© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

OR-10-058

Relationship between HVAC Airflow Rates and Noise Levels, and Noise Control in a Mechanically-Ventilated University Building

Murray Hodgson, PhD, CEng

ABSTRACT

An investigation was conducted into HVAC-related airflow rates and noise levels in five classrooms in a mechanically-ventilated building at the University of British Columbia (UBC), the relationship between them, and how to control the noise. Sound-pressure levels were measured in terms of total, A-weighted level, Noise Criteria (NC) and Room Criteria (RC) ratings, and the results compared with established acceptability criteria. Airflow velocities were measured at each ventilation outlet in each classroom and corresponding volume airflow rates determined. The volume airflow data were compared with UBC design standards and specifications in previous balancing records. Comparison of the measured volume airflow rates and noise revealed a direct relationship between the two factors. Classrooms meeting the minimum airflow requirements tended to be excessively noisy. Classrooms with acceptable noise levels tended to have inadequate volume airflow rates. The main source of noise was excessive turbulence over dampers and diffusers, intensified by poor damper placement and high airflow velocities. To make noise levels acceptable, three of the classrooms could have their ventilation systems balanced to lower volume airflow rates to the applicable minimum standard. In addition, face dampers could be replaced by volume extractors; branch takeoffs could be lengthened and their ducts enlarged. Two of the classrooms could have their volume airflow rates increased to comply with the ventilation standards. The results suggest that achieving both acceptable ventilation and noise quality in mechanically-ventilated buildings can be a challenge; they also confirm that environmental factors are not independent, and must be optimized from a multi-disciplinary perspective if high-quality environments are to be achieved for building occupants.

INTRODUCTION

A study of the classrooms in a university building was undertaken, with the goal of investigating the acceptability of, and the relationship between, airflow rates and noise levels from the heating, ventilation and air-conditioning (HVAC) system, and how the noise can be controlled. The HVAC noise was evaluated according to three different criteria for rating background-noise levels. Measured volume airflow rates were compared to their specifications, and related to the background-noise levels. This paper reports the tests done and the results. The objective was not to perform a detailed, exhaustive investigation of system performance. It was to provide direct evidence of the relationship between ventilation performance and noise, and to discuss how HVAC noise can be controlled. This paper is directed at ventilation engineers who may not always be aware of the acoustical consequences of their work, not to acoustical engineers.

There is no one way to rate the acceptability of classroom noise. Existing and proposed standards use a variety of background-noise rating methods to quantify suitable classroom-noise levels. The classroom noise in the study building was evaluated according to three methods. An ANSI standard uses an A-weighted sound-pressure-level rating (dBA), and limits classroom noise to below 35 dBA for classrooms with volumes less than or equal to 20,000 ft^3 (566 m^3), and 40 dBA for those of greater volumes (ANSI 2002). A second standard under consideration uses the Noise Criteria (NC) method of evaluating noise, and recommends a limit of NC 30 (Lilly 2000). The Room Criteria (RC) rating is recommended by ASHRAE when assessing HVAC noise, and limits classroom noise to RC 35 (N), the "N" indicating that the noise spectrum must be of neutral quality (broadband noise) (ASHRAE 2007).

Murray Hodgson is Professor of Acoustics in the School of Environmental Health and the Department of Mechanical Engineering, University of British Columbia, Vancouver, BC, Canada.

 ©2010 ASHRAE

© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

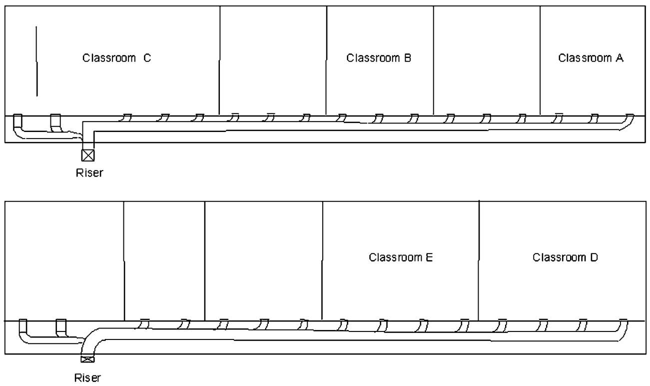

Figure 1 *Plan layout drawings of the classrooms and HVAC systems of the second (top) and fourth (bottom) floors of the study building, before renovation of classrooms A and C.*

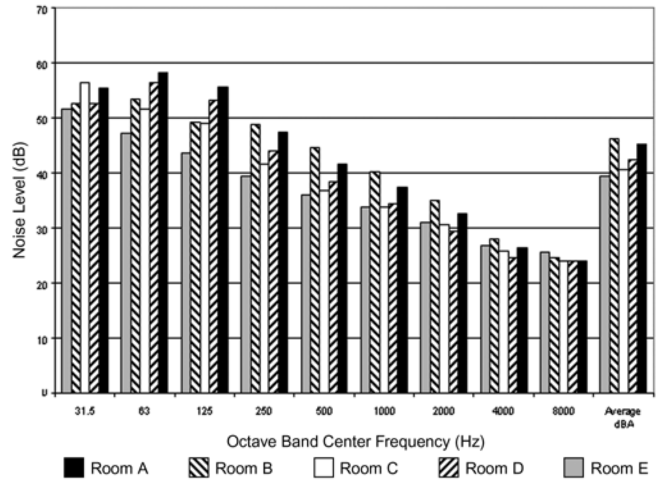

Figure 2 *Measured octave-band and total A-weighted HVAC noise levels.*

Of these criteria, the Room Criteria method is the only one which evaluates sound quality as well as level.

The building investigated in this study was chosen because previous work had shown it to contain classrooms with high HVAC noise [Hodgson 2002]. It had five classrooms (here called Rooms A-E) on the second and fourth floors. It was built in 1962. The three unrenovated classrooms and their ventilation systems had negligible sound absorption. However, classrooms A and C on the second floor were renovated in 1999 and 2001, respectively. Renovations included upgrading wall and ceiling sound absorption, and upgrading the HVAC systems by increasing the number of branch ducts supplying air to the classrooms and adding duct liner inside them, to provide sound absorption and control internal noise. All five classrooms were located on the west side of the building's main hallways – three on the second floor and two on the fourth floor. A fan located in the first-floor mechanical room supplied the west side of the building. A second fan, positioned in the penthouse, supplied all other rooms. Plan layout drawings of the classrooms and HVAC systems of the second and fourth floors are shown in Figure 1.

EXPERIMENTATION

Measurement Procedures

In each of Rooms A-E of the building volume airflow rates were measured across each ventilation outlet. HVAC noise levels were also measured, using a sound-level meter. Measurements were made at four to nine equally-spaced positions throughout the seating areas of the rooms, and energy-average, 31.5- to 8000-Hz octave-band levels calculated; from these, total A-weighted levels were calculated. As per the requirements of the ANSI standard (ANSI 2002), measurements were made with windows closed, and computers and audio-visual equipment turned off.

The airflow velocity across each classroom's HVAC-duct outlets was measured using a hot-wire anemometer. The data were collected over four equally-spaced areas over each diffuser face, and the results averaged to obtain the effective airflow rate. These were converted into volume airflows by multiplying the measured velocity by the outlet's effective cross-sectional area. The effective cross-sectional area was the fraction of the outlet's cross-sectional area not blocked by diffuser vanes, and was taken, since values for the actual diffusers were not available, from manufacturer's data for similar diffuser grills, to be 0.78 of the total cross-sectional area.

Noise-Level Results

Figure 2 shows the classroom-average HVAC noise levels. It shows that noise levels were highest at low frequencies; in the two renovated classrooms—Rooms A and C—noise levels decreased with increasing frequency, whereas the three unrenovated classrooms displayed peak noise levels in the 63-Hz octave band. Room E had the highest levels in the 31.5 to 125 Hz octave bands, while Room B had the highest levels in the 250 to 4000 Hz bands. In general, the two renovated classrooms, A and C, had the lowest levels of background noise. Though Room E had the highest noise levels at low frequencies, Room B had the highest total A-weighted level (46 dBA), since it had the highest levels at higher frequencies.

The three noise-rating methods were applied to all five classrooms; the results are shown in Figure 3. For Rooms A and C the acceptable noise-level limit was 40 dBA, whereas for Rooms B, E, and D it was 35 dBA due to their smaller volumes (ANSI 2002). With regards to the NC criterion (Lilly 2000), the limit of NC 30 applied to all five classrooms, as did the limit of RC 35 (N) for the RC rating method (ASHRAE 2007; ASHRAE 2009). These limit values are displayed in Figure 3.

© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

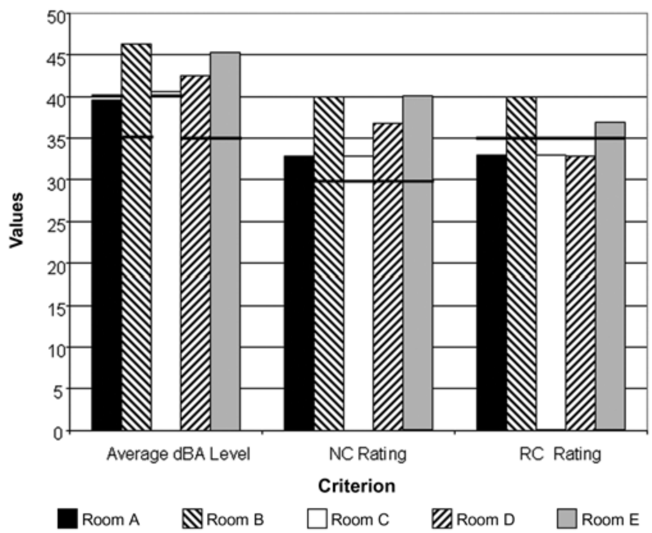

Figure 3 *Comparison of measured noise levels in five classrooms with three acceptability criteria.*

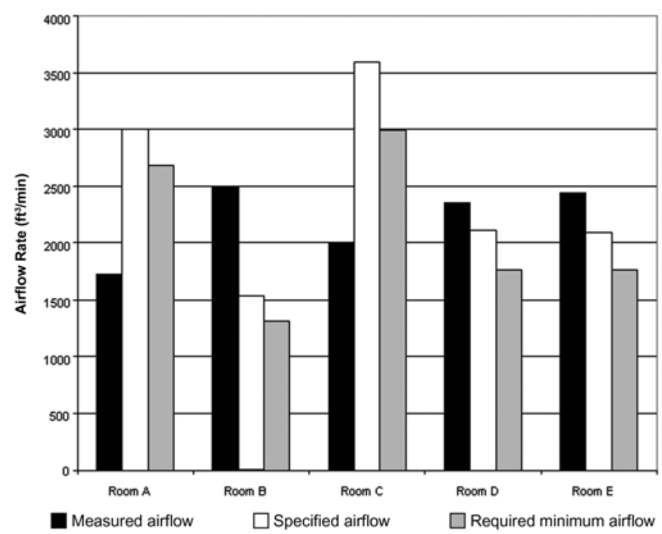

Figure 4 *Comparison of measured total volume airflow rates in five classrooms with two acceptability criteria.*

When rated by the total, A-weighted level (ANSI 2002), Room A was slightly below the 40 dBA limit (with a value of 39.5 dBA), and Room C marginally exceeded it (at 40.6 dBA). Rooms B, E, and D had unacceptably high noise levels relative to the recommended limit of 35 dBA, with values of 46, 45 and 43 dBA, respectively.

With regards to the NC criterion (Lilly 2000), all five classrooms had NC levels exceeding the NC 30 limit. Rooms B and E had ratings of NC 40, Room D had a rating of NC 37, and both Rooms A and C had ratings of NC 33.

When evaluated according to the RC criterion (ASHRAE 2007), only Rooms B and E exceeded the specified RC 35 limit. Considering the quality of the noise spectrum, all of the classrooms except for Rooms A and B had the desirable 'neutral' quality. The noise spectra of both Rooms A and B rated "H (hiss)". Although Room A was below the specified RC 35 level, it did not have a "neutral" noise spectrum. However, in this case the noise quality was determined to be marginally acceptable, given that its high-frequency component was likely not perceptible to occupants (ASHRAE 2007). Note that the values obtained from these measurements are an approximation to the actual RC ratings, as the proper procedure requires the measurement of sound-pressure levels in the 16-Hz octave band, which exceeded the capabilities of the sound-level meter used.

All three methods of evaluating the classroom noise resulted in the conclusion that Room B was the most problematic, followed by Room D, then Rooms E, C, and finally A. The two renovated classrooms, Rooms A and C, were either below, or slightly above, their acceptability limits for all three criteria.

The decibel attenuations required in each octave band (in particular, to meet the NC 30 criterion) were determined. Room B had the highest noise levels, and thus required the greatest attenuation. Attenuation was needed mainly in the 250 to 2000 Hz frequency range, with maximum values of 9.6 and 9.3 dB at 500 and 1000 Hz, respectively. Room E had the second highest noise levels; it required attenuation in the 125 to 2000 Hz octave bands, with the maximum attenuation required (7.6 dB) at 125 Hz. Room D was slightly less noisy than Room E, but showed a similar pattern of required attenuations. Its maximum required attenuation was 5.2 dB, again in the 125 Hz octave band. Rooms A and C required the least attenuation to meet the NC 30 standard; the required reductions were in the range 500 to 2000 Hz, and were a maximum of 2.9 dB at 1000 Hz.

Airflow Results

The average volume airflow rate across each diffuser was calculated from the airflow-velocity measurements for all five classrooms. Then, for each room, the total volume airflow rate from the HVAC system was found by summing the individual outlet values. These results are summarized in Figure 4. Room B had the highest volume airflow rate, with an average total value of 2500 ft^3/min (1180 L/s), even though it contained the smallest number of outlets (three). Room E had the next highest at 2443 ft^3/min (1153 L/s), followed by Room D at 2357 ft^3/min (1112 L/s), Room C at 2000 ft^3/min (944 L/s) and, finally, Room A at 1722 ft^3/min (813 L/s). The calculated volume airflow rates for each classroom were compared with the corresponding values specified by the most recent balancing report or the renovation specifications. These specified values are also shown in Figure 4. The noisiest classrooms—most noticeably Room B—had measured volume airflow rates well above those specified. Conversely, the renovated classrooms (Rooms A and C) had volume airflow rates below their specifications.

© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

The University of British Columbia requires a minimum of 8 to 10 air changes per hour (ACH) for classrooms (Lis 2002). The volume airflow rate corresponding to the absolute minimum of 8 ACH was determined for each classroom and is also shown in Figure 4. According to Figure 4, Rooms B, E, and D could have their volume airflow rates reduced below their current specifications, and still meet minimum requirements. Conversely, renovated classrooms A and C currently have volume airflow rates below the regulated minimum values.

It is difficult to determine how much of the apparent noise reductions in Rooms A and C were the result of their renovations, and how much was due to their extremely low volume airflow rates. There are two other rooms supplied by the west-zone fan on the second and fourth floors; one houses graduate-student offices, the other is a seminar room. The former room had also recently been renovated. Even though neither was a classroom, the airflows into these rooms were measured for comparison. As before, the renovated room had volume airflow rates which were below its specifications, while the other room had volume airflow rates above those indicated in the balancing report. It seems that there was a lack of proper balancing after renovations were completed.

The speed of the supply fan was also measured, to determine whether it was operating as intended. When the fan was originally selected in 1962, it was to operate at 425 rpm. However, during balancing in 1987 for energy-conservation purposes, the fan's speed was lowered to 405 rpm. The current operating speed of the fan was measured as 413 rpm, an approximately 2% increase since 1987, which is negligible. Assuming the fan was originally selected to operate at or near peak efficiency, it should still be operating in this range; thus it would not be expected to be generating excessive noise (ASHRAE 2007). Furthermore, the airflow output of the fan increased from 27,300 ft^3/min (12 880 L/s) in 1987 to approximately 27,890 ft^3/min (13 160 L/s; an increase of 2.2%); this does not account for the magnitude of the increased airflow in the non-renovated rooms. The problem was likely not the fan, but rather the reduced airflows in Rooms A and C (and the graduate-student area) which cause excessive amounts of air to be re-directed to Rooms B, E, and D (and the seminar room).

RELATIONSHIP BETWEEN VOLUME AIRFLOW RATES AND NOISE LEVELS

In general, the results of this airflow analysis confirm a positive correlation between the volume airflow rate entering each classroom (i.e., the ventilation quality) and its noise levels. This is shown in Figure 5. Classrooms meeting the minimum airflow requirements (i.e., acceptable ventilation quality—Rooms B, D, and E) tended to have excessively noisy. Classrooms with acceptable noise levels (Rooms A and C) tended to have inadequate volume airflow rates (i.e., unacceptable ventilation quality).

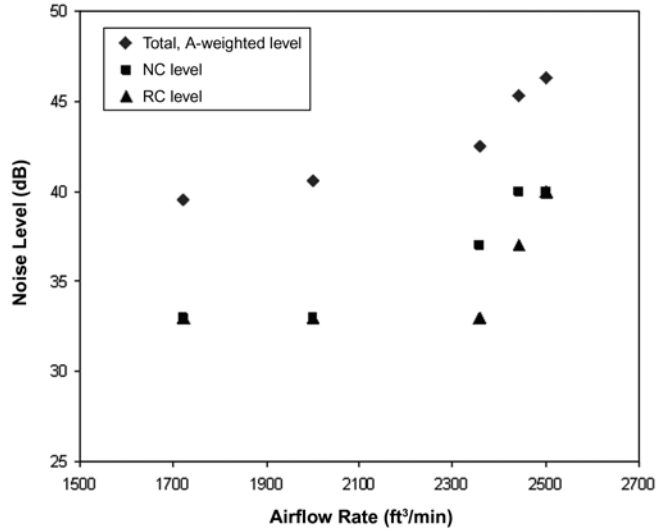

Figure 5 *Variation of measured noise levels with airflow rate in Rooms A to E.*

NOISE-SOURCE IDENTIFICATION

Both 2009 *ASHRAE Handbook—Fundamentals* and 2007 *ASHRAE Handbook—HVAC Applications* provide charts to assist in determining the dominant source of background noise. Different sources produce noises of different frequencies; thus the frequency spectra of the noise and of the required attenuation give indications of the likely source of the noise. Here, the required attenuation was mainly in the 500 to 2000 Hz bands; thus, the likely HVAC noise sources were fans and pumps, VAV units, reciprocating and centrifugal chillers, dampers and diffusers. The fan was determined to be operating at or near peak efficiency, and not generating excessive amounts of noise. The building contained three pumps, all of which were located, along with the supply fan, in the first-floor mechanical room. Two of these pumps were components of the building's plumbing system, while the third was part of its heating system. The three pumps were not very large, and were unlikely to be significant noise sources. Furthermore, Room C was closest to the first-floor mechanical room and required attenuations mainly in the 500 to 2000 Hz octave bands. If fan or pump noise were a problem, it should have been greatest in this room; however, the octave bands in which noise attenuation was required indicated otherwise. The building was not equipped with either variable-air-volume (VAV) units or chillers of any kind; thus, these were also eliminated as potential noise sources.

Aerodynamically generated noise in ducts is proportional to duct airflow velocity to the fifth or sixth power near a duct fitting (ASHRAE 2007), since this regenerated noise is caused by turbulence. Outlet diffusers and volume dampers are examples of such duct fittings. Diffusers are designed to distribute the airflow entering a room evenly; however, as established above, turbulence across these elements generates noise in strongly relation to the duct airflow velocity. This noise is produced at mid and high frequencies. Volume dampers are

© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

frequently used in HVAC systems to regulate the amount of airflow entering or exiting duct branches, typically at branch takeoffs to outlets. Because they restrict airflow, dampers increase air turbulence and noise levels, even when they are completely open, especially at mid and high frequencies. In order to minimize this turbulence, and the noise it generates, volume dampers should be installed as far as possible from the duct termination (ASHRAE 2007). Rooms B, E, and D were equipped with face dampers in each duct outlet. These were volume dampers, installed directly upstream of the outlet's diffuser; therefore, they were likely sources of noise.

Given the strong correlation between volume airflow rates and noise levels, and the fact that aerodynamically-generated noise caused by turbulence varies so strongly with airflow velocity, it can be hypothesized that most of the noise was of this type. That is, the dominant source of classroom noise was turbulence over dampers and diffusers, exacerbated by the non-optimal locations of the dampers, and high volume airflow rates.

Rooms E and D required a peak attenuation in the 125-Hz octave band. This is relatively low-frequency noise, usually caused by fan noise propagating through the HVAC-system ducts. However, fan noise was unlikely in this case. Duct walls attenuate noise in HVAC systems, this attenuation increasing with distance. The fan was located on the first floor of the building, whereas these two rooms were at the end of the fourth-floor hallway. This distance was substantial enough that it was highly unlikely that a significant amount of fan noise reached these two rooms. Moreover, if fan noise were a problem, this would also have been expected to be the case in those classrooms that were much closer to the fan location, such as Rooms C and B. However, this was not so. Duct-breakout noise from the second fan located on the building's roof was another unlikely source. If it was the cause, a low-frequency rumble should have been perceptible in the hallway, which was not the case. This noise may not have been caused by the HVAC system, but by some other background-noise source—for example, different types of electrical equipment.

NOISE CONTROL

The dominant source of classroom noise in the building was air turbulence over the dampers and diffusers at the duct outlets. Therefore, the recommendations for noise control focused on ways to minimize the amount of turbulence generated. Since this noise is created directly ahead of the duct termination, traditional approaches—such as silencers and main-duct lining—were not appropriate. Acoustical lining of the branch takeoffs to the classrooms was also not an effective solution in this case, since these were of a very limited length; any noise-level reduction due to the lining would likely be imperceptible to the occupants.

Rebalance the System

Since air turbulence in ducts varies significantly with the airflow velocity, re-balancing the HVAC system should reduce the classroom noise. Rooms B, E, and D could have their duct airflow velocities reduced to reduce volume airflow rates to the minimum allowable levels. On the other hand, Rooms A and C could have their volume airflow rates increased to comply with the recommendations. Manufacturer's data for diffusers similar to those in Rooms B, E, and D were obtained, to give an indication of the noise reductions expected to occur from these changes. These were used to estimate the noise levels associated with current volume airflow rates, and the reductions expected if the rates were lowered to minimum requirements. In Room B, reducing the volume airflow rates from each of the three outlets from the current 774, 847, and 880 ft^3/min (365, 400, and 415 ℓ/s) to 460 ft^3/min (217 ℓ/s)—three grills at 460 ft^3/min (217 ℓ/s) equals 1380 ft^3/min (651 ℓ/s), which is greater than the minimum required volume airflow rate of 1317 ft^3/min (621 ℓ/s)—the reductions varied from 12 dB at 63 Hz, to 20 dB at 4000 Hz. This would reduce noise to NC 29 (34 dBA), meeting both the NC 30 and 35 dBA criteria. In Room E, reducing the volume airflow rates from each of the four outlets from 657, 594, 592, and 600 ft^3/min (310, 280, 279, and 283 ℓ/s) to 450 ft^3/min (212 ℓ/s)—four outlets at 450 ft^3/min (212 ℓ/s) each gives a total of 1800 ft^3/min (849 ℓ/s), which is greater than the minimum level of 1763 ft^3/min (832 ℓ/s)—would result in reductions varying from 6 dB at 63 Hz to 9 dB at 4000 Hz; the resulting noise would decrease to NC 31 (37 dBA). This would almost allow Room E to meet the NC 30 criterion, but would be inadequate to reduce its total A-weighted level to below 35 dBA. In Room D, reducing the flow rates from each of the four outlets from 697, 568, 653, and 439 ft^3/min (329, 268, 308, and 207 ℓ/s) to 450 ft^3/min (212 ℓ/s)—four outlets at 450 ft^3/min (212 ℓ/s)—each gives a total of 1800 ft^3/min (849 ℓ/s), which is greater than the minimum level of 1763 ft^3/min (832 ℓ/s)—the resulting noise level is again NC 30 (36 dBA), meeting the NC criterion, and almost meeting the total, A-weighted noise-level criterion of 35 dBA. This analysis shows that lowering the volume airflow rates should significantly reduce the noise in Rooms B, E, and D.

Remove Face Dampers

Rooms B, E, and D have ducts equipped with face dampers. Dampers increase the noise level because they increase the amount of air turbulence generated. Face dampers in particular are problematic; they create turbulence directly at the duct outlet, where there is no opportunity for the airflow to "straighten" before passing over the diffuser. Face dampers can increase the noise levels generated at a diffuser by 5 to 24 dB, depending on the damper's pressure ratio (ASHRAE 2007). Thus, the removal of face dampers in Rooms B, E, and D is recommended. An alternative volume-control device—volume extractors—should be installed in their place. Volume extractors are placed at the neck of the branch takeoff, and are specially designed to minimize turbulence. Coupled with the noise-level attenuation achieved by reducing the airflow rates,

© 2010, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Published in ASHRAE Transactions 2010, Vol. 116, Part 1. For personal use only. Additional reproduction, distribution, or transmission in either print or digital form is not permitted without ASHRAE's prior written permission.

removing the face dampers would likely bring the noise in Rooms B, E, and D to values below all three criteria.

Lengthen Branch Takeoffs

The branch takeoffs to Rooms B, E, and D were very short (at most, 1 m long). The greater the length of these ducts, the less turbulence is generated at the duct outlets, as the airflow "straightens" in the branch ducts. Thus, another potential noise-control solution is to lengthen these ducts. Longer branch ducts would also allow the possible installation of acoustical lining, further reducing noise levels. However, this would require extensive modification to the existing ductwork and the classrooms, and could be quite expensive.

Increase Diameter of Branch Takeoffs

According to the law of mass conservation, for a given volume airflow rate, a greater duct diameter corresponds to a lower duct velocity. As aerodynamically-generated noise is a function of airflow velocity, less classroom noise results. Thus, a fourth recommendation was to enlarge the ducts leading to the classroom outlets.

CONCLUSION

The noise survey of the university building revealed unacceptably high levels of HVAC noise in three of the five classrooms. Three rooms also had volume airflow rates above the specified minimum of 8 air changes per hour, whereas two had airflow rates below this criterion. Comparison of the noise and airflow measurements revealed a direct correlation between the two factors in the rooms. Classrooms meeting the minimum airflow requirements tended to be excessively noisy. Classrooms with acceptable noise levels tended to have inadequate volume airflow rates. To make noise levels acceptable, three of the classrooms could have their ventilation systems balanced to lower airflows to the applicable minimum standard; in addition, face dampers could be replaced by volume extractors (and grills), and branch takeoffs could be lengthened and their ducts enlarged. Two classrooms could have their volume airflow rates increased to comply with the ventilation standards.

The unrenovated classrooms in this study had adequate volume airflow rates, but excessive noise—likely because the building design considered ventilation, but not noise. After some classrooms were renovated to reduce noise, their volume airflow rates were inadequate. The results suggest that achieving both acceptable noise levels and ventilation can be a challenge. This study confirms that environmental factors in buildings interact. The design of one affects the quality of others. Factors must not be designed independent of one another—a multi-disciplinary approach to building design is essential. Quality building design requires all disciplines to be involved in the design process in an integrated way from the start, and, of course, an adequate budget.

REFERENCES

ANSI. 2002. *Acoustical performance criteria, design requirements and guidelines for schools*. ANSI Standard S12.60-200. Melville, NY: American National Standards Institute.

ASHRAE. 2007. 2007 *ASHRAE Handbook—HVAC Applications*, Chapter 46: Sound and vibration control. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 2009. 2009 *ASHRAE Handbook—Fundamentals*, Chapter 7: Sound and vibration. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Hodgson, M. 2002. Rating, ranking and understanding acoustical quality in university class-rooms. *Journal of the Acoustical Society of America* 112(2):568-575.

Lilly, L.G. 2000. Understanding the problem: Noise in the classroom. The *ASHRAE Journal*, New York.

Lis, M. 2002. Personal communication. University of British Columbia, Vancouver, BC, Canada.

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.