supplements this report. More information on the operation of the controller and a detailed annotated listing of the controller software can be found in the report, "APEC System Functional Description and Operator's Manual," [1]. Detailed drawings of all components including assembly drawings are available in the document, "APECS Drawing Package" [2].

2. SYSTEM DESIGN

2.1 THE NOISE REDUCTION REQUIREMENTS

The goal of the program is to achieve 10 dBA of exhaust noise reduction from a passenger locomotive. To achieve this goal, the following should be determined for the active and passive systems: the minimum noise reduction requirements from each system; the functional frequency range of each system; and whether only tonal noise or broad band noise control is required for the active system.

Figure 8 shows the spectrum of the A-weighted sound pressure level (SPL) at approximately 3 feet from the edge of the exhaust stack at throttle 8 under full load. The data were measured on a stationary F40PH locomotive operated by the Massachusetts Bay Transportation Authority (MBTA). Since the data were measured very near to the stack, it is expected that most of the noise in Figure 8 comes from the diesel engine exhaust. Although there may be some contributions from the dynamic brake fans and the radiator cooling fans, earlier studies indicate that these other sources would only be significant in a few frequency bands and then only for measurements far from the locomotive [3].

Table 1 examines whether reduction of only the tones in the exhaust signature is sufficient or whether broad band noise reduction is required. The table focuses on the low frequencies where the active system would function. It shows the estimated noise reduction that would occur if an active noise control system were used to suppress tones to varying degrees in various frequency bands. For example, the first line in the table shows that if all the tones below 250 Hz were eliminated the overall noise reduction from 0 to 250 Hz is approximately 16.8 dBA. If the tones are not eliminated but reduced by 10 or 20 dB the reduction is less but still significant. This result indicates that just controlling the tones from 0 to 250 Hz will result in significant overall noise reduction and broad band noise control is not needed.

If the control band was extended to 500 Hz, reducing or eliminating the tones has much less effect on the overall noise in that band. If the tones are eliminated the overall noise level from 0 to 500 Hz is reduced by only 5 to 6 dBA. To achieve additional noise reduction would require broad band noise control. Just controlling the tones is not enough.

The results in the table show that a tonal noise reduction system will be very effective in reducing the overall noise up to 250 Hz. If the control band was extended to higher frequency, controlling only the tones will not be sufficient and the control system will need to reduce broad band noise as well. This is an advantageous result because active control technology is especially well suited to controlling tones at low frequency. Thus, the active system was designed to control just the tones from 0 to 250 Hz.

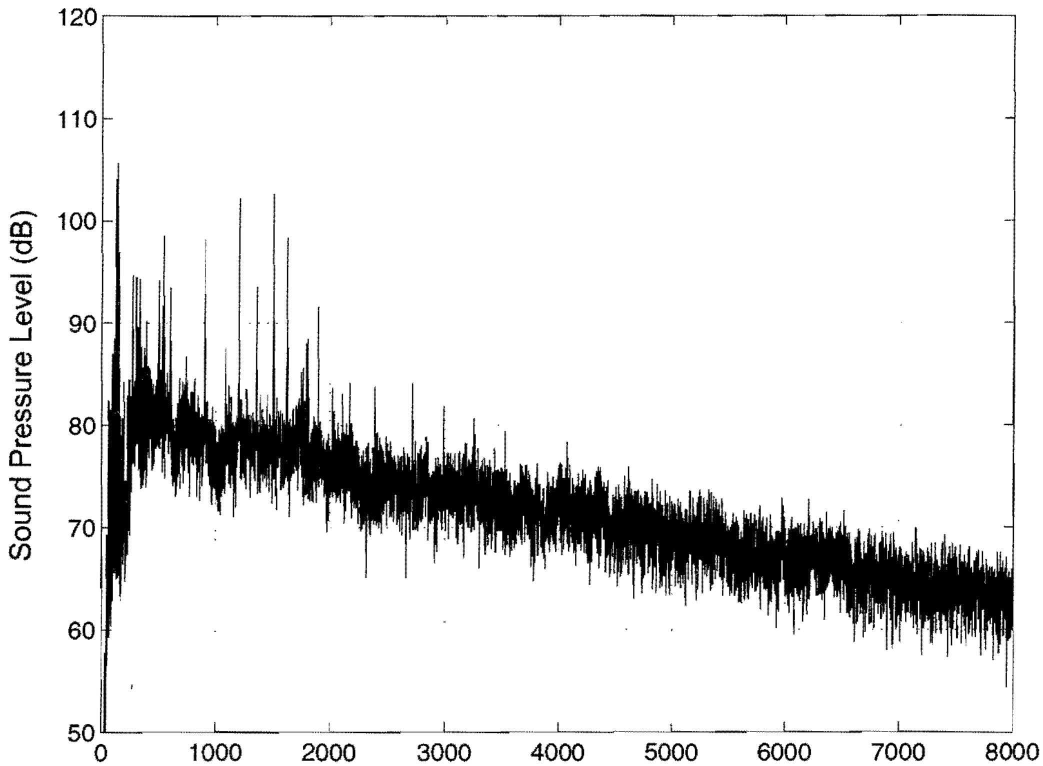


Figure 8. The A-Weighted Exhaust Noise Spectrum 3 Feet Aft of the Exhaust Stack at Throttle 8 at Full Load, Sound Pressure Level [dB] Versus Frequency [Hz]

Table 1. Total Noise Reduction in the Control Band for Various Control Bands and Tonal Noise Reductions

Frequency Range for Tonal Noise Reduction (Control Band) [Hz]	Amount of Tonal Reduction [dB]	Total Noise Reduction in the Control Band [dBA]
0-250	total removal	16.8
0-250	20	15.1
0-250	10	9.2
0-500	total removal	6.5
0-500	20	6.4
0-500	10	5.2

The following exercise estimates the overall reduction in locomotive exhaust noise if control of exhaust tones below 250 Hz is combined with passive (broad band) noise control above that frequency. The goal in this exercise will be to define the level of broad band noise reduction required and the frequency range over which it must extend in order to obtain an overall reduction in exhaust noise of 10 dBA.

Figure 9 shows the estimated A-weighted overall reduction in exhaust noise that would result if all of the tones below 250 Hz were reduced by 10 dB. This might occur if an active noise control system was employed to suppress the tones in the exhaust, and, in addition, 15 dB of broad band passive noise control were employed above 1000 Hz.

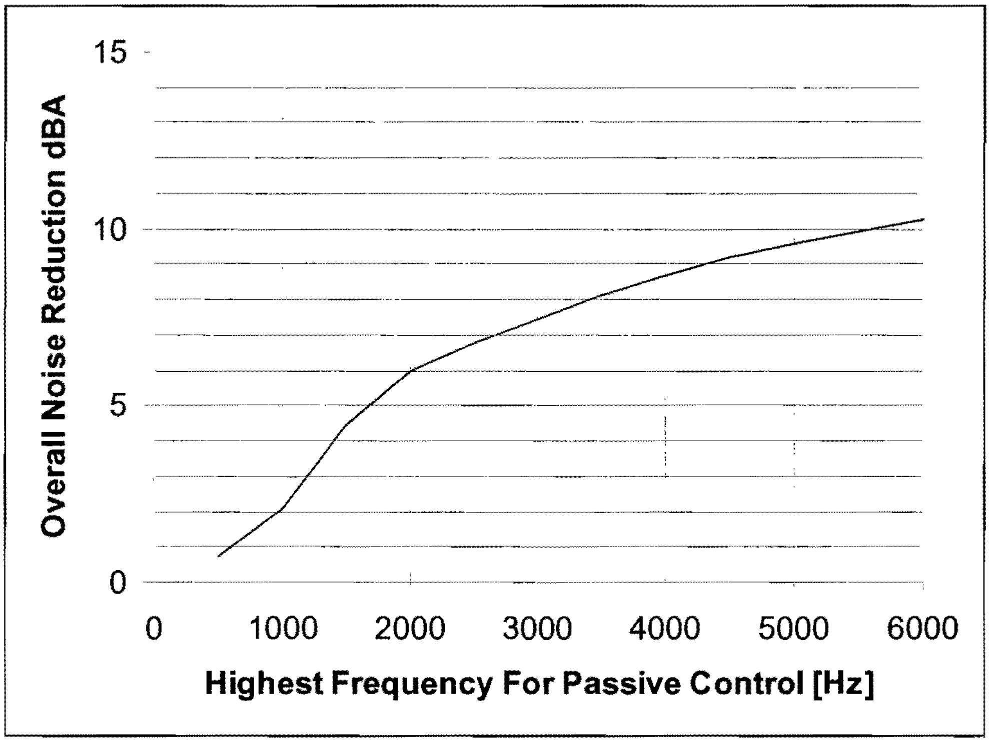


Figure 9. A-Weighted Overall Noise Reduction as a Function of the Highest Frequency of Effective Broad Band Noise Control

The estimates in broad band noise reduction required for the passive silencer in Figure 9 were based on the fact that passive silencers are less effective at low frequencies. To account for this, the insertion lost (IL) of the silencer was assumed to be zero below 250 Hz and 5 dB from 250 to 500 Hz. Figure 9 shows the increase in overall A-weighted exhaust noise reduction as the highest frequency at which the silencer is effective is increased. The curve crosses 10 dBA at approximately 5.5 kHz indicating that the passive silencer should provide 15 dB of insertion loss up to at least 5.5 kHz. Thus, if no low frequency active control were employed, the overall noise reduction from the passive silencer would be limited to approximately 5 dBA no matter what its effective frequency range.

This leads to a number of conclusions:

- Active control of tones in the exhaust below 250 Hz with no broad band control at the higher frequencies will result in less than 1 dBA of noise reduction

- No control of tones in the 0 to 250 Hz band will limit the maximum reduction of exhaust noise to approximately 5 dBA

- 10 dBA of exhaust noise reduction can be achieved with 10 dB reduction of exhaust tones below 250 Hz along with 5 dB of broad band noise reduction from 250 to 500 Hz and 15 dB reduction from 500 to 5500 Hz.

Consequently, the control of overall exhaust noise requires an active system to control tones below 250 Hz and a passive silencer to control the high frequencies above 250 Hz. This need for significant passive silencing at high frequency makes the roof mounted active noise control system more attractive since it separates the passive and active functions and allows more flexibility in the design of the passive silencer since it does not need to also accommodate the control actuators.

2.2 ACTUATOR AND SENSOR NUMBER AND PLACEMENT

The feedforward active control system, illustrated in Figure 10, was anticipated to control low frequency exhaust tones. The block diagram in the figure shows a typical feedforward controller utilizing the Filtered X algorithm to generate and update the control filter coefficients. A tachometer signal from the locomotive diesel engine is used as a single reference signal. The reference signal is applied to the control filters, which drive the control speakers through power amplifiers so as to cancel the noise from the locomotive exhaust in the control microphones. The Filtered X algorithm takes the reference signal and the signal from the control microphones and adjusts the coefficients in the control filters, W, to ensure that the noise generated by the control speakers will continue to effectively cancel the exhaust noise in the control microphones. In short, the system will adapt to changes in the reference signal and to a limited extent to changes in the transfer functions, P, relating the control speaker output to the residual microphone response. With proper control actuator and control microphone placement, the signal that cancels the exhaust noise in the control microphones will also cancel the exhaust noise in the far field.

Figure 11 shows a layout of eight control speakers and eight residual microphones in the roof mounted feed forward system. It is anticipated that the control speakers will be clustered around the exhaust stack and placed in ported enclosures. The ported enclosures serve two purposes. They allow tuning the frequency response of the speakers to concentrate the energy in the frequency band of interest and they protect the speakers from physical damage and from the weather. Control microphones will most likely be placed near the edge of the locomotive hood.

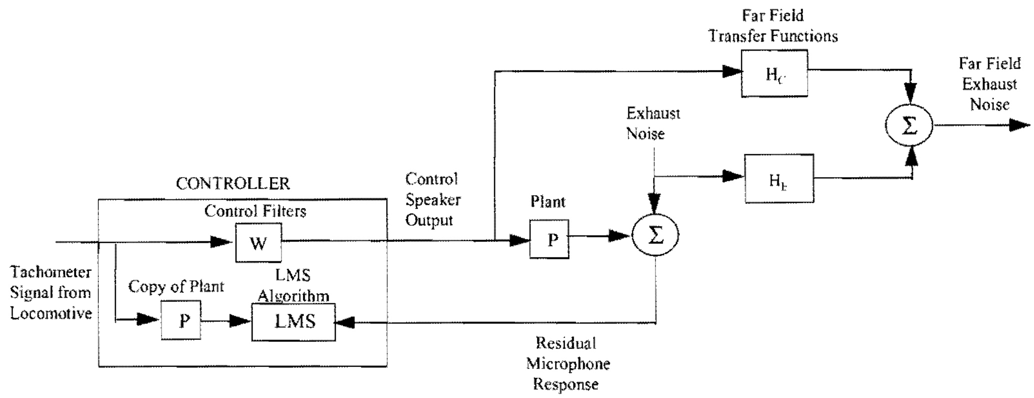


Figure 10. Block Diagram

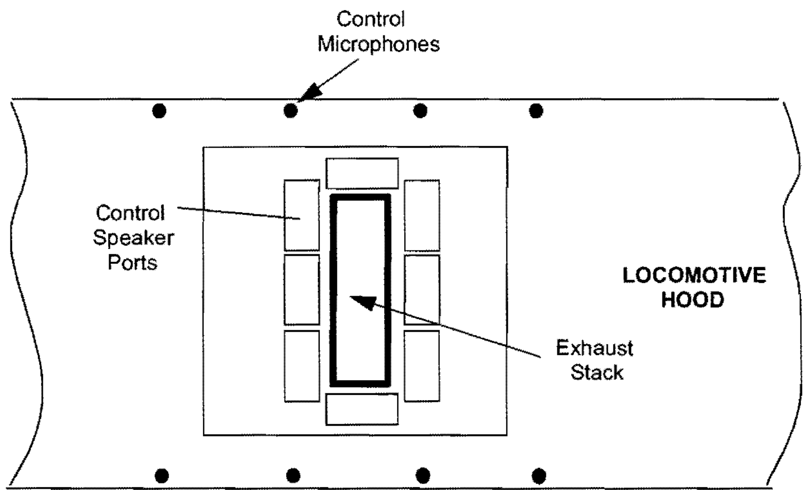


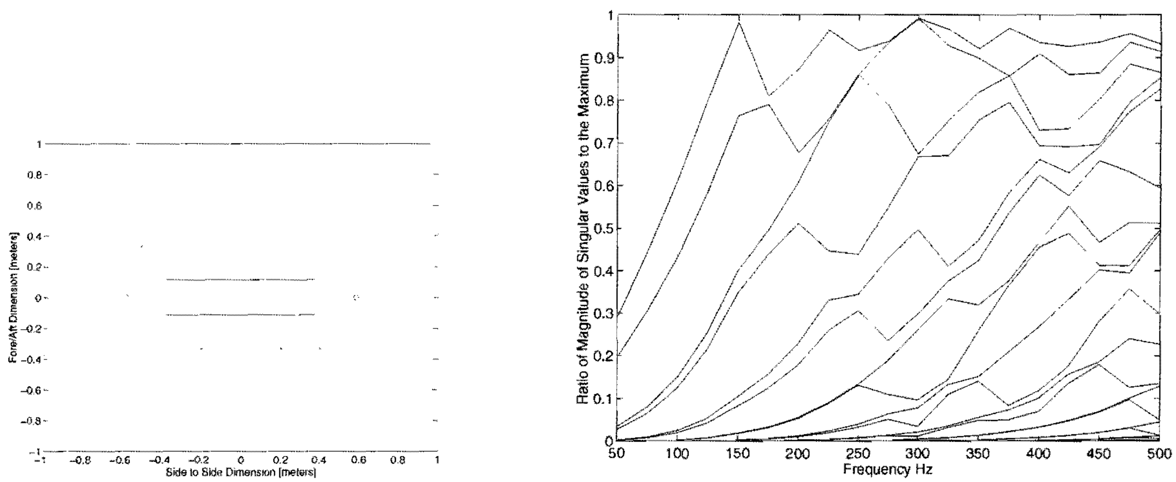
Figure 11. Placement of Control Speakers Around the Exhaust Stack Outlet

2.2.1 Number and Placement of Control Speakers

Because the exhaust duct is large (typically 30 inches by 9 inches) the control speakers cannot be placed as close to the sources of noise as would be desirable for the best noise control performance. Consequently, cancellation of exhaust noise in all directions may not be possible at all frequencies of interest without a very large number of control speakers. This section examines the number of control speakers required to obtain good global exhaust noise reduction and the best location for those speakers.

The number of required control speakers can be estimated by examining the number of significant singular values in the transfer function matrix relating the output of the control speakers near the exhaust stack to the sound pressure in the far field. The number of significant singular values in the matrix equals the minimum number of control actuators required to obtain significant noise reduction.

To estimate these singular values a simulation was performed in which 32 point sources were placed around the exhaust stack as shown in Figure 12 (a) and computed the transfer functions to 90 locations spaced evenly on a circle of 30 meters in radius, centered on the exhaust stack. By assuming free field propagation, the singular values were obtained as shown in Figure 12 (b). As shown in Figure 12 (b), at up to 100 Hz there are only four significant singular values[1]. The number gradually increases with increasing frequency until at 250 Hz there are eight and by 500 Hz there are 13. This result implies, not unexpectedly, that as frequency increases the number of control sources increase.



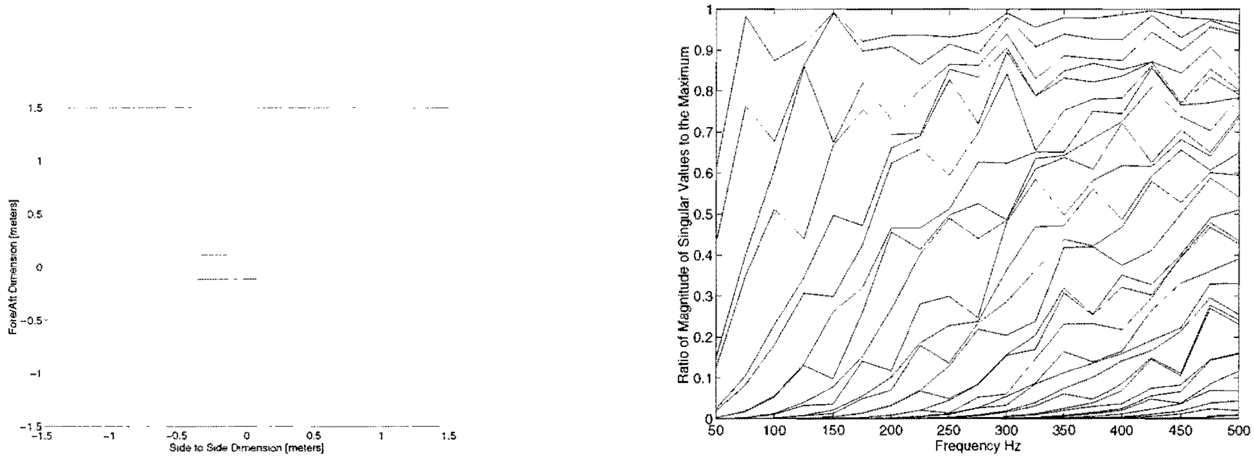
(a) Source Configuration *(b) Ratio of Singular Values to Largest vs. Frequency*

Figure 12. The Ratio of Each Singular Value to the Largest Singular Value as a Function of Frequency

If the distance of the sources from the center of the exhaust stack is doubled as shown in Figure 13, then the singular values are obtained shown in Figure 13 (b). Doubling the distance of the control sources from the exhaust stack nearly doubles the number of significant singular values, implying that the number of control sources needed to control the sound in the far field will double.

It is difficult to use the singular value distribution in Figures 12 and 13 to specify precisely the number of sources required to control the far field sound. It is clear, however, to place as many sources as close as possible to the exhaust stack. From a practical view point, the number of control speakers that can be placed next to the exhaust stack is limited by the physical size of the

[1] In this calculation, a singular value is arbitrarily considered significant, if it is 10 percent of the largest singular value or greater.



(a) Source Configuration *(b) Ratio of Singular Values to Largest*

Figure 13. The Ratio of Each Singular Value to the Largest Singular Value as a Function of Frequency for a Source Configuration at Greater Distance from the Exhaust Stack

speakers required to generate the necessary volume velocity to cancel the exhaust tones. In addition, if the number of control speakers becomes too large the complexity of the control system needed to generate canceling signals to the speakers will increase, resulting in significant increases in cost. Consequently, speakers should be strategically placed as close as possible to the exhaust stack using a configuration similar to that shown in Figure 11. Based on the results in Figure 12, if the centers of the eight control speaker ports were placed as shown in Figure 14, then good performance is expected up to approximately 250 Hz. Beyond that frequency, good far field noise reduction will not be possible.

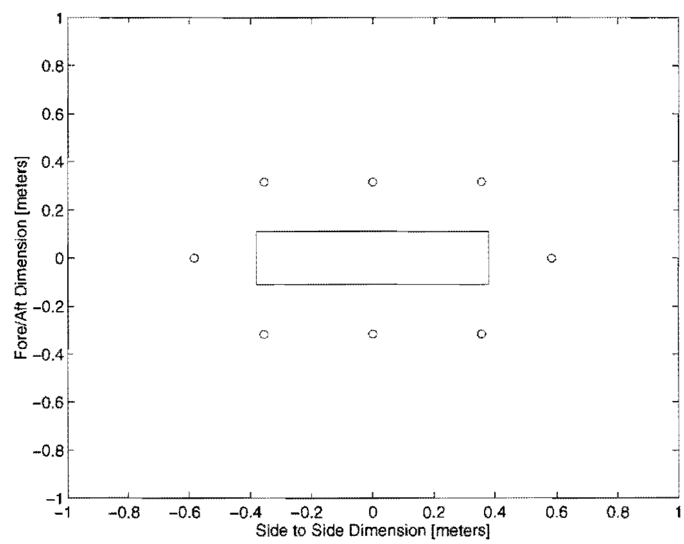


Figure 14. Control Speaker Locations Around the Exhaust Stack

Another simulation computed the optimum source volume velocities from each of the eight control speakers required to cancel optimally (in a least squares sense) the mean square, far field sound at 30 meters from the exhaust outlet. For these calculations, a single point source in the center of the exhaust duct was assumed. The reduction in the far field, mean square, angular-averaged sound pressure is shown in Figure 15. Very large noise reductions are obtained at low frequency with gradually degraded performance as the frequency increases.

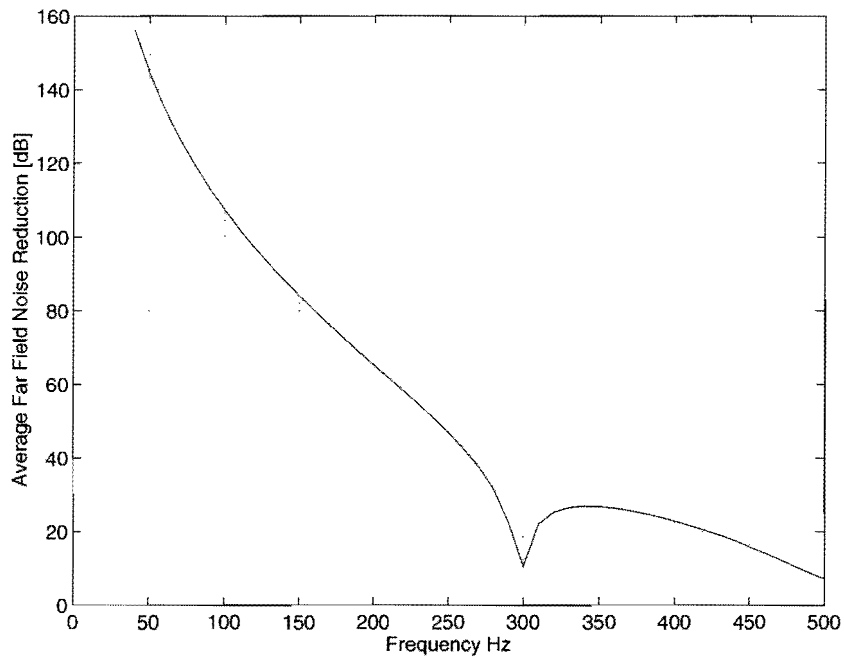


Figure 15. Reduction in the Mean Square, Angular-Averaged Sound Pressure at 30.4 Meters from the Exhaust

The dip in the noise reduction at 300 Hz is related to the distance of the control speakers away from the center of the exhaust duct. If the distance is doubled, the frequency of the dip will decrease to approximately 150 Hz and if the distance is halved, the frequency will increase to approximately 600 Hz with a significant increase in noise reduction. Note that the results in Figure 15 are not representative of the noise reduction achievable with this system configuration. Later sections will discuss other factors will intervene to reduce system performance.

2.2.2 Control Microphones

As part of the active exhaust noise control system, there will be residual (control) microphones to monitor the performance of the control system. Ideally, the control microphones would be placed in the far field, but for practical reasons, they will have to be installed on the roof of the locomotive. Consequently, the control system will drive the control actuators to minimize the signals in these microphones and not the far field sound. Relying on near-field microphones will adversely impact the achievable noise reduction in the far field. This section examines the best placement of these microphones and estimates the achievable noise reduction.

To investigate the required number of residual microphones, the noise reduction with the residual microphones in the far field will be examined, first. Figure 16 shows the simulated reduction in the mean square, angular averaged, sound pressure at 30.4 meters for twelve and eight residual microphones in which a single point source has been used to characterize the locomotive exhaust. The eight and twelve microphones have been uniformly distributed in a 120-degree sector on each side of the locomotive[2]. Also shown, for comparison purposes, is the previous calculation in Figure 15 in which 90 microphones were uniformly spaced around the exhaust at a radius of 30.4 meters. For twelve microphones, the noise reduction is nearly comparable to the result in Figure 15. Eight residual microphones on the other hand show some degradation in performance.

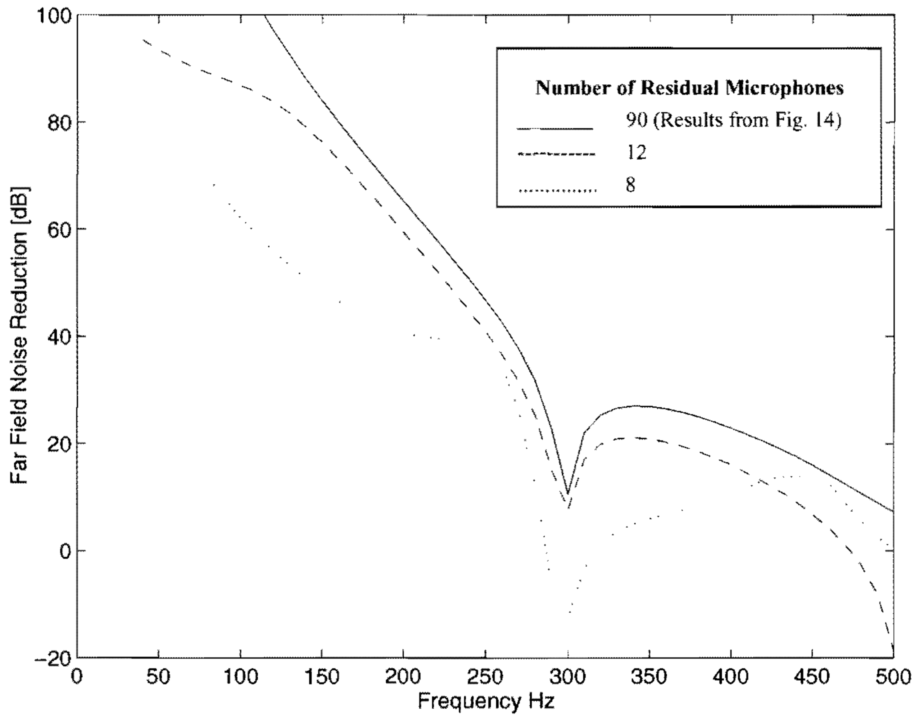


Figure 16. Reduction in the Mean Square, Angular-Averaged Sound Pressure at 30 Meters from the Exhaust Outlet with Eight Control Speakers and Various Numbers of Residual Microphones in the Far Field at 30.4 Meters from the Exhaust

If the residual microphones are placed on the locomotive hood at a radius of 1.5 meters from the center of the exhaust stack, as illustrated in Figure 17 for the twelve microphone case, the noise reduction is obtained as shown in Figure 18. It is interesting to note that with the microphones in the closer position, the performance of the system below 300 Hz with eight residual microphones becomes very similar to the performance with twelve. Four residual microphones provide clearly inferior performance.

[2] Each sector extends from 30 degrees aft of the forward direction to 150 degrees aft of forward.

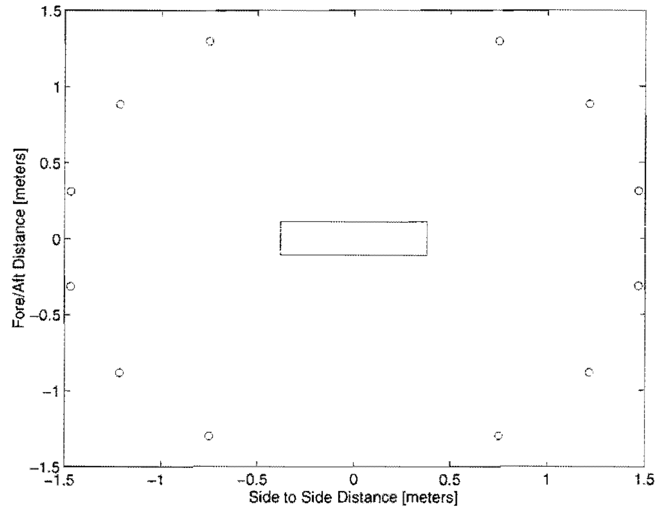


Figure 17. Residual Microphone Locations

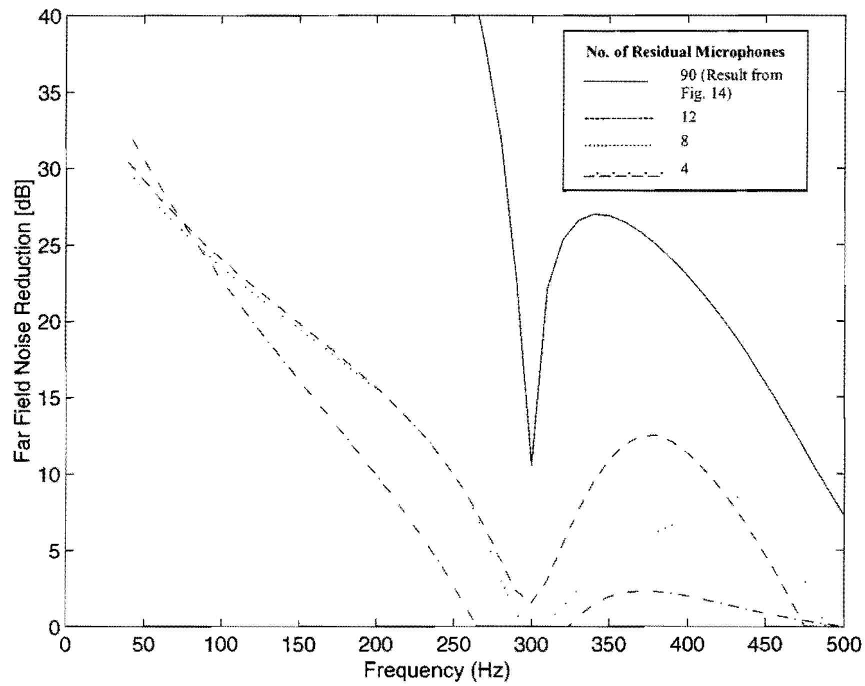


Figure 18. Reduction in the Mean Square, Angular-Averaged Sound Pressure - Noise Reduction [dB] Versus Frequency [Hz] - at 30.4 Meters from the Exhaust Outlet with Eight Control Speakers and Various Numbers of Residual Microphones at 1.5 Meters from the Exhaust

2.2.3 SOURCE COMPLEXITY

The calculations in Figure 15 assume that the exhaust can be modeled as a single point source at the center of the exhaust duct. If in actuality the source is more complex and distributed over the exhaust outlet, the performance may be affected. This section examines distributed source models to determine if there is significant degradation in performance. In Figure 19 a ten point source model is illustrated that will be used in the calculations to follow. If it is assumed that there is a single measurement of the sound in the near field of the exhaust (approximately 1 meter away from the center of the exhaust stack), then an estimate can be made of the ten point source velocities required to generate the measured sound pressure level. Once the source velocities have been estimated, the anticipated noise reduction can be computed with eight control speakers arranged as shown in Figure 14 can be computed. Various numbers of residual microphones are considered, all located at 1.5 m from the center of the exhaust stack as illustrated in Figure 17. Figure 20 shows the resulting reduction in the mean square, angular averaged sound pressure level at 30.4 meters from the center of the exhaust stack (ten point sources were used to model the exhaust). The figure shows that the noise reduction is essentially unchanged from the case in which the exhaust was modeled using a single point source, implying that for this case distributing the source across the exhaust outlet does not materially affect the noise control performance.

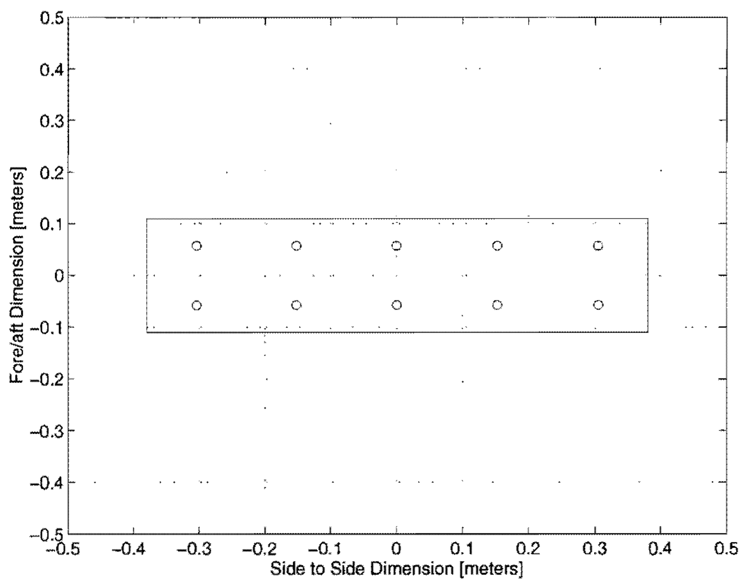


Figure 19. Assumed Source Locations in the Outlet of the Locomotive Exhaust

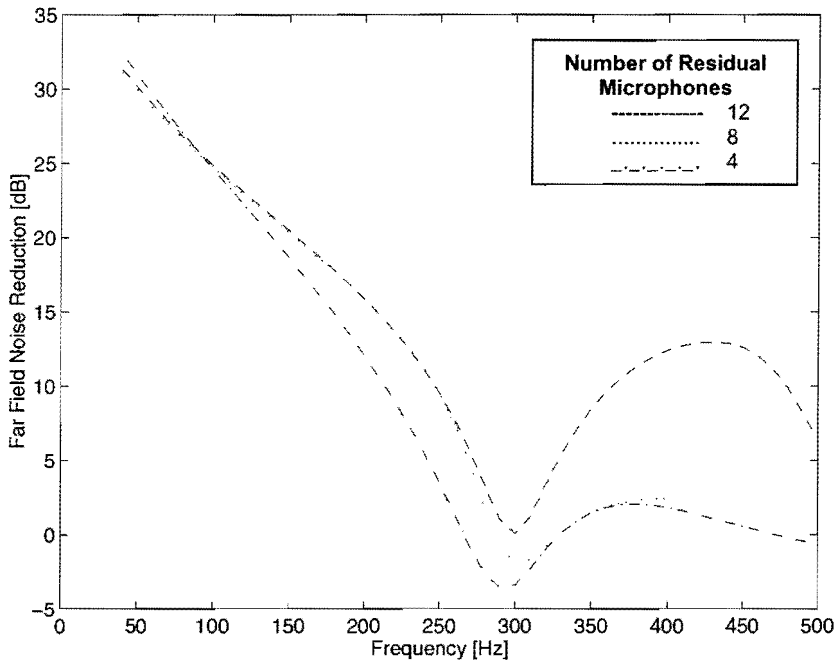


Figure 20. Reduction in the Mean Square, Angular-Averaged, Sound Pressure at 30.4 Meters from the Exhaust Outlet with Eight Control Speakers and Various Numbers of Residual Microphones at 1.5 Meters from the Exhaust

2.2.4 Refinement of System Layout

The previous section examined the dependence of the system performance on the number and location of the control speakers and residual microphones. This section examines refinements on the system layout that might improve performance. Figure 21 shows the eight-control speaker eight-residual microphone system examined so far. Since moving the residual microphones farther from the exhaust stack and moving the control speakers closer will generally improve performance, the configuration as shown in Figure 22 will be tried. In the figure, the control speakers were moved slightly closer to the exhaust stack. For this configuration, it is assumed that the control speakers will be placed in ported enclosures with 12 inches by 6 inches ports (0.3 m by 0.15 m). The residual microphones were originally laid out on the arc of a circle of 1.5 m radius centered on the exhaust stack and arranged to span a 120° sector. Now they have been arranged along a straight line and moved 1.5 m away from the center of the exhaust stack to the edge of the locomotive hood. They still span a 120° sector.

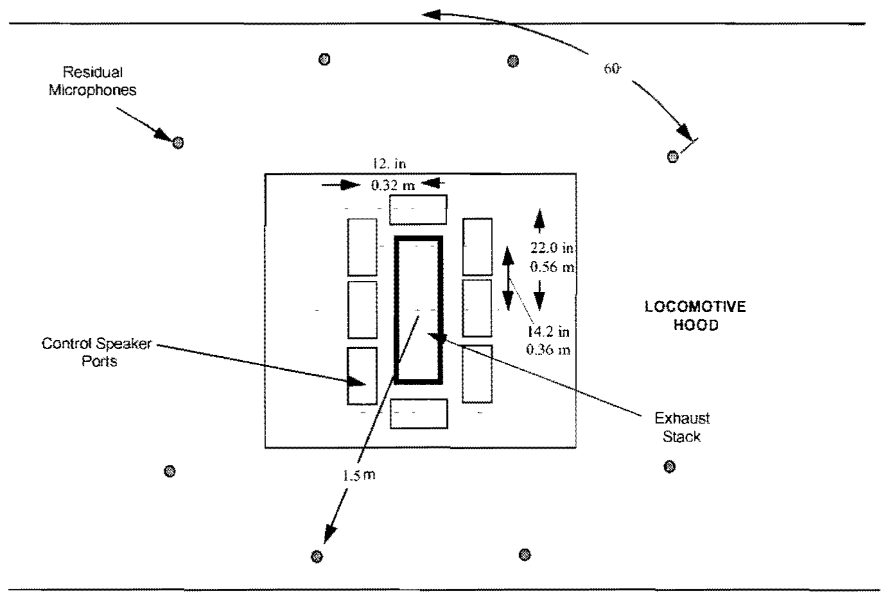


Figure 21. Plan View of the Locomotive Hood Showing the Control Speaker and Residual Microphone Layout Used to Estimate System Performance

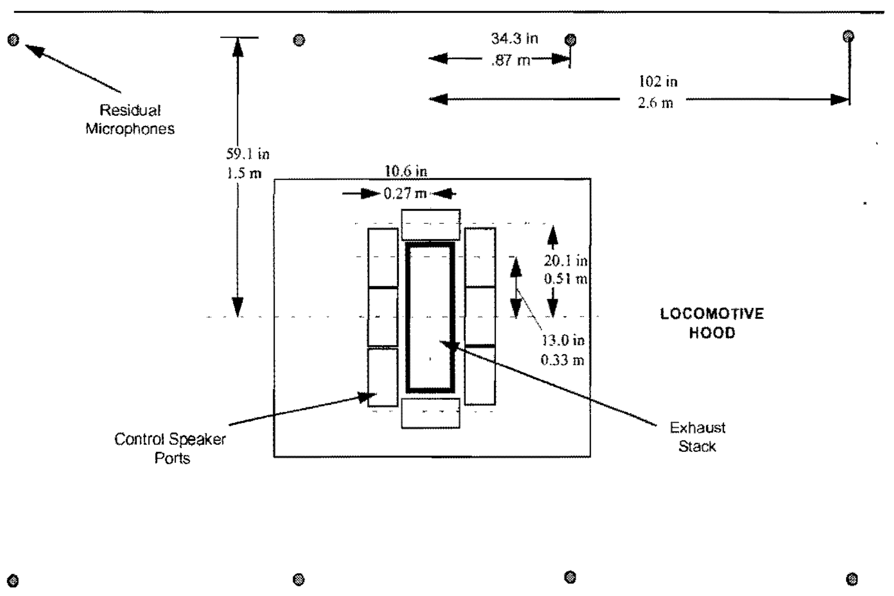


Figure 22. Plan View of the Locomotive Hood Showing the Refined System Layout

Figure 23 shows the improvement in performance when first the microphones are moved to the edge of the locomotive hood and, then, when the control speakers are moved closer to the exhaust stack. Moving the microphones to the edge of the locomotive hood gives about a 4 dB improvement in noise reduction below 250 Hz. Moving the control speaker ports closer to the exhaust stack gives additional 4 dB improvement and extends the effective range of the system to higher frequency.

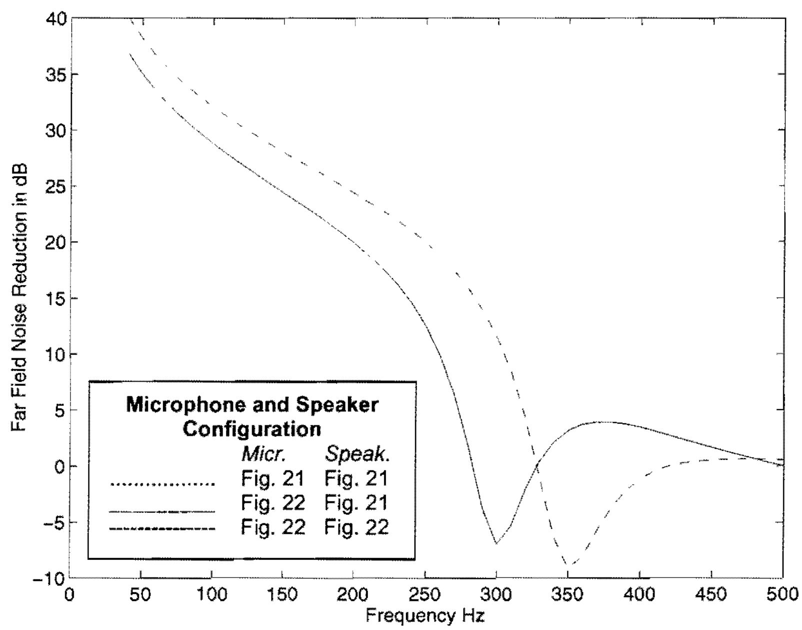


Figure 23. Reduction in the Mean Square, Angular-Averaged Sound Pressure at 30.4 Meters

The calculations so far have been for the sound pressure in the horizontal plane at 30.4 meters from the center of the exhaust stack. For completeness, the sound pressure was examined at various angles out of the horizontal plane and at different distances from the exhaust stack. Those results are shown in Figures 24 and 25, respectively. They show little effect except for large angles away from the horizontal plane.

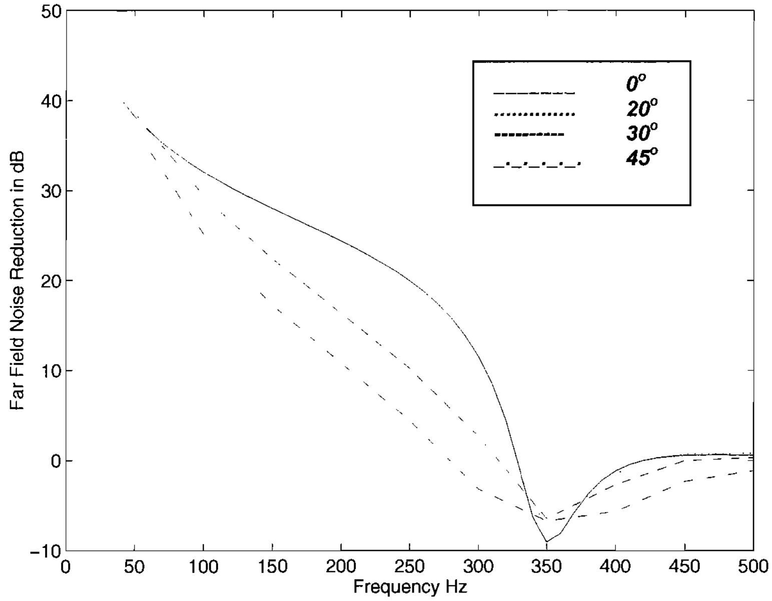


Figure 24. The Effect of Angle Out of the Horizontal Plane on the Reduction in the Mean Square, Angular-Averaged Sound Pressure at 30.4 Meters from the Exhaust

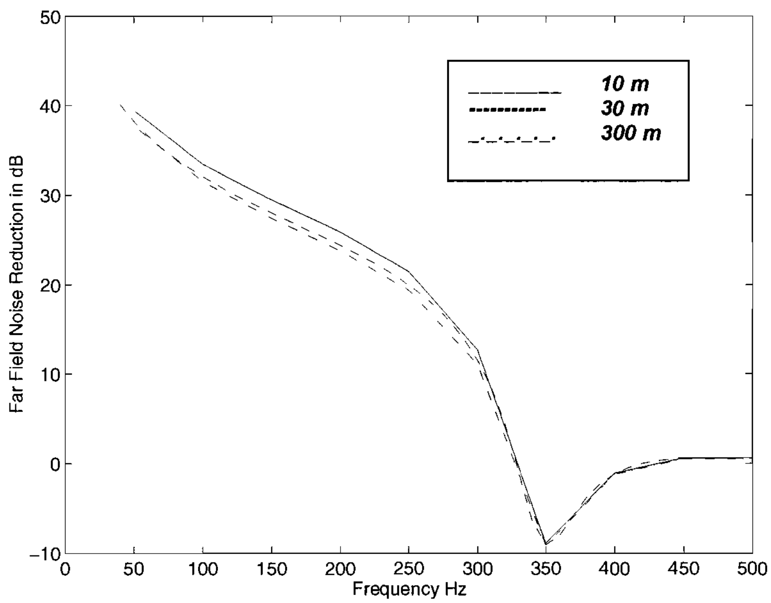


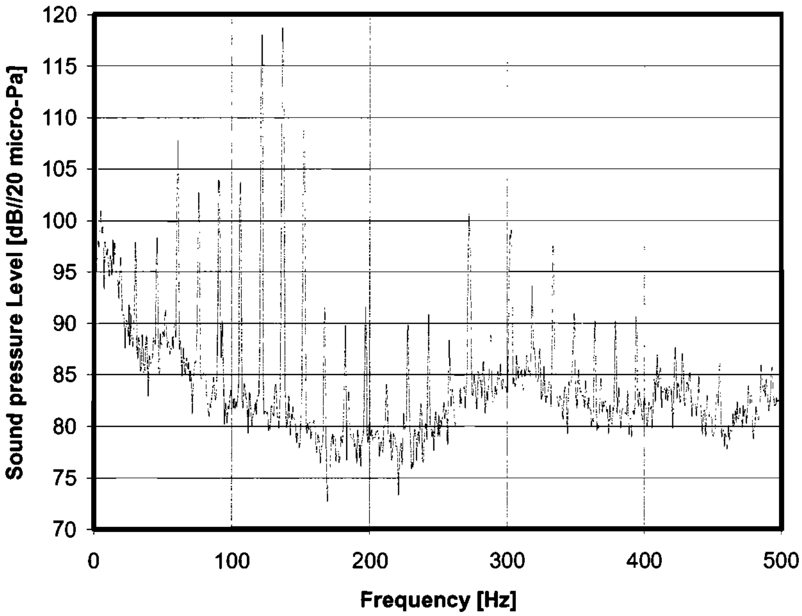
Figure 25. The Effect of Distance on the Reduction in the Mean Square, Angular-Averaged Sound Pressure at 30.4 Meters from the Exhaust

2.3 CONTROL SPEAKER, ENCLOSURE AND POWER AMPLIFIER SIZING

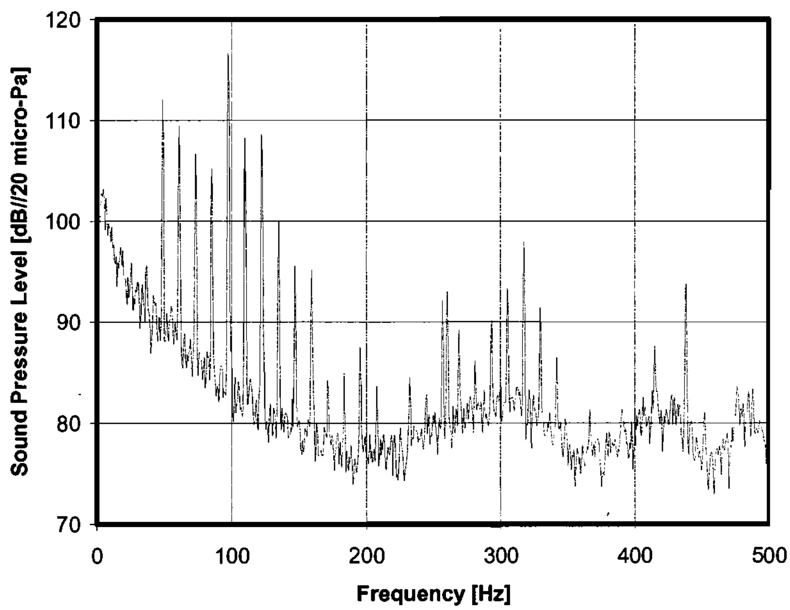
The previous section showed that over 20 dB of noise reduction below 250 Hz can be potentially obtained from the active noise control system. To achieve this noise reduction requires that the control speakers provide sufficient volume velocity to cancel the sound produced by the locomotive exhaust. To estimate the required volume velocity, measurements of the sound pressure obtained during tests carried out on an F40PH locomotive operated by the MBTA as part of their commuter rail service in Boston, Massachusetts, were used (For further details of the testing on the MBTA see Section 3.1.).

The narrow band spectra obtained at throttle 8, throttle 6 and throttle 4 are shown in Figure 26. The data were obtained from measurements made with a 1/2 inch Bruel and Kjaer condenser microphone placed 0.46 m above and 0.88 m aft of the exhaust stack. The locomotive was operated in self-load. The data were recorded on a TEAC RD 200T DAT recorder and analyzed using a Hewlett Packard two channel (800 line) FFT analyzer. Using the total sound pressure in each tonal peak to characterize the source strength of the exhaust, the volumetric velocity was estimated for each of the eight control speakers. Those volumetric velocities for each control speaker are shown in Table 2 versus the tonal frequencies at each throttle setting (4, 6 and 8) and idle. The numbering of the control speakers in the table is explained in Figure 27. Also shown in Table 2 are two measures of the overall volume velocity for each speaker. The coherent sum makes a worst case assumption that all tones are in phase. The incoherent sum assumes that the relative phases of the tones are random. The table shows that the overall volume velocities are comparable for all three-throttle settings. The volume velocities required at idle, however, are considerably lower. The largest overall volume velocity is demanded from control speakers 2 and 5 at throttle 4, a somewhat surprising result. The higher overall volume velocity required at throttle 4 is due to the large number of tones to be controlled.

Table 3 shows a number of performance requirements for control speakers 2 and 5, the two that are driven the hardest. The table shows rms volume velocity, rms volumetric displacement, rms speaker cone displacement, rms speaker cone acceleration and on-axis rms sound pressure level at 1 meter. The calculations are based on using a single 12-inch speaker (10.5 inches effective diameter) at each location. For the higher throttle settings the requirements on speaker cone acceleration seem to be the most stringent. Accelerations in excess of 200-g's rms are required for throttle 8 if the tones add coherently, for example. For throttle 4 and idle the displacement of the speaker cone makes the most demands on the control speaker. At throttle 4, speaker cone displacements of 12.8-mm (18-mm peak) are required if the tones add coherently.

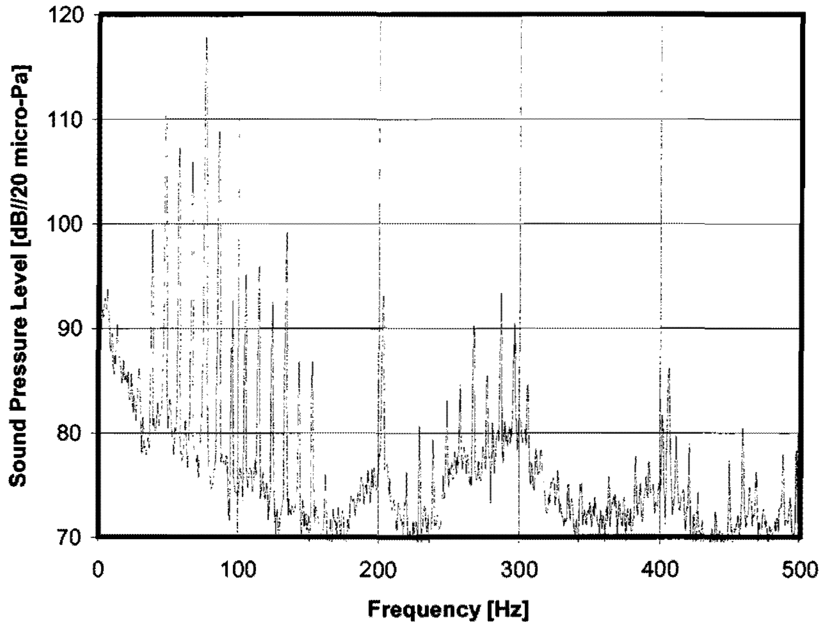


(a) Throttle 8

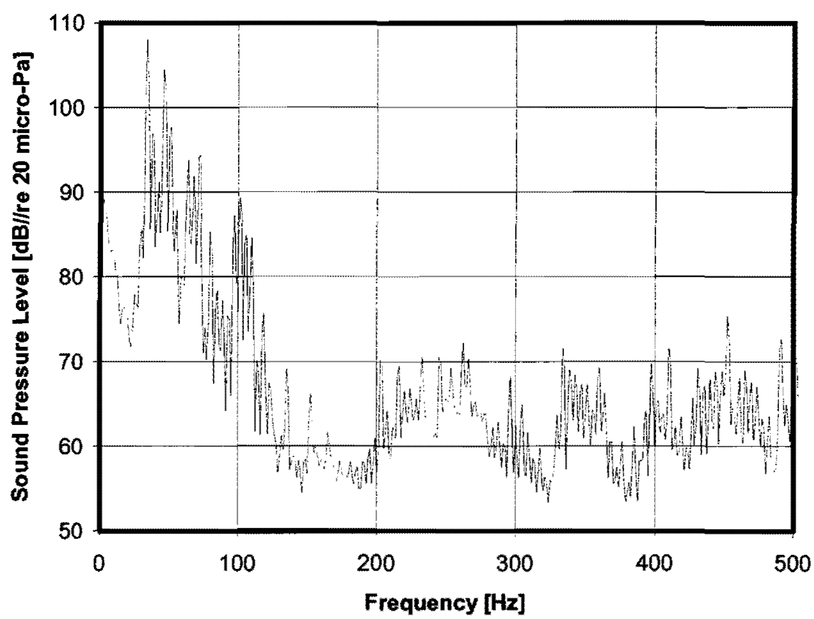


(b) Throttle 6

Figure 26. Narrow Band Sound Pressure Level Measured Close In to the Exhaust Stack of an F40 Locomotive in Self Load



(c) Throttle 4



(d) Idle

Figure 26. Narrow Band Sound Pressure Level Measured Close In to the Exhaust Stack of an F40 Locomotive (Continued)

Table 2. Volume Velocity Required from Control Speakers

(a) Throttle 8

Freq	rms VOLUME VELOCITY M^3/s							
[Hz]	1	2	3	4	5	6	7	8
60	1.22E-02	2.04E-02	1.22E-02	1.22E-02	2.04E-02	1.22E-02	3.66E-03	3.66E-03
75	5.57E-03	9.13E-03	5.57E-03	5.57E-03	9.13E-03	5.57E-03	1.46E-03	1.46E-03
90	5.32E-03	8.65E-03	5.32E-03	5.32E-03	8.65E-03	5.32E-03	1.23E-03	1.23E-03
106	5.22E-03	8.51E-03	5.22E-03	5.22E-03	8.51E-03	5.22E-03	1.10E-03	1.10E-03
121	1.68E-02	2.76E-02	1.68E-02	1.68E-02	2.76E-02	1.68E-02	3.38E-03	3.38E-03
136	2.19E-02	3.65E-02	2.19E-02	2.19E-02	3.65E-02	2.19E-02	4.40E-03	4.40E-03
151	1.16E-02	1.97E-02	1.16E-02	1.16E-02	1.97E-02	1.16E-02	2.42E-03	2.42E-03
167	8.74E-04	1.53E-03	8.74E-04	8.74E-04	1.53E-03	8.74E-04	1.98E-04	1.98E-04
182	6.00E-04	1.08E-03	6.00E-04	6.00E-04	1.08E-03	6.00E-04	1.50E-04	1.50E-04
196	8.34E-04	1.55E-03	8.34E-04	8.34E-04	1.55E-03	8.34E-04	2.33E-04	2.33E-04
211	2.95E-04	5.65E-04	2.95E-04	2.95E-04	5.65E-04	2.95E-04	9.39E-05	9.39E-05
228	7.52E-04	1.50E-03	7.52E-04	7.52E-04	1.50E-03	7.52E-04	2.80E-04	2.80E-04
242	7.75E-04	1.60E-03	7.75E-04	7.75E-04	1.60E-03	7.75E-04	3.31E-04	3.31E-04
257	6.48E-04	1.40E-03	6.48E-04	6.48E-04	1.40E-03	6.48E-04	3.21E-04	3.21E-04
272	3.13E-03	7.03E-03	3.13E-03	3.13E-03	7.03E-03	3.13E-03	1.81E-03	1.81E-03
301	4.11E-03	9.81E-03	4.11E-03	4.11E-03	9.81E-03	4.11E-03	3.23E-03	3.23E-03
coherent sum	9.05E-02	1.57E-01	9.05E-02	9.05E-02	1.57E-01	9.05E-02	2.43E-02	2.43E-02
incoherent sum	0.034016	0.057357	0.034016	0.034016	0.057357	0.034016	0.00831	0.00831

(b) Throttle 6

Freq	VOLUME VELOCITY M^3/s							
[Hz]	1	2	3	4	5	6	7	8
48	2.68E-02	4.63E-02	2.68E-02	2.68E-02	4.63E-02	2.68E-02	8.95E-03	8.95E-03
61	1.34E-02	2.25E-02	1.34E-02	1.34E-02	2.25E-02	1.34E-02	4.01E-03	4.01E-03
73	8.06E-03	1.32E-02	8.06E-03	8.06E-03	1.32E-02	8.06E-03	2.15E-03	2.15E-03
85	7.04E-03	1.15E-02	7.04E-03	7.04E-03	1.15E-02	7.04E-03	1.69E-03	1.69E-03
97	1.77E-02	2.88E-02	1.77E-02	1.77E-02	2.88E-02	1.77E-02	3.92E-03	3.92E-03
110	7.16E-03	1.17E-02	7.16E-03	7.16E-03	1.17E-02	7.16E-03	1.49E-03	1.49E-03
122	9.37E-03	1.54E-02	9.37E-03	9.37E-03	1.54E-02	9.37E-03	1.89E-03	1.89E-03
134	4.94E-03	8.24E-03	4.94E-03	4.94E-03	8.24E-03	4.94E-03	9.92E-04	9.92E-04
147	3.71E-03	6.29E-03	3.71E-03	3.71E-03	6.29E-03	3.71E-03	7.65E-04	7.65E-04
158	1.79E-03	3.08E-03	1.79E-03	1.79E-03	3.08E-03	1.79E-03	3.86E-04	3.86E-04
171	3.87E-04	6.82E-04	3.87E-04	3.87E-04	6.82E-04	3.87E-04	8.98E-05	8.98E-05
183	3.78E-04	6.82E-04	3.78E-04	3.78E-04	6.82E-04	3.78E-04	9.53E-05	9.53E-05
195	5.27E-04	9.73E-04	5.27E-04	5.27E-04	9.73E-04	5.27E-04	1.46E-04	1.46E-04
207	3.71E-04	7.05E-04	3.71E-04	3.71E-04	7.05E-04	3.71E-04	1.14E-04	1.14E-04
231	4.25E-04	8.56E-04	4.25E-04	4.25E-04	8.56E-04	4.25E-04	1.63E-04	1.63E-04
243	3.89E-04	8.09E-04	3.89E-04	3.89E-04	8.09E-04	3.89E-04	1.68E-04	1.68E-04
256	1.02E-03	2.20E-03	1.02E-03	1.02E-03	2.20E-03	1.02E-03	5.02E-04	5.02E-04
260	1.04E-03	2.26E-03	1.04E-03	1.04E-03	2.26E-03	1.04E-03	5.32E-04	5.32E-04
268	7.68E-04	1.71E-03	7.68E-04	7.68E-04	1.71E-03	7.68E-04	4.26E-04	4.26E-04
280	5.89E-04	1.35E-03	5.89E-04	5.89E-04	1.35E-03	5.89E-04	3.70E-04	3.70E-04
293	1.05E-03	2.48E-03	1.05E-03	1.05E-03	2.48E-03	1.05E-03	7.59E-04	7.59E-04
305	1.73E-03	4.14E-03	1.73E-03	1.73E-03	4.14E-03	1.73E-03	1.42E-03	1.42E-03
coherent sum	1.09E-01	1.86E-01	1.09E-01	1.09E-01	1.86E-01	1.09E-01	3.10E-02	3.10E-02
incoherent sum	0.038872	0.06556	0.038872	0.038872	0.06556	0.038872	0.011346	0.011346

Table 2. Volume Velocity Required from Control Speakers (Continued)

(c) Throttle 4

Freq	VOLUME VELOCITY M^3/s							
[Hz]	1	2	3	4	5	6	7	8
38	1.68E-02	3.00E-02	1.68E-02	1.68E-02	3.00E-02	1.68E-02	6.00E-03	6.00E-03
48	4.24E-02	7.34E-02	4.24E-02	4.24E-02	7.34E-02	4.24E-02	1.42E-02	1.42E-02
57	1.61E-02	2.71E-02	1.61E-02	1.61E-02	2.71E-02	1.61E-02	4.98E-03	4.98E-03
67	8.72E-03	1.44E-02	8.72E-03	8.72E-03	1.44E-02	8.72E-03	2.46E-03	2.46E-03
76	2.19E-02	3.59E-02	2.19E-02	2.19E-02	3.59E-02	2.19E-02	5.68E-03	5.68E-03
85	7.90E-03	1.29E-02	7.90E-03	7.90E-03	1.29E-02	7.90E-03	1.90E-03	1.90E-03
95	1.43E-03	2.33E-03	1.43E-03	1.43E-03	2.33E-03	1.43E-03	3.21E-04	3.21E-04
105	1.66E-03	2.71E-03	1.66E-03	1.66E-03	2.71E-03	1.66E-03	3.52E-04	3.52E-04
115	1.95E-03	3.20E-03	1.95E-03	1.95E-03	3.20E-03	1.95E-03	3.99E-04	3.99E-04
124	1.04E-03	1.71E-03	1.04E-03	1.04E-03	1.71E-03	1.04E-03	2.09E-04	2.09E-04
134	4.94E-03	8.24E-03	4.94E-03	4.94E-03	8.24E-03	4.94E-03	9.92E-04	9.92E-04
144	5.95E-04	1.00E-03	5.95E-04	5.95E-04	1.00E-03	5.95E-04	1.22E-04	1.22E-04
153	2.88E-04	4.93E-04	2.88E-04	2.88E-04	4.93E-04	2.88E-04	6.08E-05	6.08E-05
163	1.57E-04	2.73E-04	1.57E-04	1.57E-04	2.73E-04	1.57E-04	3.48E-05	3.48E-05
203	1.05E-03	1.97E-03	1.05E-03	1.05E-03	1.97E-03	1.05E-03	3.10E-04	3.10E-04
220	1.18E-04	2.31E-04	1.18E-04	1.18E-04	2.31E-04	1.18E-04	4.08E-05	4.08E-05
229	2.38E-04	4.77E-04	2.38E-04	2.38E-04	4.77E-04	2.38E-04	8.97E-05	8.97E-05
239	2.17E-04	4.45E-04	2.17E-04	2.17E-04	4.45E-04	2.17E-04	8.98E-05	8.98E-05
249	3.53E-04	7.46E-04	3.53E-04	3.53E-04	7.46E-04	3.53E-04	1.62E-04	1.62E-04
258	4.11E-04	8.88E-04	4.11E-04	4.11E-04	8.88E-04	4.11E-04	2.06E-04	2.06E-04
267	8.57E-04	1.90E-03	8.57E-04	8.57E-04	1.90E-03	8.57E-04	4.70E-04	4.70E-04
278	5.80E-04	1.32E-03	5.80E-04	5.80E-04	1.32E-03	5.80E-04	3.57E-04	3.57E-04
286	1.39E-03	3.22E-03	1.39E-03	1.39E-03	3.22E-03	1.39E-03	9.28E-04	9.28E-04
296	1.09E-03	2.58E-03	1.09E-03	1.09E-03	2.58E-03	1.09E-03	8.11E-04	8.11E-04
306	6.23E-04	1.49E-03	6.23E-04	6.23E-04	1.49E-03	6.23E-04	5.19E-04	5.19E-04
coherent sum	1.33E-01	2.29E-01	1.33E-01	1.33E-01	2.29E-01	1.33E-01	4.17E-02	4.17E-02
incoherent sum	0.05479	0.093841	0.05479	0.05479	0.093841	0.05479	0.017549	0.017549

(d) Idle

Freq	rms VOLUME VELOCITY M^3/s							
[Hz]	1	2	3	4	5	6	7	8
34	3.75E-02	6.77E-02	3.75E-02	3.75E-02	6.77E-02	3.75E-02	1.36E-02	1.36E-02
38	7.53E-03	1.34E-02	7.53E-03	7.53E-03	1.34E-02	7.53E-03	2.68E-03	2.68E-03
43	4.21E-03	7.39E-03	4.21E-03	4.21E-03	7.39E-03	4.21E-03	1.46E-03	1.46E-03
46	1.25E-02	2.17E-02	1.25E-02	1.25E-02	2.17E-02	1.25E-02	4.23E-03	4.23E-03
51	4.00E-03	6.86E-03	4.00E-03	4.00E-03	6.86E-03	4.00E-03	1.31E-03	1.31E-03
55	1.32E-03	2.24E-03	1.32E-03	1.32E-03	2.24E-03	1.32E-03	4.17E-04	4.17E-04
64	2.29E-03	3.80E-03	2.29E-03	2.29E-03	3.80E-03	2.29E-03	6.63E-04	6.63E-04
73	2.27E-03	3.73E-03	2.27E-03	2.27E-03	3.73E-03	2.27E-03	6.06E-04	6.06E-04
80	6.62E-04	1.08E-03	6.62E-04	6.62E-04	1.08E-03	6.62E-04	1.66E-04	1.66E-04
84	3.18E-04	5.17E-04	3.18E-04	3.18E-04	5.17E-04	3.18E-04	7.70E-05	7.70E-05
88	1.93E-04	3.13E-04	1.93E-04	1.93E-04	3.13E-04	1.93E-04	4.53E-05	4.53E-05
92	3.70E-04	6.01E-04	3.70E-04	3.70E-04	6.01E-04	3.70E-04	8.45E-05	8.45E-05
98	7.00E-04	1.14E-03	7.00E-04	7.00E-04	1.14E-03	7.00E-04	1.54E-04	1.54E-04
102	8.53E-04	1.39E-03	8.53E-04	8.53E-04	1.39E-03	8.53E-04	1.83E-04	1.83E-04
106	5.86E-04	9.55E-04	5.86E-04	5.86E-04	9.55E-04	5.86E-04	1.24E-04	1.24E-04
110	4.03E-04	6.58E-04	4.03E-04	4.03E-04	6.58E-04	4.03E-04	8.36E-05	8.36E-05
coherent sum	7.57E-02	1.33E-01	7.57E-02	7.57E-02	1.33E-01	7.57E-02	2.59E-02	2.59E-02
incoherent sum	0.04082	0.073287	0.04082	0.04082	0.073287	0.04082	0.014638	0.014638

Table 3. RMS Requirements for a 12-Inch Control Speaker at Locations 2 and 5

(a)

THROTTLE 8					
	Volume	Volume	Cone	Cone	SPL
Freq	Velocity	Displ.	Displ.	Accel.	at 1 m
[Hz]	m^3/s	m^3	mm	g's	dB
60	2.04E-02	5.41E-05	0.97	14.07	97.41
75	9.13E-03	1.94E-05	0.35	7.87	92.36
90	8.65E-03	1.53E-05	0.27	8.95	93.48
106	8.51E-03	1.28E-05	0.23	10.36	94.75
121	2.76E-02	3.63E-05	0.65	38.36	106.12
136	3.65E-02	4.28E-05	0.77	57.13	109.58
151	1.97E-02	2.08E-05	0.37	34.20	105.12
167	1.53E-03	1.46E-06	0.03	2.94	83.80
182	1.08E-03	9.45E-07	0.02	2.26	81.52
196	1.55E-03	1.26E-06	0.02	3.48	85.27
211	5.65E-04	4.27E-07	0.01	1.37	77.18
228	1.50E-03	1.05E-06	0.02	3.94	86.35
242	1.60E-03	1.06E-06	0.02	4.46	87.43
257	1.40E-03	8.66E-07	0.02	4.13	86.75
272	7.03E-03	4.11E-06	0.07	21.97	101.28
301	9.81E-03	5.19E-06	0.09	33.92	105.05
coherent					
sum	1.57E-01	2.18E-04	3.91	249	122.38
incoherent					
sum	0.057357	8.57E-05	1.54	90	113.51

(b)

THROTTLE 6					
	Volume	Volume	Cone	Cone	SPL
Freq	Velocity	Displ.	Displ.	Accel.	at 1 m
[Hz]	m^3/s	m^3	mm	g's	dB
48	4.63E-02	0.000154	2.75	25.54	96.56
61	2.25E-02	7.46E-05	1.34	15.77	90.29
73	1.32E-02	3.46E-05	0.62	11.11	87.77
85	1.15E-02	2.5E-05	0.45	11.19	88.08
97	2.88E-02	5.4E-05	0.97	32.14	97.41
110	1.17E-02	1.92E-05	0.34	14.79	90.73
122	1.54E-02	2.23E-05	0.40	21.64	94.23
134	8.24E-03	1.08E-05	0.19	12.69	89.68
147	6.29E-03	7.47E-06	0.13	10.63	88.14
158	3.08E-03	3.34E-06	0.06	5.60	82.76
171	6.82E-04	6.87E-07	0.01	1.34	70.27
183	6.82E-04	6.35E-07	0.01	1.43	70.96
195	9.73E-04	8.47E-07	0.02	2.18	74.64
207	7.05E-04	5.75E-07	0.01	1.68	72.39
231	8.56E-04	6.58E-07	0.01	2.27	74.60
243	8.09E-04	5.57E-07	0.01	2.26	75.06
256	2.20E-03	1.44E-06	0.03	6.47	84.19
260	2.26E-03	1.41E-06	0.03	6.76	84.89
268	1.71E-03	1.04E-06	0.02	5.25	82.57
280	1.35E-03	8.02E-07	0.01	4.34	80.79
293	2.48E-03	1.41E-06	0.03	8.35	86.46
305	4.14E-03	2.25E-06	0.04	14.53	91.32
coherent					
sum	1.86E-01	0.000417	7.48	217.97	114.37
incoherent					
sum	0.06556	0.000187	3.35	60.10	103.17

Table 3. RMS Requirements for a 12-Inch Control Speaker at Locations 2 and 5 (Continued)

(c)

THROTTLE 4					
	Volume	Volume	Cone	Cone	SPL
Freq	Velocity	Displ.	Displ.	Accel.	at 1 m
[Hz]	m^3/s	m^3	mm	g's	dB
38	0.03004	0.000126	2.26	13.12	90.78
48	0.07336	0.000307	5.51	40.47	98.53
57	0.027138	9E-05	1.61	17.78	91.93
67	0.014434	4.03E-05	0.72	11.12	87.93
76	0.035866	8.52E-05	1.53	31.33	97.24
85	0.012857	2.69E-05	0.48	12.56	89.43
95	0.002329	4.36E-06	0.08	2.54	75.56
105	0.002709	4.54E-06	0.08	3.27	77.84
115	0.003197	4.85E-06	0.09	4.23	80.15
124	0.001715	2.37E-06	0.04	2.44	75.53
134	0.00824	1.06E-05	0.19	12.69	89.82
144	0.001005	1.19E-06	0.02	1.66	72.21
153	0.000493	5.45E-07	0.01	0.87	66.65
163	0.000273	2.84E-07	0.01	0.51	62.05
203	0.001969	1.92E-06	0.03	4.59	79.76
220	0.000231	1.81E-07	0.00	0.58	63.06
229	0.000477	3.45E-07	0.01	1.26	70.05
239	0.000445	3.1E-07	0.01	1.22	69.80
249	0.000746	4.97E-07	0.01	2.13	74.65
258	0.000888	5.68E-07	0.01	2.63	76.52
267	0.001898	1.17E-06	0.02	5.82	83.43
278	0.001322	7.89E-07	0.01	4.22	80.59
286	0.003221	1.84E-06	0.03	10.59	88.67
296	0.002578	1.44E-06	0.03	8.77	86.98
306	0.001494	8.03E-07	0.01	5.25	82.54
coherent sum	0.228925	0.000714	12.81	201.68	113.48
incoherent sum	0.093841	0.000358	6.42	62.45	102.98

(d)

IDLE					
	Volume	Volume	Cone	Cone	SPL
Freq	Velocity	Displ.	Displ.	Accel.	at 1 m
[Hz]	m^3/s	m^3	mm	g's	dB
34	0.067673	0.000317	5.68	26.45	102.89
38	0.013418	5.62E-05	1.01	5.86	89.80
43	0.007386	2.74E-05	0.49	3.65	85.69
46	0.021674	7.5E-05	1.35	11.46	95.62
51	0.006861	2.14E-05	0.38	4.02	86.53
55	0.002241	6.49E-06	0.12	1.42	77.47
64	0.003802	9.46E-06	0.17	2.80	83.37
73	0.003731	8.14E-06	0.15	3.13	84.35
80	0.00108	2.15E-06	0.04	0.99	74.38
84	0.000517	9.81E-07	0.02	0.50	68.41
88	0.000313	5.67E-07	0.01	0.32	64.45
92	0.000601	1.04E-06	0.02	0.64	70.50
98	0.001138	1.85E-06	0.03	1.28	76.60
102	0.001388	2.17E-06	0.04	1.63	78.67
106	0.000955	1.43E-06	0.03	1.16	75.75
110	0.000658	9.52E-07	0.02	0.83	72.84
coherent sum	0.133437	0.000532	9.54	66.13	110.85
incoherent sum	0.073287	0.000333	5.96	30.37	104.09

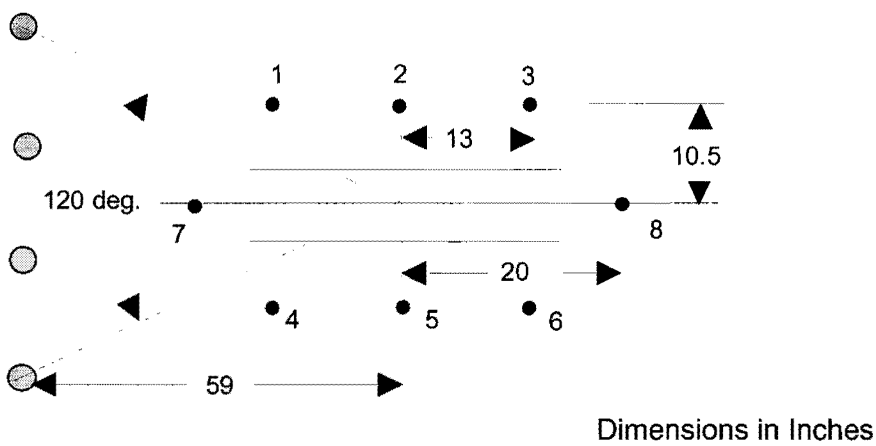


Figure 27. Numbering of the Control Speakers

To provide some margin of safety in the design of the control speakers, two speakers were placed in ported enclosures. Two speakers will reduce the speaker cone displacement requirements nearly in half and the ported enclosures will allow time to tune the speaker output in frequency to place the maximum volume velocity where it is needed. This is especially useful in enhancing the low frequency volume velocity. McCauley model 6232 12-inch high fidelity speakers were chosen as the actuators. These have a maximum peak speaker cone displacement of 7.75 mm. This is less than half of the required 18-mm peak amplitude at throttles 4 at locations 2 and 5. Consequently, at those two locations, two speaker enclosures (four drivers) were used, which should reduce the required maximum peak speaker cone displacement to approximately 4.5 mm. This provides some margin, since achieving a doubling of the volume velocity by using two speakers in each enclosure may not be possible. At the other locations the requirements on volume velocity and speaker cone displacement are typically half or less than those required at locations 2 and 5. Consequently, a single enclosure will be adequate at the other locations. This will mean a total of 10 speaker enclosures instead of only 8 with the two enclosures at location 2 being driven by the same controller channel and the two enclosures at location 5 also driven by the same controller channel. In other words there will be 10 control sources but only eight independent controller outputs. Thus, the previous analysis still applies.

Figure 28 shows the control speaker volume velocity output versus frequency estimated using the LEAP program, a commercially available computer code for predicting the performance of high fidelity speakers in enclosures. The calculations were performed for two McCauley 12-inch speakers mounted in a ported enclosure having the geometry of the enclosures that were ultimately built for the program. Detailed dimensions of those enclosures can be found in the document, "APECS Drawing Package"[2]. Also shown in the figure, are the required volume velocities at idle and throttles 4, 6 and 8 versus frequency from Table 3 for locations 2 and 5, the locations requiring the highest volume velocities. The figure shows that the design provides a volume velocity that approximately follows the required volume velocity versus frequency and emphasizes the volume velocity from 40 to 100 Hz.

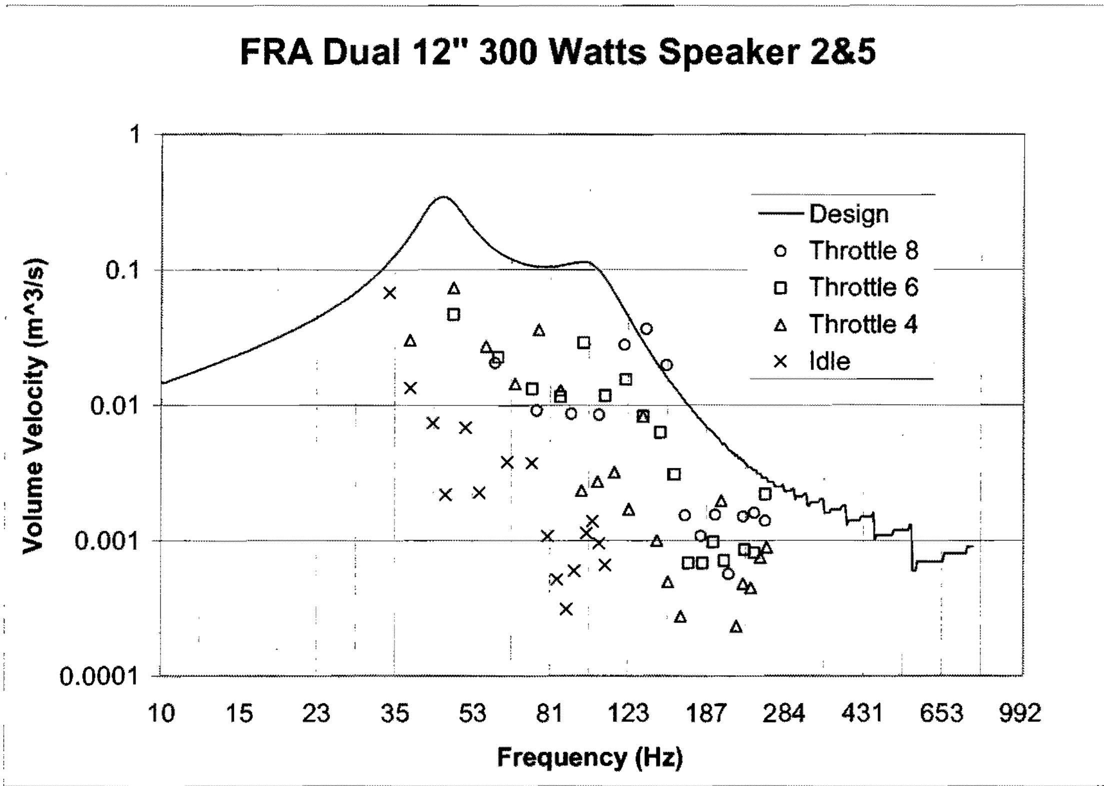


Figure 28. Predicted Speaker Volume Velocity Versus Frequency for Two McCauley 12-Inch Speakers in a Ported Enclosure Each Driven at 300 Watts

2.4 PASSIVE MUFFLER DESIGN

2.4.1 RATIONALE FOR SILENCER GEOMETRY

In Section 2.1, it was determined that the passive silencer should provide an IL that increases from zero at 250 Hz, achieves 5 dB between 250 and 500 Hz and reaches approximately 15 dB at 1000 Hz and above. With the limited space available in the locomotive, these performance requirements present a real challenge. In addition, the turbo-charged diesel engine has very stringent backpressure requirements. To meet the requirement for very low backpressure, a straight through resistive silencer was designed. While somewhat better performance might be obtained from a reactive silencer, the backpressure from such a design might be excessive.

The resulting muffler design is sketched in Figure 29. The design consists of an exhaust duct, both surfaces of which communicate with cavities covered with flow resistive material. In the middle of the duct is a center-body with cavities on both surfaces each covered with flow resistive material. The center body also contains a tuned labyrinth. Both the large cavities in the walls of the duct and the small cavities in the center body have baffles perpendicular to the duct axis. The baffles are designed to prevent the propagation of sound energy in the cavities in the direction parallel to the duct axis. Sound propagating in this way would compromise the IL of the muffler.

The deep cavities (6.5 inches and 2.56 inches) in the outer walls of the exhaust duct are designed to provide low frequency noise suppression. These cavities would ideally have been the same depth but space constraints prevented making them so. The shallower cavities in the center body (approximately 1.5 inches) control high frequency noise. The tune labyrinth is designed to control a 250 Hz tone at throttle 8 that pre-prototype testing showed the active system could not control. It is covered with 25 percent open perforated metal to provide some broadening in frequency of the IL peak at the cavity resonance.

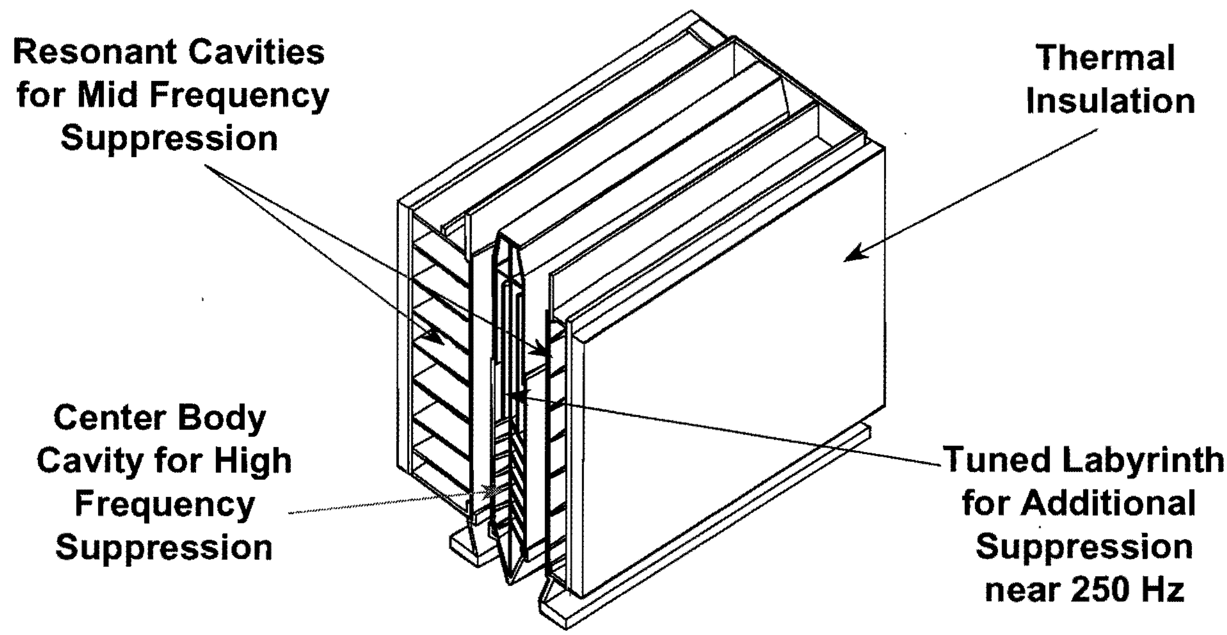


Figure 29. Passive Silencer Design, a Straight-Through Resistive Silencer

2.4.2 Analytical Evaluation of the Silencer Design

The insertion loss of the silencer was estimated using SARA 2D, a vendor proprietary, general-purpose two dimensional finite element code for structural acoustic analysis. The air in the exhaust duct and in the muffler cavities and labyrinth was modeled using rectangular quadratic acoustic elements. The coupling of those elements to the air outside the exhaust outlet was modeled using infinite acoustic elements. The inlet to the silencer was treated as having the impedance of an infinitely long duct, ρc, where ρ is the density of air and c is the acoustic wave speed. The walls of the silencer and baffles in the cavities were all treated as rigid.

The flow resistances of the material used to cover the cavities and the tuned labyrinth were estimated using analytical procedures developed by the National Aeronautics and Space Administration (NASA) for jet engine nacelle liners [3]. The materials considered for providing the flow resistance were perforated metal sheets, multi-layers of fine screening and sintered metal sheet[3]. The procedures include nonlinear flow resistance effects, the effects of mean flow and pressure and the mass reactance of the air in the holes in the sheeting or screening. Figures 30 and 31 show the comparison of the predicted and measured impedance of various flow resistive sheeting materials over shallow cavities [4][5]. The figures show two different mean flow and sound pressure level conditions. The theoretical calculations agree quite well with the measured data. This adds confidence in utilizing these analytical estimation techniques for predicting the liner flow resistance needed in the Finite Element Analysis (FEA) calculations.

The flow resistance predictive tools discussed above were used to estimate the flow resistance for two different options for flow resistive materials for the exhaust silencer: perforated metal and multi-layers of screening. Note that sintered metal sheet was not considered because of the very high cost of that type of material. Figure 32 shows the real and imaginary part of the impedance of a layer of 5 percent open perforated metal with 1/16 inch-diameter holes, and Figure 33 shows the same information for five layers of 60 wires/inch screening with 0.011-inch diameter wire. The difference is striking. The real part of the impedance of the perforated metal drops off at high frequency and the imaginary part increases rapidly. The screening on the other hand maintains an impedance whose real part is nearly ρc for air through out the frequency range of interest while maintaining a small imaginary part. To obtain high insertion loss to high frequency the flow resistive liner must have an impedance whose real part remains near ρc throughout the frequency range and whose imaginary part does not get too large. If the imaginary part gets too large, the inertia of the air at high frequency prevents it passing through the holes in the liner. If the air cannot pass through the holes in the liner, the liner cannot dissipate energy from sound waves propagating down the duct. On the other hand, if the real part of the liner impedance gets too small, little energy will be dissipated in the liner even if the air can pass through. The best way to avoid these problems is to go to holes much smaller in diameter than the thickness of the sheet. Such geometry reduces the non-linear part of the flow resistance and reduces the inertial effect of the air in the holes on the impedance. Based on these criteria the multi-layer screening is clearly the option of choice, although the possibility of plugging the small holes in the screening due to soot in the exhaust needs to be addressed during in-service testing.

[3] Feltmetal is the product name for a thin flow resistive sintered metal sheeting in which stainless steel material is sintered on a stainless steel screen to provide reproducible flow resistance through a matrix of very fine voids.

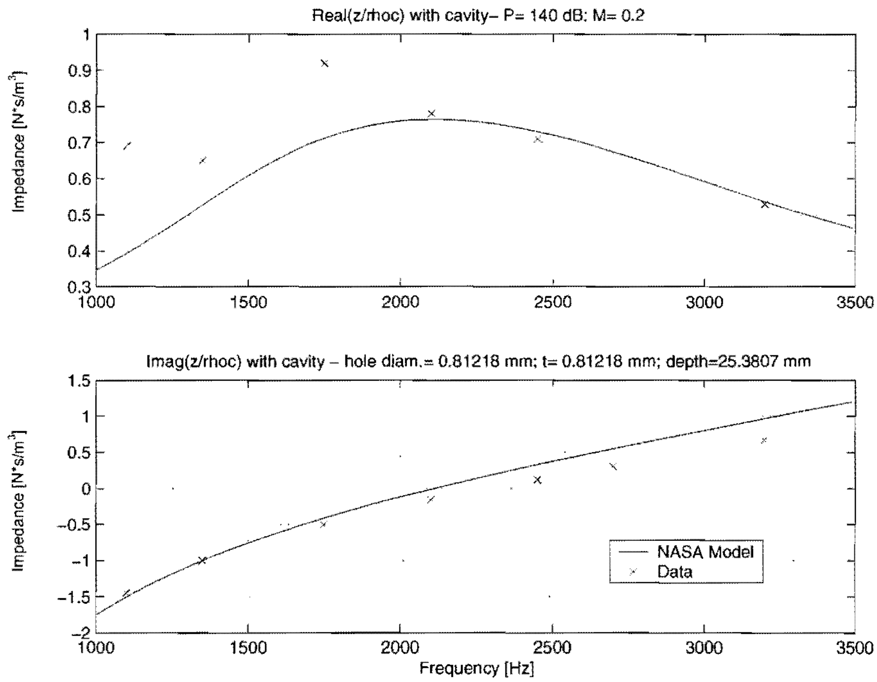


Figure 30. Comparison of Liner Impedance [N•s/m^3] Predictions and Measurements (6.7 Percent Open Area; 0.032-Inch Dia. Holes; 0.032-Inch Thick Sheet Over 1 Inch Cavity; Mach No. = 0.2; SPL 140 dB)

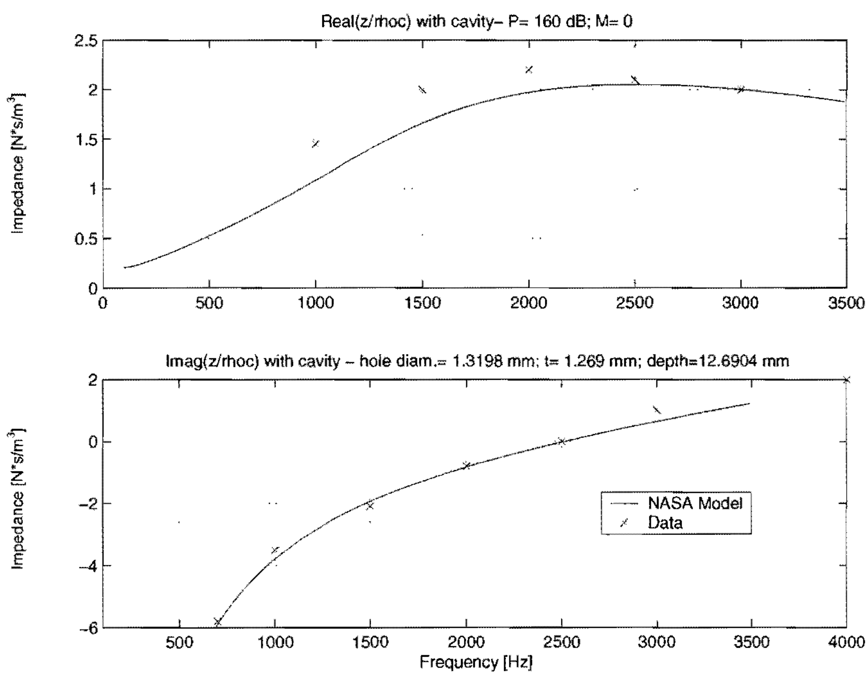


Figure 31. Comparison of Liner Impedance [N•s/m^3] Predictions and Measurements (5.2 Percent Open Area; 0.052-Inch Dia. Holes; 0.05-Inch Thick Sheet Over ½ Inch Cavity; Mach No. = 0; SPL 160 dB)

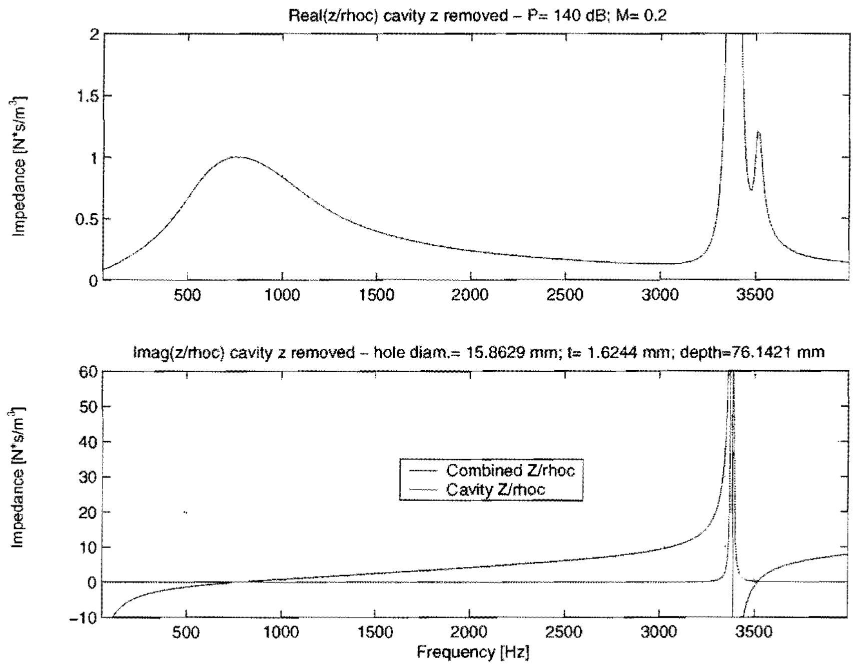


Figure 32. Predicted Impedance [N•s/m3] of Perforated Metal Sheeting (5% Open Area; 1/16-Inch Dia. Holes; 0.064 Inches Thick; Mach No. = 0.2; SPL= 140 dB)

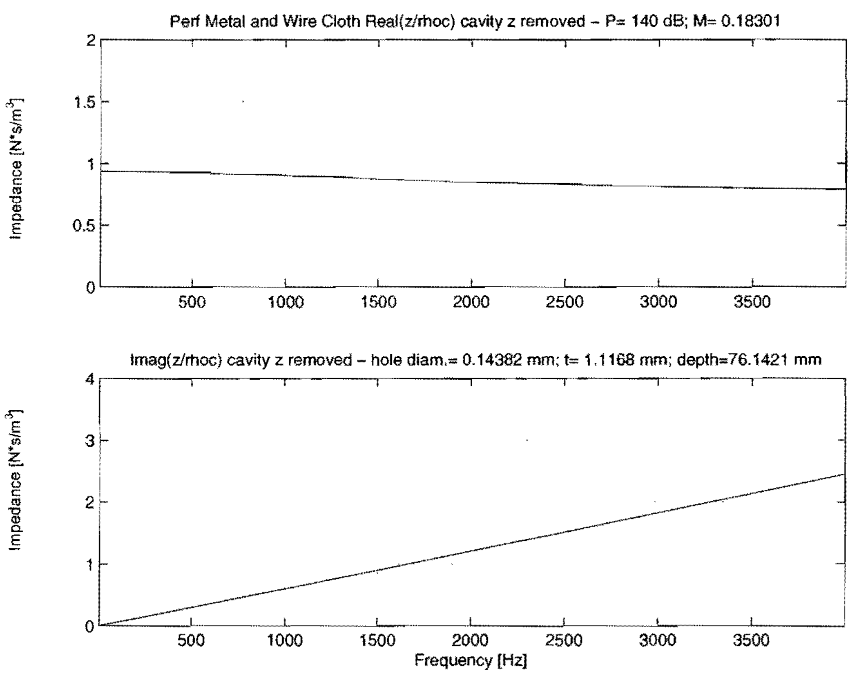


Figure 33. Predicted Impedance [N•s/m^3] of Five Layers of Screening on Perforated Metal Sheet Screening (60 Wires/Inch; 0.011 Inch Dia. Wire; 5 Layers Perforated Sheet: 20% Open Area; 0.1-Inch Dia. Holes; 0.1 Inches Thick; Mach No. = 0.2; SPL= 140 dB)

In order to improve confidence in the ability of the FEA model to predict the muffler IL, the predictions were compared with wind tunnel measurements of the IL of an early version of a duct silencer made from ¾-inch plywood. The dimensions and geometry of the silencer as installed in the wind tunnel are shown in Figure 34. The test setup and instrumentation are shown in Figure 35 and the silencer as tested is pictured in Figure 36. Although the particular geometry of the silencer in the figure is of no interest here, the test results did allow comparison of the predictions of IL with measurements for a number of test conditions and liner materials.

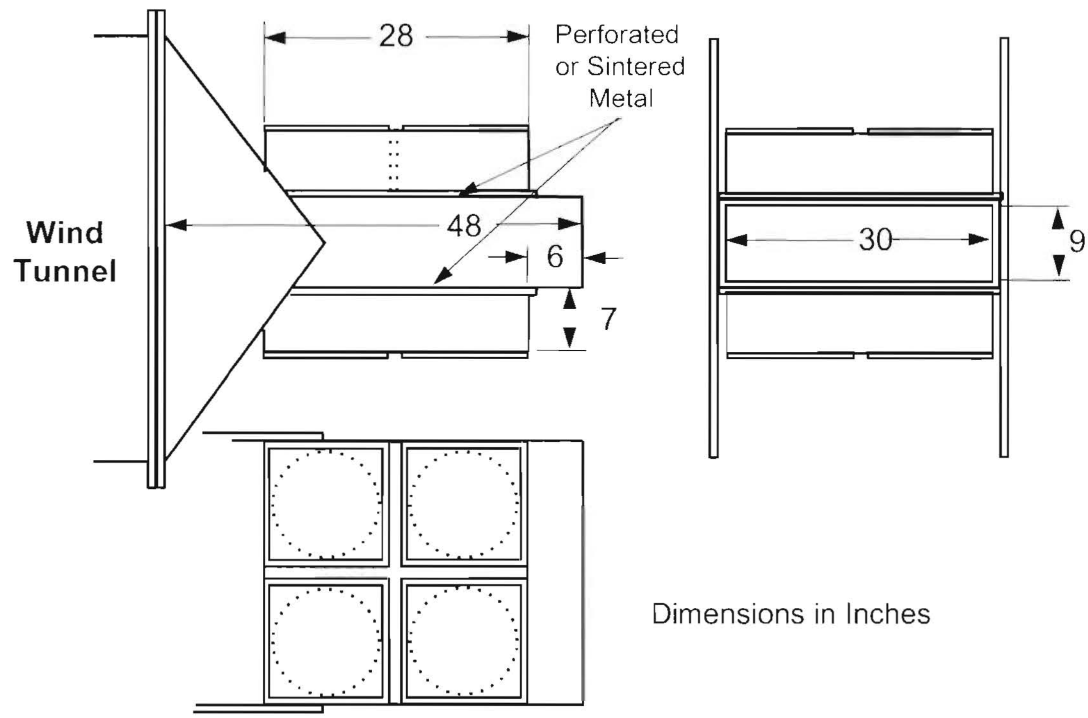


Figure 34. Setup for the Wind Tunnel Test of Insertion Loss for the Active Liner Concept

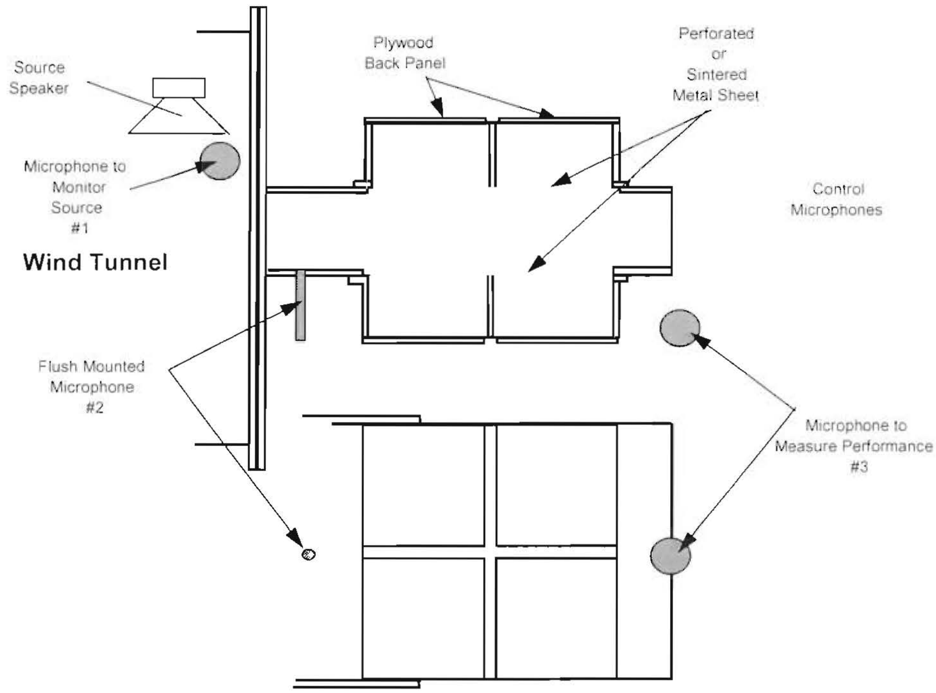


Figure 35. Test Setup and Instrumentation for the IL Test of the Early Silencer Design

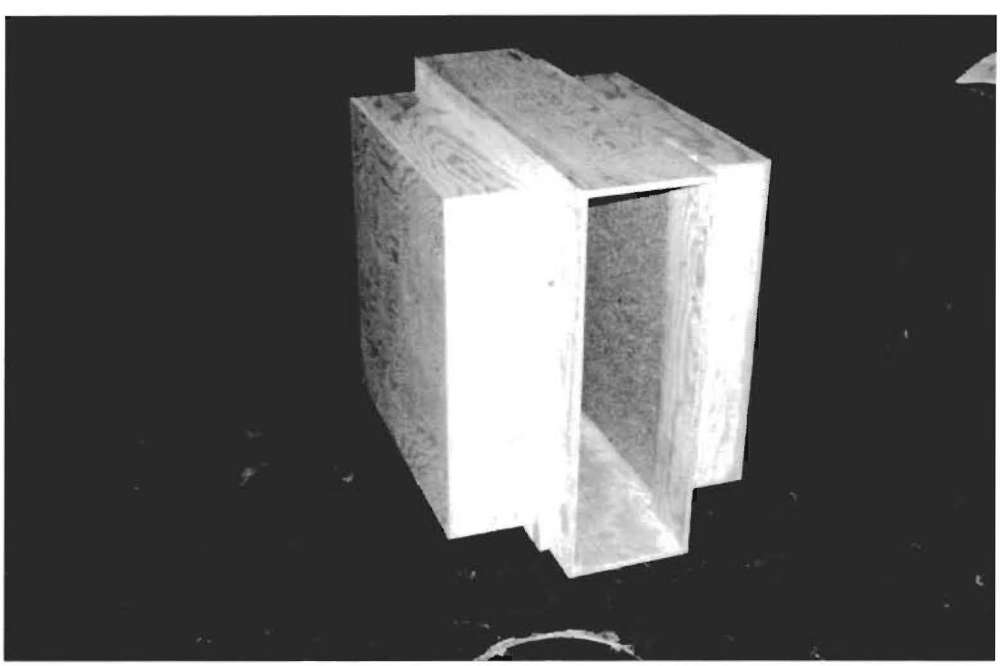


Figure 36. The Silencer Prepared for Testing with Sintered Metal Liner

Testing was carried out with two different liners and with plywood sheeting substituted for the flow resistive liners. Flow through the simulated exhaust duct was provided by the wind tunnel. The sound pressure level (SPL) was measured at the outlet, microphone 3 in Figure 35, for a number of different flow conditions. The IL of the silencer was calculated by taking the difference between the SPL at the outlet with a plywood liner and the SPL at the outlet with the liner of interest. The FEA code was then used to estimate the same quantities. The comparison of predictions and measurements is shown in Figure 37 and 38. Figure 37 shows the comparison for 2 percent open perforated metal with 0.125-inch diameter holes with a flow velocity of Mach 0.1. Figure 38 is for the no flow case with a sintered metal liner. The predictions and measurements agree reasonably well providing some confidence in the validity of using this predictive approach to help in the design process. Some of the differences between the predictions and measurements are due to the finer resolution in frequency of the measurements compared to the predictions.

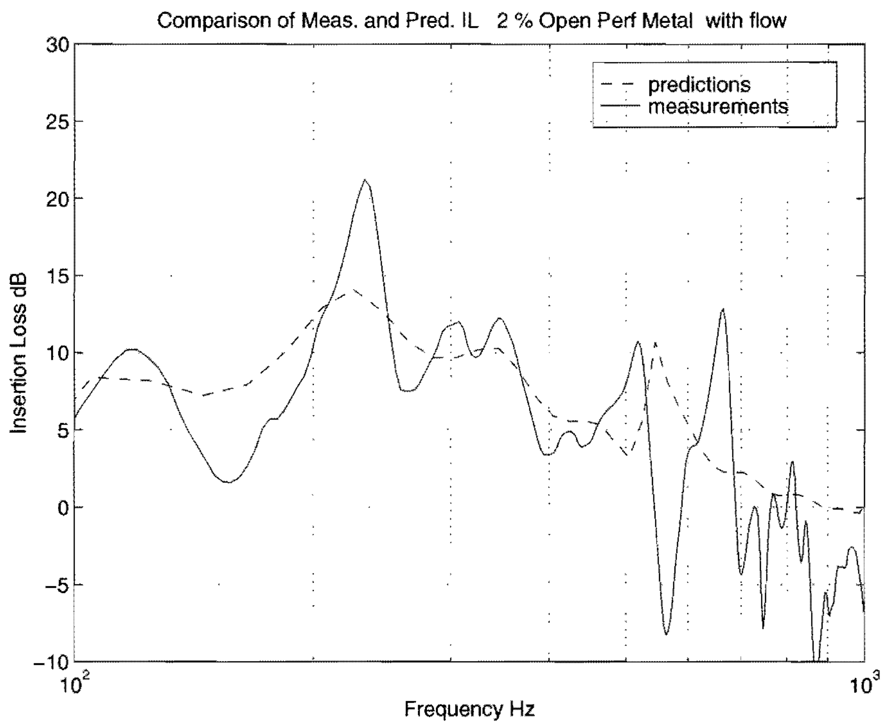


Figure 37. Comparison of FEM Predictions with Wind Tunnel Measurements; 2 Percent Perforated Metal Liner 3/32 Inch Thick with 0.125 Inch Dia. Holes with Flow (Mach No. =0.1)

Next, using the FEA modeling approach described above, the insertion loss of the prototype silencer in Figure 29 is predicted. In the calculation, the flow resistance calculation for the five layers of stainless steel screening will be used (see Figure 33). The flow conditions will be based on test results on an F40PH and an F59PHI locomotive. Those tests will be described in more detail in Section 3.

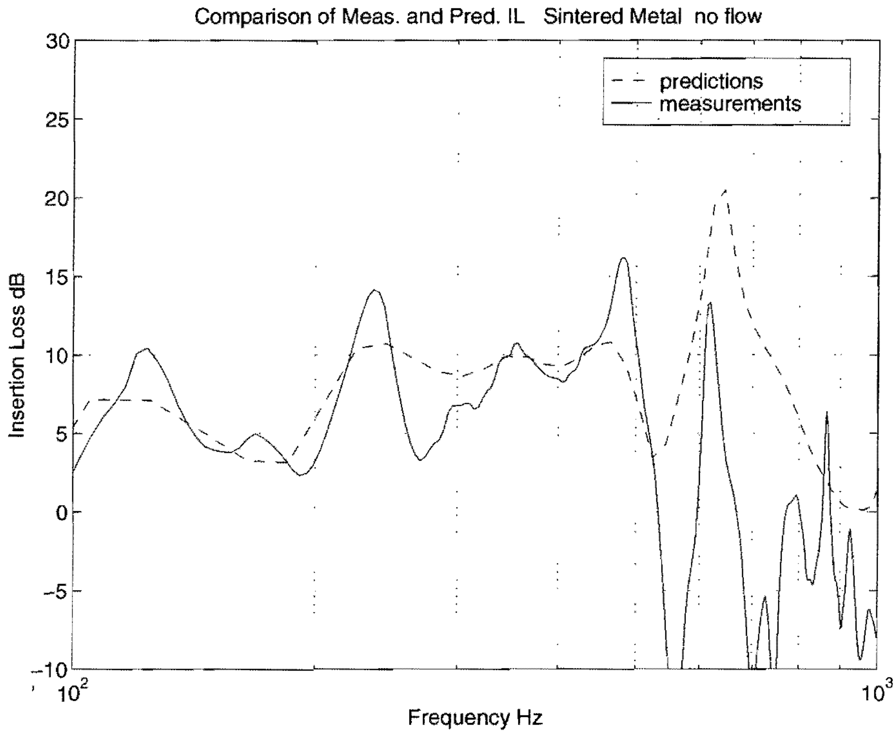


Figure 38. Comparison of FEM Predictions with Wind Tunnel Measurements Sintered Metal Liner without Flow (Flow Resistance = 100 N s/m^3)

The IL predictions are given in Figure 39 along with the assumed flow conditions. The figure shows that the low frequency requirements of 5 dB IL from 250 to 500 Hz is satisfied and the high frequency requirement of 15 dB above 500 Hz is also satisfied except for a few narrow frequency bands up to 3 kHz. Up to 4 kHz the IL lies for the most part between 10 and 15 dB. Finally at 250 Hz a slight peak was seen in the IL due to the tuned labyrinth. All in all the design comes very close to satisfying the requirements for the passive silencer.

2.5 CONTROL MICROPHONES

As discussed earlier the system will require eight control microphones, mounted on the roof of the locomotive. Since the microphones will be exposed to wind, weather and potential physical damage, it was necessary to select a robust microphone design that includes suitable protection. Past experience has shown that electret microphones are very durable, inexpensive, very resistant to adverse weather conditions and will continue to function even when wet.

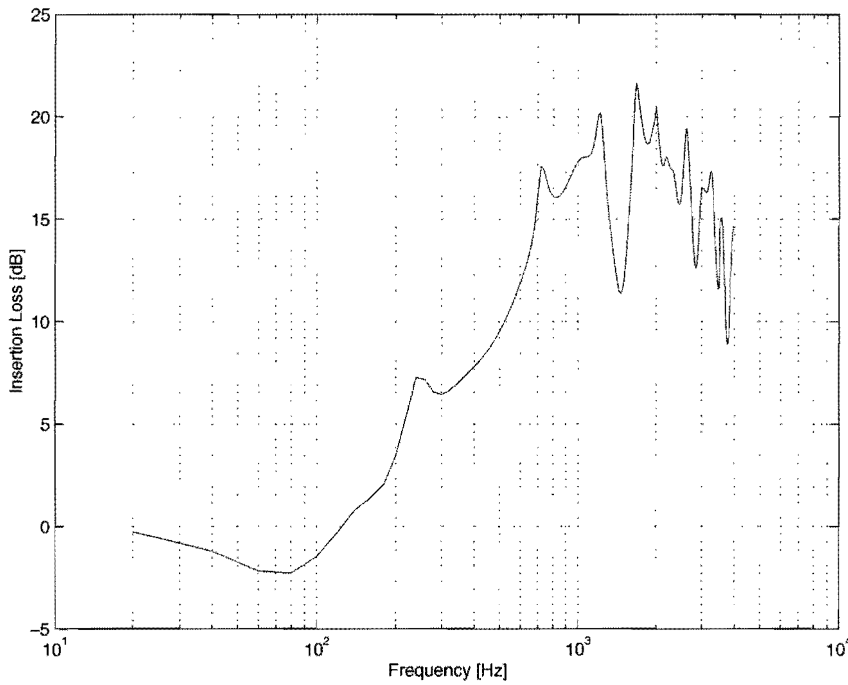


Figure 39. Predicted Insertion Loss of the Prototype Silencer (Flow Velocity 60 m/s; SPL 145 dB)

Consequently, a prototype was designed, based on an electret microphone. A package was developed with a preamplifier and microphone in a small sealed aluminum enclosure with a hole to expose the face of the microphone. The enclosure was then encased in foam and enclosed in a protective aluminum housing with numerous louvers. The package was assembled and tested by mounting it on the roof of a car. The car was then driven at a variety of speeds and the test microphone output, the output of a monitor microphone and an accelerometer were all recorded. The purpose of the test was to determine if the background noise from the microphone, exposed to wind and vibration, was sufficiently low to allow the measurement of locomotive noise on the roof of the locomotive. The microphone in its protective housing, mounted on the roof of the automobile, is shown in Figure 40. The data were analyzed using an HP 3562 Spectrum Analyzer. The analysis was carried out over the frequency range of 20 Hz to 420 Hz, resulting in 0.5 Hz nominal analysis bandwidth. The data are plotted versus frequency for four speeds in Figure 41. If the data in Figure 41 is compared with the data in Figure 26, it is clear that the microphone/housing design is a good one for measuring exhaust tones on the locomotive roof at speeds up to 80 mph for throttles 4, 6 and 8. At idle the background noise should be even lower, since the locomotive will be stationary (no wind noise).

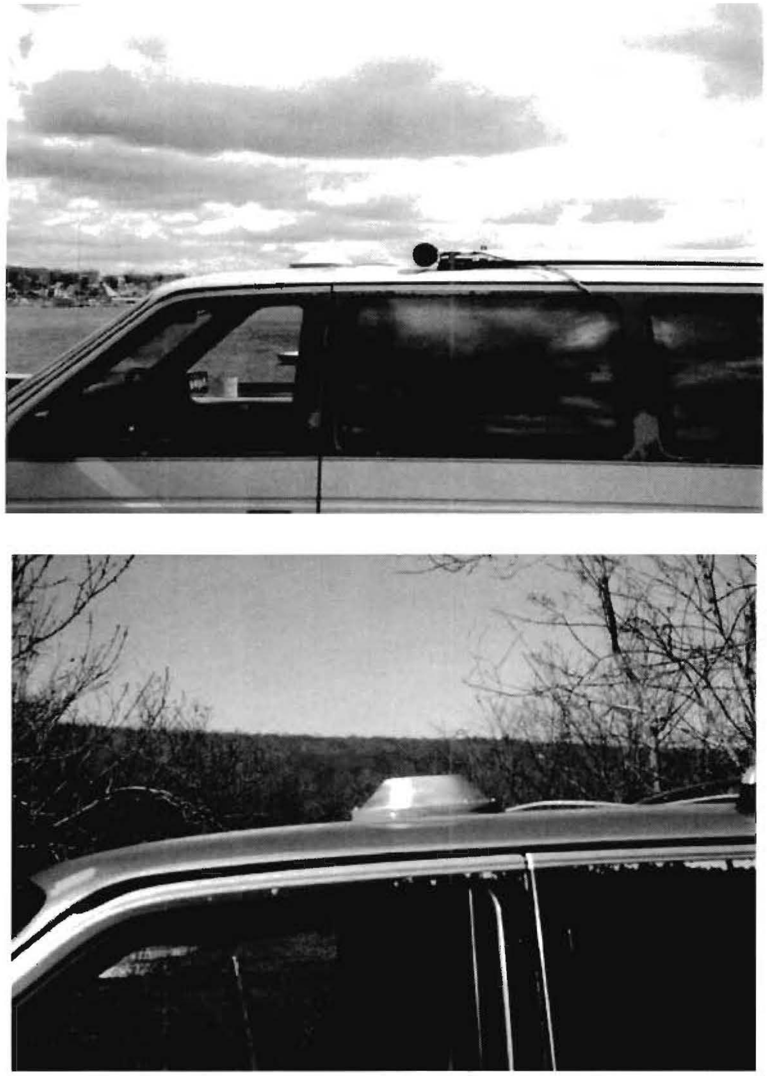


Figure 40. The Test Microphone Mounted on the Car Roof

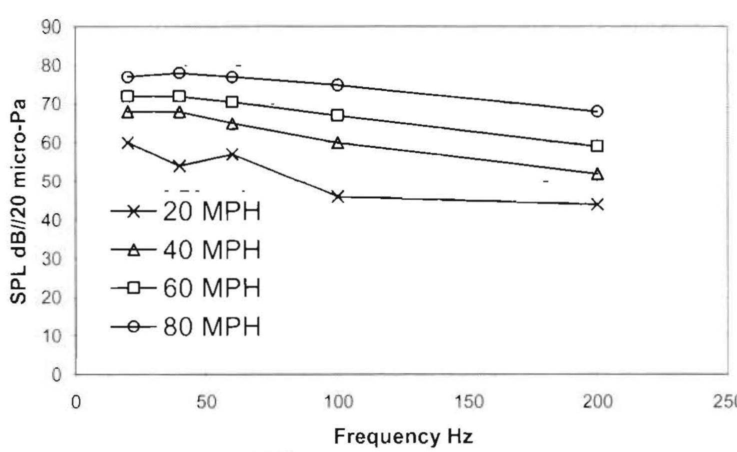


Figure 41. Output of the Test Microphone as a Function of Speed and Frequency

2.6 CONTROLLER

2.6.1 CONTROLLER ALGORITHM AND SYSTEM IDENTIFICATION

Figure 42 illustrates the structure of the feedforward controller. The controller uses a tachometer on the engine as the only reference signal. The output of the controller drives eight speaker channels. The signals from the eight control microphones mounted on the roof of the locomotive are brought back to the controller where, along with the tachometer signal, they provide inputs to the LMS algorithm to modify the coefficients of the (1 x 8) control filter W. The plant P is an (8 x 8) transfer function matrix relating the eight control filter outputs to the control speakers to the control microphone signals at the controller. A copy of the plant in the controller is required to ensure convergence of the algorithm.

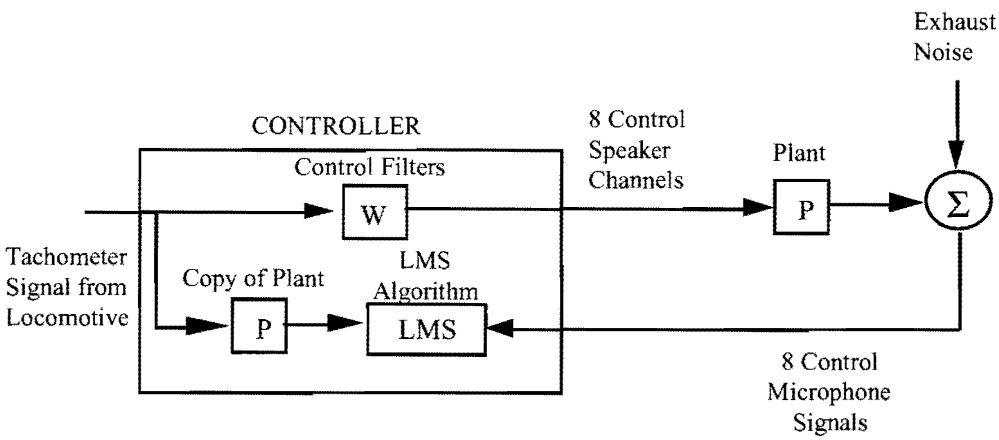


Figure 42. Controller Block Diagram

Since P will change, it must be measured periodically and its representation in the controller as a FIR filter updated. The block diagram for the system to do so is shown in Figure 43. The top part of the block diagram is the same as in Figure 42. The part of the block diagram below the dashed line is the system identification module.

In the system identification module a probe signal is injected into the control loop. This is a broad band signal whose average level is 6 dB below the existing noise. To prevent the probe signal interfering with the LMS algorithm, the probe signal is applied to a filter, which is a copy of the plant transfer function P, and subtracted from the control microphone signal before that signal is used in the LMS algorithm that adapts the control filter W.

The reference signal, the probe signal and the control microphone signals are then used in the two LMS algorithms at the bottom of the block diagram. The LMS algorithm on the left is the core algorithm designed to converge on the plant transfer function, P. The LMS algorithm on the right is designed to converge on T+WP and is designed to further reduce the residual tones in the signal that is applied to the plant identification LMS algorithm. The system identification algorithm runs continuously but updates the plant transfer functions at discrete intervals of time after the LMS algorithm has converged. The probe signals are applied to one control speaker at a time. When the algorithm converges those eight transfer functions are updated and the next control speaker is driven with the probe. The process continues until all 64-transfer functions have been updated and then starts again.

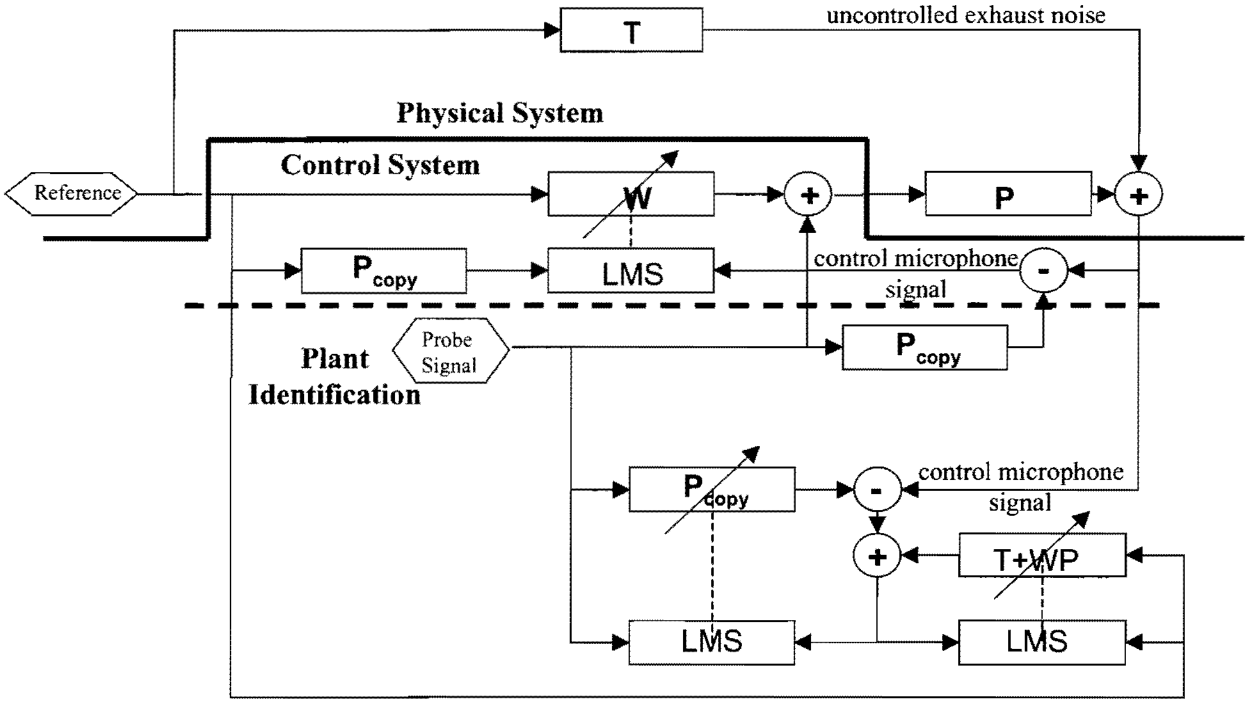


Figure 43. System Block Diagram Showing System Identification Module

2.6.2 Controller Component Selection

The primary issues in the design of the controller were the following:

- system sampling rate and anti-aliasing filtering
- computational capacity of the DSP (digital signal processor) for the in-line calculations
- computational capacity of the DSP to handle the system identification calculations and
- A/D and D/A word length

Because anti-aliasing filters can be a very expensive component in a controller, it was decided to use the low order filters available on the chosen A/D-D/A I/O boards. To do so meant that the input signals in the A/D converters had to be sampled at a rate considerably higher than needed .

to achieve the 250 Hz control bandwidth. Consequently, a physical sampling rate of 2 kHz was chosen. However, this rate decimated the samples by a factor of four in the controller when doing the computations. Using the higher sampling rate allowed low order filters on the I/O boards to be used that still obtained good rejection of aliased components of the signals.

During proof of principle testing on a F59PHI locomotive at the Southern California Regional Rail Authority (SCRRA) it was found that an AT&T DSP32C digital signal processing chip did not have sufficient computational capacity (see Section 3.2). As a consequence, Table 4 was developed that estimates the number of floating point operations per second that would be required for a system for in-line and system identification processing.

Table 4. DSP Computational Requirements

Multi Channel Filtered X LMS Processing Estimator

A/D Input Parameters	
Effective Sample Rate (Hz)	500
Decimation filter	31

Control Parameters	
Number of References	1
Number of Residuals	8
Number of Outputs	8
Control Filter Length (W)	180
Control Update Rate	1
Leaky LMS (0 for no 1 for yes)	1

Plant Parameters	
Plant Filter length (H)	200
On-Line P-Filter Length (P)	200
On-Line Plant Update Rate (0 to #chs per sample)	1

Memory	Bytes	
Sizeof Float	4	
Size of Int	2	
Control	6480	
LMS Control Update	51872	
Reference Filtering	52256	
On-Line Plant Update	6400	
Total	117014	
	114	**kbytes**

Floating Points/Inst 1

Operation	MACs	Mips	Mflop
Control Filtering	2880	1.4	1.4
Control Filter Update	34560	17.3	17.3
Leaky LMS	1440	0.7	0.7
filter X	25600	12.8	3.0
On-line Plant Filter Update	19200	9.6	9.6
Decimation filter	558	0.3	0.3
Total	83680	41.8	32.0

32 Bit Floating Point Processors	Chip Memory	Memory Word	cycles/clock	flops/cycle	Clock Speed	Max MFLOPS
Analog Devices ADSP-21020 (33)	0	32	1	2	33	66
Analog Devices ADSP-2106x (SHARC) (33,40)	512k	32	1	3	40	120
AT&T DSP32C (40,50)	1536	24	0.25	2	50	25
Texas Instruments C31 (27,33,40)	8k	32	0.5	2	40	40
Texas Instrumetns C32 (27,33,40)	512	32	0.5	2	40	40
Texas Instruments C40 (40,50,60,80)	8k	32	0.5	2	60	60
Motorola DSP96002 (33,40)	4k	32	0.5	2	40	40

In Table 4 under "Operation," the "Total" indicates a minimum of 32 million floating point operations per second (Mflops) is needed. To size processors, the estimate is ordinarily doubled and a processor is selected with a computational capacity considerably greater than the resulting number. The DSP 32C used during the testing at SCRRA is clearly too small with only 25 Mflops throughput. To be on the safe side two Texas Instruments C44 DSPs were used. These are similar to the C40 in the table. Two were chosen so that the system identification calculations could be performed on one and the in-line control filter calculations could be performed on the second, separating the two processes. This arrangement clearly provides more than enough computational capacity with sufficient excess capacity to deal with any unmodeled overhead in the computations.

For the I/O boards, the approximately 60 dB dynamic range available from 12 bit A/D's was more than adequate to deal with the approximately 35 dB dynamic range of the tones (highest to lowest) and the approximately 45 dB dynamic range of the tonal to broad band levels in the exhaust spectrum.

3. PRELIMINARY TESTING

3.1 MBTA TESTING

One of the first series of tests carried out was on the Massachusetts Bay Transportation Authority's (MBTA) Commuter Rail system. The system is operated by Amtrak under contract to the MBTA and utilizes F40PH locomotives for the passenger rail service. The MBTA agreed to allow measurements for a half-day on December 3, 1995, on the locomotive shown in Figure 44 at the Amtrak Maintenance Facility in South Boston, Massachusetts. These were preliminary tests and focused on measuring the temperatures in the exhaust stream and on the OEM exhaust silencer, the sound pressures within the exhaust silencer, and the sound pressure just outside the exhaust stack. At this point in the program, the feasibility of the active liner system, described in Section 1.1, was being tested. For that approach, exhaust gas temperature and exhaust duct sound pressure were critical parameters. Although the active liner approach was ultimately rejected, the temperature data and sound pressure data exterior to the exhaust duct were important information for the roof mounted feedforward approach ultimately implemented.

Figure 44. The MBTA F40PH Locomotive Used for Testing at the Amtrak Maintenance Facility

The site where the locomotive was placed for the test is shown in Figure 45. Though the site was not suitable for far field measurements because of a large number of reflection surfaces and the high background noise in the area, it was suitable for the close exhaust stream measurements.

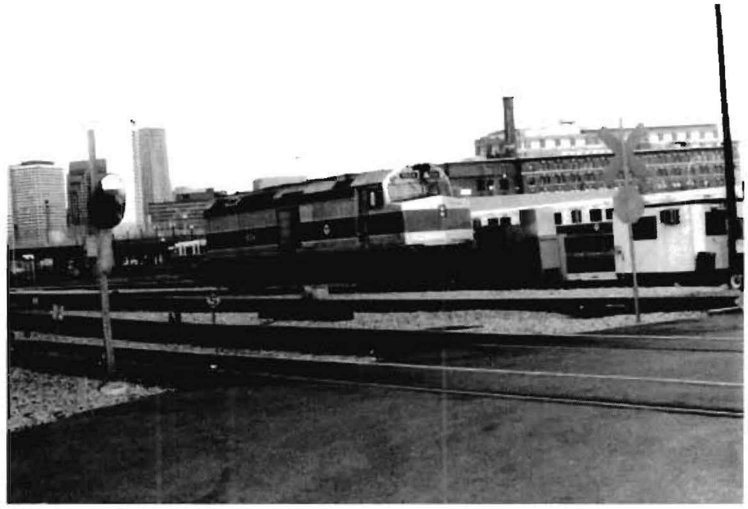
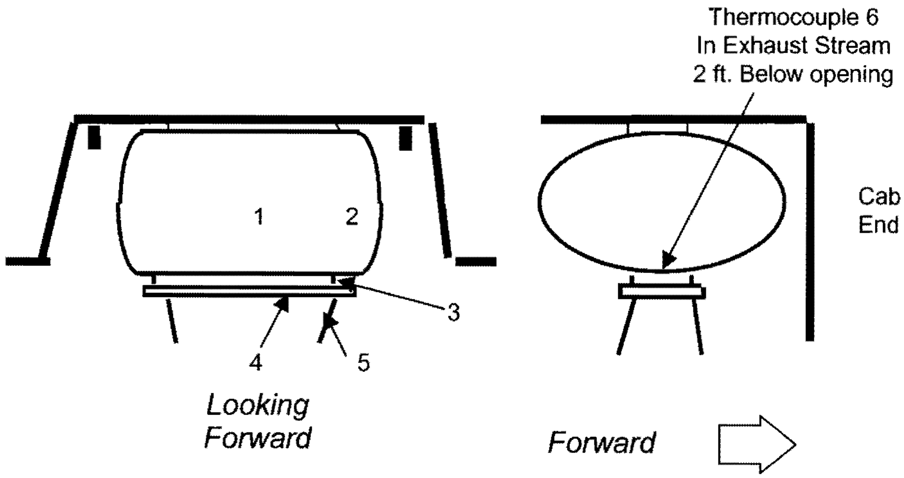


Figure 45. The MBTA/Amtrak Test Site

The temperature data acquired during this test are shown in Figure 46. Thermocouples were installed at the locations shown in the figure and the locomotive was run in self-load at throttle 8 until all thermocouples stabilized (about 30 minutes to one hour). All thermocouples were type E, and an Omega Model DP 460 digital read out was used to acquire the data. The temperatures are generally higher than measured in subsequent tests on other locomotives, especially the exhaust gas temperatures. In later tests discussed in Section 3.2 and Section 4 the exhaust gas temperature was generally around 650°F much less than the approximately 950°F measured during this tests. The reasons for the generally higher temperatures in this test are unknown.



Temperature Measurements

Degrees Fahrenheit

Location	1	2	3	4	5	6
Ist Measurement	597	481	775	703	624	953
2nd Measurement	590	454	775	707	621	956

Figure 46. Temperature Measurements

The sound pressure external to the exhaust was also measured approximately 3 feet away at a variety of throttle settings. These data have already been discussed in Section 2.3 where they were used to estimate the required volume velocity from the control speakers.

A device was also developed, sketched in Figure 47, for measuring the pressure within the exhaust stack. The device consists of a 36 inches of ¼ inch thick stainless steel tubing one end of which is filled with steel wool and terminated with a probe tip to reduce flow noise. This end is placed in the exhaust stream. At the other end of the tube, a ¼ inch condenser microphone is inserted with the axis of the microphone perpendicular to the stainless steel tube. A T-junction with compression fittings on the three ends of the T allowed an air tight seal to be made around the microphone and the stainless steel tube. A third tube consisting of 30 feet of plastic tubing was inserted in the compression fitting on the third leg of the T. The plastic tubing contained several strands of coarse twine to provide absorption for attenuating sound waves propagating down the plastic tube thereby reducing any sound waves reflecting back to the microphone that might contaminate the measurement (i.e., the tubing provided an anechoic termination). The probe microphone was calibrated by placing it in a band limited random noise sound field in an anechoic chamber and comparing its response to a standard ¼ inch instrumentation microphone place right beside the probe tip. The ratio in decibels of the probe microphone output to the

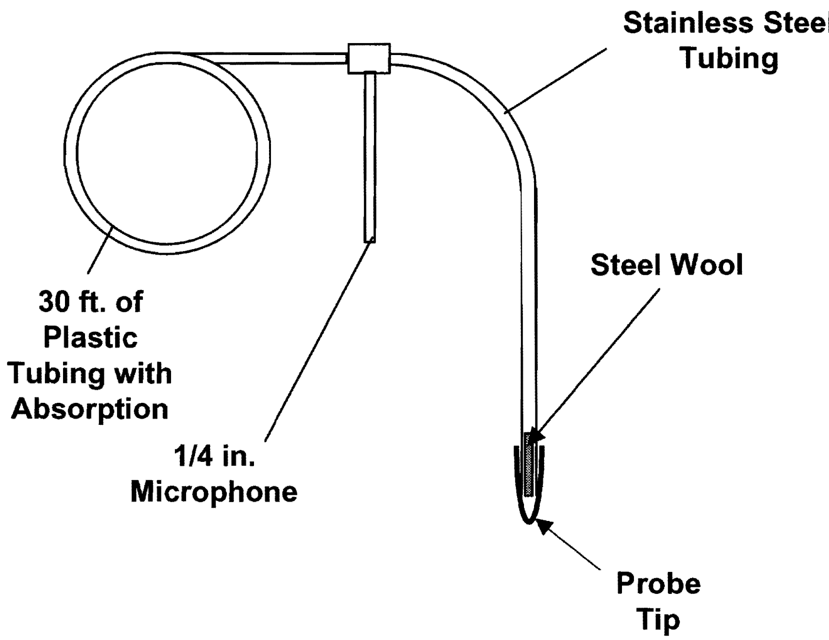


Figure 47. Probe Microphone for Measuring the Sound Pressure in the Exhaust Duct

reference microphone output is shown in Figure 48. Above 40 Hz the probe appears to behave in a fairly consistent way. The large peaks below that frequency may be due in part to reflections in the long plastic tube. At high enough frequency the absorption in the tube reduces the effect of those reflections. Data in Figure 48 was used to calibrate the probe measurements.

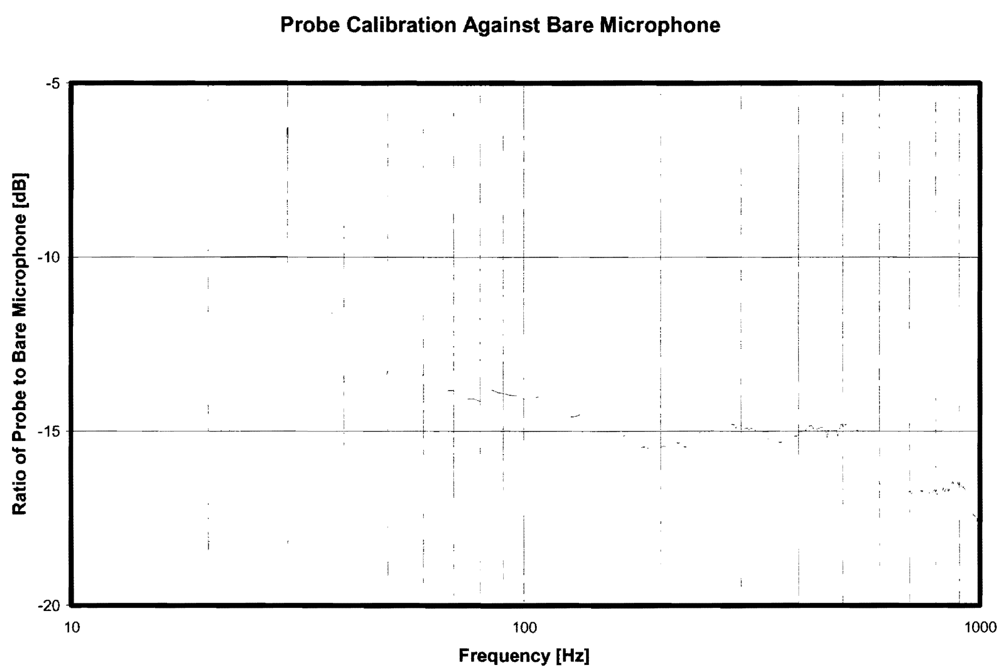


Figure 48. Calibration of the Microphone Probe

Figure 49 shows an example of the measurements taken with the probe. The sound pressure level in the exhaust duct in the exhaust stream is compared to the sound pressure level external to the duct (approximately 3 feet away) and to the sound pressure within the exhaust duct but mostly out of the exhaust stream. The figure shows the intensity of the sound pressure level in the exhaust duct (over 140 dB for some tones) and the masking of some of the weaker exhaust tones by flow noise. Not surprisingly, the exhaust noise sound levels external to the exhaust duct are substantially less than within the duct.

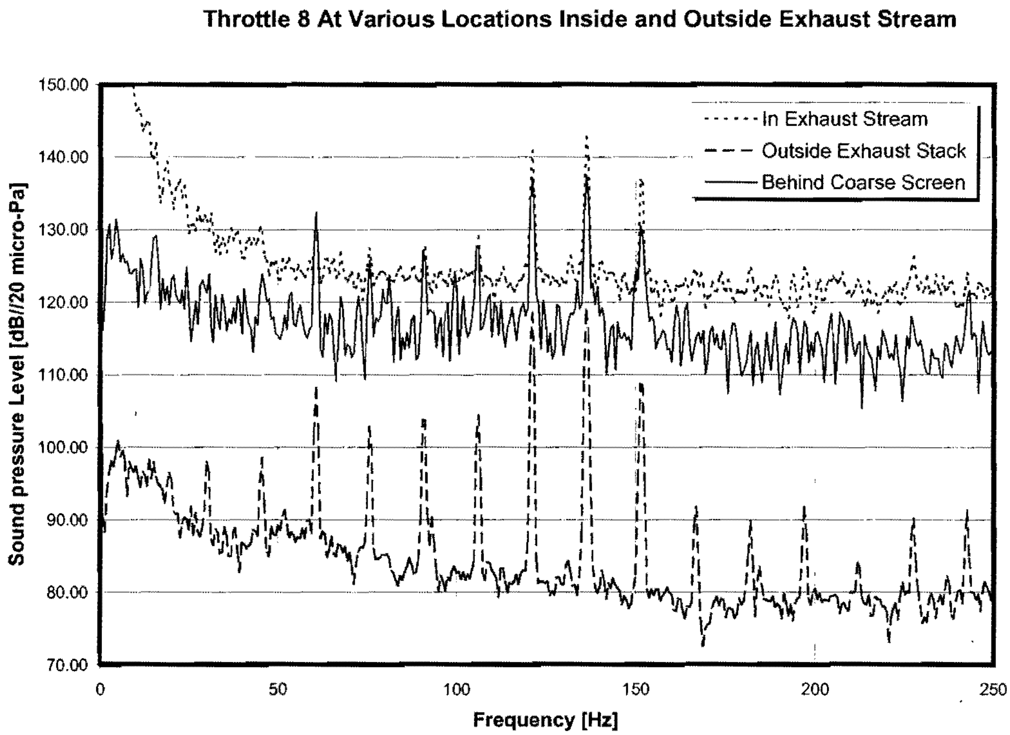


Figure 49. Measurements of the Sound Pressure in the Exhaust Duct – Approximately 18 Inches Below the Outlet and External to the Outlet

3.2 TESTING ON THE SOUTHERN CALIFORNIA REGIONAL RAIL AUTHORITY (SCRRA)

3.2.1 OVERVIEW

To aid in the development of a hybrid active/passive locomotive exhaust reduction system a series of tests was carried out at the SCRRA Metrolink yard in Los Angeles, California, during the week of July 15, 1996. Arrangements with the railroad were made by the Electro-Motive Division of General Motors (EMD) who also participated in the tests, provided personnel to operate the locomotive and provided some test equipment. FRA, through a designated contractor, was responsible for supervising and conducting the testing. SCRRA provided the test locomotive, the test site and personnel to provide support services.

These tests had the following objectives:

1. Measure the reduction achievable in locomotive exhaust tones below 250 Hz with an experimental roof mounted active noise control system

2. Acquire transfer function data necessary to evaluate analytically alternate system configurations

3. Acquire acoustic, temperature and flow velocity data to facilitate the design of an integrated active/passive silencer

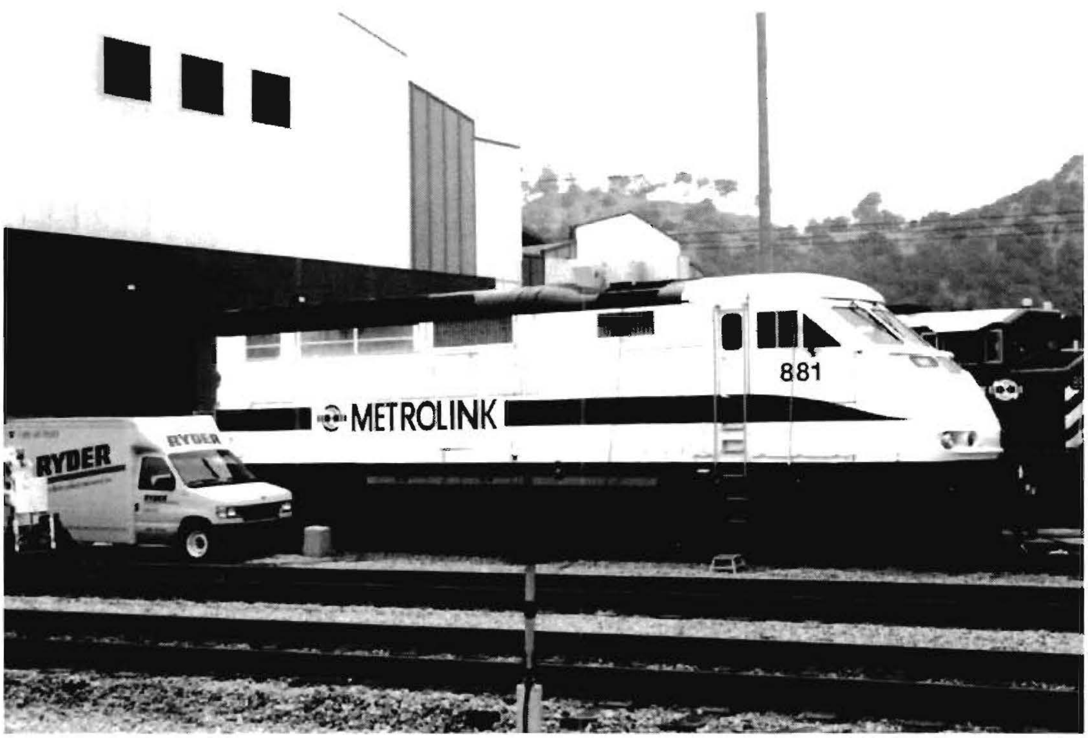
The first two objectives were by far the most important and the first four days of testing were spent acquiring these data. The remaining objective was fulfilled in the remaining two days of the six-day test period.

3.2.2 TEST SITE

All the testing was carried out at the Metrolink Yard of the SCRRA in Glendale, California near Los Angeles. The test locomotive, which is described more fully in the next section, was positioned just outside of the locomotive repair facility. Figure 50 (a) shows the locomotive at the test site from the front, and Figure 50 (b) shows it as seen from one of the far field microphones located 100 feet from the locomotive. The figure also shows the instrumentation van, which was placed just behind and on the right side of the locomotive. The van contained all of the recording, analysis and signal processing instrumentation. Placing it close to the locomotive facilitated the stringing of cables to connect on-board sensors and speakers to the instrumentation van.

The area to the right of the locomotive was essentially free field as shown in Figure 51 making it an excellent location for the far field microphones. Unfortunately, during the course of the day locomotives and trains were parked in this area making significant portions of the day unsuitable for acoustic measurements as illustrated in Figure 52 (the test locomotive is out of the picture to the right). To allow for timely acquisition of noise data, the far field microphones were moved to a closer location. Figure 53 illustrates the far field microphone locations that were ultimately used. There were initially three far field microphones located 100 feet from the locomotive, one directly opposite the exhaust stack and two 45° to each side. In the course of the measurements two additional microphones were located closer to the locomotive to reduce background idling noise and prevent trains from blocking the line of sight from the far field microphones.

(a) Front View



(b) Seen from the 100-Foot Far Field Microphone

Figure 50. The Test Locomotive Positioned at the Test Site

(a) Seen from the Locomotive Hood

(b) Looking Along the Tracks with the Test Locomotive out of the Picture to the Left

Figure 51. The Far Field Microphones

Figure 52. Obstruction of Line of Sight Between Microphones and Locomotive Caused by the Presence of Idling Trains

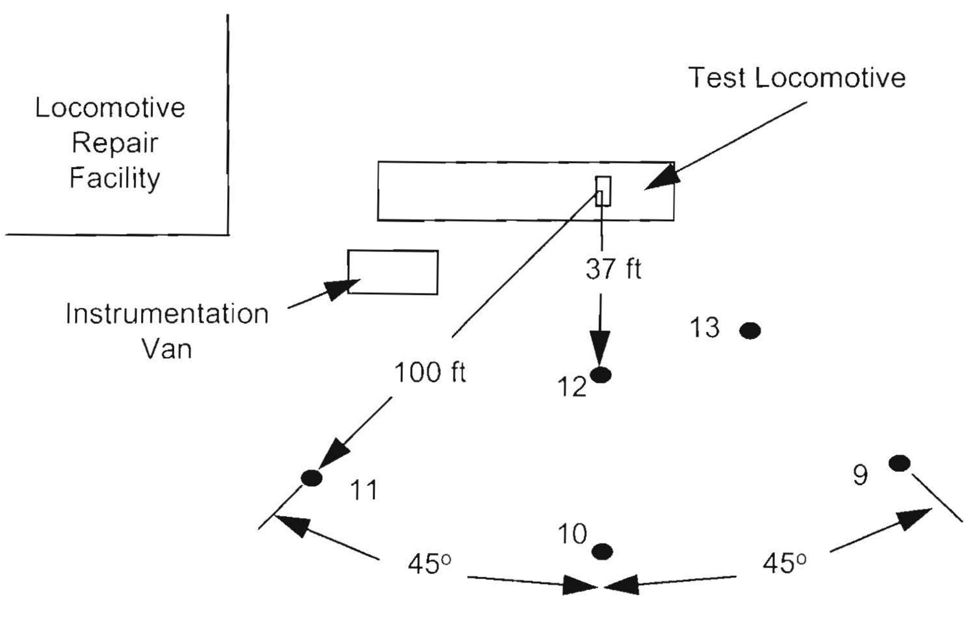


Figure 53. Test Site with Far Field Microphone Position

3.2.3 TEST LOCOMOTIVE

An F59 PHI locomotive, used for commuter rail service in the Los Angeles area and manufactured by EMD, was provided by SCRRA for six days of testing. The locomotive was a 12-cylinder 3000 HP turbocharged two stroke per cycle diesel electric locomotive. It was equipped with a self-load capability through the dynamic brake system allowing testing to be carried out with the engine fully loaded without the need for an external resistive load bank. While this made setting up the locomotive for testing easier, the dynamic brake cooling fan that forced cooling air over the dynamic brake resistive grids did produce significant noise that could have interfered with the measurements. Fortunately, tones from the dynamic brake fans are at a somewhat higher frequency than the important tones from the exhaust. Figure 54 shows the noise from the locomotive at microphone 10 located 100 feet from the locomotive at various throttle settings. The exhaust tones that needed to be controlled are primarily below 250 Hz. The large tones seen above this frequency are primarily due to the dynamic brake fans, radiator cooling fans and turbocharger.

3.2.4 ACTIVE NOISE SUPPRESSION TESTING

In Section 2, an analytical study determined that a roof mounted active tonal noise suppression system would be capable of suppressing locomotive exhaust tones up to approximately 250 Hz. To be effective, it was determined that the system would require eight independent control speakers, placed as close to the exhaust stack as possible, and eight independent residual microphones, placed as far from the exhaust stack as possible. During this test, the speakers and microphones could not be placed in precisely the patterns examined in Section 2 because of the presence of various cooling fan grills and other obstructions on the locomotive hood. Also it was decided at this early stage not to modify the hood of the locomotive to allow for flush mounting of the speaker ports with the hood of the locomotive. This significantly reduced costs and eased obtaining the cooperation of a participating railroad. The consequence of this decision was that because of the physical size of the speakers the speaker ports could not be place in precisely the locations desired.

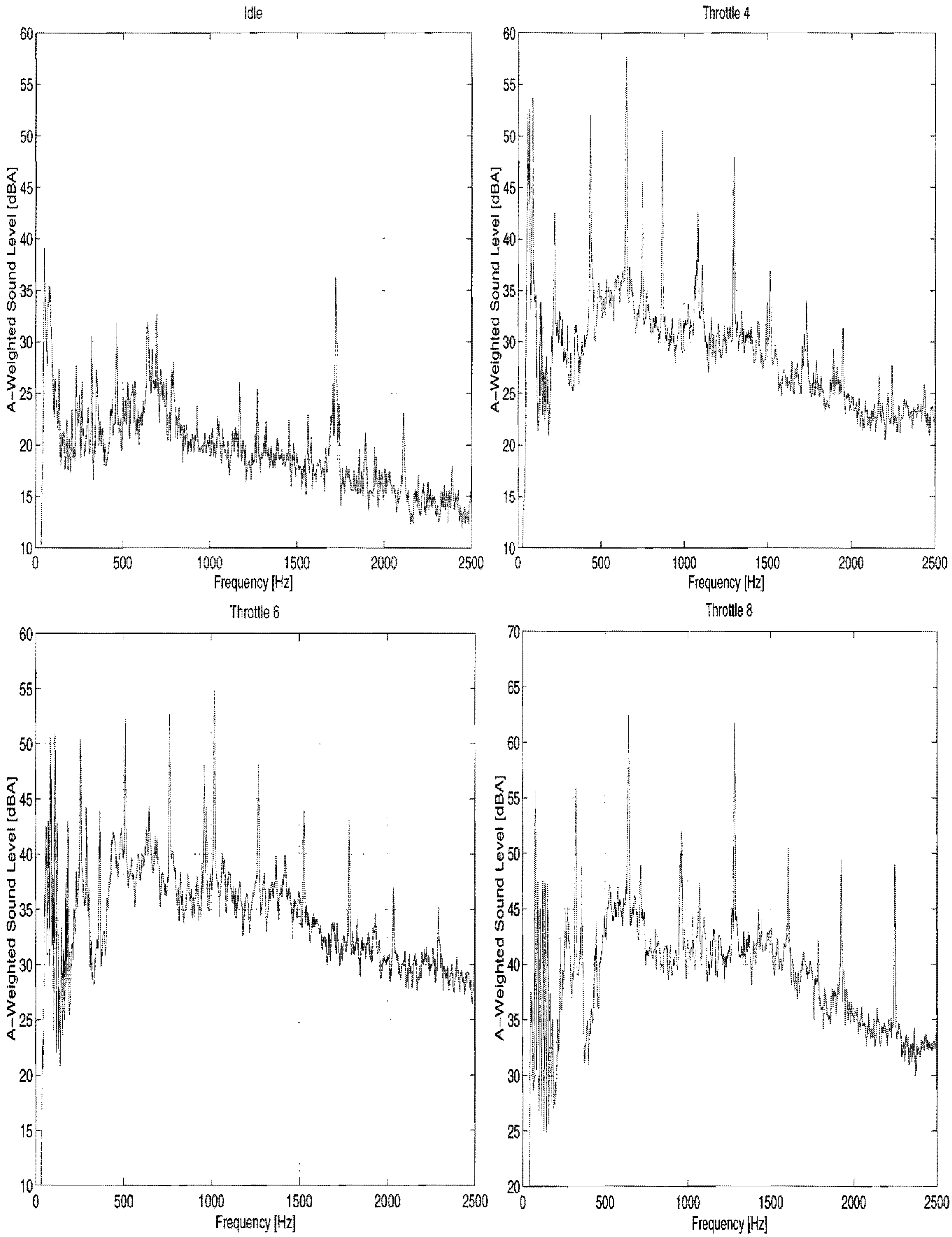


Figure 54. A-Weighted Locomotive Noise at Microphone 10, 100 Feet to the Side of the Locomotive at Idle and Various Throttle Settings at Full Load

3.2.5 INSTRUMENTATION

The controller was the Professional Active Noise and Vibration Controller (ProANVC) system built by BBN and based on the AT&T DSP32C digital signal-processing chip. The ProANVC system is capable of dealing with up to 32 inputs (residual microphones and reference signals) and sixteen outputs (control speakers). It implements the filtered-X feedforward control algorithm. Anti-aliasing filters for the input signals to the controller and reconstruction filters for the outputs of the controller were two Rockland model 716-11 sixteen-channel systems. The reference signal was provided by a Monarch Tach IV optical tachometer. The tachometer was clamped to the locomotive diesel engine at an access hole that allowed line of sight to the flywheel. A white spot painted on the flywheel allowed the tachometer to sense the rotation of the flywheel and provided one pulse per engine revolution. The residual microphones were all 1/2-inch condenser microphones B&K 4133, B&K 4134 or ACO Pacific 4012. The B&K microphones utilized General Radio 1560-P42 microphone preamplifiers and the ACO Pacific microphones ACO Pacific PS9200 microphone preamplifiers.

The control speakers were either McCauley 12-inch model 6232 or McCauley 15-inch model 6242 moving coil extended low frequency transducers. These speakers were chosen because of their very high power handling capacity (400 W continuous). All speakers were mounted in ported enclosures to provide maximum volume velocity in the 50 Hz to 250 Hz frequency range. The enclosures are shown in Figure 55.

Four different two-channel amplifiers were used to power the speakers: These are indicated in Table 5. The power ratings are for a 4-ohm load. Because of the additional impedance load due to the long cables to the speakers and voltage losses in the cables the actual power delivered to the speakers was significantly less than the rated power. The power at the speakers is also shown in the table.

Far field microphones were General Radio ½-inch piezoelectric microphones Model No. 1962-9601 with General Radio 1560-P42 microphone preamplifiers. The microphones were mounted on tripods 5 feet above the ground.

Table 5. Power Ratings of the Amplifiers Used to Drive the Control Speakers

Amplifier	Power Rating W/Channel	Power at Speaker W/Channel
BGW 8000	350	112
QSC 3800	540	173
QSC 3500	425	136
Carver PM-300	150	48
Total	**2930**	**938**

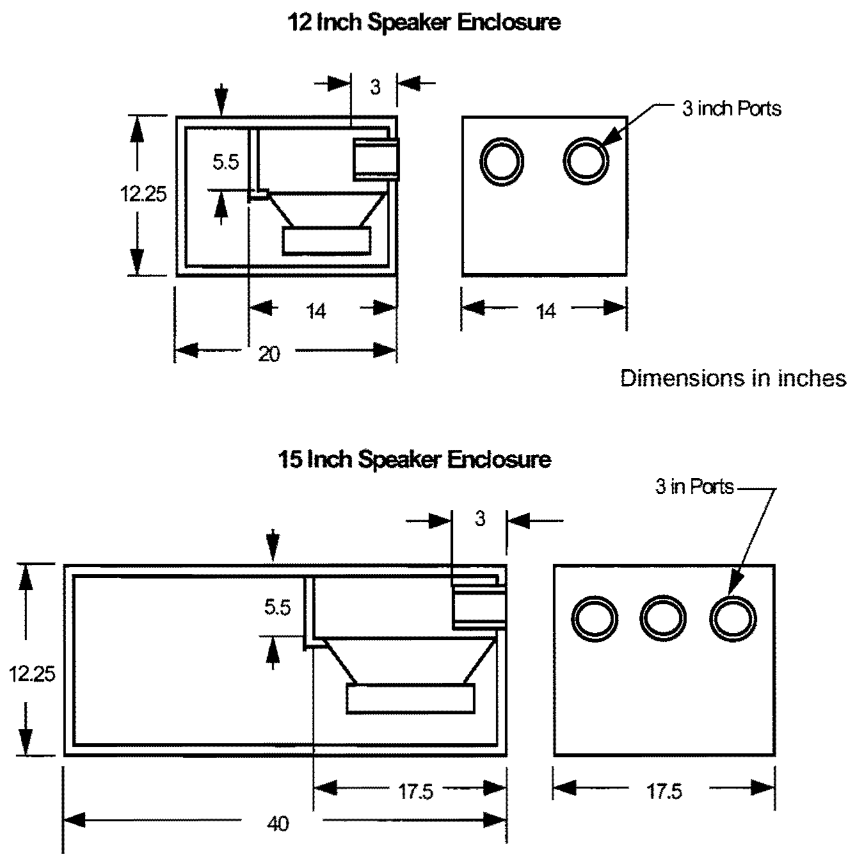


Figure 55. Speaker Enclosures

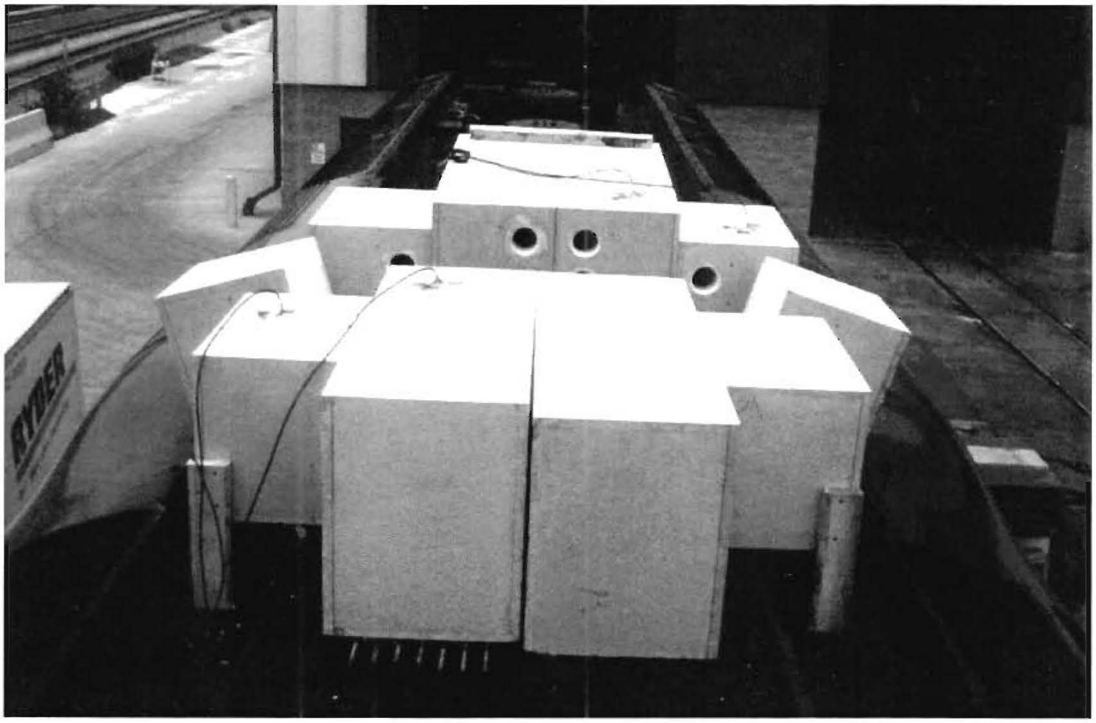
3.2.6 SYSTEM CONFIGURATION

A number of control speaker, residual microphone and controller configurations were tried to obtain the best system performance. The first configuration tried is illustrated in Figure 56. Eight of the ten speakers have their ports aimed at the exhaust stack, which lies between the speakers. The two speakers to the side of the exhaust stack have their ports aimed vertically. Two 15-inch speakers, located forward of the exhaust stack near the centerline, were driven by the same control signal. Their counterparts aft of the exhaust stack were also driven by a single control signal. The remaining six 12-inch speakers were each driven by an independent control signal for a total of eight independent control speaker channels. The residual microphones were taped to the hood of the locomotive as illustrated in Figure 56 (b). To prevent hood vibration from contaminating the microphone signals, each microphone was wrapped in compliant layer before taping to the hood. Figure 57 shows a plan view of the locomotive hood with locations of the microphones and speakers relative to the exhaust stack. The figure also shows the number of the microphones and speakers that will be used to refer to each in later sections of this report.

For this speaker and microphone arrangement, the controller was configured with a sampling rate of 630 Hz, 60 taps for each of the 64 control filters and 60 taps for each of the 64 plant filters. Measurement of far field sound showed that this system configuration provided little or no reduction in the exhaust tones. To improve performance alternate arrangements of the speakers and microphones were tried and the configuration of the controller was adjusted, as well. The final configuration settled on is shown in Figures 58 and 59. Note that the speakers all have their ports oriented in the vertical direction and that four 12 inch speakers have been substituted for the four 15-inch speakers. The latter change was made because the 15-inch speaker enclosures are 40 inches long. Consequently, if the ports were to be oriented vertically, they would stand nearly 40 inches above the exhaust outlet. Since, from previous analytical studies, increasing the separation distance of the speakers from the exhaust outlet has an adverse effect on performance, 12-inch speakers whose enclosures are only 20 inches long were substituted to minimize that separation distance.

The vertical orientation of the speakers in the Final Configuration was chosen, because, after measuring the transfer function between speaker input voltage and sound pressure at the microphones, the transfer function with vertically oriented speakers to be less complex with lower dynamic range as illustrated in Figure 60. Less complex transfer functions are advantageous because their amplitude and phase characteristics can be accurately matched with shorter digital filters in the controller.

The implementation of a filtered-X algorithm in the controller requires that there be a digital representation of all 64 speaker to microphone transfer functions in the controller - the more accurate this representation the better the control. The filter could not be lengthened to improve the performance, since the controller's capacity was near its limit. When the microphone and speaker arrangements were changed, the digital representation of the plant transfer functions improved without increasing filter length. The controller was reconfigured to further improve



(a) Control Speakers with Ports Oriented Towards the Exhaust

(b) Microphones Taped to the Locomotive Hood

Figure 56. The Original Control Speaker and Residual Microphone Configuration

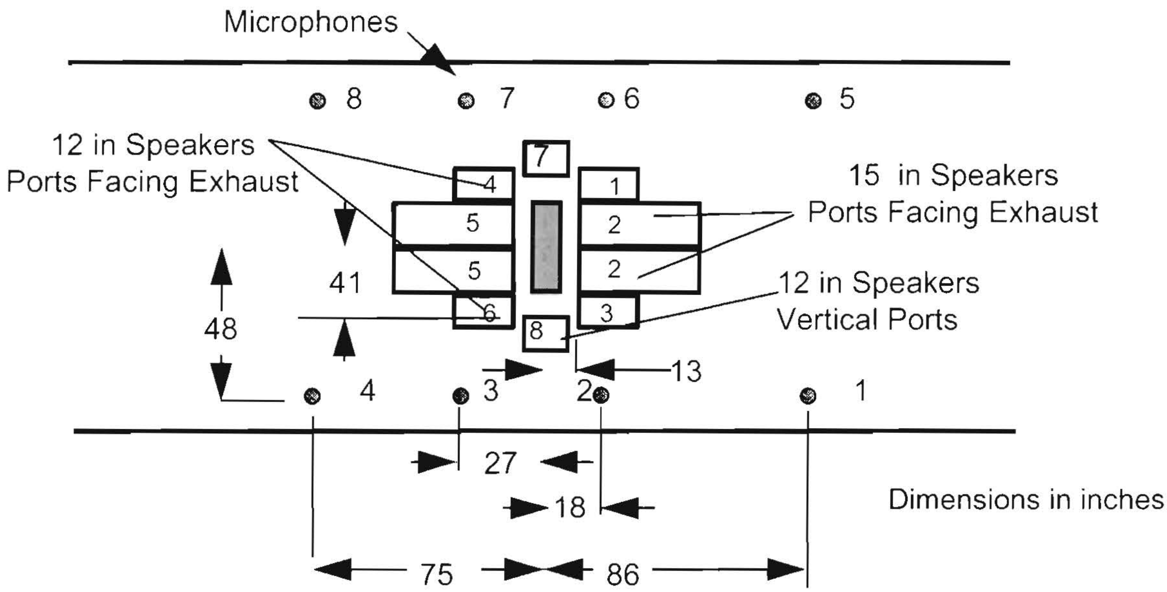


Figure 57. Original Microphone/Speaker Configuration

the performance of the control system. The sampling rate was reduced to 320 Hz. This allowed both plant transfer function filters and control filters from 60 to 90 taps to be increased. The disadvantage of reducing the sampling rate was that the control bandwidth was reduced from over 250 Hz to less than 150 Hz. Fortunately, the predominant tones are below this frequency in this locomotive.

(a) The Control Speakers with Vertically Oriented Ports

(b) The System Seen from the Roof of the Locomotive

Figure 58. The Final Control Speaker and Residual Microphone Configuration

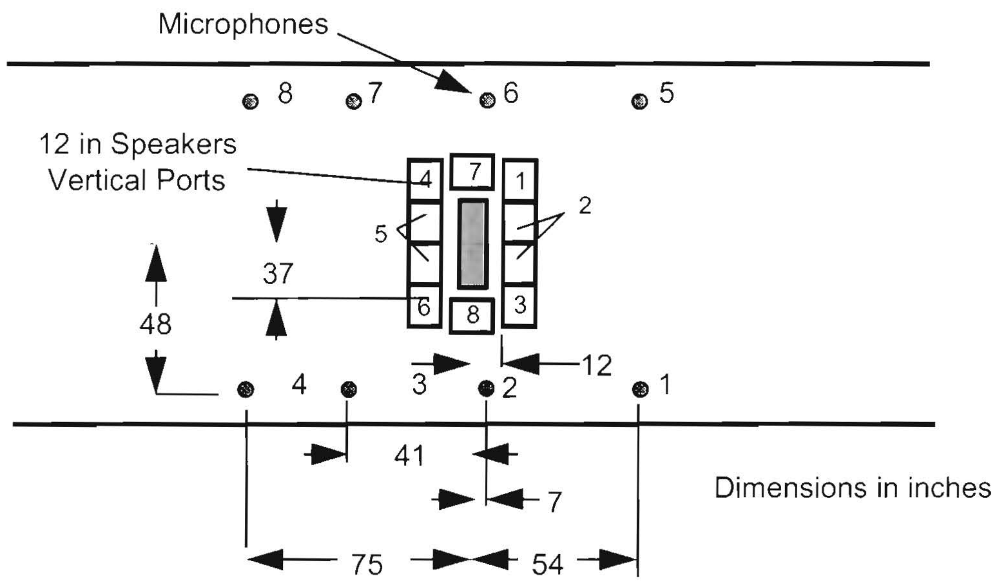


Figure 59. Final Microphone Speaker Configuration

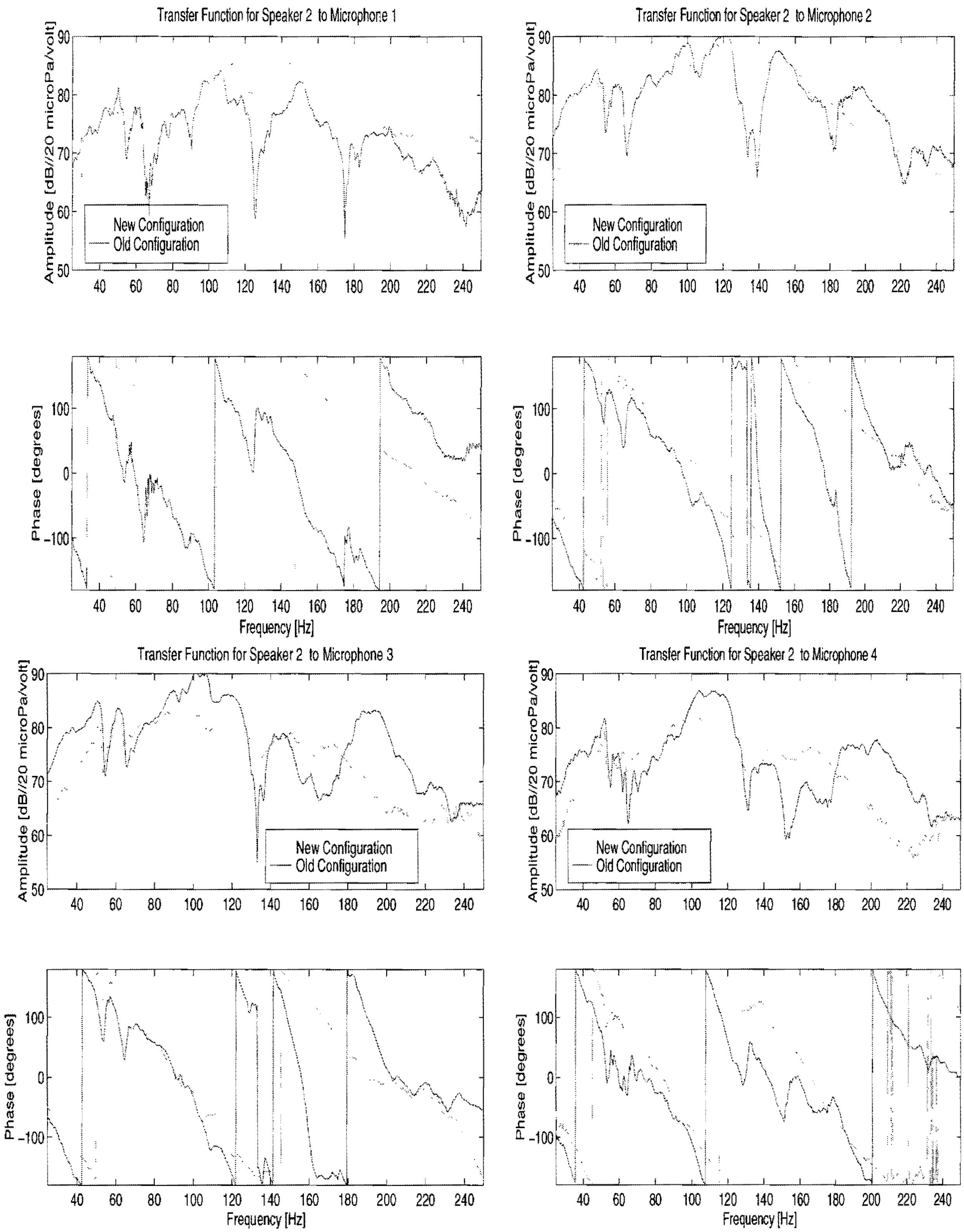


Figure 60. Comparison of Transfer Functions for the Original and Final System Configurations

3.2.7 System Performance

The performance of the final system configuration is shown in the narrow band spectra of Figures 61 through 65 for the various throttle settings under full load and for idle and fast idle. The overall reduction in sound below 150 Hz is shown in Table 6 along with the reduction in the largest tone. For the high idle condition other trains in the yard blocked the far field microphone at position 10 located 100 feet from the locomotive. Consequently, no data are shown for that location.

At idle, there is a peak in the sound pressure level spectrum, shown in Figure 61, every 3.2 Hz. Since, typically two taps per tone are needed to control each tone, and there were only 90 taps in each control filter, the controller was not expected to control the nearly 50 tones below 150 Hz. With a faster digital signal-processing chip in the controller, longer filters could have been used, and better performance would have been obtained. While Table 6 shows that the highest tones were substantially suppressed at two of the microphone locations and Figure 61 shows that other tones were suppressed as well, some tones were actually increased in amplitude. The tones with increased amplitude degraded the overall reduction in sound pressure level as shown in the table.

During the measurements at fast idle there was a train parked on the yard tracks that blocked the line of sight to microphone 10 at 100 feet. Consequently, no data is shown for that location. There are peaks in the sound pressure level spectrum every 5.9 Hz as shown in Figure 62. Unfortunately there is a profusion of peaks at other frequencies as well due perhaps to noise from other sources on the train parked in the yard. In any event the reduction in overall sound pressure level between 0 and 150 Hz is between 7 and 12 dB at microphones 12 and 13 both 40 ft from the locomotive with reductions in the largest tone at approximately 52 Hz of between 14 and 21 dB. The reductions were sufficient that to an observer the low frequency throbbing of the locomotive was completely suppressed.

Table 6. Overall Far Field Noise Reduction from 0 to 150 Hz With the Active System Operating [dB]

Throttle Setting	Mic. 10 100 ft		Mic. 12 40 ft		Mic. 13 40 ft	
	Noise Reduction 0-150 Hz	Noise Reduction Maximum Tone	Noise Reduction 0-150 Hz	Noise Reduction Maximum Tone	Noise Reduction 0-150 Hz	Noise Reduction Maximum Tone
Idle	1.9	8.1	3.3	20.3	-1.6	1.9
High Idle	-	-	12.3	20.8	7.2	13.9
Throttle 4	9.0	11.0	8.0	10.7	9.3	10.5
Throttle 6	1.9	9.3	1.4	0.2	2.8	6.0
Throttle 8	5.5	8.3	2.0	0.7	3.5	10.2

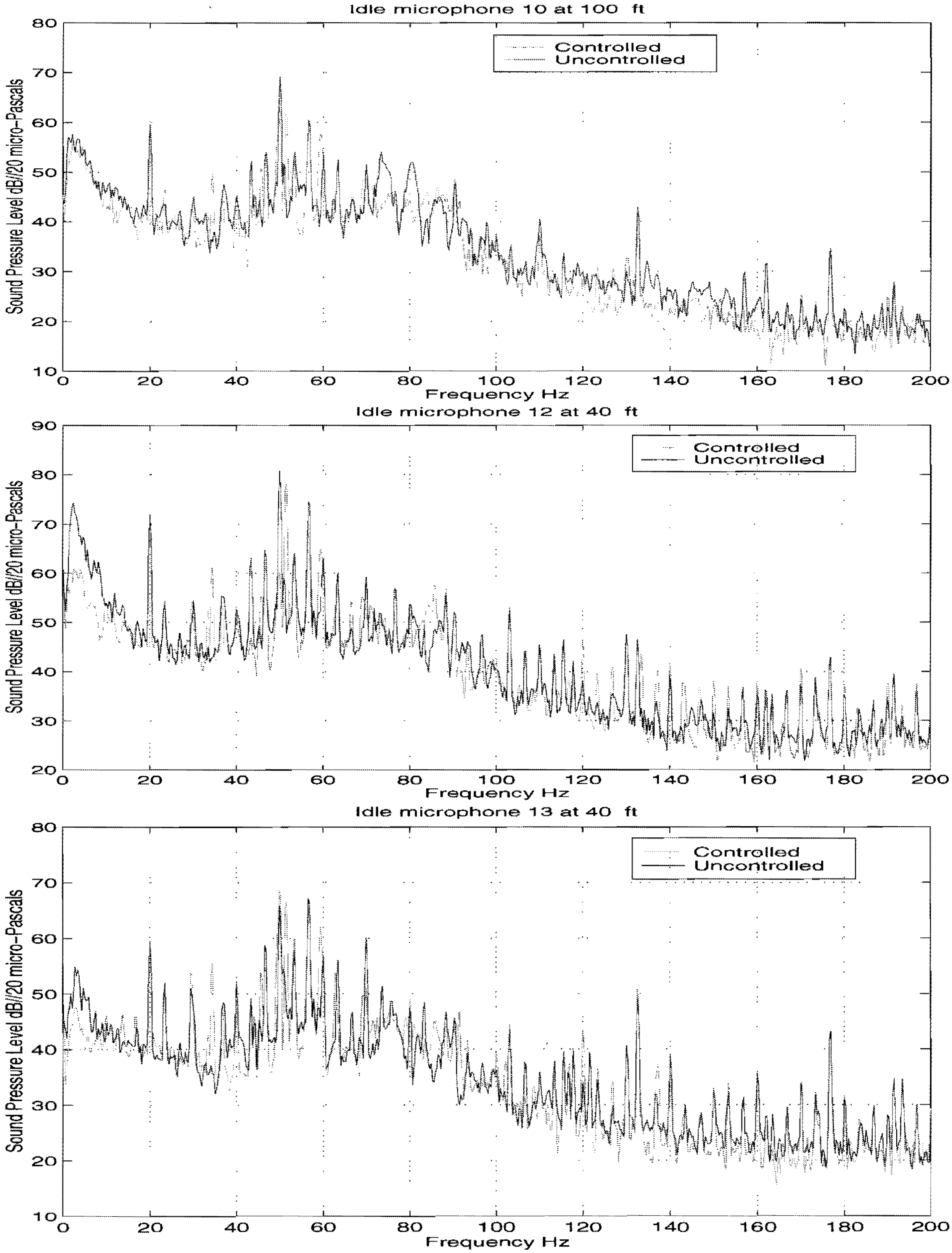


Figure 61. Far Field Sound at Idle

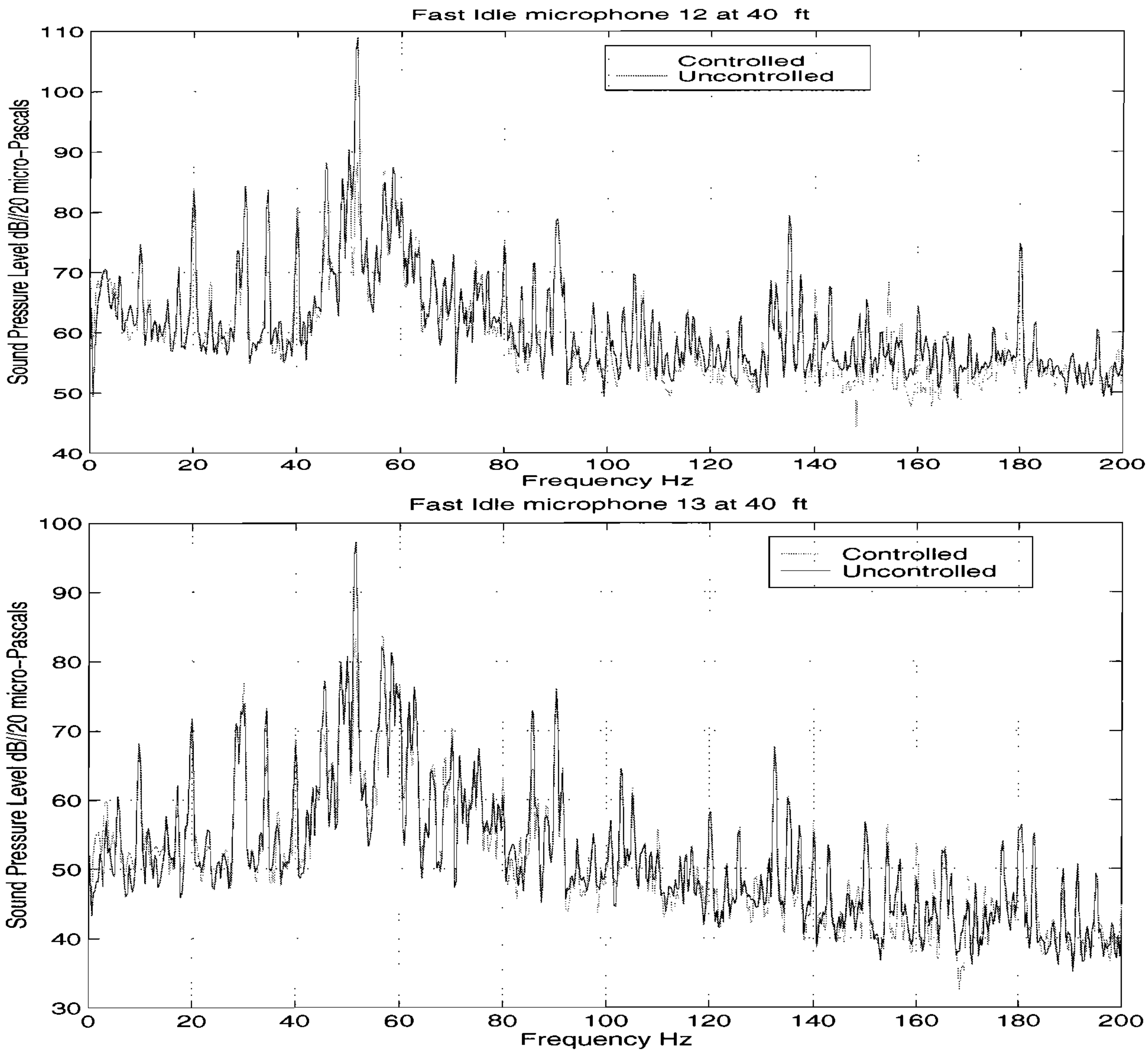


Figure 62. Far Field Sound at High Idle

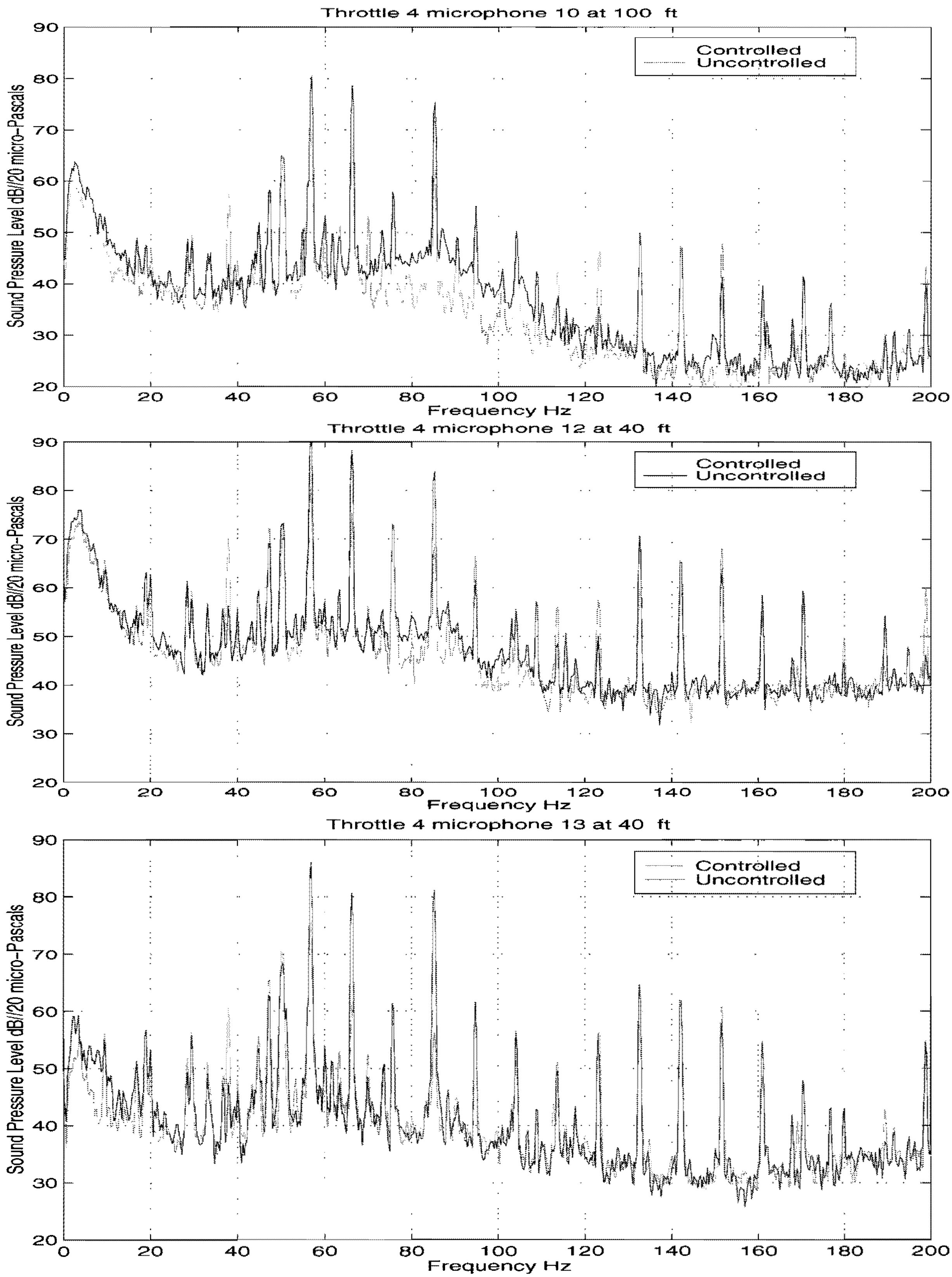


Figure 63. Far Field Sound at Throttle 4

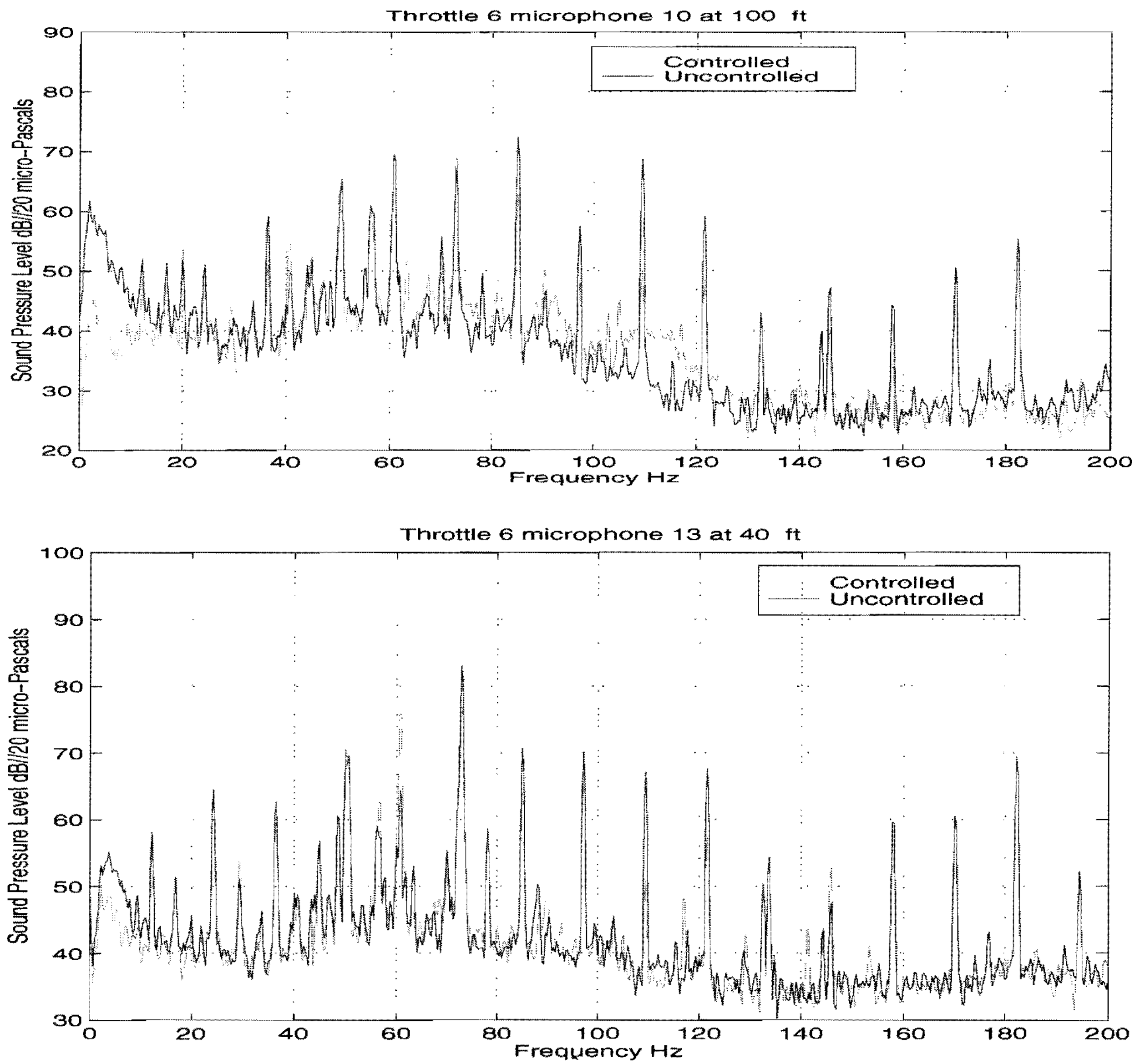


Figure 64. Far Field Sound at Throttle 6

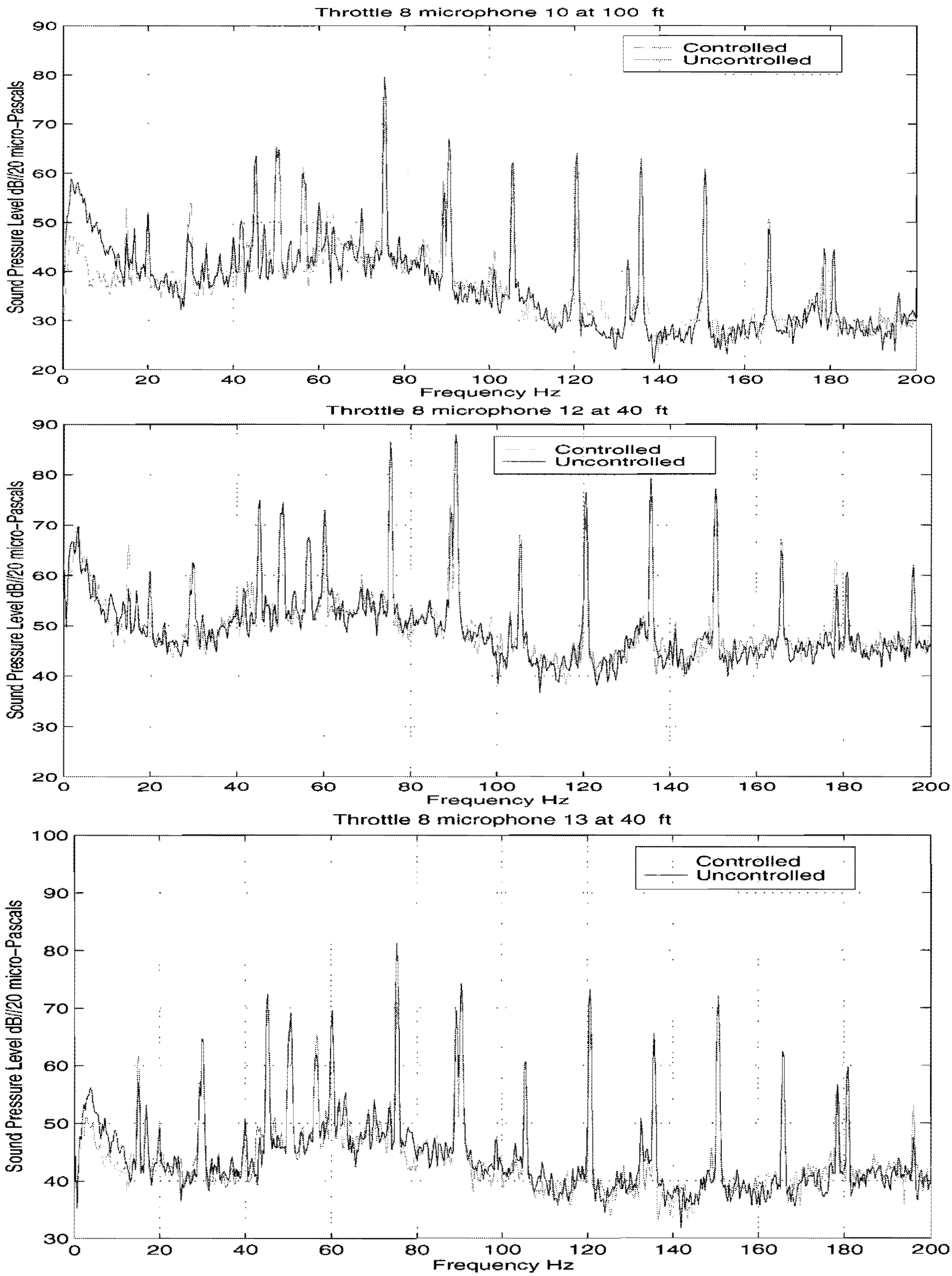


Figure 65. Far Field Sound at Throttle 8

By far the most significant reductions were obtained at throttle 4. At that throttle setting, there are peaks in the spectrum every 9.6 Hz. Figure 63 shows that the three largest tones at 57, 66 and 75 Hz are reduced between 10 and 25 dB. Table 6 shows that the overall sound pressure level between 0 and 150 Hz has been reduced by 9 dB at the 100-foot microphone. To an observer at the site the low frequency throbbing of the locomotive was completely suppressed.

The sound pressure level spectrum for throttle 6, shown in Figure 64, has a peak in the spectrum every 12.1 Hz. The overall reductions in sound pressure level in Table 6 are disappointing with generally only 2 to 3 dB of reduction. At each microphone location the system failed to control all of the largest tones although some individual tones were substantially reduced. The sound pressure levels at this throttle setting were sufficiently high that the controller began to demand more power from the amplifiers than they could supply without limiting. Consequently, some distortion in the amplifiers may have contributed to the reduced performance.

At throttle 8, Figure 65 shows that there are peaks in the sound pressure level spectrum every 15.3 Hz with the first significant peak occurring at approximately 46 Hz. The figure also shows that the overall sound pressure level between 0 and 150 Hz was reduced by approximately 6 dB. In addition the strongest tone at 76 Hz was reduced by over 8 dB. To an observer at the site the low frequency throbbing of the locomotive was substantially reduced. At this throttle setting as at throttle 6 the controller began to demand more output from the power amplifiers than they could provide without distortion. Here, however, the demands were even greater than at throttle 6. Consequently, to prevent limiting of the amplifiers that would have compromised the performance of the system, the convergence of the controller was stopped before it was finished. While this reduced distortion in the power amplifiers, it meant some loss in performance because the controller filter coefficients had not converged to their optimum values. Unfortunately it is not known how much improvement in performance would have been achieved if more powerful power amplifiers were used and if the control filter coefficients were allowed to finish converging.

3.2.8 Temperature Measurements

Exhaust gas temperature is a critical parameter in the design of exhaust silencers. Consequently, a series of measurements were carried out with a number of thermo-couples placed on a special exhaust duct[4] installed in the locomotive for this test and in the engine compartment to define the temperature environment where the control speakers would have to operate. The thermocouple locations are shown schematically in the special exhaust duct in Figure 66. All thermocouples were type E, and an Omega Model DP 460 digital read out was used to acquire the data. The resulting temperature measurements after running at throttle 8 under full load until all temperature measurements had stabilized are given in Table 7.

[4] The special exhaust duct was an early design for the passive silencer to be used in conjunction with the active system. Its exterior dimensions and the gauge of the metal used its construction were similar to the passive silencer described in Sec. 2.4 but the internal structure was somewhat different.

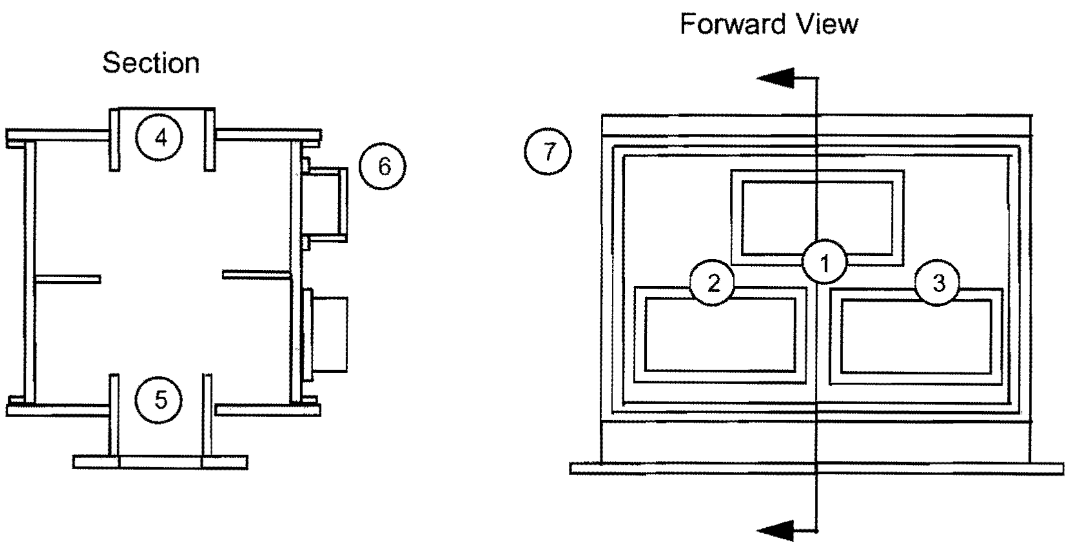


Figure 66. Location of Thermocouples on the Special Exhaust Duct

Table 7. Temperature Measurements at Throttle 8 Under Full Load

Thermocouple Number	Location	Temperature °F
1	Under bolt	459
2	Under bolt	526
3	Under bolt	406
4	In Exhaust Stream at Outlet	650
5	In Exhaust Stream at Inlet	662
6	In Air in Engine Compartment	320
7	In Air in Engine Compartment	305

The temperatures on the exhaust duct are slightly higher than those measured on the OEM exhaust muffler on a F40 locomotive in December at the MBTA. The exhaust temperatures, however, are nearly 300°F lower. The very high temperatures measured in the air around the exhaust duct were a matter of some concern. It was apparent from these test results that considerable thermal shielding and perhaps forced air cooling of the control speakers would be required when they were mounted in the engine compartment.

3.2.9 FLOW AND BACK PRESSURE MEASUREMENTS

Two pitot tubes and associated manometers were provided by EMD for measuring back pressure and flow velocities in the exhaust. One pitot tube was placed near the exhaust outlet and one near the inlet to the exhaust duct. The back pressure and flow velocities for throttles 4, 6 and 8 are shown in Table 8. At all throttle settings backpressure was so low as to be difficult to measure. It is surprising that the flow velocities measured at throttle 6 are nearly the same as those measured at throttle 4. However, it should be emphasized that the flow from the exhaust stack was highly irregular, varying substantially with location in the outlet plane of the exhaust.

Table 8. Backpressure and Flow Velocities at Full Load in the Alternate Exhaust Duct with 25 Percent Open Perforated Metal Liner Installed

Throttle Setting	Flow Velocity m/sec	Back Pressure in/H_2O
4	40	< 0.1
6	41	< 0.1
8	67	0.2

3.2.10 CONCLUSIONS FROM THE TESTING AT SCRRA

The active locomotive exhaust noise control system tested at SCRRA on the F59 PHI locomotive demonstrated conclusively that an active noise control system could provide substantial reduction in low frequency exhaust noise. The system provided substantial reduction in overall exhaust noise between 0 and 150 Hz at fast idle (12.3 dB), throttle 4 (9.0 dB) and throttle 8 (5.5 dB) with lesser reductions at idle (1.9 dB) and throttle 6 (1.9 dB). The reduced performance at idle was due to limitations in the controller. In the case of idle these limitations prevented the use of sufficiently long control filters to deal with the multiple tones in the exhaust noise signature. In the case of throttle 6 the reduced performance was due in part to insufficient control authority from the control speakers; however, why the system performed more poorly at throttle 6 than throttle 8 is unclear.

The control system required ten 12-inch speakers packaged in tuned ported enclosures of approximately 2 cubic feet each with approximately 1 kW of amplifier power available at the speaker inputs. More power was required by the control system at throttle 8 and throttle 6 to achieve the desired cancellation. Limiting the convergence of the filters at throttle 8 to prevent overdriving the amplifiers and overdriving the amplifiers at throttle 6 may have led to some performance degradation at these two throttle settings.

Measurements of back pressure and flow velocity and temperature were also made with the special exhaust duct silencer. These data were used later in the passive silencer design and in the integration of the passive silencer and active noise control system design.

4. PROTOTYPE SYSTEM DESCRIPTION

As part of the program, a prototype system was designed for installation on a locomotive. Chicago Metra, the commuter rail system in Chicago Illinois, agreed to provide an F40PH locomotive for installation of the prototype system and to assist in the installation. EMD provided assistance in interfacing the system to the locomotive and oversight in ensuring that the system would survive the locomotive environment. They also provided assistance in the installation. This section describes each of the major components of the prototype system.

4.1 SPEAKERS AND ENCLOSURES

The Active/Passive Exhaust Noise Control System (APECS) uses ten speaker enclosures, each containing two 12-inch McCauley model 6232 high fidelity speakers. These are ported enclosures designed to enhance the speaker output in the 40 to 100 Hz frequency range as discussed in Section 2.3. The enclosures came in two different geometries as illustrated in Figure 67. Two different geometries were necessary to fit the speakers in the limited space beneath the locomotive hood. Four of the L shaped enclosures, the enclosures to the left in the figure, were installed forward of the locomotive exhaust stack and the remaining six enclosures with the more rectangular shape shown on the right were installed on each side and aft of the exhaust stack. The cylindrical cap on the top of each enclosure is a rain shield to minimize rain and foreign material from entering the enclosures. Drains were also provided in the enclosures to allow any water that enters to escape. The enclosures were made of aluminum and considerable effort was made to ensure that there were no panel resonances in the operating frequency band of the speakers (38 to 250 Hz).

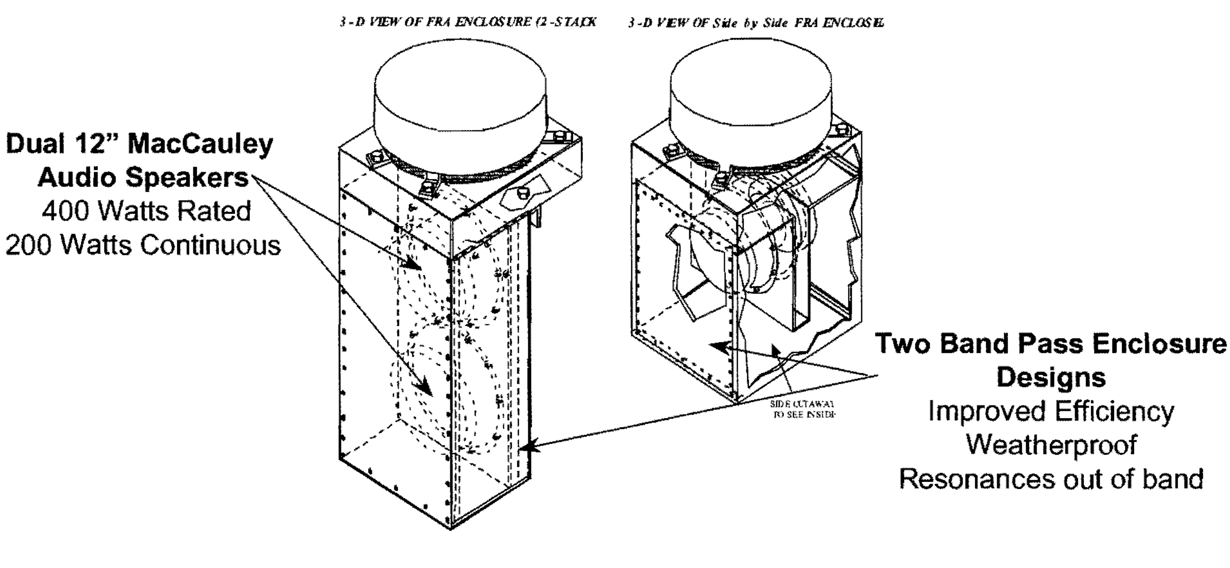


Figure 67. Two Different Geometry Speaker Enclosures

4.2 COOL BOX

The speaker enclosures were designed to fit inside the *cool box* illustrated in Figure 68. The cool box fits around the passive silencer beneath the locomotive hood. The forward end of the cool box fits flush against the rear wall of the air compartment where the traction motor blower and alternator blower are located. The cool box is designed to accept cooling air through three openings at the base of the box on each side.

The original plan was that the alternator blower in the air compartment would supply the cooling air. The alternator blower in its normal configuration does not have enough capacity to supply both the cool box and the alternator. However by changing the blower wheel it is possible to increase the capacity of the blower such that it can supply both. However for the prototype demonstration, a test with the locomotive stationary, this modification was unnecessary. A simpler approach, described in Section 5, using the traction motor blower was implemented instead. While this approach would not be acceptable for moving tests where all of the traction motor blower air is required for the traction motors, it provided a temporary solution for the prototype demonstration.

Cooling air flows from the openings at the base of the cool box up around each speaker enclosure. Each enclosure is placed in the cool box such that there is a ½ to 1 inch spacing around all sides of each enclosure. This allows cooling air to flow around each speaker enclosure to prevent its overheating due to the high temperatures in the engine compartment. The exceptions to this are the L-shaped enclosures at the front of the cool box whose forward walls rest directly against the cool rear wall of the air compartment.

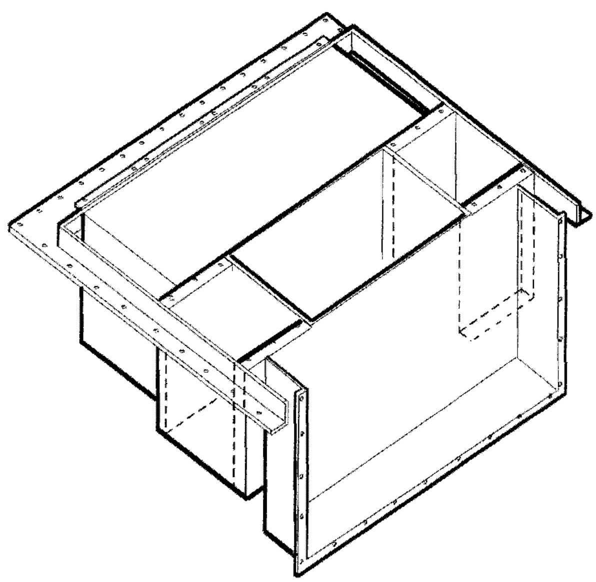


Figure 68. Cool Box

The ten speaker enclosures are shown mounted in the cool box in Figure 69. The left photograph shows the rear of the assembly and the right photograph shows the view from the front. Also shown in the front view is a large angle iron that spans the width of the locomotive hood. This was the one piece of hardware that had to be replaced in the locomotive structure. It is simply installed in place of a similar piece in the locomotive. Aside from this one change the cool box drops into the locomotive engine compartment with no modification other than the unbolting and removal of the hatch cover and the replacement of the OEM silencer with the hybrid silencer.

(a) Rear View

(b) Front View

Figure 69. Speaker Enclosures Mounted in the Cool Box

4.3 CONTROL MICROPHONES

The APECS uses eight identical microphone assemblies to sense the acoustic signals. The assembly, shown in Figures 70 and 71 consist of the microphone element, signal conditioning electronics and the housing.

The microphone element is a Gentek Model 3304-1 electret microphone that is mounted to a custom board along with power and signal conditioning. The microphone is insensitive to humidity and can even tolerate a water droplet on the active surface without a drastic change in the sensitivity and without producing extraneous noise. The signal from the microphone is passed through a preamplifier that provides some amplification, and frequency shaping to attenuate noise below 25 Hz. The microphone is powered with 24 VDC from the electronics cabinet in the locomotive cab.

The microphone and electronics are packaged in a small sealed aluminum box with an access hole for the microphone element to fit flush with the surface of the box. This box is encased in open celled acoustic foam and placed within a protective aluminum housing. The housing is designed to attach to the locomotive hood on the angled transition piece between the top flat roof and vertical sides. As is shown in Figure 70, the face of the microphone is mounted in the side of the box. When the assembly is installed on the roof of the locomotive, the long dimension of the protective housing is parallel to the locomotive axis and the microphone face is oriented away from the locomotive towards the ground.

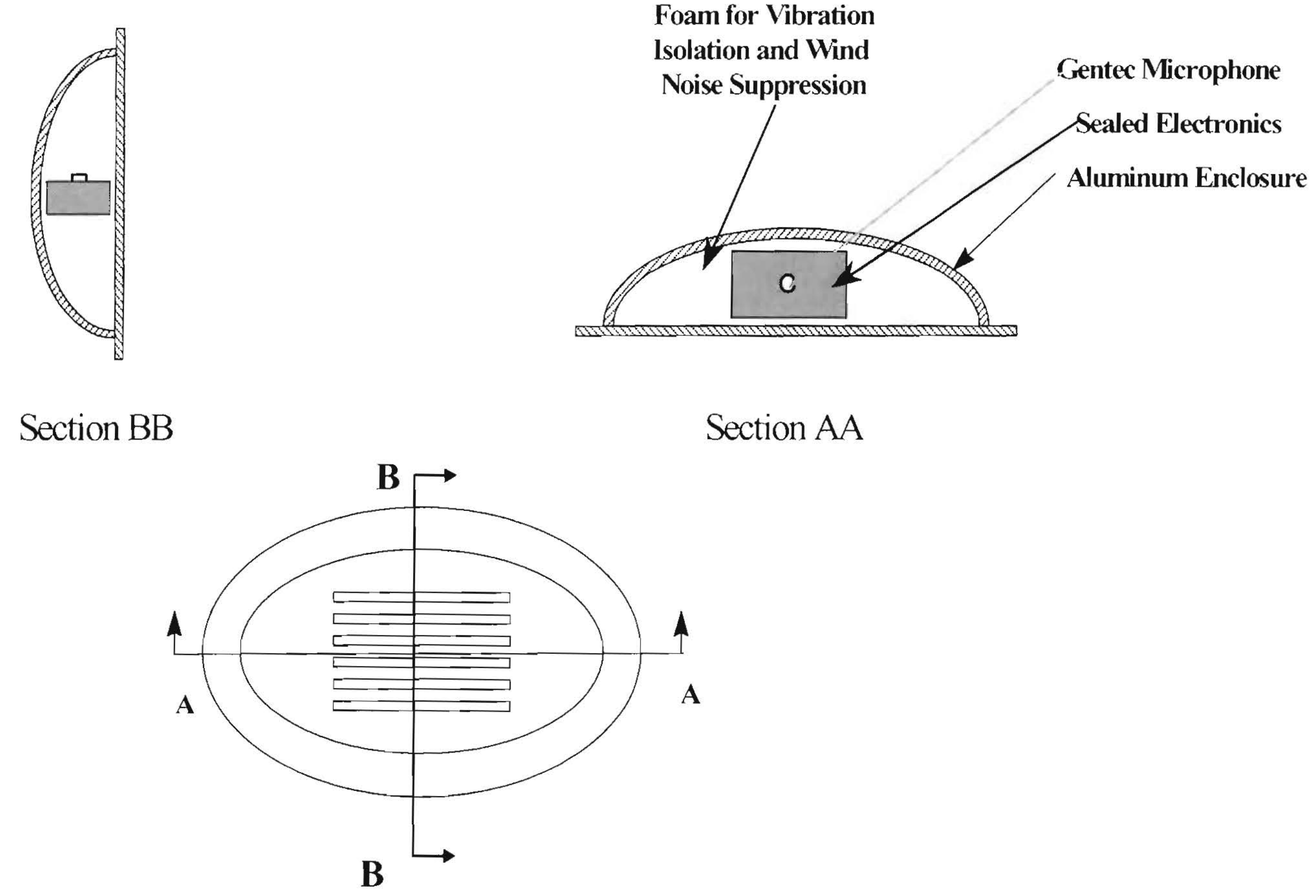


Figure 70. Control Microphone Package

This arrangement minimizes any puddling of water on the face of the microphone. The acoustic foam serves three functions. The first is to reduce vibrations from the hood from exciting the microphone element. The second purpose is to attenuate pressure fluctuations due to turbulent flow over the microphone housing. The third function is to provide some environmental protection from the elements (e.g., water, soot, etc.).

The microphone sensitivity has been chosen to allow the measurement of signals with maximum peak amplitude of 148 dB re 20 micro-Pascals.

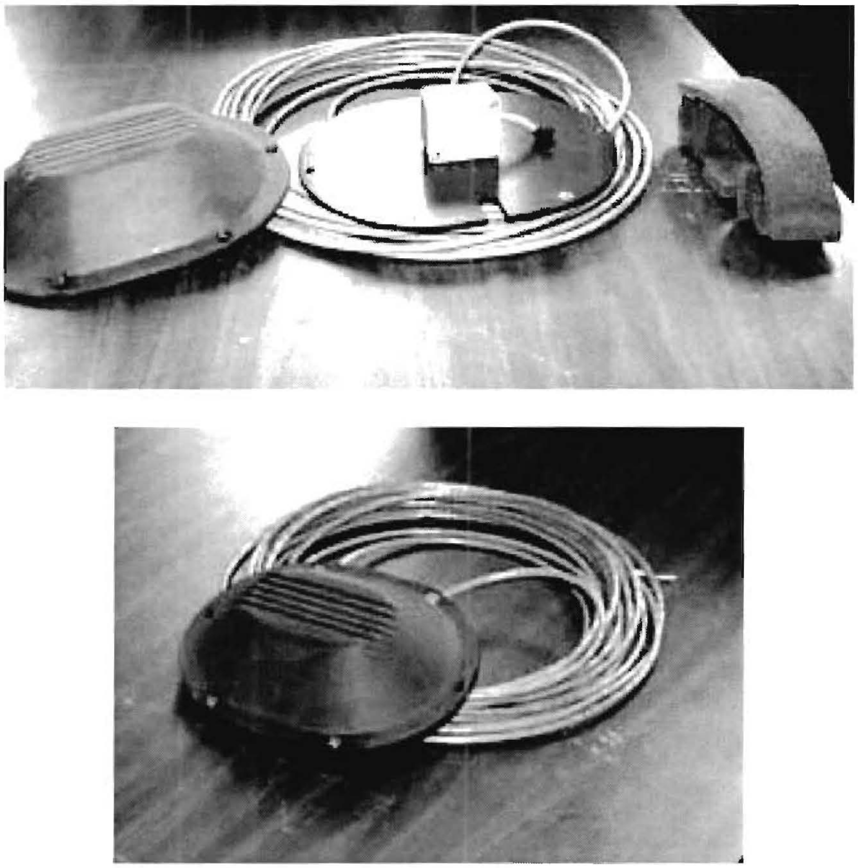


Figure 71. Microphone Assembly and Components

4.4 TACHOMETER

The tachometer, shown mounted near the locomotive flywheel in the engine compartment in Figure 72, provides the reference signal to the controller. It provides one pulse to the controller for each rotation of the engine. The unit is a Monarch ROS-5W optical tachometer powered by 5VDC from the controller electronic cabinet in the locomotive cab. The tachometer provides a voltage output when it illuminates a highly reflective surface. The tachometer mounting may be placed up to 30 inches from the reflective surface and at an angle of 45 degrees from a perpendicular to the surface. For this application the tachometer is aimed at a portion of the flywheel at the rear of the locomotive. A small piece (0.5 inches square) of reflective tape is affixed to the flywheel to provide a reflection. The size of the tape is chosen such that it is illuminated for only a very short time as it passes the tachometer. This arrangement provides a signal containing the fundamental rotation rate of the engine and its entire harmonics through 250 Hz at nearly the same magnitude.

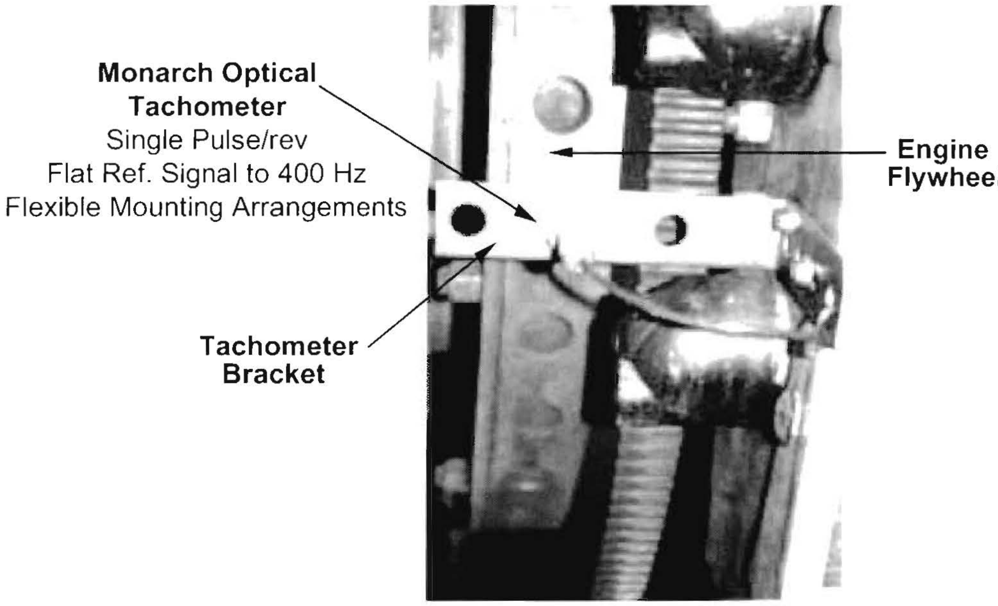


Figure 72. Tachometer Mounted Near the Locomotive Flywheel

4.5 ELECTRONICS ENCLOSURE

The electronics enclosure is shown in Figure 73. It is designed to fit in a compartment in the cab on the front wall to the left of the entrance to the nose of the cab. Drawer slides and cable guides are provided to allow the cabinet to be pulled out of the compartment to provide access to all of its components. The plan was for cooling air from a duct in the nose compartment to be ducted to the back of the compartment containing the electronics enclosure and to exhaust through the front. For the prototype test, however, the cabinet was not installed in the compartment and separate fans provided cooling.

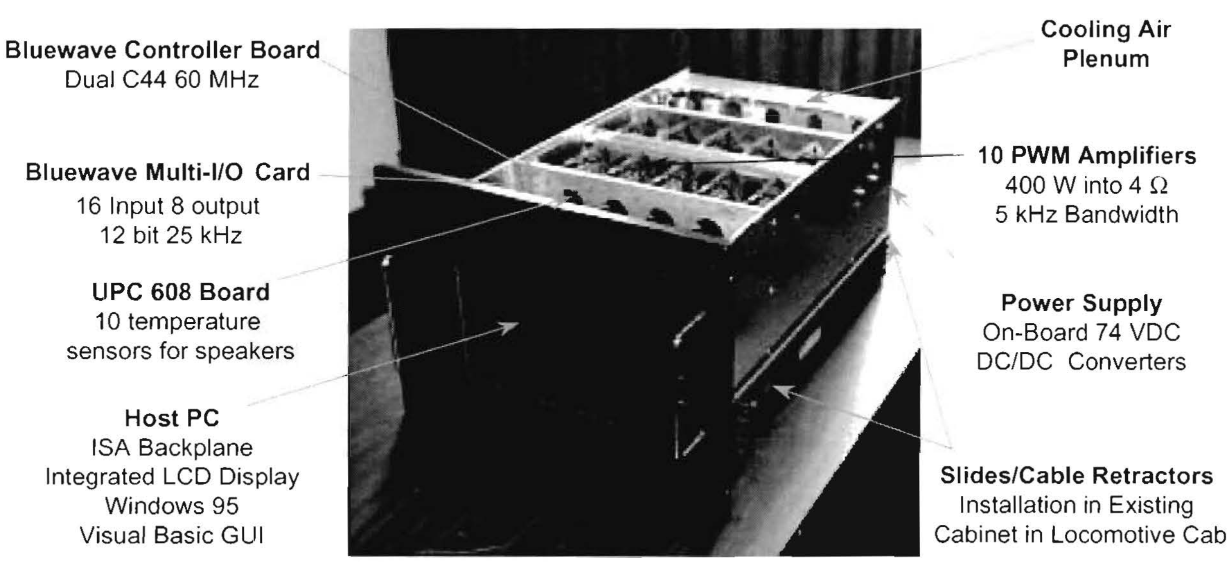


Figure 73. The Electronic Enclosure

The enclosure contains a number of components. There is a host computer with integral LCD 10-in. flat panel display incorporated in a NEMA 4/12 enclosure (oil and watertight). The computer is a CyberResearch NAP 10 80586 133 MHz processor running Windows 95 with 32 Mbytes of RAM. Programs are stored on a standard IDE hard disk. The three ISA expansion slots in the computer hold the DSP controller cards and a temperature measurement card.

A Loughborough Sound Images (now Bluewave) QPC-40/C40S1-60 carrier board performs the signal processing with two TIM-40 modules each carrying a Texas Instruments TMS320C44 Digital Signal Processor (MDC44S3-60) running at 60 MHz. One of the DSP's is for in-line control functions and one is for all plant identification functions.

A Loughborough Sound Images (now Bluewave) PC/16IO8 Multi-channel IO board with 16-inputs and 8-outputs handles the signal conditioning and analog to digital conversion functions. The A/D's and D/A's are both 12 bit and there are 3 pole Butterworth anti-aliasing filters on the board for each channel.

There is also provision for monitoring of temperature in the speaker enclosures. A third expansion card, a UPC 608 board allows the computer to monitor the outputs of 10 resistive temperature devices (RTDs) mounted near the voice coil in one speaker in each of the speaker enclosures. Finally the cabinet contains 10 pulse width modulation (PWM) amplifiers for driving the speakers. One amplifier drives each pair of speakers in the enclosures. The amplifiers are 400 Watts each and drive two 8-Ohm speakers wired in parallel (4-Ohm load). Power to drive the electronics is provided by the locomotive's 74 VDC supply from the auxiliary generator. The standard auxiliary generator on the F40PH locomotive has a capacity of 18 kW. This is adequate for the planned stationary test; however, for in-service testing the 18 kW unit will need to be replaced with an available 22 kW unit to ensure adequate power to the APECS and other locomotive services such as battery charging and cab heating. The electronics cabinet also contains a number of DC to DC converters to power the host computer and the control microphone preamplifiers.

5. PERFORMANCE TESTING OF THE PROTOTYPE

5.1 BACKGROUND AND OVERVIEW OF RESULTS

During the two week period beginning July 26, 1999, the APECS was installed and tested on the Ernest Marsh #100 an F40PH-2 passenger locomotive, shown in Figure 74. Chicago Metra provided the locomotive, and removed it from regular commuter rail passenger service for over two weeks to make it available for testing. The design of the APECS was a joint effort between the FRA (through a contracted vendor) and EMD. FRA was responsible for the design and fabrication of the system and EMD dealt with interface issues with the locomotive. Both organizations supervised the installation, which was carried out with the help and cooperation of Chicago Metra at the 51st St. Yard in Chicago. The purpose of the test reported on here was to demonstrate the acoustic performance of the system and to perform a number of tests to evaluate the ability of the system to perform properly in the railroad environment.

Figure 74. The Test Locomotive (Ernest Marsh #100)

On Monday, July 26, baseline measurements of the noise from the locomotive were carried out before system installation. Although similar measurements were made two years earlier, it was felt that too much time had past and that repeat measurements were necessary. The installation required from Tuesday, July 27 through Monday, August 2 to accomplish. Check out of the system required most of the week of August 2. Successful noise reduction performance of the system was finally tested on the afternoon of Friday August 7. The APECS was removed from the locomotive the following week.

Measurements of the noise on the roof of the locomotive with the active system turned on and off showed reductions of over 20 dB in the primary tones under many of the operating conditions. Overall reductions are shown in Table 9 versus throttle setting. The reductions were very respectable and reflected the fact that human observers could easily distinguish the reduction in the low frequency noise when the system was turned on and off.

The passive IL was determined by measuring the noise on the roof of the locomotive approximately 4 feet from the exhaust outlet (where exhaust noise is likely to dominate). These measurements were performed during the baseline tests when the OEM silencer was still installed and were then repeated after replacing the OEM silencer with the new silencer. The results of those measurements are shown in Table 10. The table shows the change in the A-weighted sound level above 200 Hz where the silencer was designed to function. The performance is quite respectable. The performance appears to be somewhat better under unloaded conditions. This performance under loaded conditions is believed to be understated because of the contaminating noise from the dynamic brake fans, which were operating to control the resistor grids during self load. These fans were only operating during the loaded tests.

Table 9. Active System Noise Reduction

Throttle Setting	Overall Noise Reduction in Control Band dB
Idle	13.2
High idle Unloaded	22.9
Throttle 4 Unloaded	13.6
Throttle 6 Unloaded	13.4
Throttle 8 Unloaded	10.7
Throttle 4 Loaded	12.3
Throttle 6 Loaded	8.7
Throttle 8 loaded	9.6

Table 10. A-Weighted Noise Reduction of the Passive Silencer above 200 Hz

Throttle Setting	Overall Noise Reduction dBA
Idle	8.0
High idle Unloaded	8.9
Throttle 4 Unloaded	7.6
Throttle 6 Unloaded	7.8
Throttle 8 Unloaded	6.3
Throttle 4 Loaded	5.4
Throttle 6 Loaded	4.3
Throttle 8 loaded	5.4

Table 11 shows the overall A-weighted noise reduction for the passive and active components of the APECS combined over the full frequency range of interest. The reductions under all conditions, with the exception of throttle 6, are very encouraging. The goal of 10 dB in noise reduction has been almost achieved for most of the operating conditions. Note the noise reduction under load may be understated by 1 to 2 dB because of dynamic brake fan interference as noted above.

All components of the APECS functioned very well after installation and system check out. The cool box with cooling air supplied by the traction motor blower provided excellent thermal protection to the control speakers. At no time during running of the locomotive, even at throttle 8 full load, did the speaker temperatures exceed 130°F. The control microphones after some adjustment to remove unwanted DC signal components functioned well. The power amplifiers initially showed some tendency to overheat; however, after adjusting the cooling airflow all ten amplifiers functioned flawlessly. One of the two DSP chips on the controller board did fail near the end of the test period. The failure is thought to be mostly due to mishandling of the board rather than an operational failure. However, software adjustments were made to continue the test.

None of the problems mentioned above was a result of the locomotive environment but rather were the normal difficulties encountered when placing a system in operation for the first time.

Table 11. Overall A-Weighted Noise Reduction of the APECS

Throttle Setting	Overall Noise Reduction 25 Hz to 5 kHz dBA
Idle	5.1
High idle Unloaded	8.7
Throttle 4 Unloaded	6.9
Throttle 6 Unloaded	7.7
Throttle 8 Unloaded	5.8
Throttle 4 Loaded	6.4
Throttle 6 Loaded	4.3
Throttle 8 loaded	6.9

5.2 BASELINE TESTS

In April of 1997, the noise from the test locomotive shown in Figure 74 was measured at the locations shown in Figure 75. It was felt, however, that too much time had past since those measurements were performed and that, at a minimum, limited repeat measurements should be carried out. Consequently the measurements were repeated at the locations shown in Figure 75. Comparison of the two sets of baseline measurements at a limited number of operating conditions showed them to be essentially the same.

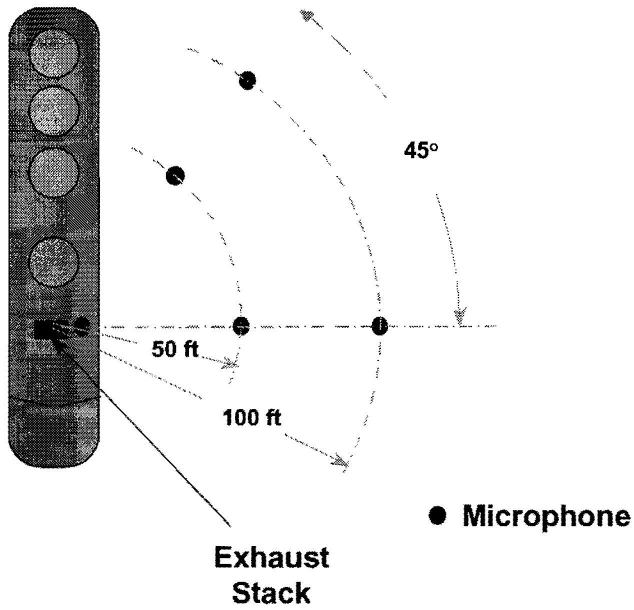


Figure 75. Microphone Locations for the April 1997 Baseline Measurements and the Repeat Measurements on July 26, 1999

5.3 INSTALLATION AND SYSTEM OVERVIEW

Installation of the APECS began on Tuesday, July 27, 1999, and continued through August 2, 1999. The first step in the installation was the removal of the existing exhaust silencer and the installation of the new APECS silencer. Figure 76 shows the new silencer installed on the engine as seen through the hatch cover opening. The new silencer is covered with a layer of rock wool thermal insulation, which in turn is covered, with a protective layer of barium sulfate, a rubber-like material. The insulation was designed to protect the cool box with its associated speaker enclosures from the high temperatures of the walls of the silencer (over 650°F). The protective layer originally designed to provide physical protection of the insulation during installation proved to be a small problem. When the engine was operated at throttle 8 under full load for any length of time the protective cover began to melt. In future installations the protective cover will not be used at all or the material will be changed. In fact the insulation may itself be unnecessary if sufficient airflow can be obtained between the exhaust silencer and the inner walls of the cool box "chimney."

Figure 76. The New APECS Exhaust Silencer Installed on the Ernest Marsh

After completing the installation of the silencer, the cool box was installed with its associated speaker enclosures as shown in Figure 77. The installation proved to be quite simple. The cool box with its speaker enclosures was simply lowered into the hatch cover opening and guided around the silencer. As expected, some shimming was required where the cool box rests on the hatch cover flange to align everything, but in general the installation proceeded without any problems.

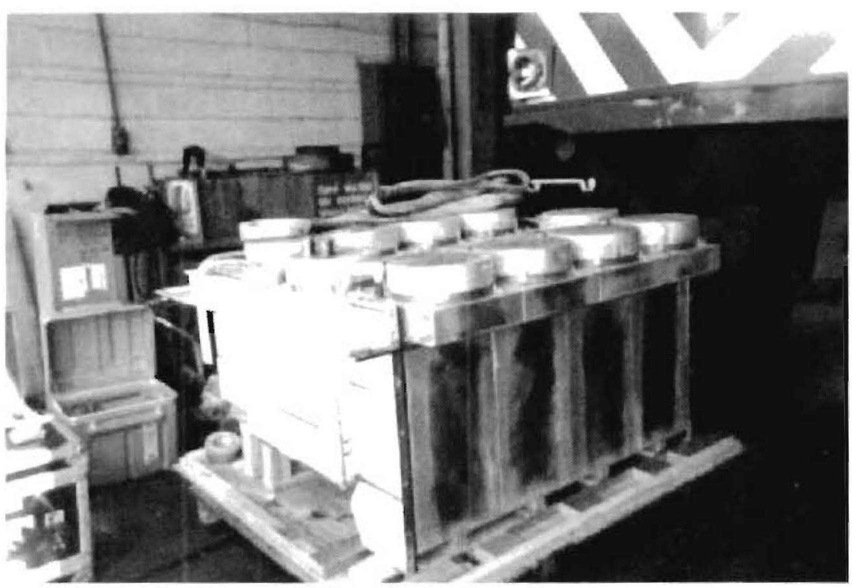


Figure 77. The Cool Box and Associated Speaker Enclosures

The cool box provides a protective enclosure around the ten speaker enclosures. The speaker enclosure walls are spaced approximately 1/2 inch from the walls of the cool box. Cool air from the traction motor blower was injected at six locations at the bottom of the cool box. The cooling air flowing between the walls of the cool box and the speaker enclosures keeps the speaker enclosures and their associated speakers at a safe temperature even through the engine compartment of the locomotive itself may reach as high as 260°F.

Figure 78 shows the cool box and speakers being raised for installation in the locomotive and Figure 79 shows the unit installed in the locomotive as seem from inside the engine compartment looking forward towards the locomotive cab. To the left is the engine and exhaust manifold which extend forward under the cool box. The white material seen surrounding the cool box is 1-inch thick mineral wool insulation with a fiberglass cloth layer (Feratex) for physical protection. This layer performed very well except above the exhaust manifold were the very high temperatures (up to 1200°F) caused the fiberglass to darken. A heat shield is likely needed in this area to reduce the heat loading on the cool box due to the exhaust manifold.

Figure 78. The Cool Box Being Raised for Installation in the Locomotive

As mentioned above the cool box was provided with cooling air from the traction motor blowers to keep the speaker enclosures and the control speakers cool in the engine compartment.

Figure 79. The Cool Box Installed in the Locomotive

Figure 80 shows the arrangement for bringing the cooling air from the air compartment forward of the engine compartment to the cool box. A 6-inch flexible duct was routed to the traction motor blower duct in the air compartment. The other end of the flexible duct was fed to a distribution manifold. From the distribution manifold, shown in the figure, six 2-inch flexible ducts lead to six openings in the base of the cool box. Three of the ducts can be seen connecting to the base of the cool box. There are three symmetric connections on the other side of the unit. While this arrangement appeared to provide adequate cooling, future installations should have eight openings. Two additional openings should be installed on each side of the cool box at the base at the aft end. This is the area where the exhaust manifold comes close to the cool box and additional cooling could be helpful in preventing future problems.

The cool box and associated speaker enclosures can be seen after installation from the top of the locomotive in Figure 81. The figure shows the 10 speaker enclosures with their weather protective covers and conduit for bringing the speaker cables from the power amplifiers in the cab. The conduit also contained wires for the resistive thermal devices (RTD) that monitored the temperature on the speaker magnets. Figure 81 also shows the control microphones taped to the hood of the locomotive. Rather than bolt the microphone housings to the locomotive hood, it was decided to simply tape them in place to provide the flexibility in changing their location during the test if necessary.

Figure 82 shows the tachometer installation. A commercially available optical tachometer was attached to a bracket to enable it to shine on the engine flywheel where a reflective strip was placed. The tachometer provided pulses at the engine rotation rate and its entire harmonics through out the frequency range (0-250 Hz). It performed reliably throughout the test period.

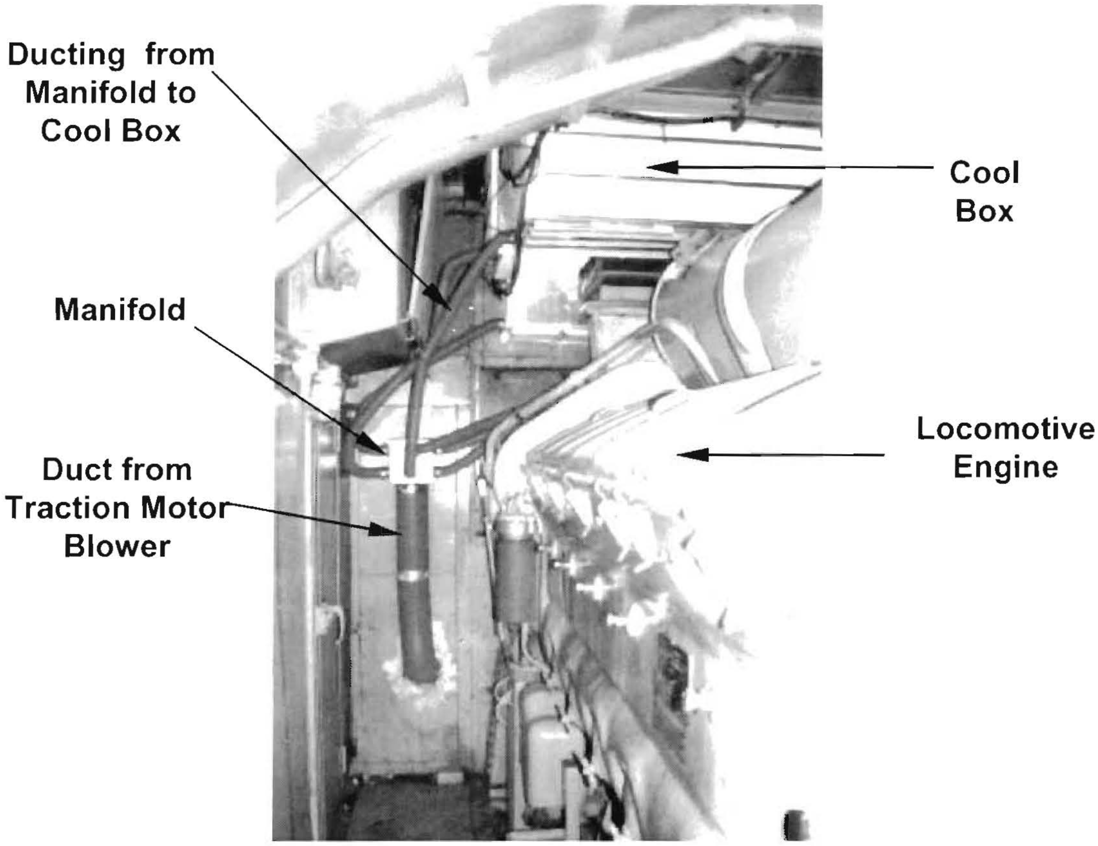


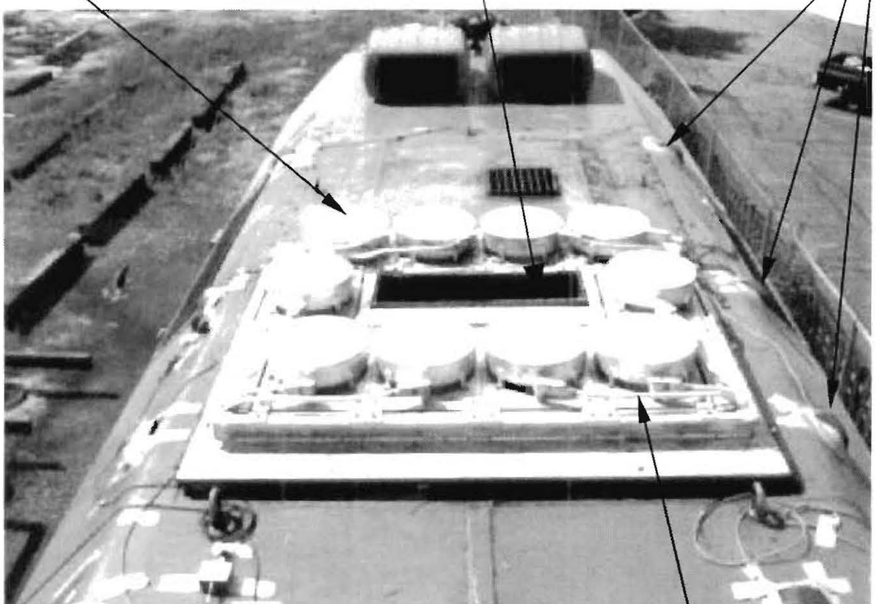
Figure 80. The Arrangement for Bringing Cooling Air to the Cool Box

The test site at the Chicago Metra 51st St. Yard is shown in Figure 83. The pictures show three views from the locomotive towards the far field microphones. The railroad was very helpful in keeping the tracks mostly free of cars during the test periods. Unfortunately, the railroad could do nothing about the background noise at the test site. The 51st St. Yard is located in an urban area in the south of Chicago and is next to the Dan Ryan expressway, which also contains a transit line in the median strip. The system performance was determined primarily by using measurements near the exhaust outlet, since there were high background noise levels and the presence of other noise sources on the locomotive. This was especially true for the determining the passive silencer performance. For the active system, which can be tested by turning it off and on, performance measurements were carried out at the far field microphones. However, there may still have been background contamination of the measurement since the measure of acoustic performance is the overall sound level and not the reduction of individual tones.

Control Speakers

Locomotive Exhaust Stack

Control Microphones



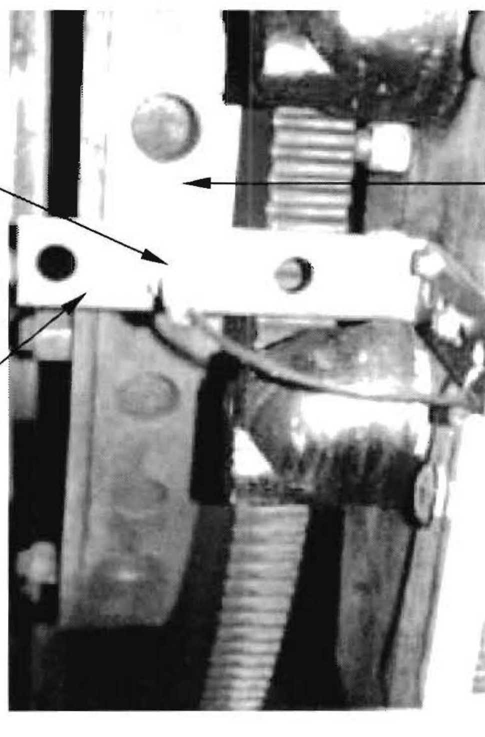
View Looking Forward

Conduit for Speaker and RTD Cables

Figure 81. The APECS Installed in the Locomotive as Seen From the Roof

Optical Tachometer

Engine Flywheel



Tachometer Bracket

Figure 82. The Tachometer Reference

Figure 83. The Test Site at the Chicago Metra 51st St. Yard

5.4 THE PERFORMANCE OF THE ACTIVE SYSTEM

Measurements of locomotive noise with the active system turned off and on were carried out at microphone positions 1 through 5 shown in Figure 75. Initially, the system performance was to be measured in the locomotive cab at the engineer's ear position with windows and doors closed. Unfortunately, the initial heating problems encountered with the power amplifiers (which were corrected) prevented measurements while running with the windows closed. After the heating problems were corrected, there was not enough time to repeat the cab measurements.

The system was designed to have concurrent, covert system identification and continuous adaptation. Plant identification (measurement of the 64 transfer functions between speakers and control microphones) was performed with a broad band random signal applied to the control speakers one at a time. The probe signal was set to be 6 dB below the broad band ambient noise levels in the control microphones. A complete plant measurement required about 10 to 15 minutes. During measurements, noise reduction performance began to degrade after about 30 minutes. When one of the DSP chips failed during the test, the software was changed that enabled the system to operate with only one DSP. Originally the system was designed so that one DSP performed the in-line control function and the other the concurrent, covert system identification. With only one DSP running, plant identification measurements could only be performed when the controller was off. However, the plant identification measurements were performed with the locomotive running. In general, plant measurements at throttle settings up to throttle 4 resulted in good performance.

Some degradation in performance was found with plant identification measurements at the higher throttle settings. Unfortunately, time restrictions did not allow further pursuit of this issue. Please note, however, that the plant identification software used in this test represents a significant advance in technology over that used during the original proof of principle test on the F59PHI locomotive at the SCRRA yard in Glendale, California. There, all plant identification measurements had to be made with the locomotive engine off.

The sound pressure level in microphone 5 on the roof of the locomotive is shown in Figures 84 through 91 with the active system on and off. The narrow band power spectra cover the range from 0 to 400 Hz. The control bandwidth of the active system was from 0 to 200 Hz. Consequently, reduction above 200 Hz was not expected. However, broader bandwidth is shown to display any out of band amplification that might have occurred. Since one of the goals is to reduce the A-weighted level of the exhaust noise, any out of band amplification of higher frequency exhaust noise would be undesirable because higher frequency noise contributes more strongly to the A-weighted noise level than low frequency noise.

All of the figures show the intense low frequency tonal energy characteristic of locomotive exhaust noise. The tones become significant just below 40 Hz, peak in magnitude near 100 to 150 Hz and then decline in magnitude. The figures show a dramatic reduction in the magnitude of these tones due to the active system. In some instances, the reduction of individual peaks is as much as 30 dB. At idle, an order of 20-dB reduction of low frequency tones was found, but also, unfortunately, evidence of out of band amplification in the vicinity of 200 Hz was also found. Out of band amplification is less of a problem at the other throttle settings, although, at high idle, there is more than preferred.

Another problem became apparent while the system was operating at the higher throttle settings. The controller began to drive the power amplifiers and control speakers near their limits. Consequently, to prevent distortion and the resulting out of band amplification, the leakage factor (control effort weighting) was increased to limit the amount of control. This is apparent in the gradual decrease in noise reduction with increased throttle setting. Improvements in performance at the higher throttle settings would require control speakers capable of larger cone displacement than the 12-inch McCauley's. Adding additional speakers is not an option because of space limitations. Note, however, that the current speaker and enclosure design represents a considerable improvement over that employed during the SCRRA proof of principle tests.

A final issue that should be addressed in future modifications of the system is the need for increased bandwidth. It is apparent from the results at throttle 6 and 8 that control of the tones above 200 Hz would be desirable, especially a tone at 250 Hz in a few of the traces. A resonant cavity in the passive silencer was designed to control this tone. It would be desirable to attempt to control this tone with the active system as well. In addition there are a number of other tones between 200 and 400 Hz in the spectra from the higher throttle settings that could potentially be controlled with the active system. To increase the bandwidth would require modifications to the speaker enclosures as well as increased computational capacity in the controller. Increasing the bandwidth will be productive only up to about 300 Hz. To go higher and achieve far field noise reduction comparable to that on the hood will require more control speakers, the addition of which are not possible due to space restrictions.

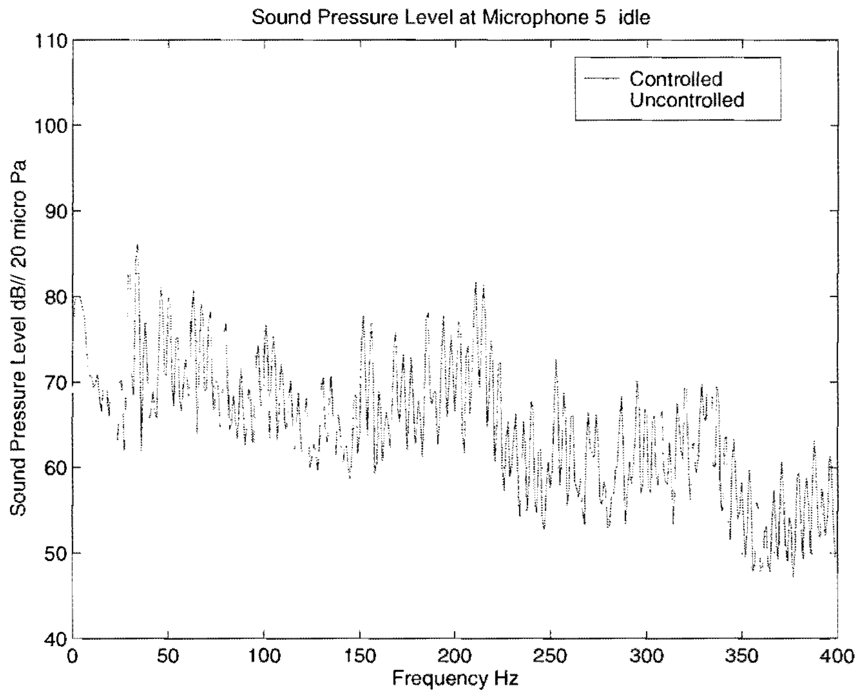


Figure 84. Sound Level at Low Idle at the Roof Microphone with the Active System On and Off

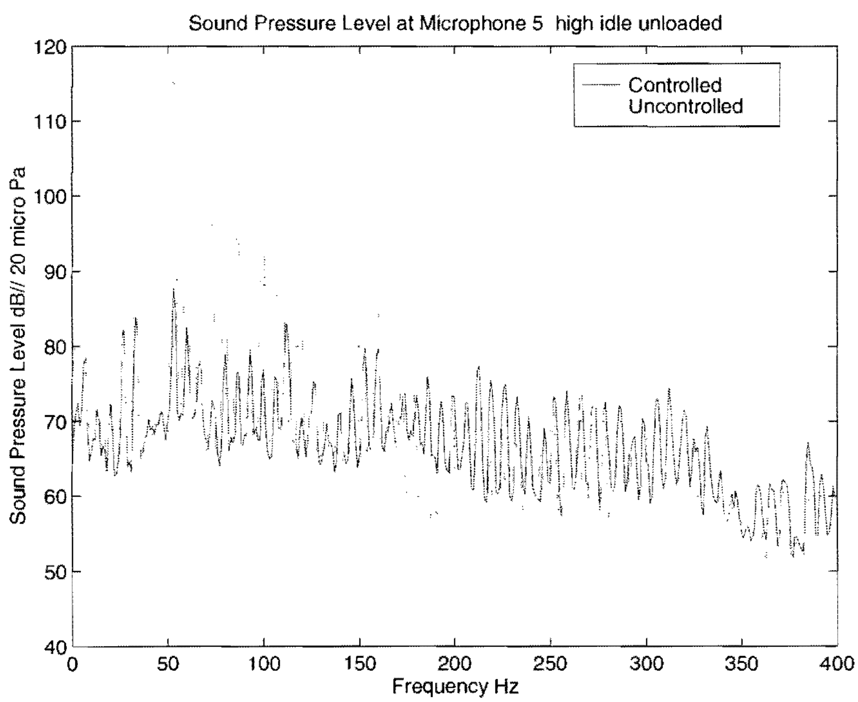


Figure 85. Sound Level at High Idle at the Roof Microphone with the Active System On and Off

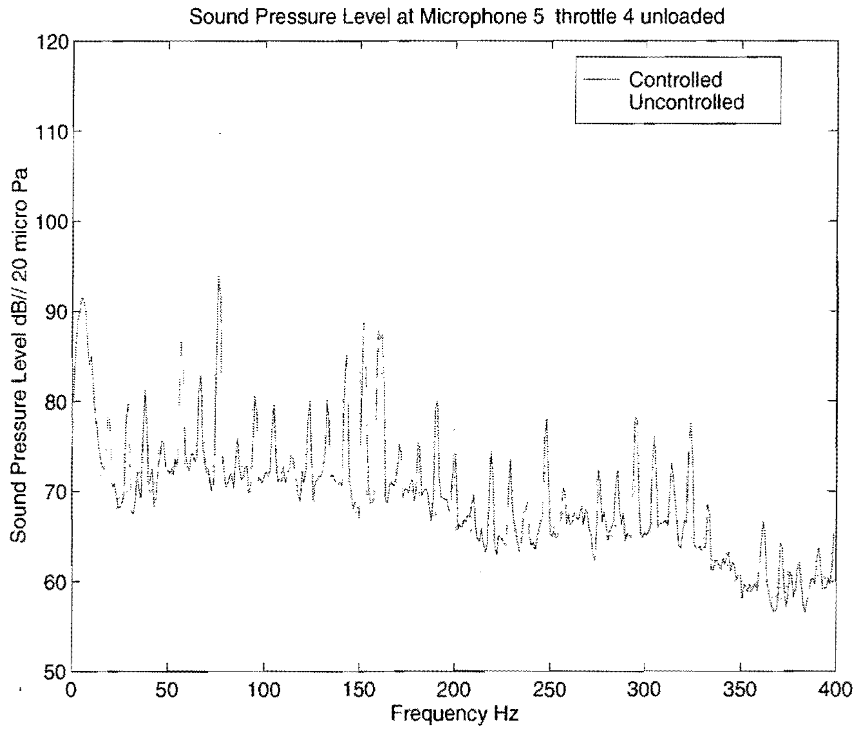


Figure 86. Sound Level at Throttle 4, Unloaded, at the Roof Microphone with the Active System On and Off

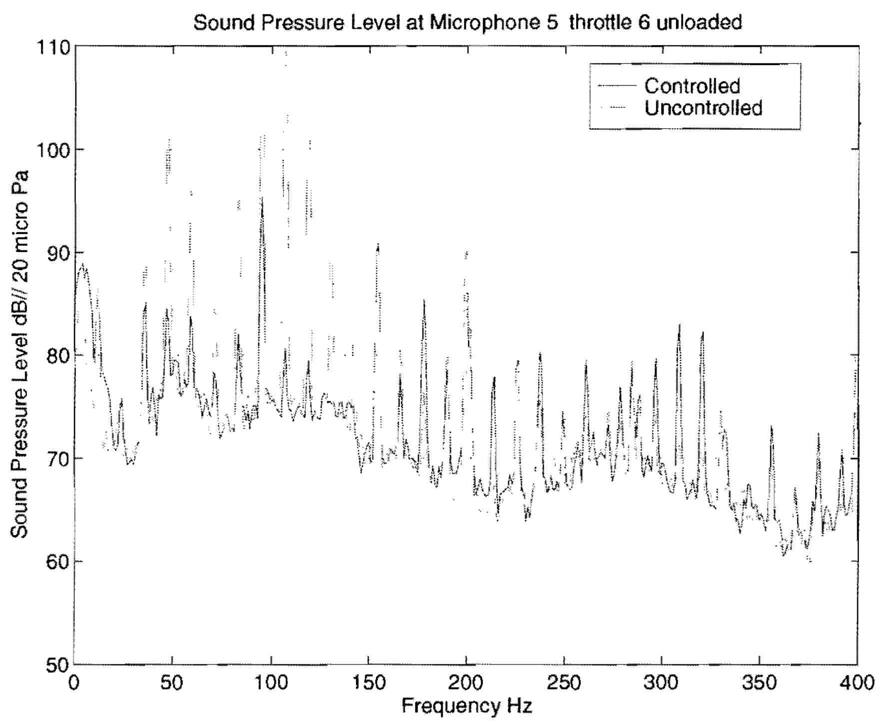


Figure 87. Sound Level at Throttle 6, Unloaded, at the Roof Microphone with the Active System On and Off

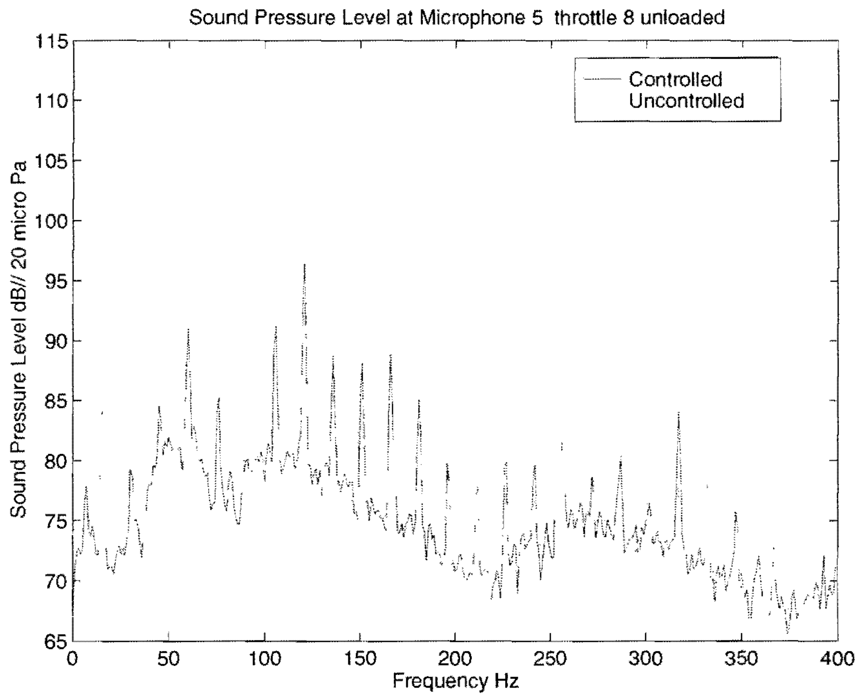


Figure 88. Sound Level at Throttle 8, Unloaded, at the Roof Microphone with the Active System On and Off

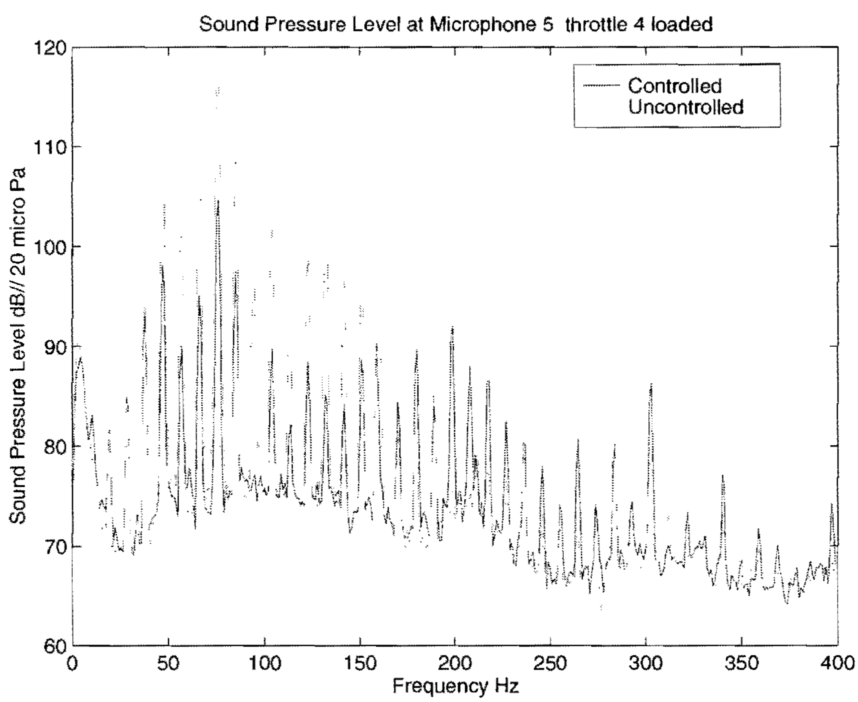


Figure 89. Sound Level at Throttle 4, Loaded, at the Roof Microphone with the Active System On and Off

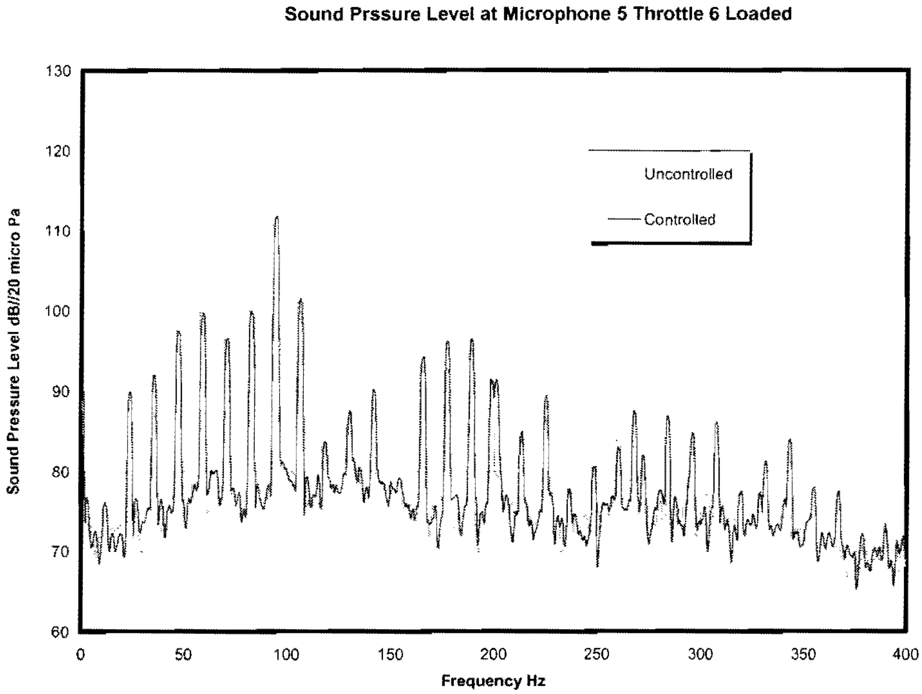


Figure 90. Sound Level at Throttle 6, Loaded, at the Roof Microphone with the Active System On and Off

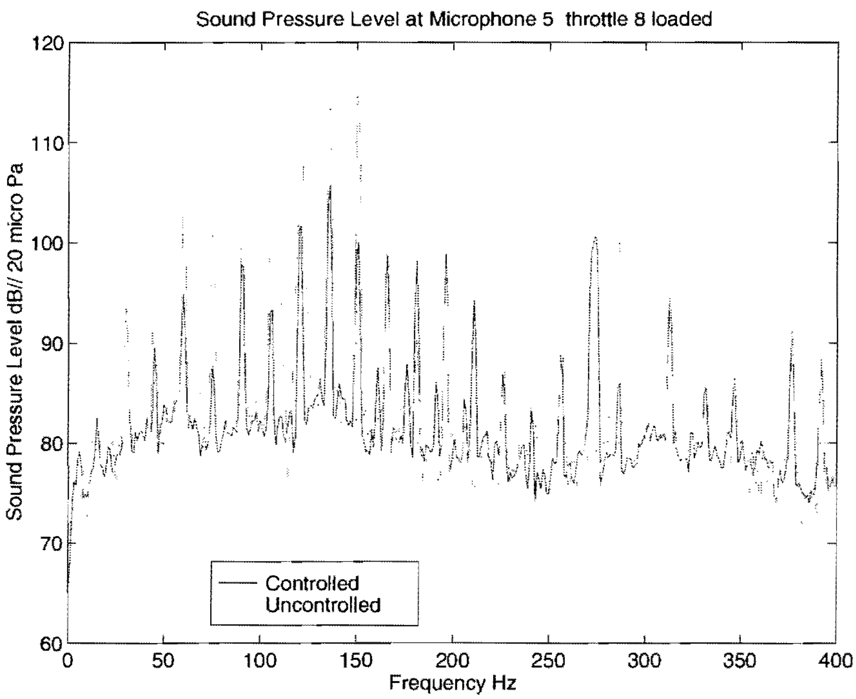


Figure 91. Sound Level at Throttle 8, Loaded, at the Roof Microphone with the Active System On and Off

The reductions shown in Figures 84 through 91 and the overall reductions in Table 10 were measured on the roof of the locomotive and one would expect the reductions to be less in the far field. Table 12 summarizes the overall noise reduction.

Table 12. Overall Noise Reduction [dB] of the Active System Below 200 Hz at All Microphone Locations

Throttle Setting	Mic. #1 100 ft 90°	Mic. #2 50 ft 90°	Mic. #3 100 ft 45°	Mic. #4 50 ft 45°	Mic. #5 4 ft Roof
Idle	4.0	7.1	4.3	4.7	13.2
High idle	11.5	16.5	10.2	16.3	22.9
Throttle 4 Unloaded	5.7	4.0	7.0	6.7	13.6
Throttle 6 Unloaded	2.4	0.9	3.7	1.8	13.4
Throttle 8 Unloaded	-5.5	-1.3	4.0	1.8	10.7
Throttle 4 Loaded	5.7	6.5	7.7	7.1	12.3
Throttle 6 Loaded	2.2	3.7	4.7	4.5	8.1
Throttle 8 Loaded	1.5	2.5	3.9	3.7	9.6

As the table shows the reductions observed at the far field microphones are generally less than those found at the roof. There could be a number of reasons for this. Background noise was quite high at the site. These reductions are simply the change in the integrated energy below 200 Hz. At the more remote microphones the background could influence the integrated energy. Also there are many sources of tonal energy from the locomotive. Microphone 5 was on the roof and was dominated by the exhaust. At other microphone locations other sources not addressed by the APECS could be contributing to the signature. In general one uncontrolled tone is all it takes to badly compromise the measurement of noise reduction performance. The third reason for the reduced performance at the far field microphones is that one expects the noise reduction at the remote microphones to be somewhat less than observed on the roof. Controlling the noise at eight locations on the roof does not guarantee that the control will be as good at remote locations in the far field. In the past, less than 5-dB difference has been observed between the roof and the far field. At some locations and throttle settings in the table the difference is larger than this, indicating that other factors are at work. Original plans were to move the control microphones to alternate positions on the roof of the locomotive and observe the change in far field noise reduction performance. Unfortunately time constraints prevented this from happening. Future versions of the system should consider adding additional control microphones and measuring the effect of control microphone placement and number on the far field noise reduction performance.

5.5 THE PASSIVE SYSTEM

The passive system designed to work with the active system was described in Section 2.4. It is designed to suppress high frequency noises of 200 Hz and above. However, the actual design performance of the OEM silencer was a very large unknown. There were no insertion loss data available either from EMD or the silencer manufacturer, Universal Silencer. Since the OEM silencer was removed and replaced with the new APECS silencer, it would have been very useful to know in advance any loss in IL performance.

The difference in insertion loss of the new silencer and the OEM by acquiring noise data was measured at microphone position 5 on the roof approximately 4 feet to the side of the exhaust stack. Measurements at other microphone locations were too contaminated by background noise and noise from other sources to rely on them to quantify the silencer IL. The measurements at microphone position 5 are shown along with background noise in Figures 92 through 100. Examining the data carefully, especially for the loaded cases, evidence of the tuned labyrinth doing its job at 250 Hz can be seen where it enhances the insertion loss. Figure 101 compares the change in insertion loss between the two silencers for the loaded and unloaded operating conditions. There the IL difference between the OEM and new silencer has been averaged over 250 Hz bands to smooth what would otherwise be a very jagged curve. The loaded cases cluster together and the unloaded cases cluster at an IL difference that is about 1-2 dB higher. Also shown in the figure is the design insertion loss. If the OEM silencer IL is as small as expected then the measured difference should be close to the design IL. The measurements are 10 dB or so below the theoretical IL above 1 kHz, and are close to predicted below 1 kHz. It is not certain whether the lower performance above 1 kHz is, in fact, true or whether other sources on the locomotive prevent measurement of the exhaust noise reductions greater than 10 dB. Thus, the IL from 300 to 3000 Hz was measured at 5-12 dB, less than desired but still excellent.

The fact that the IL measured under load is less than when measured unloaded is not surprising. During the loaded tests the dynamic brake fan was running to cool the resistor grids, which were dissipating the locomotive power. The fan is noisy and would contaminate the measurement, preventing our observing the full noise reduction from the new silencer.

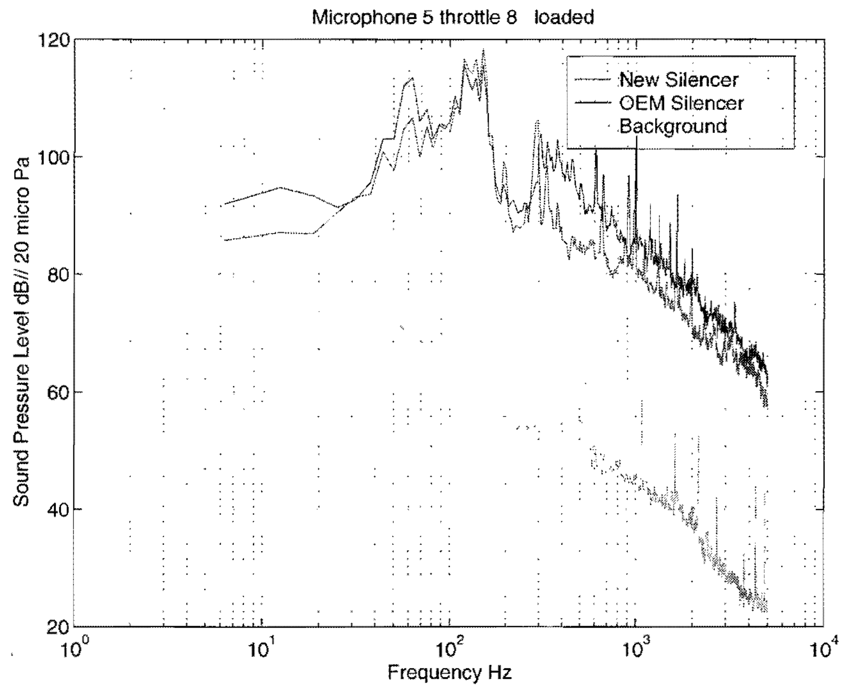


Figure 92. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 8 Loaded

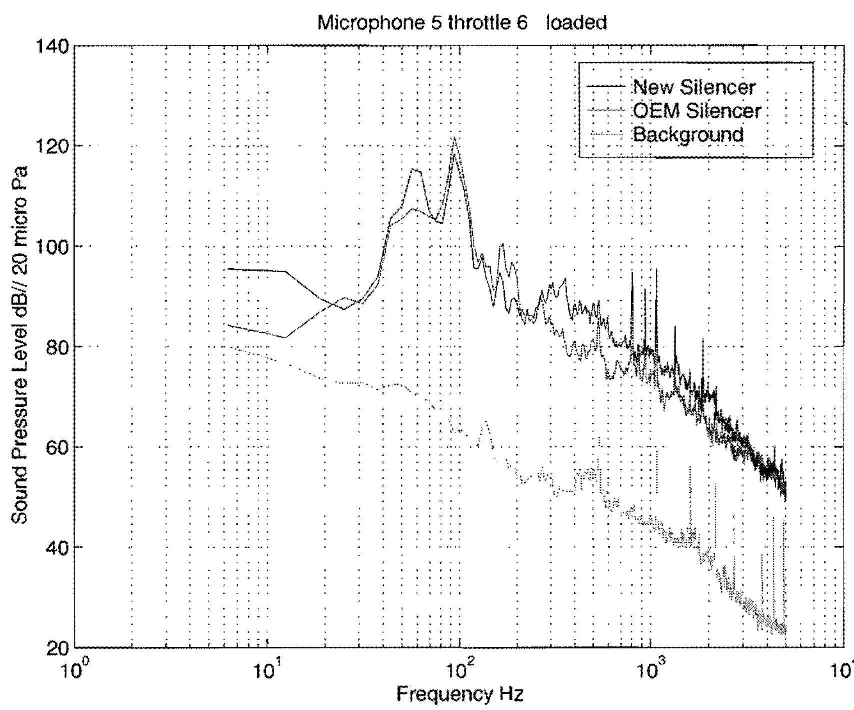


Figure 93. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 6 Loaded

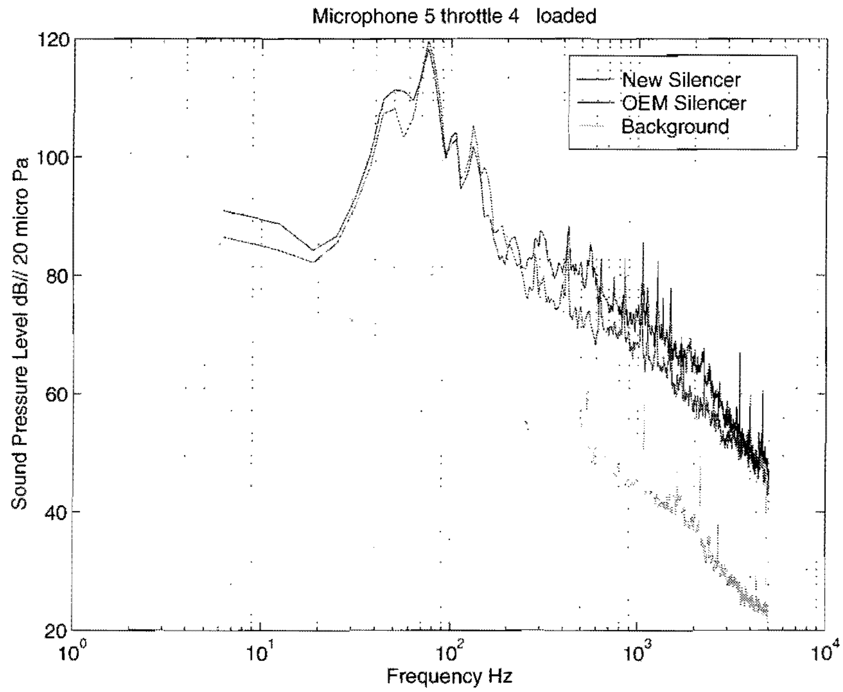


Figure 94. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 4 Loaded

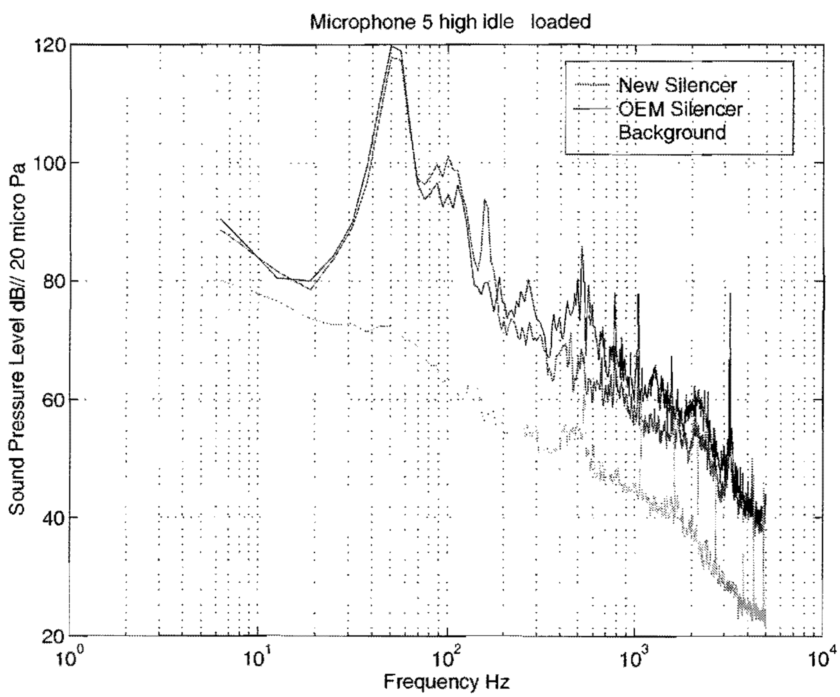


Figure 95. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 1 Loaded

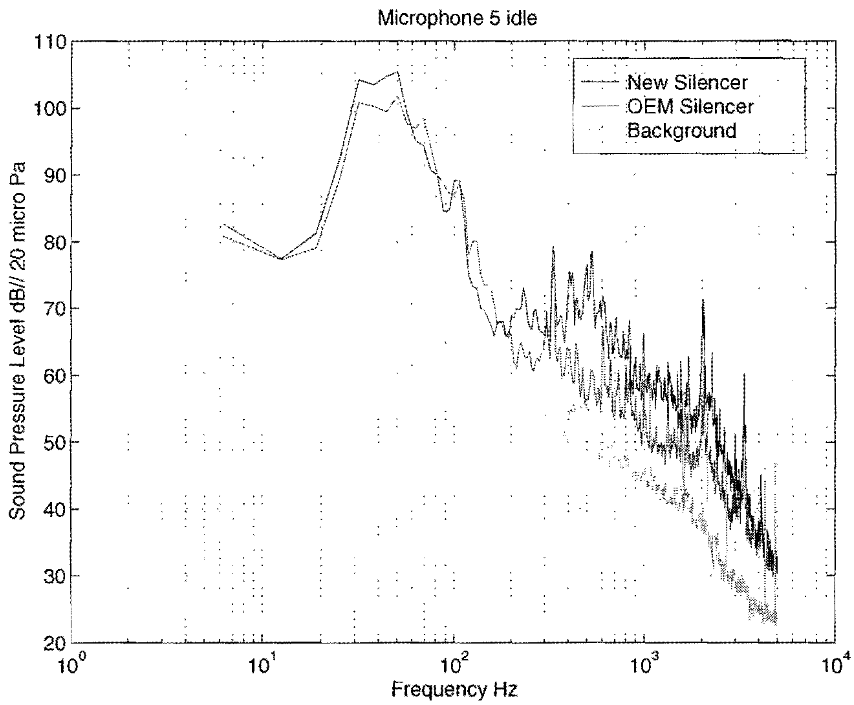


Figure 96. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Idle

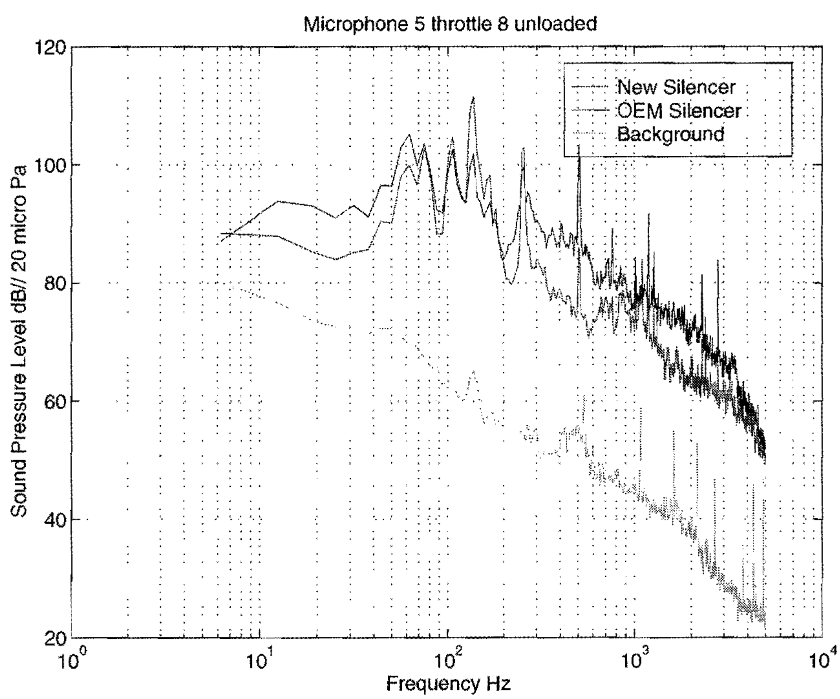


Figure 97. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 8 Unloaded

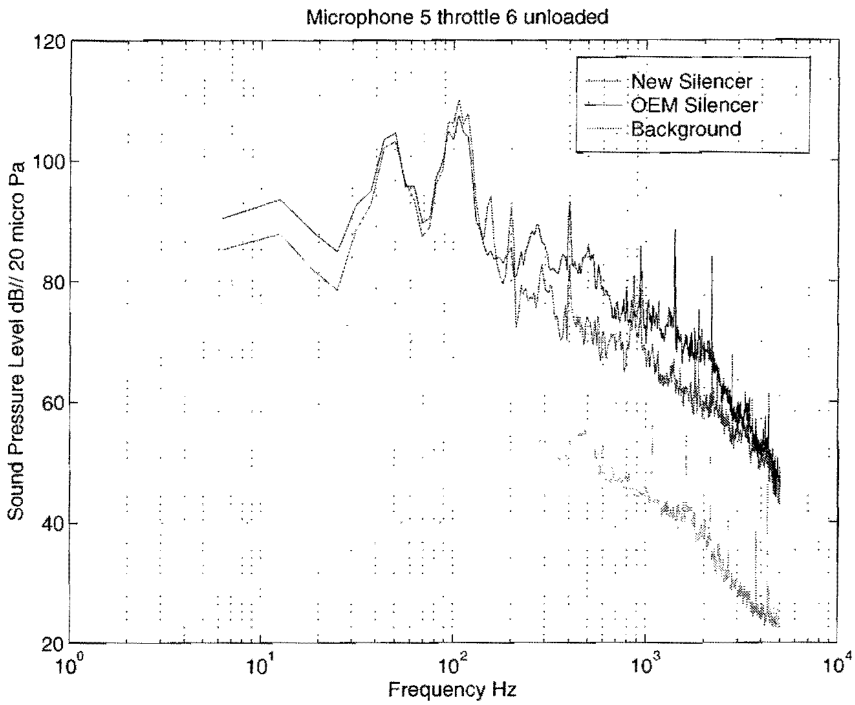


Figure 98. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 6 Unloaded

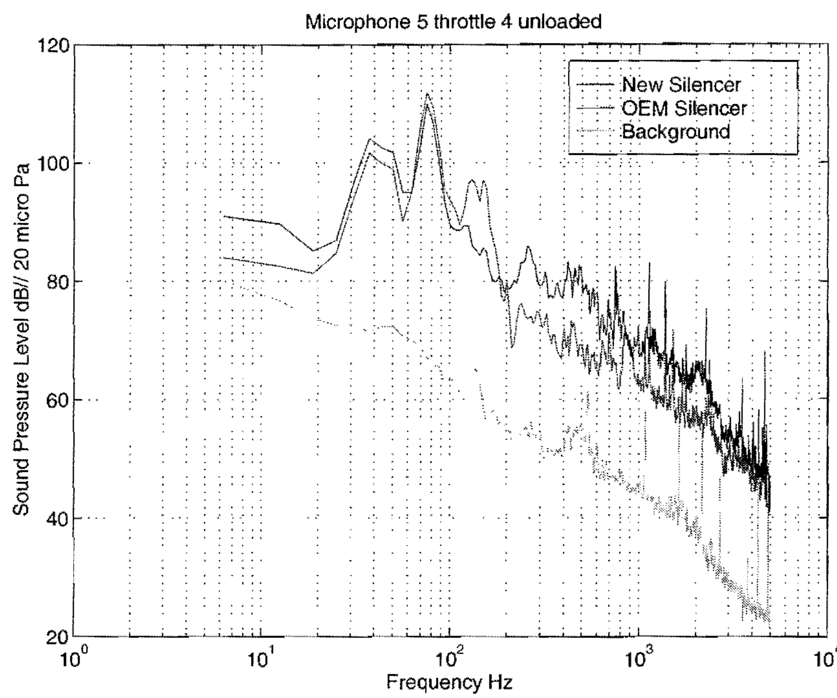


Figure 99. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers Throttle 4 Unloaded

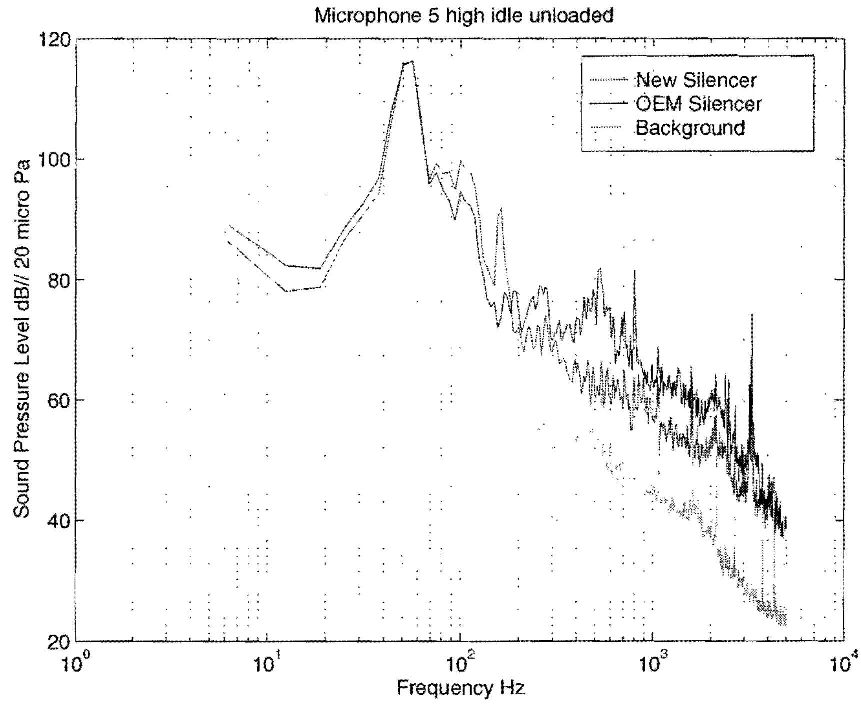


Figure 100. Comparison of Noise Levels at Microphone 5 with the OEM and New Silencers High Idle

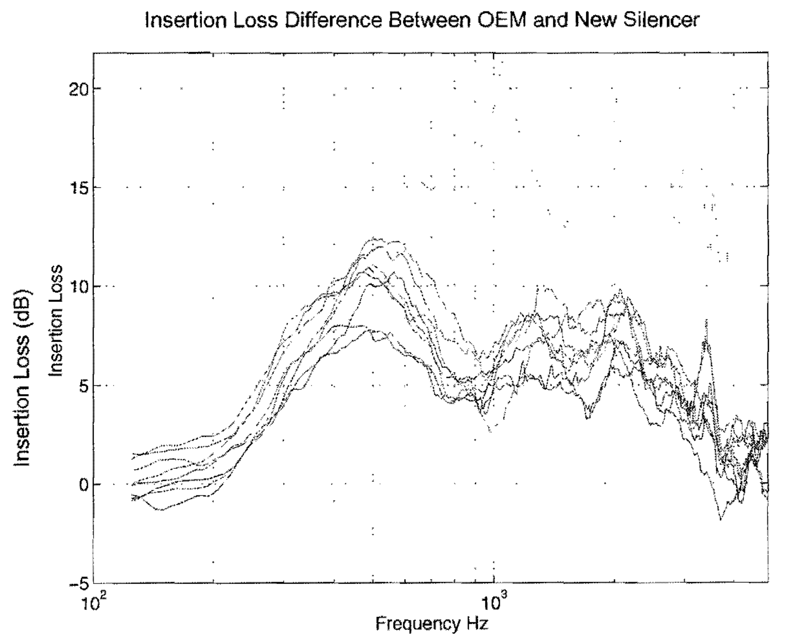


Figure 101. The Average Insertion Loss Difference Between the OEM and New Silencer

In the course of the acoustic measurements EMD personnel measured the backpressure due to the new silencer using a water manometer. It varied from 6 to 8 I
n. H_2O at throttle 8 full load. Although this is above the specification of 5 inches H_2O, it is not clear whether this represents a problem or not. Flow velocities in the silencer outlet at throttle 8 full load were found to be approximately 60 meters/second.

5.6 OVERALL LEVELS

The bottom line of this program is the reduction in the overall A-weighted locomotive exhaust noise. The final result is illustrated in Table 13. In this table the spectrum at the roof microphone position 5 was measured and the A-weighted level was computed from the measured data in two parts: one from 25 Hz to 400 Hz and one from 400 Hz to 5 kHz. The active system is providing most of the noise reduction (and occasionally a little out of band noise amplification) in the former and the passive silencer provides most of the noise reduction in the latter. The first two columns in Table 13 show the A-weighted levels in the two frequency ranges for the APECS for nine locomotive operating conditions. The third column shows the overall level for the two frequency ranges combined. The next three columns show the same information for the original locomotive, and the last column is the A-weighted noise reduction. For all of the operating conditions the results are very close to achieving the 10-dBA-noise reduction goal. In addition to the measurements the reduction in sound level and the improvement in sound quality was universally noted by human observers at the site.

It was not possible to make the same comparisons as shown in Table 13 at the far field microphone locations, because of contamination due to background noise and other noise sources, e.g., the dynamic brake fans. This contamination is of primary concern for the noise reduction due to the passive system. The change in the tones at the far field microphones due to the active system can usually be measured with little difficulty. Thus, the measured data was used to compute the A-weighted noise reduction due to the active system at each far field microphone. In general this was less than the noise reduction at the roof microphone, position 5. Then the change in the noise reduction at microphone 5 was computed if the active system noise reduction were reduced to correspond to the active system noise reduction achieved at the selected microphone. The implicit assumption in this approach is that the IL of the passive silencer is the same on the top of the roof and at the far field locations, a reasonable assumption. Table 14 presents the results. Except for idle, the noise reductions are less at each microphone position for each operating condition. However, they are still quite respectable. Note again that the loaded noise reductions may be underestimated. Recall that the IL difference for the loaded operating conditions was 1 to 2 dB lower than for the unloaded conditions. The two operating conditions should have shown similar IL. The Loaded IL measurements may indeed be contaminated by the dynamic brake fan noise. Especially for throttle 8 where, as Table 13 shows, the high frequency noise dominates the controlled A-weighted level, an increase in silencer IL would have a direct effect in increasing the final overall noise reduction.

Table 13. The Change in A-Weighted Sound Level [dB] Due to the APECS as Measured At Microphone #5

Roof Microphone		Active Control/New Silencer			OEM Silencer Alone			
Throttle	Load	A -weighted Level 25 Hz-400Hz	A -weighted Level 400Hz-5 kHz	A-weighted Overall Level 25 Hz-5kHz	A -weighted Level 25 Hz-400Hz	A-weighted Level 400Hz-5 kHz	A-weighted Overall Level 25 Hz-5kHz	Overall Noise Reduction dBA
idle	unloaded	83	79.6	84.6	83.3	88.6	89.7	5.1
hi idle	unloaded	83.8	83.2	86.5	91.8	92.6	95.2	8.7
t4	unloaded	86.8	89	91.0	92.5	96.5	98.0	6.9
t6	unloaded	90.2	92.7	94.6	97.1	100.8	102.3	7.7
t8	unloaded	99.2	97.6	101.5	100.1	106.3	107.2	5.8
t4	loaded	92.5	93.5	96.0	99.8	99	102.4	6.4
t6	loaded	98.8	101	103.0	103.6	105	107.4	4.3
t8	loaded	102.7	106.2	107.8	110.2	112.8	114.7	6.9

Table 14. Estimated Noise Reduction [dB] at the Far Field Microphones

Throttle	Load	mic 5	mic 1	mic 2	mic 3	mic 4
idle	unloaded	5.1	5.2	6.5	6.1	6.4
hi idle	unloaded	8.7	6.5	8.9	5.8	7.4
t4	unloaded	6.9	5.2	4.1	5.6	5.4
t6	unloaded	7.7	6.6	5.6	6.4	6.0
t8	unloaded	5.8	-0.1	3.7	6.2	4.7
t4	loaded	6.4	5.2	4.1	4.7	4.6
t6	loaded	4.3	-1.9	1.2	2.8	1.7
t8	loaded	6.9	2.9	6.5	6.6	6.6

6. MODIFICATIONS REQUIRED FOR IN-SERVICE TESTING

The goal of the testing described in Section 5 was to obtain data on the operational and acoustic performance of the APECS in a locomotive under stationary conditions. The system as tested will require a number of modifications before it can be released for in-service testing. It was originally planned to put the test locomotive with the APECS into service for 6 months. Because of funding constraints it was decided to install the APECS in the locomotive without making all of the modifications necessary for in-service operation. The success of this preliminary test indicates that taking the next step to an in-service test is both feasible and advisable. Areas that need to be addressed before the system can be placed in service for an extended time include:

Installation of a higher capacity auxiliary generator to provide system power

The existing 18 kW auxiliary generator on the F40 test locomotive does not have enough excess capacity to power the APECS, estimated at approximately 4 kW. An alternate 22 kW auxiliary generator is available, which can be installed in place of the existing generator. It would have more than enough excess capacity to power the APECS.

Installation of a higher capacity alternator blower wheel to provide cool box cooling air

For the APECS test cooling air was supplied by tapping into the traction motor blowers. Since the test was stationary, using this air presented no cooling problems for the locomotive. In operation the traction motor blowers do not have enough excess capacity to provide cooling air for the traction motors and the APECS. By replacing the blower wheel on the alternator blower with one with higher capacity sufficient cooling air can be bled off for the APECS without compromising the cooling of the main alternator.

The addition of two openings for cooling air injection at the rear of the cool box

Airflow at the rear of the cool box seemed somewhat restricted. Although, for the current test, there were no observed cooling problems in the speaker enclosures, additional openings for the injection of cooling air at the rear of the cool box are advised.

Improved thermal isolation material on the lower rear of the cool box and a heat shield covering the exhaust manifold

The heat insulation material (mineral wool) used in the prototype silencer had a protective coating on the outside surface that under the intense heat of the silencer began to deteriorate. For in-service testing mineral wool insulation is still a good choice because of its ability to accept very high temperatures. However, expanded metal should be used in place of the protective coating to contain and protect the insulation.

Design of a thermal management system for the electronics enclosure to better control hot and cold environments

The tests of the prototype were performed in the summer and cooling fans directed air to the power amplifiers to keep them and the associated electronics, DSPs, I/O boards, RTD boards and host computer cool. A thermal management system should be developed that senses the air temperature in the electronics enclosure and regulates the flow to keep the temperature within specific bounds. In the winter due to the use of commercial grade electronics it will be necessary during cold start up and potentially during normal operation to heat the air to keep it above 32°F.

Development of self-booting software to allow for autonomous operation of the controller

For the prototype test an operator was present to boot the host computer and start the controller. The software for autonomous operation was not available for this test. That software would be essential for in-service testing. It will need to be able to automatically boot the host computer, check for system faults and restart the controller in the event of a shutdown.

Development of software and the hardware interface to monitor the locomotive throttle setting and adjust leakage based on the setting

During the prototype testing the controller attempted to drive the control speakers beyond limits set in the controller. To prevent distortion, the leakage in the controller was adjusted to prevent over driving the system. Leakage is similar to control effort weighting, which penalizes the controller for driving the control actuators too hard. The amount of leakage depended on the throttle setting. Consequently provision will have to be made for monitoring throttle setting and adjusting leakage in the control algorithm based on that setting. This may be unnecessary if more acoustic output can be obtained from the speakers so that the controller will tend to drive them within allowable limits.

Development of software to provide a basic self-monitoring system

Basic self monitoring system needs to be developed that monitors the control speaker performance and temperature, the control microphone signals and the reference signal from the tachometer and takes appropriate action if faults are found. The action might include shutting down the system completely or shutting down a drive channel or control microphone and reconfiguring the system. There should also be a means for the user to interrogate the system and identify the fault.

Shock isolation of the electronics cabinet

To ensure long life of the electronics the electronics cabinet should be vibration isolated from the locomotive.

Other Improvements

In addition to the above changes to improve reliability and durability of the system, improved performance could be obtained if the following were carried out:

- Replace the current speakers with speakers with greater cone displacement
- Increase the bandwidth of the system from 200 Hz to 300 or 400 Hz, modifying the speaker enclosures to increase their bandwidth and providing additional computation hardware to allow the system to go to a higher sampling rate
- Increase the number and optimize the placement of control microphones for better far field control
- Redesign the passive silencer for reduced back pressure and improved insertion loss

7. CONCLUSIONS

The APECS was installed on an F40PH-2 locomotive. A-weighted exhaust noise reductions for most of the operating conditions were within a few decibels of the 10-dBA goal. Once up and running, the system performed flawlessly, providing audible reductions in locomotive exhaust noise. The control speakers in their protective cool box proved to be very durable even in the hostile environment in the locomotive engine compartment. Cooling air requirements were approximately 700 CFM. The passive silencer performed well acoustically, and except for the protective cover on the thermal insulation which began to melt at high temperature, also performed well mechanically, and the thermal insulation provided the needed thermal protection to the cool box. Backpressure was somewhat higher than specified for the engine. It is not known at this time if this is a problem or not. The installation of the cool box/control speaker assembly and the silencer went smoothly with few fitting problems. The optical tachometer performed well throughout the test. The control microphones also performed well after a problem with a DC offset was corrected. Cooling problems with the electronics enclosure were encountered, due primarily to the power amplifiers. Proper temperature control (both hot and cold) needs to be addressed before the system can be placed in service for any length of time. One of the DSP chips in the controller failed near the end of the test. It is believed that the chip was damaged during the time that repairs were being made to the electronics to deal with the power amplifier-heating problem. The failure probably did not occur as a consequence of normal operation of the system. Otherwise, the controller performed flawlessly running on 74 VDC locomotive power throughout the test. The signal input/output boards functioned with no problems during the test, including the RTD boards.

Technology developments that proved to be successful in this phase were:

- The thermal management system of the loud speakers through the use of a cool box and approximately 700 CFM of cooling air provided by the traction motor blower

- Inexpensive microphones that adequately measured sound pressure in the presence of roof vibration

- Significant improvement in low frequency noise reduction especially at idle and the higher loaded throttle settings where it was not possible to obtain good performance before

- An improved passive silencer that performed near to its design but utilizing maintainable screening

- An improved controller that allowed for the measurement and covert simultaneous update of the plant transfer functions

- An improved controller that allowed for operation over a broader frequency band and for the control of more exhaust tones, and

- A low-impact design that allowed for installation in the locomotive with only minor modifications.

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