The 40th International Conference on
Mechanics of Solids, Acoustics and Vibrations &
The 6th International Conference on
"Advanced Composite Materials Engineering"
ICMSAV2016& COMAT2016
Brasov, ROMANIA, 24-25 November 2016

EXPERIMENTAL INVESTIGATION OF NOISE PARAMETERS IN HVAC SYSTEMS

Marcelin Benchea[1], Carmen Bujoreanu[2]
[1] "Gheorghe Asachi" Technical University of Iasi, Romania, marcelin.benchea@tuiasi.ro
[2] "Gheorghe Asachi" Technical University of Iasi, Romania, carmen.bujoreanu@gmail.ro

Abstract: *Heating, ventilating and air-conditioning (HVAC) systems represent a major source of inside buildings noise with a negative effect on the acoustical environment, so beneficiaries are in particular annoyed by the noise generated from the radiant unit and the air circulating ducts. The technical data presented in manufacturer's catalogues highlight the relations between the air flow, pressure, power-charging and the sound level. The measurements presented in this paper are carried out at different fan's speeds, ranging the power-charge from 30% to 100% and static air pressure is slowly adjusted between 0 Pa to 500 Pa. Third-octave band analysis of random noise of the handling units is realized in an anechoic room and agrees the requirements of ISO 3744:2011 and ISO 5136:2010 standards. The experimental methodology used in this paper is of real interest for the HVAC manufacturers in order to rate the sound level of their products.*
Keywords: *HVAC systems, sound level, acoustic, air flow, third-octave band analysis.*

1. INTRODUCTION

HVAC systems are indispensable equipments for the modern buildings in order to ensure the human thermal comfort and admissible air quality. The HVAC system includes air handling unit (AHU), 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. Their performances must also agree the imposed noise regulations.
The air handling equipment involves any type of HVAC system that contains a fan and it is used to condition and move air through a duct system [2]. 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.
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 boxes and air outlets such as diffusers, grilles, dampers and registers. Therefore the most significant noise sources are the air handler equipment and the air circulating ducts [4]. 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 [3]. 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. 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.
Numerical techniques for acoustic propagation in dissipative ducts and radiant unit, such as two-port transfer matrix method [3, 6], finite and boundary elements methods [5, 7] have been developed. There are also algorithms and procedures based on sound power computation [5].
In this paper are investigated two types of commercial air handler units with the same ducts diameters, but different inlet-outlet configuration. 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 [11, 12].

2. EXPERIMENTAL TESTS AND PROCEDURES

Software 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 [7]. Therefore, potential errors may appear and the calculated results could be quiet far from the real situation.
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 [1, 13, 14]. Sound power levels are usually determined in acoustics laboratory, by measuring sound pressure levels in test facilities using some specific procedures [11, 12] in order to realize data compatibility between different manufacturers.

2.1. Terminology

HVAC system manufacturers evaluate the sounds in the frequencies between 45 Hz 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. In terms of sound perception, the human ear is most sensitive in the frequency range 1000 Hz to 4000 Hz than 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 [10].
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.

$$SWL = 10 \cdot lg \frac{p}{p_0} \tag{1}$$

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

$$SPL = SWL + 10 \cdot lg\left(\frac{Q}{4 \cdot \pi \cdot r^2} + \frac{4}{R_C}\right) \tag{2}$$

where *SWL* (dB) is the sound power re 10^{-12} W; *SPL* (dB) is the sound pressure re $2 \cdot 10^{-5}$ 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 α_W the average absorption coefficient [1].
Relation (3) is simplified and becomes:

$$SPL = SWL - 11 \tag{3}$$

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 measured background noise in our anechoic room is: $SWL = 28.9$ dB or $SPL = 17.9$ (distance from the source $r = 1$ m).

2.2. Experimental Tests

The experimental setup for acoustic measurements [8] consists of:
- device for measuring the pressure drop (Fluke 922 pressure differential meter),
- the test object (AHU),
- the transition elements on either side of the test object (air ducts),
- special receiving-sound equipment consisting of:
  1) microphone Bruel&Kjaer type 4133 (sensitivity 4 - 16 mV on N/m^2; 0.4 - 1.6 mV on µbar; 36 - 150 dB(A)),
  2) sonometer Bruel&Kjaer type 2209 (amplification: 2 Hz or 10 Hz at 70 kHz; standard filters according IEC R179, IEC R179A and ANSI Type 1) connected with the microphone and NIDAQ board,
  3) multifunctional external data acquisition board type-NIDAQPad-6015 on USB,
  4) laptop with LabVIEW soft compatible with National Instruments DAQPad for data processing.

The experimental investigation is performed in the anechoic room using a test facility that agrees with ISO 3744:2011 and ISO 5136:2010 [11, 12].
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. There is analyzed 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 1 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, while figure 2 shows the frequencies and a descriptive terminology at which different types of HVAC components influence the sound spectrum in a room.

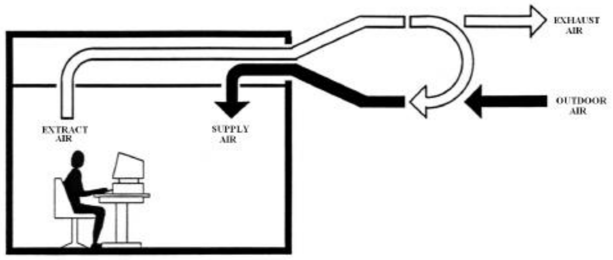

Figure 1: The sound paths in the air ducts

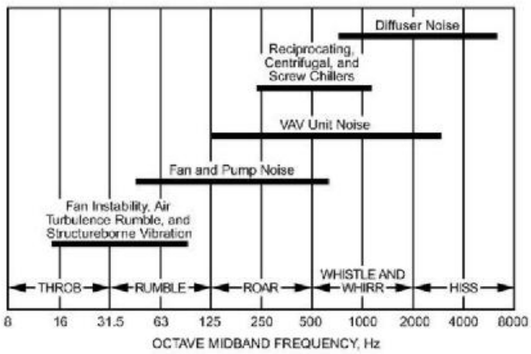

Figure 2: Frequency ranges for HVAC components

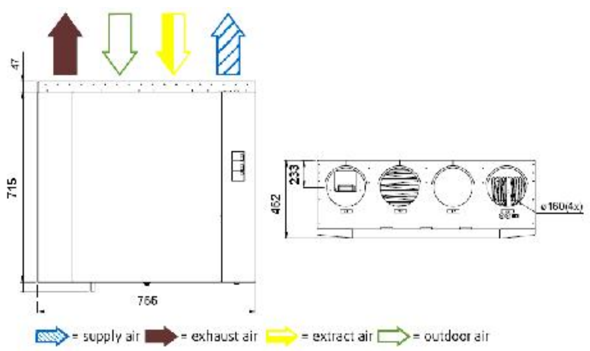

Figure 3: AHU 1 unit

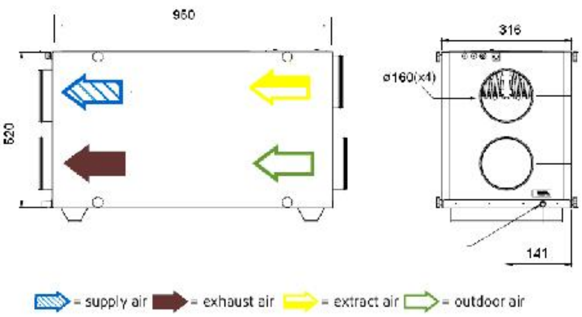

Figure 4: AHU 2 unit

The experimental setup is presented in figure 5.

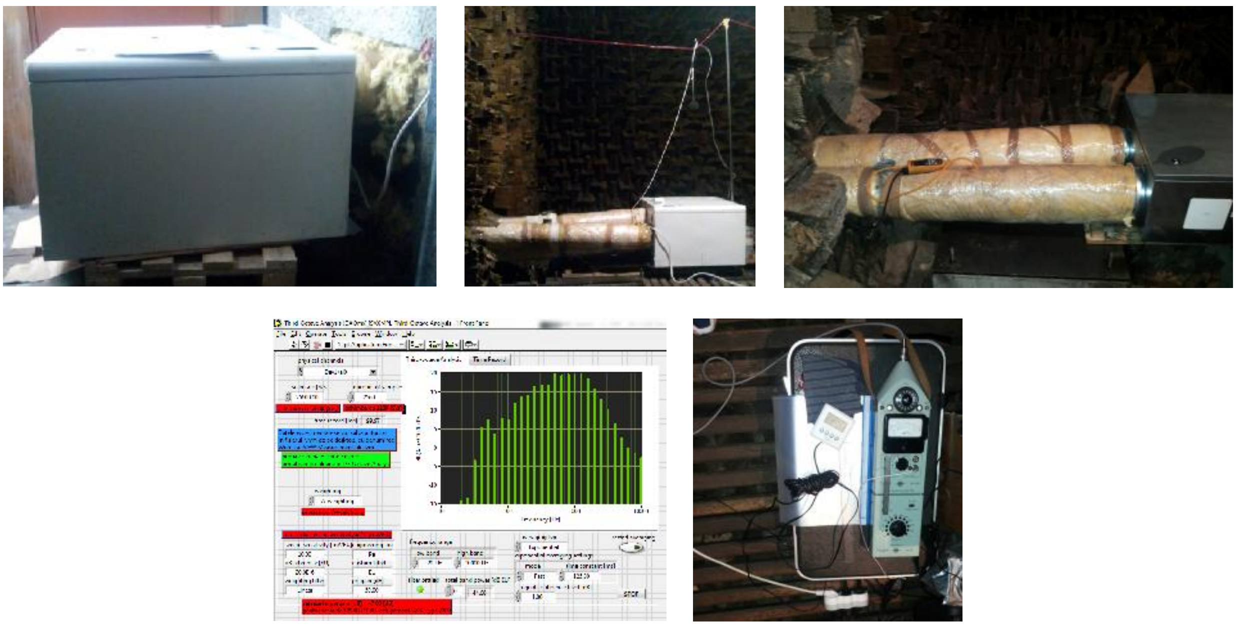

Figure 5: Experimental setup

3. EXPERIMENTAL RESULTS

Figures 6 and 7 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 0 Pa 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).

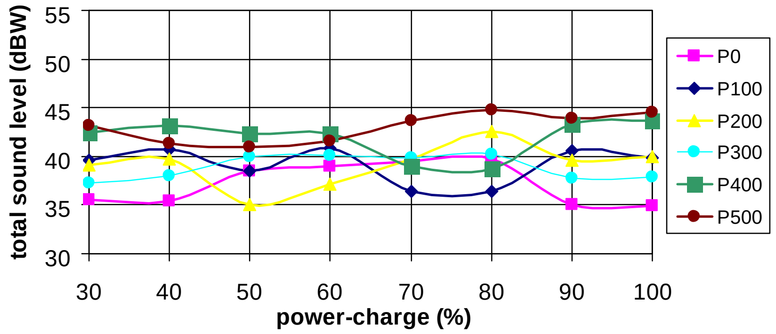

Figure 6: Sound level for the AHU 1 radiant unit

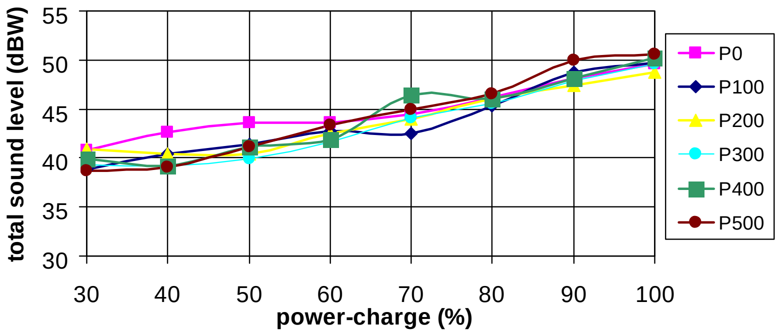

Figure 7: Sound level for the AHU 2 radiant unit

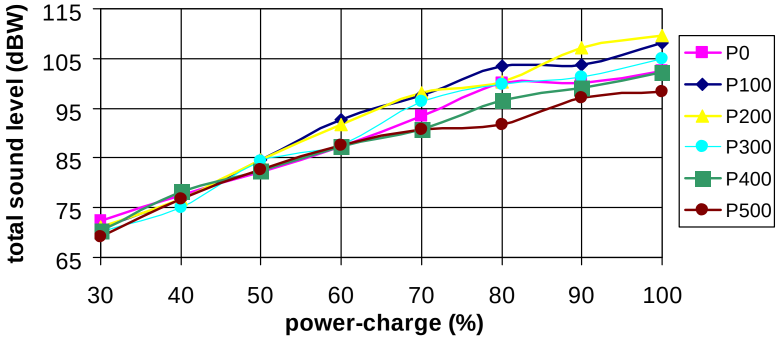

Figure 8: Sound level for AHU 1 supply air duct

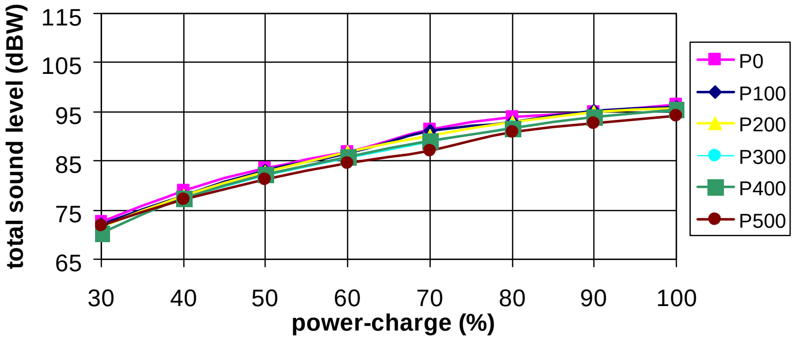

Figure 9: Sound level for AHU 2 supply air duct

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 figures 8 and 9 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 10 to 13). 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.

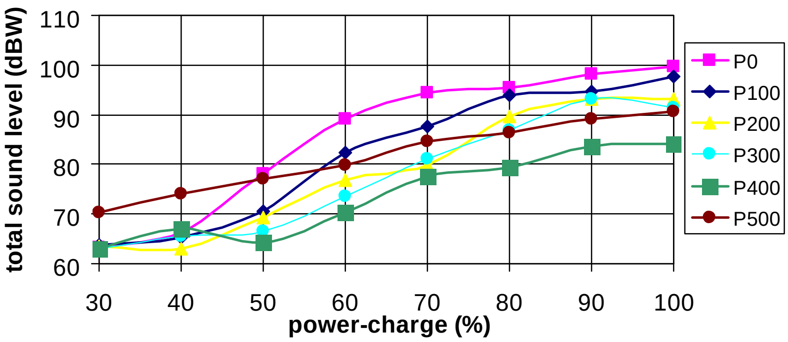

Figure 10: Sound level for AHU 1 exhaust air duct

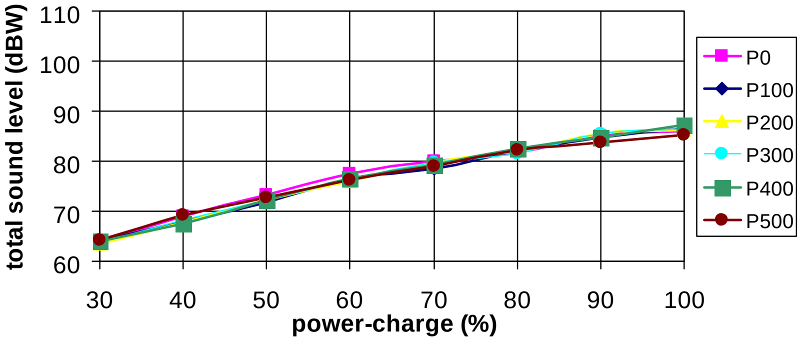

Figure 11: Sound level for AHU 2 exhaust air duct

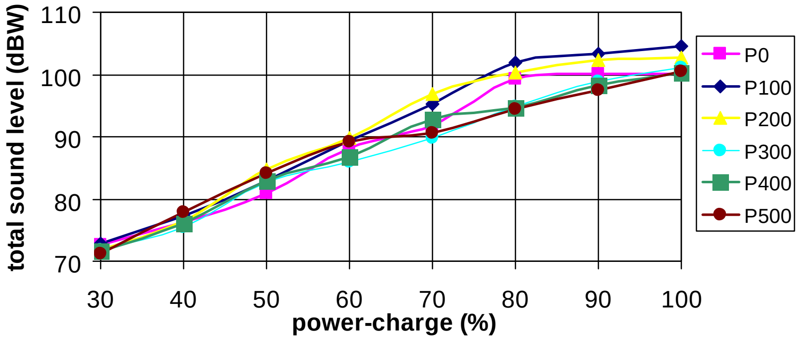

Figure 12: Sound level for the AHU 1 extract air duct

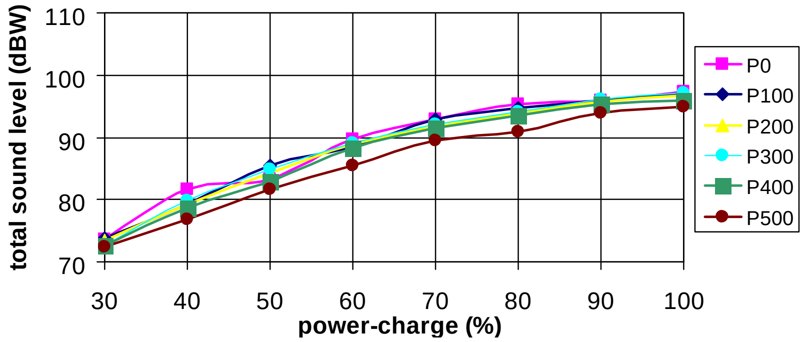

Figure 13: Sound level for the AHU 2 extract air duct

4. CONCLUSION

The sources of noise in HVAC systems are multiple, but 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 presents 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 (NR 35 - NR 40) and the ducts sound level tendency emphasizes the importance of the ducts inlet-outlet mounting. They are influencing the air flow direction and the sound level. The fan power-charging, the air flow pressure, the noise generated in the system constitute acoustic performances of the HVAC systems. 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. 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.

REFERENCES

[1] Air Change Catalogue - Next-AHU™ Technical Data pp 1-4, 2014.
[2] ASHRAE - Handbook-Sound and Vibration Control, chapter 48, pp.279, USA, 2011.
[3] Badino A., Borelli D., Gaggero T., Rizzuto E. and Schenone C. "*Normative framework for ship noise: Present situation and future trends*", Noise Control Engineering Journal, 60(6), 740-762, 2012.
[4] Bhatia A., HVAC Systems Noise Control, Continuing Education and Development Inc., pp.109, 2014.
[5] Bies D.A. and Hansen C.H., Engineering Noise Control-Theory and Practice, Spon Press,NewYork, 2003.
[6] Boden, H. and Glav, R., "*Exhaust and Intake Noise and Acoustical Design of Mufflers and Silencers*", Handbook of Noise and Vibration Control, John Wiley & Sons, 2007.
[7] Borelli D. and Schenone C., "*A finite element model to predict sound attenuation in lined and parallel baffle rectangular ducts*", HVAC&R Research, 18 (3) pp. 390-405, 2012.
[8] Bujoreanu C., Benchea M., "*Experimental study on HVAC sound parameters*", IOP Conference Series: Materials Science and Engineering, vol. 147(1), 012051, 2016.
[9] Cory W., Fans & Ventilation: A Practical Guide, Elsevier, pp. 424, 2005.
[10] Desmond S.D., "*Fans in air handling unit*", Woods Air Movement Limited UK, pp. 23, 2000.
[11] ISO 5136:2010 - Determination of sound power radiated into a duct by fans and other air-moving devices. In-duct method.
[12] ISO 3744:2011 - Determination of sound power levels and sound energy levels of noise sources using sound pressure. Engineering methods for an essentially free field over a reflecting plane.
[13] Lindab Catalogue - Ventilation Guide, pp. 1-22, 2008.
[14] System Air Catalogue - Air Handling Units, pp. 1-138, 2012.
[15] Ver I.L., Beranek L.L., Noise and Vibration Control Engineering: Principles and Applications, John Wiley & Sons, 2005.