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Classroom Noise at the Applied Research Laboratory Penn State University

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The Applied Research Laboratory at Penn State is a center of acoustic research in the United States. When Marshall Long Acoustics was selected to be the acoustical engineer on its new Applied Science Building, we were particularly careful to take into account every noise source. When we went out on the final inspection, the question arose, "Why does the classroom measure NC 42 when it should be less than 30?" The answer was not so elementary, my dear Watson.

In 1992, the Applied Research Laboratory at Penn State University decided to construct a new building to house a center of acoustical education and additional research programs. They selected Kling Lindquist Partners (recently renamed KlingStubbins) of Philadelphia, PA to be the architects. Marshall Long Acoustics of Sherman Oaks, CA was selected to be the acoustical and audio visual engineers and Vibron Ltd. of Toronto, Canada was selected to do the vibration control engineering for the underwater test facilities.

The acoustical and audio visual engineering included a large lecture hall, shown in Fig. 1 and a classroom. The lecture hall design solution was interesting since the architect's original configuration was quite good with a flat ceiling and a sloped seating area. We recommended a hard ceiling for strong overhead reflections with absorbent wall panels to control reverberation. We also designed a loudspeaker system to cover the front part of the audience and added delayed ceiling speakers in the back half of the seating area. We wanted a convex reflecting surface above its small platform stage so we designed a perforated metal panel with a solid Plexiglas sheet behind the lower portion. The loudspeakers were placed behind the perforations and so were invisible, while remaining clearly audible.

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After the building was completed I returned for a final inspection. Working with the AV contractor we adjusted the volumes for each delay zone, the equalization, and the ceiling speaker delays. This work was routine and uneventful. The lecture hall behaved quite well. In spite of its large capacity (400 seats) most lecturers use it without the sound system. When that was completed, I was asked to inspect the classroom, which, I was told, was noisy. I measured the HVAC noise in the classroom and it was an NC 42, much greater than the NC 30 design goal. This was a mystery. The measured data are shown in Fig. 2.

The HVAC design employed a variable air volume (VAV) configuration wherein a remote fan operates at a constant, efficient speed, and in-duct dampers act as air valves to regulate the amount of air delivered to each register. I examined the data to try to learn what I could from the spectrum shape. In general HVAC noise falls into spectral regions according to the predominant source. Noise below the 250 Hz octave band is usually attributable to the fan itself. Between 250 and 1000 Hz the source is generally excess velocities in the ducts or VAV units. Above 1000 Hz the predominant source is excess velocities at the registers. The data pointed to a combination of register and duct velocities.

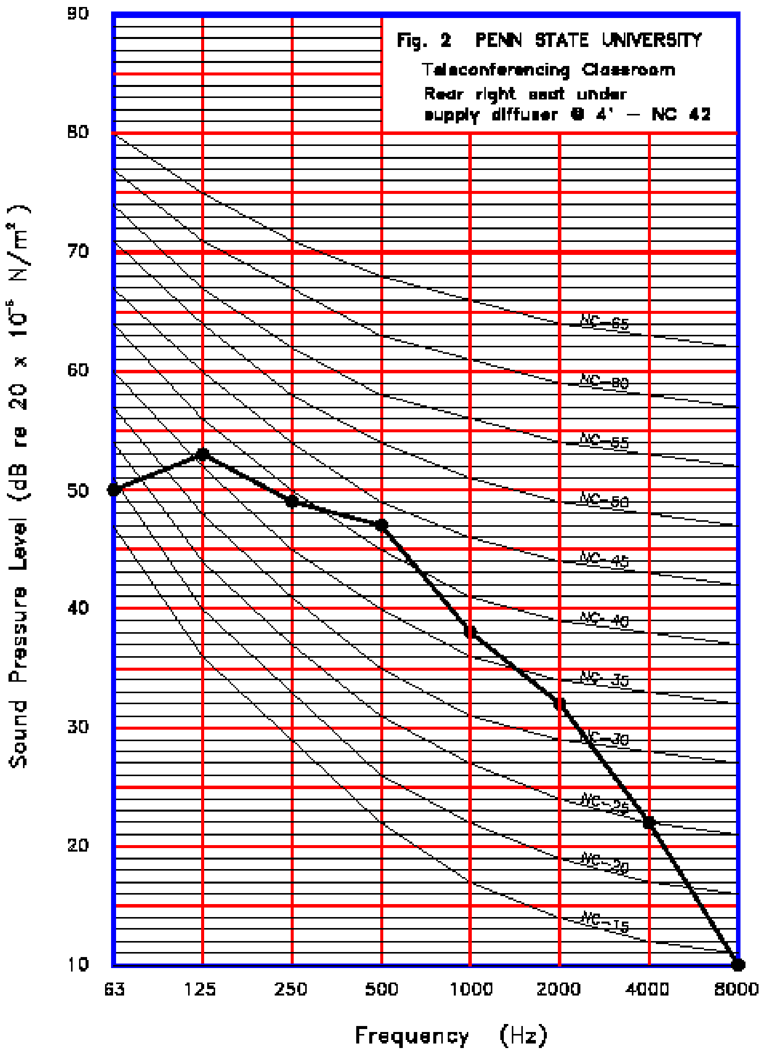
Upon checking the air coming from each register, we found that the system was significantly out of balance. For example one diffuser had a volume flow of 350 CFM, while an adjacent diffuser had only 80 CFM. I adjusted the diffusers so that they were approximately equally balanced. This procedure often cures register-generated noise. After balancing, the NC level was reduced to an NC 35. The data are shown in Fig. 3. Most of the decrease occurred at the high frequencies. Still, this was too high for a classroom, where we had anticipated an NC 30.The shape of the noise spectrum now indicated excess velocities in the ducts. This was curious to me since we had oversized the VAV units specifically to avoid this problem. After much puzzling, I climbed up into the T-bar ceiling to look at the VAV units. When my head was in the plenum, Jiri Tiche, a well known acoustics professor happened by and politely asked if I knew what I was doing. I explained that I was trying to fix the noise problem. After a few minutes he went on his way, but I am not sure he was convinced.

The result of my burrowing was that I was able to read the label on the VAV units, which turned out to be one size smaller than the units we had specified—8” diameter rather than 10”. The lower duct size resulted in a 56% increase in velocity, which theoretically can generate an 11 dB increase in sound level at the 60 log V relationship used for diffusers and dampers. This was the correct order of magnitude. My supposition is that the mechanical contractor substituted a smaller size that was rated for the flow volumes in the duct and no one had any reason to object or to question it. We submitted an inspection report requesting that the properly sized unit be retrofitted.

New VAV units were installed and the background measurements were repeated. The results, in Fig. 4, show an NC 29. It is interesting that the high frequency noise increased somewhat. This is probably due to the fact that the larger VAV unit delivered more air to the registers and increased register flow noise.

It is difficult to know how to avoid this type of problem without on site inspections by the acoustical engineer. Usually these are considered too expensive for the typical job, but they are very necessary. Substitutions are a common problem in construction and are treated through the submittal process. Because VAV units show up in the mechanical specifications and are reviewed by the mechanical engineers, acoustical engineers rarely get to see them. This type of substitution is difficult to catch because the on site architect or mechanical engineer is not familiar with these subtleties.

Figure 1 Applied Research Laboratory Lecture Hall



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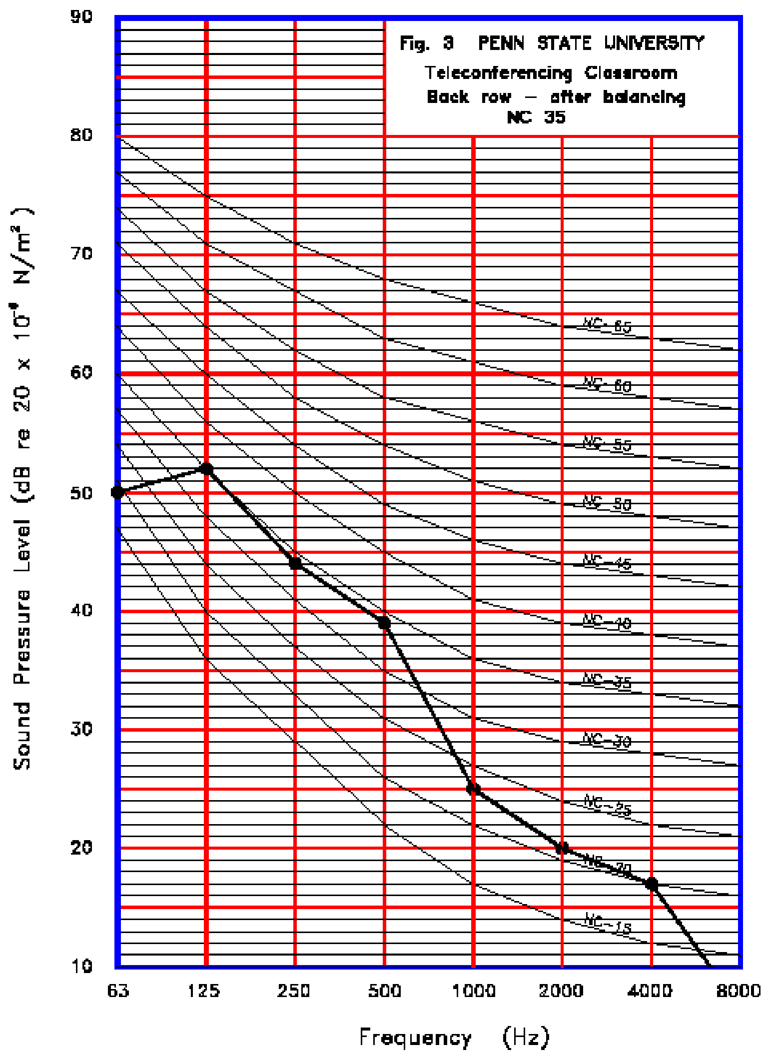

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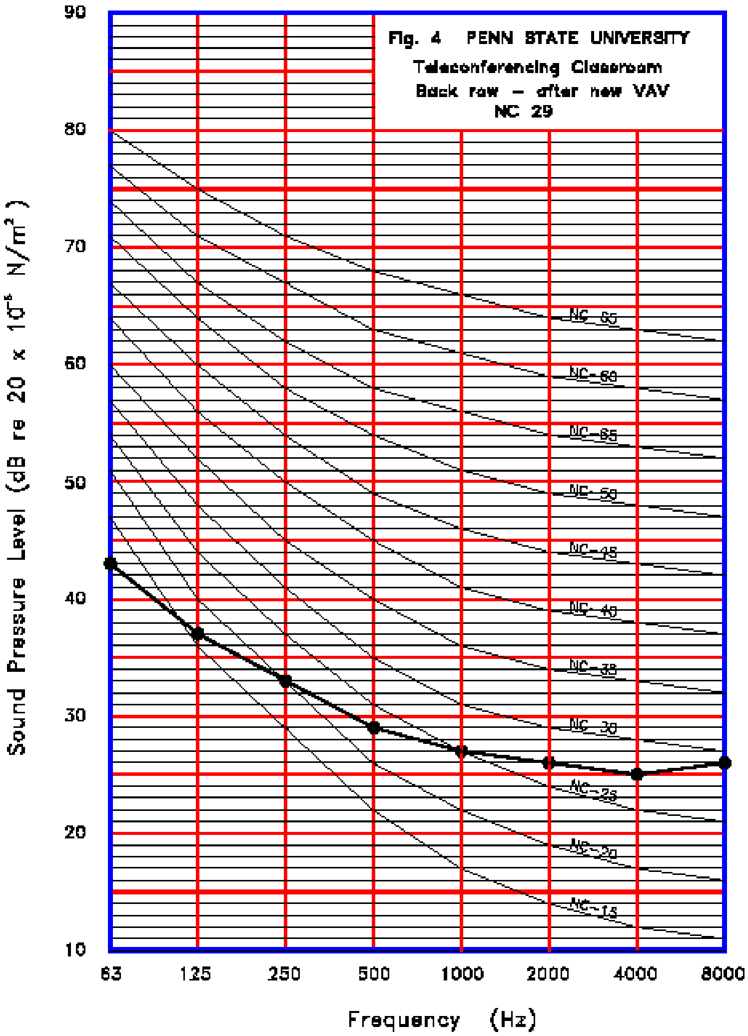

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