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PASSIVE ABSORBER NOISE REDUCTION FOR CLEANROOM FANS

Jack B. Evans, P.E.

Engineered Vibration Acoustics & Noise Solutions
1705 W Koenig Lane
Austin, Texas 78756 USA
Evans@JEAcoustics.com

Abstract

A recently constructed demonstration microelectronics cleanroom was upgraded to accommodate new generation semiconductor fabrication machine tools. The ambient noise level due to HEPA filter recirculation fans exceeded machine tool manufacturers' permissible background sound level. The existing 78(+) dBC ambient sound level in the cleanroom had to be reduced to 75 dBC. Fifteen cleanroom recirculating air handlers, each with very short discharge ducts into pressurized HEPA filter membrane plenum modules, generated noise with a significant 125 Hz 1/3 octave tone. Dominant fan blade passage tones needed reduction.

Space and budget limitations restricted possibilities for modification of cleanroom walls, HEPA filter ceiling, or recirculating air handlers. No external attenuators could fit between the AHUs and HEPA modules. Internal passive absorptive attenuation was retrofit into the plug-fan plenums to reduce discharge noise. AHU inlet plenum wall absorption was added. Absorption was installed in mechanical mezzanine return air path to change room effect.

Using only passive absorber products that were totally mylar encapsulated to meet cleanroom requirements, recirculating plenum fan discharge and return noise was reduced just enough to meet the 75 dBC criterion. A plan and section of facility with sound levels before and after absorptive noise treatment is included.

1.0 INTRODUCTION

In the process of upgrading the cleanroom facility to accommodate a new generation stepper and other microelectronics machine tools, noise and vibration validation measurements were conducted to compare existing conditions versus criteria. Results showed that structural vibration met all criteria, but continuous ambient noise exceeded two equipment manufacturers' 75 dBC permissible background noise criteria[1].

The cleanroom facility is arranged with a mechanical mezzanine above the cleanroom, and a sub-fab support space below. The facility structure is isolated from the surrounding manufacturing building. Air handlers in the mezzanine supply air via short discharge ducts into pressurized HEPA filter plenum enclosures. Air flows from the filter modules, through the cleanroom, and into the sub-fab via perforated access floor tiles on an open waffle structural floor. Air is returned from the sub-fab via vertical return air chases that connect the sub-fab to the mezzanine, into open AHU inlets.

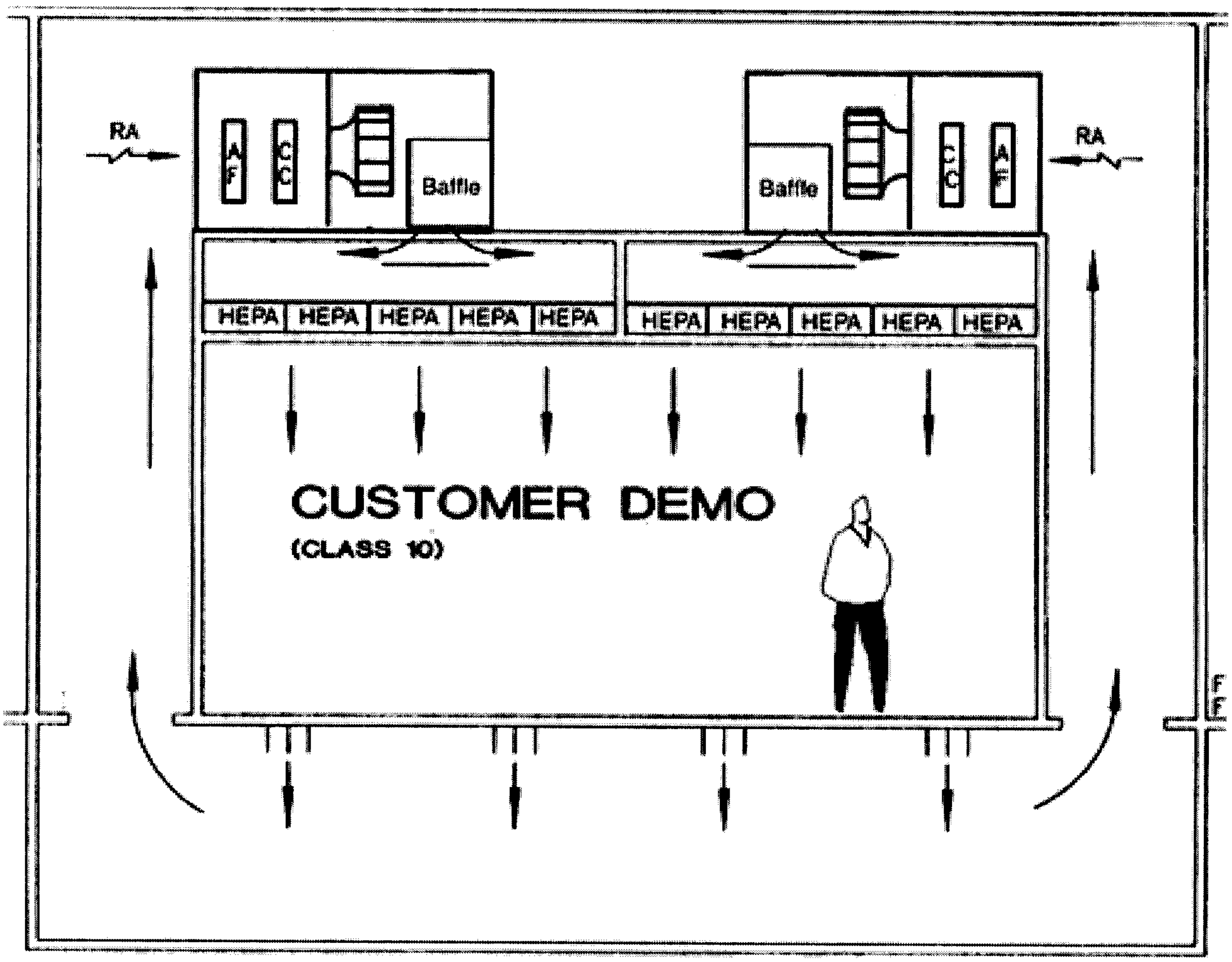

FIGURE 1: AIRFLOW SCHEMATIC

Measurements were conducted with a Larson-Davis 2900 1/3 octave integrating spectrum analyzer[2,3], with an ANSI Type I precision 1/2" microphone and pre-amplifier. The empty cleanroom (before installation of equipment) was 78 - 80 dBC, with significant tones in the 16 Hz and 125 Hz 1/3 octave bands. With (low frequency) C-weighting[4], the 125 Hz 1/3 octave is more critical than (network attenuated) frequencies below 50 Hz[5]. Prominent fan blade passage tones in the 125 Hz band were identified as dominant noise sources. Due to the reverberant nature of the cleanroom, measurements did not confirm whether the major fan noise contribution was in the discharge, return air, or both paths. Dual noise reduction strategies were selected; to focus on 125 Hz noise reduction, while attenuating noise over a broad frequency range.

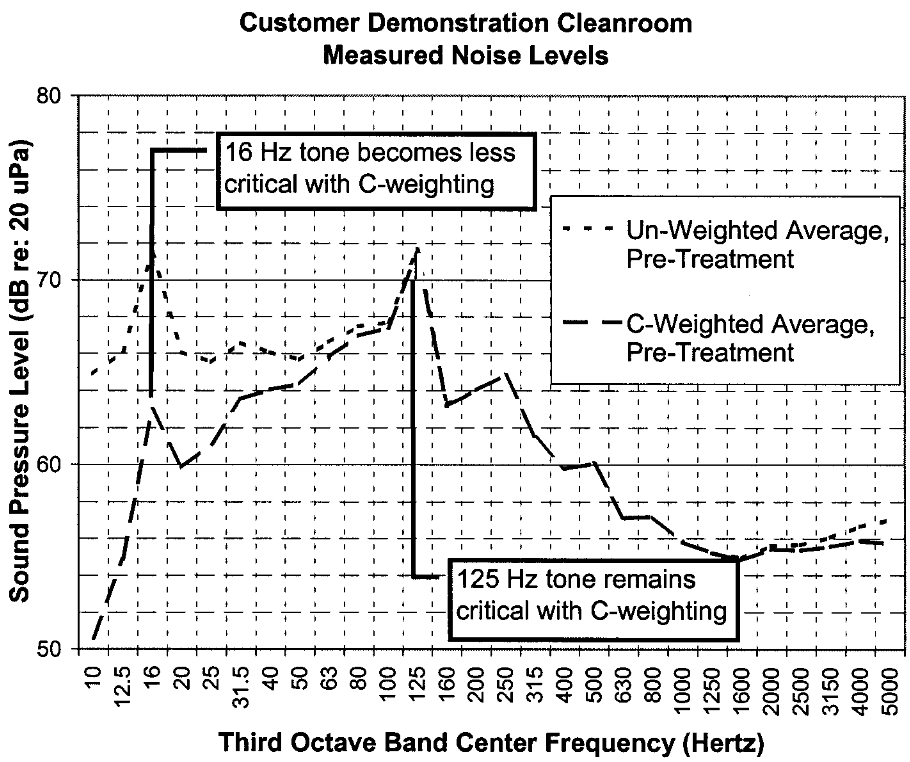

FIGURE 2: EFFECT OF C-WEIGHTING ON CRITICAL TONES

A noise attenuation solution had to be developed that would reduce the overall noise level 3-5 dB, without requiring modification of the air handler and HEPA filter configurations. Demolition in the cleanroom or mezzanine would be too dirty. Space constraints prevented insertion of noise control elements between the AHU and filter plenums. Project parameters eliminated active sound cancellation and passive duct attenuators, leading to adoption of classic acoustical absorption to achieve goals.

2.0 METHODOLOGY / OUTLINE OF THEORY

Active noise cancellation was considered, since it can be effective on strong tones and broadband low frequency noise. An experimental cancellation system was set up on a single AHU discharge to validate the conceptual design. It was moderately successful on the fan tone in the discharge, but unattenuated fan noise in the return air path remained significant. Implementation of cancellation in supply and return paths would require two cancellation systems per air handler, which would exceed the budget.

External duct attenuators were considered to attenuate fan noise, but the cost and time required to make space-accommodating modifications exceeded the budget.

Existing AHU internal sound attenuation included acoustical liner in the fan plenum, but lower frequency effectiveness was limited by the 50 mm thickness (small absorption coefficients). Splitter baffles within the air handler for discharge attenuation are isolated from the fan by a thin, lightweight wall. "Breakout" flanking through the wall allows noise to essentially bypass the baffles. For the absorption concept to work, this deficiency had to be corrected.

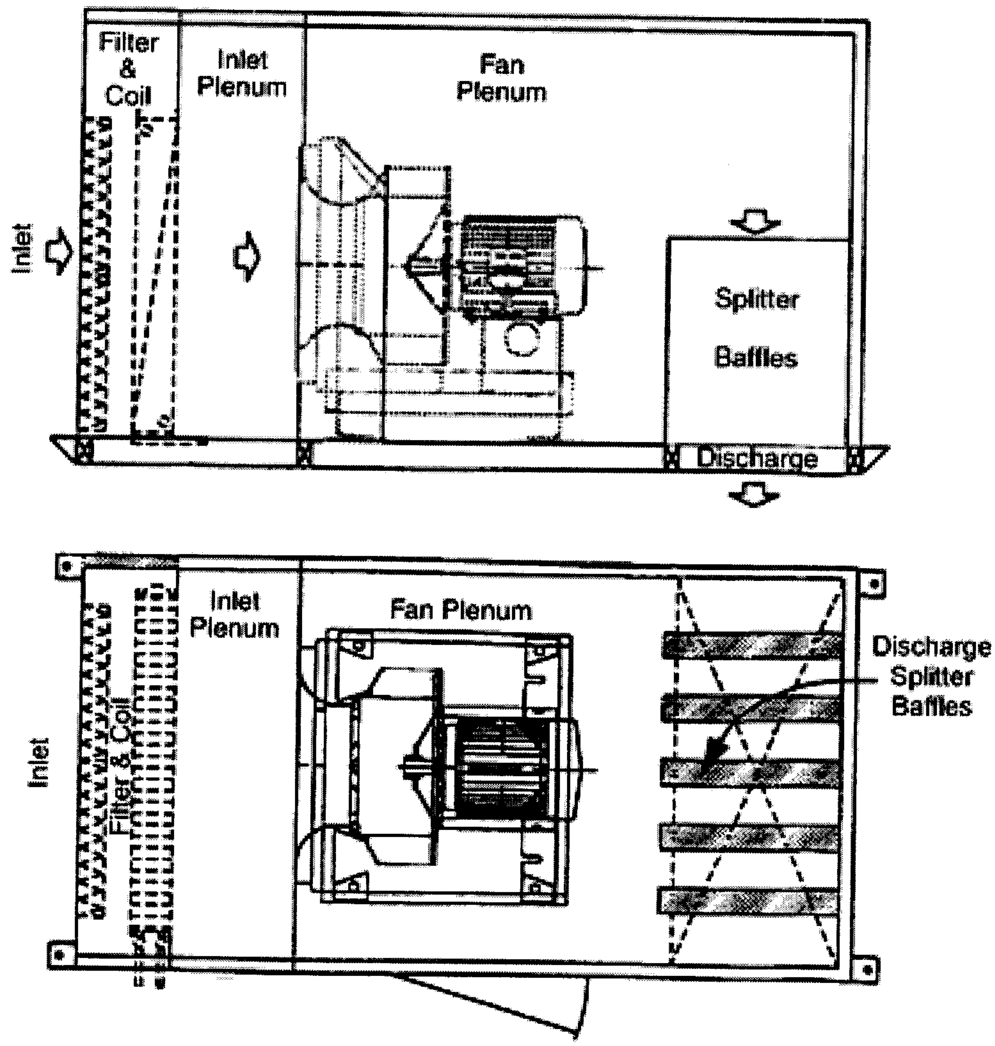

FIGURE 3: EXISTING AHU - SECTION (TOP) AND PLAN (BOTTOM)

Additional absorptive noise reduction was considered in the mechanical mezzanine for return inlet noise and in the AHU's internal plenums, after the earlier measurements and the cancellation demonstration showed approximately equivalent supply and return noise contributions. Up to 5 dB of noise reduction was required, and was considered feasible, since the reverberant field contribution to the overall noise level was 6-8 dB, based on estimate of absorption noise reduction[6]:

$$NR = 10 * \log(A_2/A_1)$$

where:

NR = reduction in reverberant sound level in decibels between two conditions of room absorption

A_1 = total absorption in square meters or square feet initially present in room (sum of room surface areas times their absorption coefficients)

A_2 = total absorption in square meters or square feet after new absorbing material is added

Surface absorption can reduce reverberant noise build-up (change room effect). Longer wavelength, low frequency noise needs greater absorber depth, for a given surface area, to achieve larger absorption coefficients[7]. The mechanical mezzanine and return air chase had no absorptive surfaces. By increasing thickness of acoustic liner in the fan plenum, adding acoustic liner in other modules of the air handler, and installing acoustical surface finish on room walls near the air handler inlets, supply and return noise could be reduced.

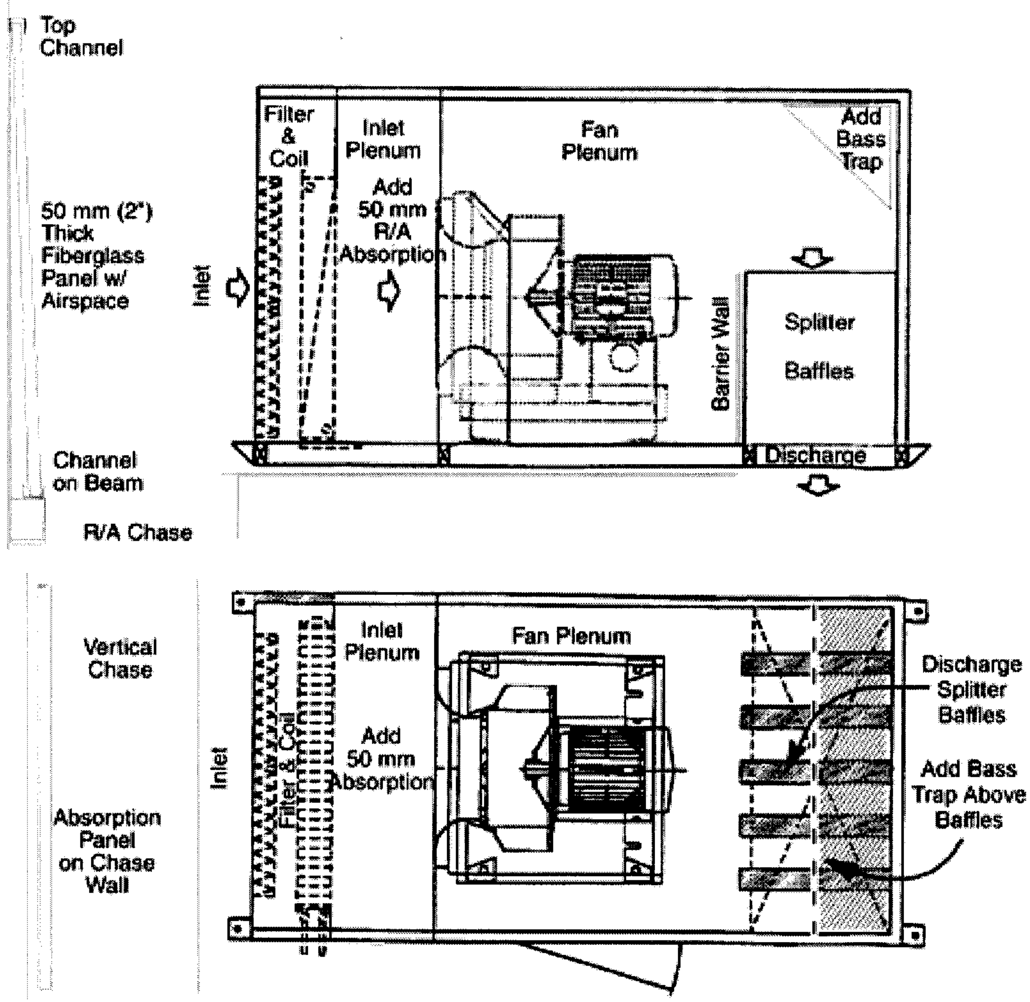

FIGURE 4: MODIFIED AHU - SECTION (TOP) AND PLAN (BOTTOM)

3.0 CASE STUDY / DISCUSSION / NOISE PREDICTION

In addition to the existing air handler fan plenum walls' 50 mm (2") fiberglass liner (mylar encapsulated for cleanliness), the following measures were implemented:

- Mylar encapsulated 50 mm (2") thick fiberglass liner panels were installed in the return air inlet plenum for return noise reduction.
- A large triangular cross-sectional shaped baffle was installed in an upper corner of the fan plenum, above the discharge attenuator section, to act as a "bass trap" or low frequency absorber. This was designed to help reduce the 125 Hz fan tone.
- A heavy sheet metal panel was installed on the interior partition between the fan and the discharge attenuator section to prevent breakout (and force the sound to travel through the splitter baffle passages, instead of partially bypassing them).
- A 50 mm (2") thick mylar encapsulated fiberglass absorber was installed over the face of the sheet metal barrier on the attenuator section.

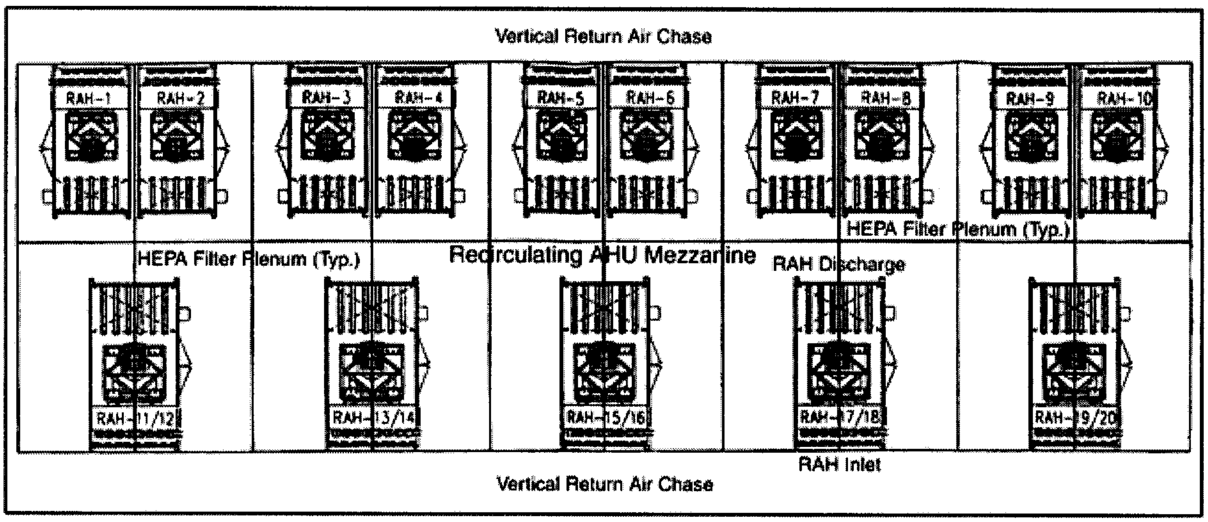

FIGURE 5: MECHANICAL MEZZANINE PLAN SHOWING MODIFICATIONS

Noise from the AHU inlets radiated into the mechanical mezzanine and return air chases. To reduce reverberation in the mezzanine and direct noise reflection into the return air chases, the following measures were implemented:

- Mylar encapsulated 50 mm (2") thick NRC 0.85 absorber panels were installed on the building walls, opposite the AHU inlets, to make approximately 50% of wall space absorptive. (The mezzanine floor and ceiling remain acoustically reflective.)
- Support angles were applied to building walls to permit the absorber panels to lean against the wall with a vertically tapered air space behind them to increase lower frequency absorption (the air space gave the panels an effective depth greater than the panels' thickness and allowed some absorptive area on the backs of the panels).

4.0 CONCLUSIONS / EXPERIMENTAL RESULTS

The AHU and mechanical mezzanine acoustical absorption modifications were made quickly with relatively small cost, meeting schedule, cleanliness and budget requirements for the cleanroom upgrade. Although physical limitations and avoidance of demolition precluded application of attenuators or noise cancellation systems, novel adaptations of classic room acoustics and passive absorbers were developed and implemented to achieve ambient noise criteria.

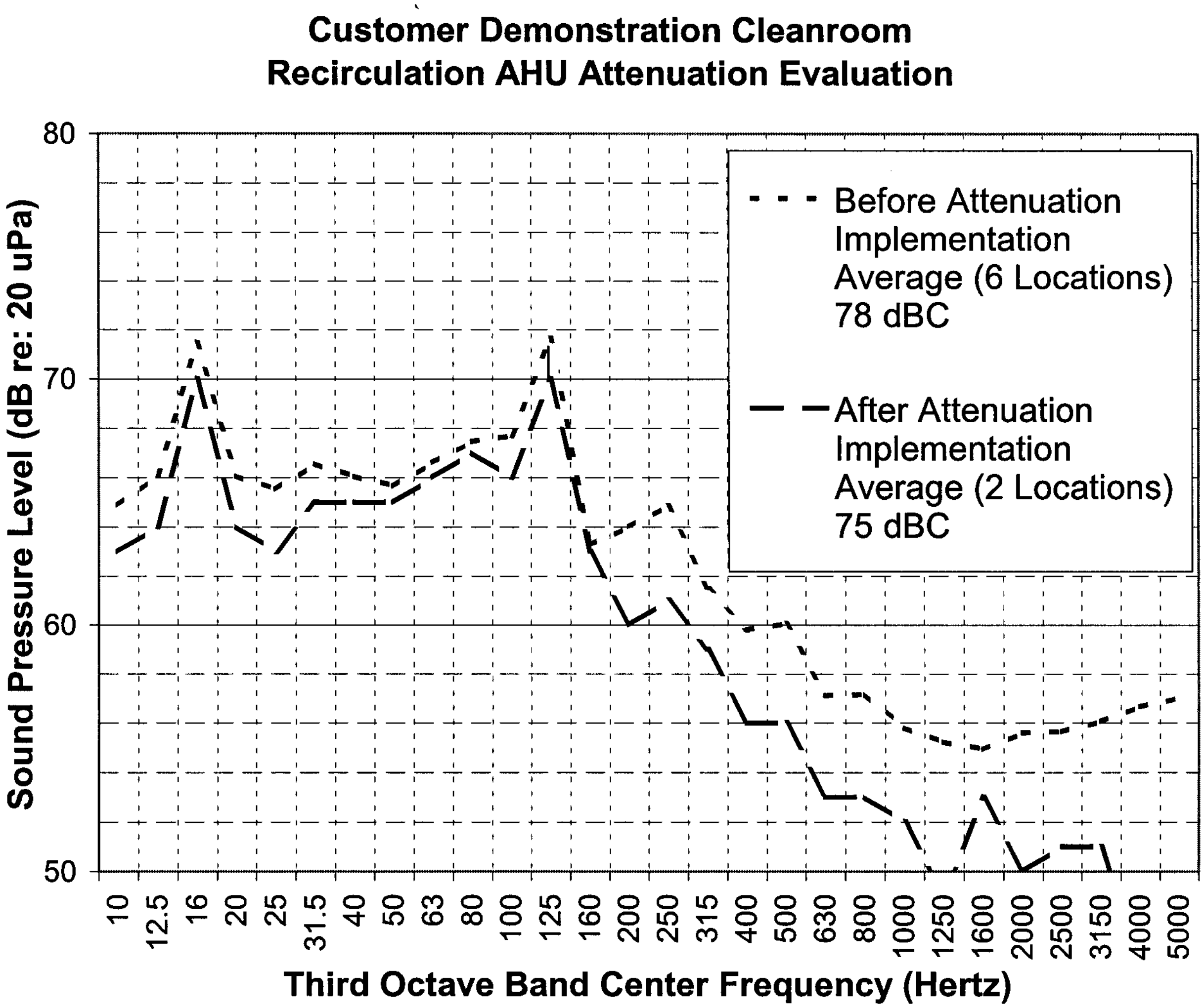

FIGURE 6: PERFORMANCE OF MODIFICATIONS

Measurements conducted after the installations of the passive absorbers in the AHUs and absorption panels on the walls showed reduction from the previous 78-80 dBC to 75 dBC to meet machine tool permissible noise criteria, including reduction of the strong 125 Hz fan blade passage tones.

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