Some considerations on noise monitoring for air handling equipments

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Abstract. The HVAC (Heating, Ventilating and Air Conditioning) beneficiaries are in particular annoyed by the noise generated from the radiant unit and the air circulating ducts, since they are located inside the rooms and buildings. The comparatively experimental results highlight the relations between the air flow, pressure, power-charging and the sound level. 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. Third-octave band analysis of random noise of the handling units is realized in an anechoic room, using the measurement procedures that agrees the requirements of the ISO 3744:2011 and ISO 5136:2010 standards. For an accurate design of the HVAC system, the designer needs to know not only the sound power of the radiant unit, but also from all of the air paths, since the sound travels along with the conditioned air. The experimental methodology used in the paper is of real interest for the HVAC manufacturers, in order to rate the sound level of their products and to improve the noise attenuation.

1. Introduction
The design for buildings, offices and rooms requires a clear specification of acoustical systems and products location, needing some quantitative information because the noise criteria regulations become more and more strictly. HVAC (Heating, Ventilating and Air Conditioning) systems from the modern buildings are indispensable for the human comfort and their performances must also agree the imposed noise regulations.

The air handling equipment (AHU) involves any type of HVAC system that contains a fan and is used to condition and move air through a duct system [1]. The air handling unit is a good example to prove the difficulty in accurate sound data evaluation since there are many possible paths between the noise source and the receiver.

Each fan, as component of an air handling product is operating at various speeds and within a range of flow and static pressure conditions. However, the fan inside the air handler changes the sound performances due to its housing that modifies the airflow pattern at the inlet and outlet [2]. Moreover, the sound may have different ways to leave the air handler, as it is reported in this paper. There are supply, extract, exhaust and outdoor air ducts along which the sound also travels. Also, the air velocity, the material roughness and the fittings of the ducts significantly increase the friction in the air handler system and obviously the noise generation [2]. Depending on the geometrical inlet/outlet openings configuration of the air handler, the sound passes through the ducts in the direction of...
airflow or against it. In this case, it travels back out with the return air inlet and the regenerated noise modifies the sound level from the source. Therefore, these paths together with the air handler radiant unit including fan and dampers provide a complex acoustic situation, difficult to be predicted by theoretical algorithms. The computer codes predicting the air handler noise may have potential errors as they calculate the sound level contributions of each element of a system. Nevertheless, numerical techniques for acoustic propagation in dissipative ducts and radiant unit, such as two-port transfer matrix method [3, 4], finite and boundary elements methods [5, 6] have been developed. There are also algorithms and procedures based on sound power computation [7].

In this paper there are experimentally investigated two types of commercial air handler units (AHU) with the same ducts diameters, but different inlet-outlet configuration. The comparatively results highlight the relations between the air flow, pressure, power-charging and the sound level. Third-octave band analysis of random noise of the handling units is realized, using measurement procedures that agree the requirements of the ISO 3744:2011 and ISO 5136:2010 standards [8, 9]. For an accurate design of the HVAC system, the designer needs to know not only the sound power of the radiant unit, but also from all of the air paths, since the sound travels along with the conditioned air.

2. Experimental methodology
The HVAC manufacturers are gathering information about the fan power, duct pressure vs. air flow variation, temperature, sound data and these performance parameters are included in their catalogues [10, 11, 12]. Sound power levels are usually determined in acoustics laboratory, by measuring sound pressure levels in test facilities using some specific procedures [8, 9] in order to realize data compatibility between different manufacturers.

HVAC system designers 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. 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].

The sound power level $SWL$, according to ISO 3744: 2011, is the sound energy constantly transferred per second from the sound source. A sound source has a constant sound power that does not change if it is placed in a different room environment. Sound power is a theoretical value that is not measurable.

$$SWL = 10 \log_{10} \left( \frac{P}{P_0} \right).$$

with: measured sound pressure $p$ and the reference sound power level $p_0 = 10^{-12}$ W.

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

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

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
The two procedures of determining the sound power of air handlers equipment are the reverberant room and free field methods.

The reverberant room method has the aim to create a diffuse sound field through multiple reflections of sound waves, realizing the same sound pressure inside the room. The free field method needs an homogenous medium without boundaries or reflecting surfaces. In this case, the sound pressure values are recorded on a hemispherical surface surrounding the equipment and then, the sound power values are calculated.

For our measurement, we have used the free field method because we dispose of an anechoic room, in 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 125Hz ÷ 20kHz frequency band. Thus, the sound pressure level in any direction may be measured and the directivity factor \( Q \) is 1.

The test facility realized in accord with ISO 3744:2011 and ISO 5136:2010 standards is presented in figure 1.

![Figure 1. Test arrangement for in-duct noise measurements, according to ISO 5136: 2010.](image_url)

1 flow direction 2 air handler unit AHU 3 transition duct 4 intermediate duct 5 test duct 6 pressure taps 7 anechoic termination with \( l \geq 4d \) or \( l \geq 1m \)

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

For various fan’s speeds, expressed in power-charging variation between 30 and 100%, the sound level data are acquired. The ducts air flow is slowly adjusted from full closed to full open for each range of charging.

The next step is to analyze the sound level dB(A), A weighted, both on radiant unit and the air ducts, using the third-octave analysis with LabVIEW soft, evaluating the pressure levels (dBA) for nominal frequencies of 63, 125, 250, 500, 1000, 2000, 4000, 8000 Hz. Also, the analysis evidenced the equivalent continuous sound level that is an averaged noise in a specified time interval.

Figure 2 describes typical air paths of handling unit, namely the ducts, in order to a better understanding of the air travel way and the sound that accompanies it.
The sound level tests where performed on two air handling commercial types denoted with AHU 1 and AHU 2. The main geometrical data are depicted in the figures 3 and 4, respectively. Also, we mention: the air flow capacity (350 m$^3$/h for AHU 1 and 300 m$^3$/h for AHU 2), the same weight (50 kg) and the air ducts diameters of Φ160 mm.

The important difference between the two air handlers consists in the air flow direction through the ducts system, which can modify the sound level due to the regenerated noise occurrence in a reverse flow. The sound may leave the air handler in multiple ways because the supply, exhaust, extract and outdoor air ducts can have different ways to cross the radiant unit.

![Figure 2. The sound paths in the air ducts.](image)

![Figure 3. AHU 1 geometrical data.](image)

![Figure 4. AHU 2 geometrical data.](image)

![Figure 5. Experimental setup - outside and inside the anechoic room.](image)
Figure 5 depicts the experimental setup realized in accord to ISO 3744:2011 and ISO 5136:2010 standards, using the anechoic room facilities from our department.

3. Results and discussions
Figure 6 represents a comparative evaluation of the total sound level for the tested radiant units running at different air flow pressure from 0 Pa to 500 Pa (the worst case in exploitation) taking into account of the fan power-charging. AHU 1 radiant unit has the lowest noise level and uniform noise behaviour with charging increase. A very quiet functioning at low charging and P0 air flow pressure can be also noted. AHU 1 sound level values range between 35÷45 dB and it can be observed that it is a little less noisy than the AHU 2 radiant unit (40÷50 dB).

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 total sound level variation related to the fan power-charging is expressed in figure 7 for the supply air ducts of the tested AHU 1 and AHU 2 units, ranging the air flow pressure from 0 Pa to 500 Pa. The total sound level values are much higher (65÷105 dB) than those for the radiant units and the sound increasing tendency with fan power-charging preserved. Also, the supply air duct of AHU 2 has the best performance in sound attenuation and similar to the supply air duct of AHU 1 at 500 Pa air flow pressure. The supply air duct of AHU 1 it is noisier than the supply air duct of AHU 2 at high fan power-charging and low air flow pressure.

The same tendency is recorded for the others air ducts (figures 8, 9 and 10). The total sound level is higher in the air ducts of AHU 1 unit at high fan power-charging and low air flow pressure, and it is decreasing at high air flow pressures, similar to the air ducts of AHU 2 unit, where the total sound level doesn’t varies very much with the air flow pressure.
A possible explanation of the fact that AHU 1 noise level in the air ducts is greater than the sound level of AHU 2 ducts consists in the air paths configuration. Figures 3 and 4 show the difference in the air flow directions through the ducts system, which modify the sound level. The regenerated noise in the AHU 1 air ducts is more intense since there is reverse air flow due to the disposition of the ducts (AHU 1 has the ducts inlet-outlet on the same side). On the other hand, the AHU 1 radiant unit is a little less noisy than the AHU 2 radiant unit, thus the inside attenuation elements do their job more efficient compared to the AHU 2 unit.

4. Conclusions
The HVAC beneficiaries are in particular annoyed by the noise generated from the radiant unit and the air circulating ducts, since they are located inside the rooms and buildings. The paper focuses on the analysis of the equivalent continuous sound level in two commercial air handlers units. The noise level of the radiant unit complies with the standards requirement (about 50 dB) and the ducts sound level tendency emphasizes the importance of the ducts inlet-outlet mounting. They are influencing the air flow direction and, consequently, the sound level. The fan power-charging, the air flow pressure, the noise generated in the system constitute acoustic performances of the HVAC. In order to optimize their design and efficiency from this point of view it is necessary the in-duct noise control, generally solved through the silencers use, thus adding adequate attenuation to the system. A suitable silencer selection is imposed by the duct noise level. Moreover, identifying the noise sources due to frictional causes inside the air handler equipment is a necessary step in the HVAC acoustic study. A future work
will be focused on the friction rate. Therefore, the experimental methodology used in the paper is of real interest for the HVAC manufacturers, in order to rate the sound level of their products and to improve the noise attenuation according to the specified standards.

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