Experimental study on HVAC sound parameters

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Abstract. HVAC system represent major source of buildings internal noise and therefore they are designed to provide a human acoustic comfort besides the thermal and air quality requirements. The paper experimentally investigates three types of commercial air handler units (AHU) with different ducts cross-section sizes and inlet-outlet configuration. The measurements are performed in an anechoic room. The measurements are carried out at different fan’s speeds, ranging the power-charge from 30-100% while the duct air flow is slowly adjusted from full open to full closed, between 0-500 Pa. The sound pressure levels of the radiant units are rated using NR curves. Also, the supply and the outdoor ducts sound levels are compared in order to point the frequencies where the noise must be reduced. Third-octave band analysis of random noise of an air handling unit from a HVAC system is realized, using measurement procedures that agrees the requirements of the ISO 3744:2011 and ISO 5136:2010 standards. The comparatively results highlight the effects of the geometry, air flow pressure and power-charging dependencies upon the sound level. This is the start for a noise reduction strategy.

1. Introduction
HVAC (Heating, Ventilating and Air Conditioning) systems are indispensable equipments for the modern buildings in order to ensure the human thermal comfort and admissible air quality. The HVAC system include air handling unit, supply and exhaust air ducts, return air path, terminal equipment, spaces served. These components are designed to achieve the raise or decrease of the building temperature and humidity, to add or remove outside air and to provide its proper filtration.

At the same time, these equipments represent major source of internal noise and therefore the acoustical environment may suffer. Sound is generated by a source, it is transmitted along one or more paths and reaches a receiver. This is the source-path-receiver concept that constitutes the basis in the system analysis for noise control. The noise generator mechanism is the sound source and the selection of a quiet source, together with an optimized room sound absorption and designed propagation paths for minimal noise transmission represent the tasks for noise control [1].

In HVAC systems, the source of noise is a combination of different processes, such as mechanical noise from fans, pumps, compressors, motors, control dampers, variable air volume (VAV) boxes and air outlets such as diffusers, grilles, dampers and registers. Therefore the most significant noise sources are the air handler equipment (AHU) and the air circulating ducts [2].

From the theoretical point of view, there have been developed many numerical techniques for acoustic propagation in dissipative ducts, silencers and radiant unit, such as two-port transfer matrix method [3, 4], finite and boundary elements methods [5, 6]. There are also algorithms and procedures based on sound power computation [1, 7].
However, computational modeling of noise generation and propagation in HVAC systems are difficult since all the components are installed and interdependent and in fact they create a new acoustical environment. For instance, fan noise determined by experimental tests may be quite different once the fan is installed in an air handler; therefore a sound test is essential in this case [8].

In this paper there are experimentally investigated three types of commercial air handler units (AHU) with different ducts cross-section sizes and inlet-outlet configuration. The comparatively results highlight the effects of the geometry, air flow pressure and power-charging dependencies upon the sound level. Third-octave band analysis of random noise of an air handling unit from a HVAC system is realized, using measurement procedures that agrees the requirements of the ISO 3744:2011 and ISO 5136:2010 standards [7, 9]. The air handler unit selection could be a difficult task for the designers which have to consider all these parameters.

2. Experimental methodology

Softwares predicting the HVAC systems noise may have potential errors as they are using algorithms that calculate the sound level contributions of each element of a system. Generally, the algorithm can be considered an empirical black box that takes an incoming sound power level and characteristic information of the element to produce an output sound power level [6]. Therefore, potential errors may appear and the calculated results could be quiet far from the real situation.

2.1. General considerations and terminology

The experimental investigation is of more interest for the HVAC manufacturers, and the technical data from their catalogues [10, 11, 12] means the evaluation of performances curves, in terms of specific fan power, duct pressure-air flow relation, temperature and sound data. Performance curve information is gathered by connecting the system to a laboratory test room. Test procedures are prescribed in ISO 3744:2011 and ISO 5136:2010 standards in order to provide accurate readings. The measurements are carried out at different fan’s speeds while the duct air flow is slowly adjusted from full closed to full open. The HVAC manufacturers use this knowledge to develop computer programs for the components selection thus helping the system designers and the beneficiaries.

In the paper, the experimental investigation on sound characteristics for some commercial air handlers unit (AHU) is performed. We have used the anechoic room, from our department, with dimensions of 10m x 10m x 8m and a large access door (2m x 1.5m). In the anechoic room the walls, roof and floor are covered in a highly sound absorptive material to eliminate any reflections. The average absorption coefficient is 99% for 125 Hz ÷ 20 kHz frequency band. Thus, the sound pressure level in any direction may be measured and the directivity factor Q is 1 [7]. This is the major advantage of this kind of room, we can exactly measure the noise emitted by the source and there are no addition due to the reflection waves.

HVAC system designers generally evaluate the sounds in the frequencies between 45 and 11200 Hz. Using the octave bands, this frequency range is separated into eight octave bands with center frequencies of 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz. [13]. Also, the human ear is most sensitive to sound in the frequency range 1000 Hz to 4000 Hz than to sound at very low or high frequencies. This means that the noise at high or low frequencies will not be as annoying as it would be when its energy is concentrated in the middle frequencies. A higher sound pressure is therefore acceptable at lower and higher frequencies. This knowledge is important in acoustic design and sound measurement [13].

We also note the difference between sound power and sound pressure notions, because equipment suppliers have often incidentally provided either sound pressure data either sound power level. This practice has confused the market more than it has informed. The sound power level SWL, according to ISO 3744: 2011, is the sound energy constantly transferred per second from the sound source. Sound power is a theoretical value that is not measurable. The sound pressure level, SPL
is a measurable sound level that depends upon environment. It is expressed in decibels at a specified distance and position [7].

Usually, both sound parameters are calculated and expressed in decibels although they are completely different and should not be confused.

\[
SWL = SPL - 10 \log \left[ \frac{Q}{4\pi r^2} + \frac{4}{R_C} \right]  \tag{1}
\]

where \(SWL\) (dB) is the sound power re \(10^{-12}\)W; \(SPL\) (dB) is the sound pressure re \(2 \times 10^{-5}\) Pa; \(r\) (m) is the distance from the source; \(Q\) is the directivity factor of the source in the direction \(r\); \(R_C\) (m\(^2\)) is the room constant of the following form: \(R_C = \frac{S \cdot \alpha_w}{1 - \alpha_w}\) with \(S\) (m\(^2\)) total room surface and \(\alpha_w\) the average absorption coefficient in the room [13].

Relation (2) is simplified and becomes:

\[
SPL = SWL - 11 \quad [\text{dB}] \tag{2}
\]

Therefore, in the anechoic room, at \(r = 1\)m distance from the source, \(Q = 1\), the sound pressure level, \(SPL\), is 11 dB less than its sound power level, \(SWL\). The microphones path is presented in figure 1. The measured background noise in our anechoic room is: \(SWL = 28.9\) dB or \(SPL = 17.9\) (distance from the source \(r = 1\)m in the 1÷8 points).

![Microphone path in the anechoic room.](image)

The selection of an appropriate noise criterion is important when it must be specified acceptable levels of noise. The noise in buildings are rated using several methods, the common used is the Noise criteria (NC) for USA or Noise rating (NR) for Europe [14].

NC is defined as single number noise rating system commonly used to rate the steady-state noise levels in a room. They consist of a family of curves which defines the maximum allowable sound pressure levels in octave bands.

NR curves plotted on a scale of frequencies vs. sound loudness represent approximately equal loudness levels to the human ear [1]. These curves define the limits that the octave band spectrum must not exceed. For instance, to achieve NR 35 rating, the sound spectrum must be lower than the curve in every octave band.

NC (NR) is particularly used in Heating, Ventilating and Air Conditioning (HVAC) work, with an NR 35 and NR 40 being the most common requirement. A short explanation of NR curves together with their numerical values can be found in British Standard BS 8233: 1991 'Sound insulation and noise reduction for buildings - Code of Practice', Appendix B.
2.2. Method and device

Figure 2 describes a typical air handler unit with fans, dampers and ducts, while figure 3 shows the frequencies and a descriptive terminology at which different types of HVAC components influence the sound spectra in a room.

The frequency range where there are possible complaints related to the sound may be known and can help identifying the issues sources.

The experimental investigation is performed in the anechoic room using a test facility presented in figure 4. It is realized in accord with ISO 3744:2011 and ISO 5136:2010 standards.

The measurement chain include: device for measuring the pressure drop (Fluke 922 pressure differential meter), microphones Brue&Kjaer type 4133, soundmeter Brue&Kjaer type 2209, connected with the microphone and NIDAQ board, multifunctional external data acquisition board type-NIDAQPad-6015 on USB, laptop with LabVIEW soft compatible with National Instruments DAQPad for data processing.

The paper analyzed the sound level dB(A), A weighted, both on radiant unit and the air ducts, using LabVIEW soft, with third-octave analysis and measuring total band power (dBW) and pressure levels (dBA) for nominal frequencies of 63, 125, 250, 500, 1000, 2000, 4000, 8000 Hz. Specifications
for the tools included in LabVIEW library are defined by ANSI and International Electrotechnical Commission (IEC) standards and the results are fully compliant to the international standards (ANSI S1.11-2004 and the IEC 1260:1995 standards).

**Table 1.** Technical details on tested air handlers units.

| AHU type | Dimensions | Duct diameter/ Weight/Capacity |
|----------|------------|-------------------------------|
| AHU 1    |            | Φ200 mm, 62 kg, 400 m$^3$/h   |
| AHU 2    |            | Φ160 mm, 50 kg, 300 m$^3$/h   |
| AHU 3    |            | Φ160 mm, 50 kg, 400 m$^3$/h   |
The sound level tests were performed on three AHU commercial types, denoted and characterized in table 1. The experimental setup is presented in figure 5.

![Experimental setup - outside and inside the anechoic room](image)

**Figure 5.** Experimental setup - outside and inside the anechoic room

3. Results and discussions

The measurements are carried out at different fan’s speeds, ranging the power-charge from 30-100% while the duct air flow is slowly adjusted from full open to full closed, between 0-500 Pa.

Figures 6, 7, 8 present the sound level results obtained on the radiant units of the tested air handlers. This sound level variation reports the dependence between the air flow pressure (from the duct full open valve- P0, to the duct full closed valve- P500), and the fan power-charge.

AHU 1 is the least noisy of the three tested commercial types, with total band power ranging from 32-52 dB related to the fan power charge (figure 5). The most used regime in HVAC exploitation corresponds to 50-70% fan power-charge according to the manufacturer, so it is of interest to monitor these data points. The sound level values for these points are slightly higher for AHU 2 compared to those of AHU 1, in the same conditions (figure 7). Also, AHU 1 has a better sound level behaviour, in terms of HVAC requirements, at low charging (30 - 40%) and the same noise characteristics at high charging of 90 - 100%. Obvious, the air pressure increasing influences the measured sound level in the same way.

Figure 8 represents the sound level for AHU 3 radiant unit using the same dependencies as in the figures 6 and 7. Even if the overall noise level keeps approximately the same values as the other two units (32-52 dB in the entire range of charging) and the air pressure increasing lead to the same increased evolution of the sound level, one can notice the measured noise fluctuations.

These sudden changes are mainly observed at P400 and P500 where the sound power level can reach 56 dB. There are noisy mechanical components inside AHU that excite the whole structure and we identified the heat exchanger creating issues. During AHU running, this component randomly comes into operation thus causing the noise jumps.

Figure 9 represents a comparative evaluation of the sound power level for the tested units running at air flow pressure P500 (the worst case in exploitation) taking into account of fan power-charging. AHU 1 radiant unit has the lowest noise level and uniform noise behaviour with charging. A very quiet functioning at low charging can be also noted.
Figure 6. Sound level for AHU 1 radiant unit.

Figure 7. Sound level for AHU 2 radiant unit.

Figure 8. Sound level for AHU 3 radiant unit.
Figure 10 presents the noise level of AHU 1 and AHU 2 from the sound pressure perspective, see eq.(3), and the noise curves for air flow pressures and power-charging of interest are depicted. The AHU 1 sound pressure values range between 29-38 dB and it can be observed that it is a little less noisy than AHU 2 radiant unit.

![Radiant unit](image1.png)

**Figure 9.** Sound power for radiant units.

![Radiant unit](image2.png)

**Figure 10.** Sound pressure for radiant units.

The HVAC manufacturers are interested in ducts noise in order to optimize their design and efficiency from this point of view. This optimization means the in-duct noise control, generally solved through the silencers use, thus adding adequate attenuation to the system. Therefore, the silencer selection is imposed by the duct noise level.

In this regard, the sound power variation related to the fan power-charging is expressed in figure 11 for the supply air ducts of the three tested AHU, at air flow pressure P500. The sound power values are much higher (70÷95 dB) than those for the radiant units and the sound increasing tendency with fan power-charging is preserved. Also, the supply air duct of AHU 1, compared to the two other tested ducts from figure 11, has the best performance in sound attenuation. Figure 12 completes the discussion related to the air supply duct and presents the sound pressure levels for the power-charging of interest. It can be observed the sound pressure decreasing trend with the air flow pressure
increasing. The sound pressure values for AHU 1 supply air duct are lower than those for AHU 2. For instance, at 100% power-charging for AHU 1, the sound level (80-84 dB) is comparable to the level recorded at 80% power-charging for AHU 2.

![Figure 11. Sound power for supply air duct.](image)

![Figure 12. Sound pressure for supply air duct.](image)

Figures 13 and 14 present the noise levels for outdoor air ducts of tested handlers. Sound power values for AHU 1 outdoor air duct, when air flow pressure is P500, range between 62 and 85 dB, as in figure 13. It can be noticed its lower sound values, and also AHU 3 lower sound values have to be considered. Figure 14 reports the sound pressure values and we notice their slightly increasing trend with the air pressure increasing. AHU 1 outdoor air duct is less noisy than AHU 2 corresponding duct, in all range of power-charging.

The noise-air pressure evolution in the supply and outdoor ducts is influenced by the flow direction. There are a regenerated noise due to the flow direction (forward or reverse) which explain the situations from figure 12 (the noise decreases with pressure increasing in the supply duct) and a contrary tendency depicted in figure 14 (noise increasing with pressure in the outdoor duct).
Outdoor air duct

Figure 13. Sound power for outdoor air duct.

Outdoor air duct

Figure 14. Sound pressure for outdoor air duct.

Noise rating for radiant unit

Figure 15. Noise rating for the radiant units.
Figure 15 have tried to fit the obtained noise levels on noise rating curves (NR), a set of "equal-loudness contours" based on the sensitivity of the human ear [1]. Therefore, for the radiant unit, we have presented the sound pressure values vs. frequency for the most inconvenient situation, air pressure P500 at 100% power-charging. It can be seen that the noise levels for the three tested radiant units are rated according to NR 35, at most NR 38 curve. The noise levels correspond to the HVAC requirements and they can be declared as compliant.

4. Conclusions

The sources of noise in HVAC systems are multiple, but the end users are in particular annoyed by the noise generated from the radiant unit and the air circulating ducts, as they are located inside the rooms and buildings. The paper focuses on the sound power/pressure study developed in these commercial HVAC components. It can be notice a good noise rating for the radiant units, due to their suitable sound insulation design. The best noise behaviour was recorded for AHU 1, both for radiant unit and for air ducts, possible because of its constructive type (the duct diameter is greater than the others tested systems, as is denoted in table 1). Even if AHU 3 has sound power fluctuations due to the heat exchanger and one can be worried about this, the noise rating is good (NR 35). The fan power-charging, the air flow pressure, the noise generated in the system may constitute acoustic performances of the HVAC. A sound attenuation strategy include the placing of in-duct silencers, a high absorptive material use lining the ducts with a special attention on regenerated noise caused by the flow direction and velocity (see supply vs. outdoor ducts noise). This will be a future work.

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