DOI: 10.61782/fa.2023.1238

forumacusticum 2023

ASSESSMENT OF HVAC-RELATED BACKGROUND NOISE IN OFFICES. ACCURACY OF SINGLE NUMBER PARAMETERS AGAINST SPATIAL DISTRIBUTION OF SOUND

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ABSTRACT

Acoustic design must ensure the HVAC noise to be sufficiently low and unobtrusive in quality, so as it does not interfere with requirements for occupant use. For instance, when background noise reduces speech intelligibility, productivity can be lost. Accordingly, LEED, WELL and BREEAM quality protocols propose for rating HVAC-related background noise parameters as SPL, NC and RC. The work aims to determine how significant single-number parameters are to the spatial distribution of perceived loudness and sound quality, considering that HVAC-related noise primarily propagates from supply air system's path or breaks out through ductwork. Simulations were performed on rooms of an office building, of the same intended use (open office) but different in volumes, which comply with protocols guidelines. Spatial distribution of SPL, NC and RC was analyzed, to evaluate the necessity to impose compliance, in addition to an average value, with thresholds at significant points. The impact of radiative noise contribution of the mechanical equipments on single number parameters and maps was estimated, considering the difficulty in simulating it compared to airborne noise. Identified background noise levels were adopted to understand how STI and derived parameters vary in open-offices and ratings following quality protocols and intelligibility standard were compared.

Keywords: *equipment noise, acoustic simulation, acoustic measurements, office buildings, international building certifications*

1. INTRODUCTION

To guarantee adequate comfort in living environments noise emitted by HVAC systems must be low enough. Such problem is very much felt in buildings intended for office use, and in order to try to reduce it as much as possible, correct plant design is necessary [1, 2].The noise reaching the environment from HVAC is due to transmission of sound air-borne by ducts, air-borne by structures and structural by structures. Plants generally consist of technical stations, distribution networks (air or water) and terminals (radiators, fan coils, etc.).Acoustic emissions due to HVAC systems cover a large part of the audible spectrum: from low frequencies (16 – 63 Hz) with the rumble due to airflow and turbulence-generated noise in a duct, to mid frequencies (125 – 500 Hz) with the roar of fans and VAV boxes, to high frequencies (1000 – 4000 Hz) with the hiss of dampers and diffusers. Occupants evaluate the comfort in an environment on two factors: perceived loudness of the noise relative to the normal activities and sound quality of the background noise. HVAC related noise must be low so as not to interfere with activities, and not to affect intelligibility of speech, reducing productivity at work. Methods for evaluating the noise produced by an HVAC system include the traditional A-weighted sound

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pressure level (dBA), tangent Noise Criteria (NC) and Room Criteria (RC). Literature provides recommended background noise levels (BNL) for spaces with HVAC systems based on intended use, e.g. offices and conference rooms are listed as NC/RC 30. The NC/RC curves refer to sound pressure levels (SPL) at octave bands selected on the basis of appropriate loudness in the speech interference range (500 - 2000 Hz) and show contours for high and low frequencies balanced at the same loudness level. Noise criterion (NC) was established in U.S. in 1957 to value indoor noise. It is based on a set of sound pressure curves ranging frequencies from 63 to 8000 Hz, which define the limits of the octave band spectra that must not be exceeded to meet the occupants' acceptance in the spaces. The noise spectrum is classified by the same NC rating as the lowest NC curve not exceeded. Room Criteria (RC) curves were proposed by Blazier for the American Society of Heating Refrigeration and Air-Conditioning Engineers (ASHRAE) in 1981. Sound pressure levels in nine bands (16 - 4000 Hz) are plotted, and, to the curve individuation, subjective judgment must be added to report the level. The RC curve is a straight line with a slope of 5 dB per octave band. Acoustic evaluation based on octave bands is recommended: because broadband SPL in dB(A) does not reflect undesirable contributions of low-frequency noise, SPL in dB(C), which takes them into account, is also evaluated. For acoustic comfort, the noise produced by the HVAC system should not have predominant frequency bands neither audible tones such as hum. Speech Transmission Index (STI) allows to evaluate acoustic comfort in confined spaces. It expresses quality of speech transmission, between a speaker and a listener, in terms of intelligibility, quantifying the combined effect of background noise interference and reverberation. STI values range from 0 to 1, which indicate respectively no and total intelligibility (Tab. 1).

Table 1. STI perceptual rating.

Value of STI	Speech quality (CEI EN 60268-16)
0 < STI ≤ 0.3	Bad
0.3 < STI ≤ 0.45	Rare
0.45 < STI ≤ 0.6	Acceptable
0.6 < STI ≤ 0.75	Good
0.75 < STI ≤ 1	Excellent

In international WELL protocol the contribution of mechanical equipments is counted in Feature S02, about BNL, defined as Leq in dB(A) or dB(C) at 1kHz considering HVAC on, and in Feature S03, about equipment contribution to Speech Privacy Potential (SPP), expressed by Noise Criterion (NC) [3].In the LEED protocol BNL limits from HVAC systems are the ones reported in 2015 ASHRAE Handbook-HVAC Applications, Chap. 48, Table 1 [4]. There, in addition to acceptable values expressed in dB(A), reference is made to RC curves [5]. In the Italian regulations the main reference standards are D.P.C.M. 14/11/1997, concerning systems' acoustic impact to the surrounding environment and D.P.C.M. 5/12/1997, which considers plants noise inside the building that houses them. The latter is joined by the recently updated UNI 11367 [6], which defines the reference classes for various acoustic parameters, including those related to noise induced by continuous and discontinuous systems. In [6] plant noise descriptors change with respect to DPCM 5/12/97: correct noise level of continuous operation systems (Lic) and correct noise level of discontinuous systems (Lid). However, they are always single-number indices, derived from the same parameters although corrected for residual noise and reverberation time. Prediction of HVAC system noise in a room is difficult both for path noise estimation and for receiver room's behavior interpretation. In many offices air supply constitute an array of distributed ceiling sound sources. The distribution depends on geometric aspects of spaces [7] and thermal loads, where these are equal outlets emit nominally equal sound power levels [4]. To calculate SPL could be very tedious for large rooms with high number of devices and air ducts. Moreover just adding calculated SPL for each element at specified locations in the room increase in inaccuracy per non-array distribution of the sources, irregular geometries of the office and presence of sound-scattering surfaces. [4] points out that the classic diffuse room equations for a source may not produce accurate results for standard furnished rooms [8], highlighting the need to improve their correction. Knowledge of the accuracy of equations used in analytical methods and their approximation to spatial distribution is fundamental in environments in which acoustical conditions are largely affected by background noise, responsible for discomfort and reduction of privacy distance, as open plan offices. From the literature structure-borne sound radiated through walls and other building's elements result of minor impact on the annoyance with respect to airborne noise [9]. That is why in the present study it was neglected. The work aimed to determine how significant the parameters described above are, comparing the single-number values obtained from the predictive analytical methods of the standards with the spatial distribution found from simulations by Odeon software. The analyses were performed on four offices of different volumes and intended uses.

2. CASE STUDY

Simulations were performed on four open office rooms of an office building, different in volumes and shapes, which comply with protocols guidelines (Fig. 1, Tab. 2). The boundaries are floating floor with stoneware finish, metal countertop at the area occupied by the HVAC system terminal units and plasterboard in the remaining space, perimeter walls and shared with stairwells (OO.t1 and OO.t15) plastered, walls shared with other offices in plasterboard and glass, wooden or glass doors and double glass windows (Rw=42 dB). The furnitures are chairs with αw=0.7 and desks with αw=0.11. As sound sources the mechanical elements shown in Table 3 were taken into account: electronic fan coil with inverter of the four-tube type with two batteries, operating, depending on the type, at speeds of 6 or 7 volts. The air network crossed by the noise from the source to the diffusers which lead it in the room, consists of the elements reported in Table 4.

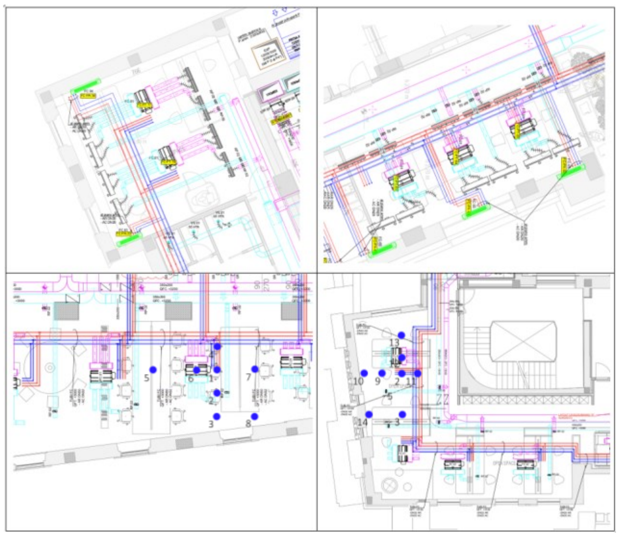

Figure 1. Investigated rooms: (a) OO.t1; (b) OO.t3; (c) OO.t12; (d) OO.t15.

Table 2. Details of the investigated rooms.

Room	Volume [m3]	∑S·α [m2]	∑S·α/V	Fsch [Hz]	Mechanical Equipement
OO.t1	151.2	48.4	0.32	126	FC01 (2)
OO.t3	149.69	38.0	0.25	127	FC01(3)
OO.t12	176.31	70.5	0.40	117	FC01 (3)
OO.t15	247.97	70.8	0.29	98	FC01 (2), FC02 (2)

Table 3. Sound sources considered for the purposes of HVAC background noise level (LEED 1).

Type	FC.01			**Speed[V]**	6			
	Lw [dB(A)]							
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**Glob**
Inlet+Rad	37.9	45.3	48.1	48.0	44.3	37.3	27.6	**53.0**
Outlet	35.6	38.1	42.3	35.1	32.4	28.0	20.6	**45.2**
Sum	39.9	46.0	49.1	48.2	44.6	37.8	28.4	**53.7**
Type	FC.02			**Speed[V]**	7			
	Lw [dB(A)]							
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**Glob**
Inlet+Rad	40.6	48.0	50.5	50.8	47.5	40.8	31.1	**55.8**
Outlet	37.9	40.5	44.5	37.5	35.1	31.2	23.0	**47.5**
Sum	42.5	48.7	51.5	51.0	47.7	41.3	31.7	**56.4**

Table 4. HVAC system.

Element	**n**	**Details**
Fan coil	1	Ductable, with electronic motor with inverter - horizontal mounting - Four-tube (2 batteries) - Speed 6.0/7.0 V
Inlet/outlet section		
Plenum (PAL)	1	1.200x0.167x0.241m
Straight duct (aluminum)	3*	Depending on the room
Flexible duct (Micro-perforated aluminium/polyester-glass wool- aluminium/polyester)	3	d=0.160m, l=0.500 m
Linear Diffuser	2	0.142x1.200m

**connection between plenum and flexible duct, not present in all rooms*

3. METHODOLOGY

HVAC noise levels mainly results from three transmission paths [4]: 1) Direct sound radiated from sound source to ear and Reflected sound from walls, ceiling and floor; 2) Air- and structure-borne sound radiated from casings and through walls of ducts and planum, transmitted through walls and ceiling into the room; 3) Airborne sound radiated through supply and return air ducts to diffusers in room and then to listener by direct or reflected path. An analytical method and a numerical model were implemented and compared.

3.1 Analytical model

The evaluation of noise transmitted from the air-to-water plant to the served environment was carried out analytically for OO.t1 both by the simplified calculation model [1, 2] and that of the UNI EN 12354-5 standard [10]. The most appropriate procedure for this type of sources was therefore defined, applying it to the remaining offices. From the detailed analytical model of UNI 12354-5, the additive criterion of the contributions due to different kind of sound generation and transmission of these sources was borrowed. Moreover the estimated sound pressure levels in accordance with this standard use the diffuse sound field theory, also at low frequencies. Some studies indicate that these sound levels overestimate the measured room average at low frequencies. The estimated levels at low frequencies are on the safe side, with Equation G.1 of Annex G giving an indication of the safety margin. For evaluating the pressure introduced by propagation in ducts it was considered that the UNI 12354-5 term $Ln,_d$ do not adequately take into account absorption coefficients and reverberation time of the environment. Known the sound power level produced by the sources, fans in this case, the amount which reaches the environment $(Lw,_1)$ was calculated taking into account the attenuation which occurs in straight sections covered with sound-absorbing material, in curves and connections, in branches, plenums, if any, and terminals by equations and graphs contained in [2]. Concerning branches, it can be assumed that the acoustic power associated with the main duct is distributed among the secondary branches proportionally to the air flows. In this case each plenum (Tab. 4), covered internally with absorbent material with α=0.7. The values of energy attenuation by reflection, at the terminal mouth of a duct, were evaluated with reference to the gross surface of the diffusers, neglecting slits and baffles which generally do not make a significant contribution in terms of reflections, except at very high frequencies with lengths comparable to their dimensions [2]. We then consider the lateral escapes(Lw,lat), namely the sound power transmitted through the duct's wall or the fan coil's case:

$$Lw,lat = \Delta Lwi - R_c + 10 * \log\frac{S_p}{S_c}$$

Where: Rc is the sound insulating power of the duct wall [dB]; Sp is the duct wall area [m^2]; Sc is the duct wall area is the duct section area [m^2].

For noise radiated by fan coil system in the suspended ceiling only direct contribution was considered:

$$Lw,_2 = Lw,lat - \Delta Lwi$$

Where: ΔLwi is the contribution of suspended=contributo di suspended ceiling and insulating mats [dB].

The equation of UNI 12354-5 for $Ln,_a$, appropriate for a receiving environment other than the one in which the noise is generated if the latter is produced in technical rooms, was considered excessive for noise produced by fancoils in false ceiling.

It is possible then to determine for $Lw,_1$and $Lw,_2$ ambient sound pressure, considering it diffusive.

$$L_p = L_w - 10 * logA + 6$$

In the end $Lp,_1$ and $Lp,_2$ were added following 12354-5.

3.2 Numerical model

By Odeon software, 3D models of the four offices were realized in order to carry out numerical simulations. Odeon is a provisional software, which uses a hybrid geometric method based on pyramid ray tracing. Low-order reflections (early reflections) are modeled using the image source method, while the reverb tail (due to later reflections or late reflections) by means of secondary sources positioned on the walls at the points of the last reflections. Moreover, being energy based, therefore not calculating resonant-modes, it is suitable for simulations of medium-large spaces, whilst in small rooms, where low frequencies may be significant, may not provide reliable results. For the sources analyzed in this work, diffuser and fan, ASHRAE Handbook [4] defines noise ranges in 1000-4000 Hz range and around 500 Hz respectively, so the model here developed could be extended to smaller offices. As a further verification, for each room the Shroeder frequency (Fsch) was calculated, to identify the watershed between the behavior of the room as resonator or as reflector. (Tab. 2).The SPL at a given location in a room caused by a specific sound source is a function of the sound power level and radiation characteristics of the source, acoustic properties of the room surface treatments, room volume and source-receiver distance. Open plan offices consist very often of zones where the ceiling materials are of different types or the furniture design differs significantly. As indicated by the standard [10] the measurements should preferably be made in each zone. For this reason, even in the case of larger open offices (OO.t1) representative areas of homogeneous zones, in materials and furnishing, were identified. A completely absorbent fictitious material was assigned to the boundaries of the area to be studied, in case they did not correspond to real surfaces, but were necessary to delimit an analysis box. Receiver positions in workstations correspond to the person's head (1.20 m from the ground). The positions shall be at least 0.5 m from tables and at least 2.0 m from walls and other reflecting surfaces. Two kinds of sound sources are typically encountered in HVAC system applications: point (grilles, diffusers, fan-coils) and line (associated with sound breakout from ducts) [4]. In the model, as sources for the aerial sound radiated through ducts, diffusers were considered. They were assessed as sources with directivity factor Q=2.Concerning air-borne sound radiated from ducts' walls and transmitted through ceiling into the room, omnidirectional point sources were set up to simulate fans, while the noise emitted by the ducts was reproduced by considering aligned omnidirectional point sources, the power emitted from which was assumed as the lateral escapes of the duct (See Section 3.1).The suspended ceiling was simulated as "transmission surface", to take into account the permanence of the sound in the gap between it and the slab of the upper floor, and the passage of part of the noise towards the main environment. The corresponding sound reduction indexes for each third octave band were assigned to it.

3.3 Validation

The values found in an experimental campaign in a typical office (12 m^2) of the same building of the analyzed ones (Fig.2), were employed to validate the methodologies and adapt the theoretical absorption coefficients to the actual of the materials used. UNI 8199 was followed for measurements. In both analytical and numerical simulations both the airborne noise through the channels and the airborne noise through the structure due to the irradiation of the fan and the lateral escapes from the ducts were considered. The fan was set to 4V speed; the room was evaluated in unfurnished configuration.

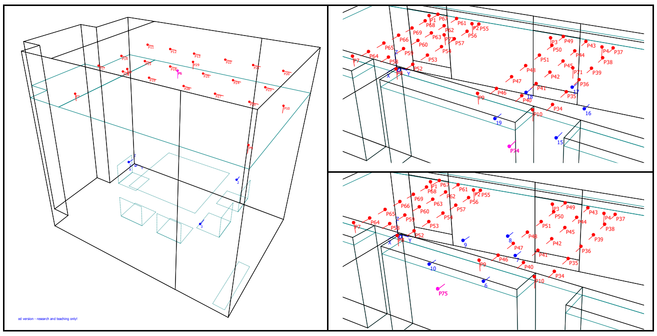

Figure 2. (a) 3D acoustic model of validation room; Source-receiver positions for STI evaluation in OO.t12: (b)Under; (c)Far.

The average SPL obtained in the office (40.6 dB(A)) was compared with that calculated by analytical method 39.3 dB(A), and the difference found is 1.3 dB (Tab. 5). Reason for this underestimation of the analytical result with respect to the average level in the room or in any case the level in the individual stations observed by measurements, is that wanting to evaluate the average pressure level in the environment and not for a specific receiver, only the diffuse contribution was considered in the equations. This according to UNI 12354-5 [10], which considers the diffused sound field, and excluding the semi-reverberated field, proposed by ASHRAE [4] as more appropriate for normally furnished rooms, because at the time of measurement lacked desks and seats. The average deviation between the sound pressure levels observed at each of the three measuring points identified according to UNI 8199 and the values obtained from the simulations at the same points is smaller and equal to 0.5 dB, although the model continues to underestimate the effect of the plant. Being the error between the real and the simulated room SPL around the JND of 1 dB, results are satisfactory.

Table 5. Detailed analytical results.

	Leq [dB(A)]			
	Measured	**Analytical**	**Numerical**	
Receiver	(12.5Hz-20kHz) (20Hz-20kHz)	Air+RadFC +RadDuct	Air+RadFC	Air+RadFC +RadDuct
R1	40.8		39.3	40.0
R2	40.4	39.3	39.4	40.2
R3	40.5		39.4	40.1

4. RESULTS

4.1 SPL

In analytical evaluations noise contributions were the airborne noise which passes through the ducts and the noise emitted by the ducts and fans. Simulations were instead carried out considering three configurations with increasingly detailed sources: only the airborne sound introduced into the rooms through air ducts ("Air"), the airborne sound introduced into the rooms through air ducts and radiated by fans ("Air+RadFC"), the airborne sound immitted in rooms through air ducts and radiated by fans and duct walls ("Air+RadFC+RadDuct"). As can be seen from the analytical results in Table 6, all offices are compliant with the background noise limits imposed by the LEED protocol. The offices were considered furnished and not occupied. This constitutes a limit at the design stage, because it risks imposing excessively demanding interventions compared to those necessary, also taking into account that the values referred to, in case of compliance verification with the limits, are the average ones in the analyzed space.The numerical method also allows to observe the variation of levels inside the spaces, which for rooms of volume between 150 and 250 m^3 as those examined, is around a range of 3 dB (configuration Air+RadFC+RadDuct). The greatest variability is described in the methodology, allows to get results very close to the values you would get from measurements (see Section Validation) but it is expensive, both in terms of time spent upstream for the evaluation of the share of radiated noise to be associated with point sources, and in relation to the simulations' duration. The difference that is found between the average SPL "Air+RadFC+RadDuct" and the average SPL "Air+RadFC" without ducts is of about 0.5, dB with a minimum of 0.2 dB in the case of compact spaces such as OO.t1 (Tab. 7). Simulating only the "Air" airborne noise from the ducts, neglects a contribution to the average sound pressure level of 12.3 dB compared to the "Air+RadFC+RadDuct" configuration.

Table 6. Detailed analytical results.

Open Office t1 (OO.t1)									
	Leq [dB(A)]								BNL_{max} [dB(A)]
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**A**	
A-Dif	21.4	29.7	34.7	34.6	30.1	23.2	14.4	**38.6**	
A-Rad	27.4	33.8	36.5	35.9	32.1	24.8	14.9	**40.4**	
Total	28.3	35.2	38.7	38.3	34.2	27.1	17.7	**42.6**	45
Open Office t3 (OO.t3)									
	Leq [dB(A)]								BNL_{max} [dB(A)]
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**A**	
A-Dif	22.4	30.8	35.7	35.6	31.1	24.2	15.5	**39.6**	
A-Rad	24.4	30.5	33.7	32.3	27.9	19.8	11.1	**41.4**	
Total	26.5	33.7	37.8	37.3	32.8	25.5	16.8	**43.6**	45
Open Office t12 (OO.t12)									
	Leq [dB(A)]								BNL_{max} [dB(A)]
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**A**	
A-Dif	20.7	29.1	34.1	33.9	29.5	22.5	13.8	**38.0**	
A-Rad	26.7	33.1	35.9	35.2	31.4	24.2	14.3	**39.7**	
Total	27.7	34.6	38.1	37.6	33.6	26.4	17.0	**41.9**	45
Open Office t15 (OO.t15)									
	Leq [dB(A)]								BNL_{max} [dB(A)]
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**	**A**	
A-Dif FC01	18.9	27.3	32.3	32.2	27.7	20.7	12.0	**36.2**	
A-Rad FC01	24.9	31.3	34.1	33.4	29.6	22.4	12.5	**37.9**	
A-Dif FC02	21.5	30.0	34.7	34.9	30.8	24.2	15.3	**38.9**	
A-Rad FC02	27.0	33.8	36.3	36.1	32.7	25.8	15.7	**40.5**	
Total	30.2	37.3	40.6	40.4	36.6	29.7	20.2	**44.6**	45

A-Dir=Airborne Diffuse *BNLmax= Maximum allowed BNL*
A-Rad=Airborne Radiated

Table 7. Broadband analytical and numerical results.

	Leq [dB(A)]									
	Analytical	**Numerical**								
Room	Air+RadFC	Air			Air+RadFC			Air+RadFC+RadDuct		
		Max	Mean	Min	Max	Mean	Min	Max	Mean	Min
OO.t1	42.6	27.9	26	23.9	41.7	40.3	38	41.85	40.5	38.85
OO.t3	43.6	28.2	26.8	25.5	43.8	42.2	41.1	44.2	42.8	41.7
OO.t12	41.9	30.8	28.9	26.6	38.9	37.6	35.7	39.4	38.2	36.5
OO.t15	44.6	33.5	31.1	29.1	41.8	39.7	37.7	42.4	40.3	38.3

This error is all the more significant the smaller the size of the studied environment: around 150m^3 reaches 15 dB. This means that whereas it is essential to simulate the noise radiated by fan coil fans, the contribution radiated through the duct walls can be omitted.Simulations allow to visualize the spatial distribution of SPL in the four office spaces. Figure 3 shows the results of the source configuration "Aereo+meccFC+meccDuct". The highest frequency contribution is at mid-low in the Air+RadFC+RadDuct configuration; space distribution in OO.t1 is between 34 dB(A) and 37.2 dB(A) in OO.t1.

4.2 NC, RC, STI

Assuming receiver points in a standard configuration with respect to fan coil source (Fig. 1), in them spatial distribution of SPL in the different offices (furnished but not occupied) was evaluated. Linear spectral distributions, considered the equal height of the rooms and the positioning of the receivers under fan coils of the same type (FC01), were compared with the NC and RC curves (Figg. 4,5). In offices of larger volume, OO.t12 and OO.t15, NC and RC are lower, settling on 35-curve. It should also be highlighted that in these cases the values at central frequencies (1000 Hz), decisive for the tangency with the characteristic NC/RC curve, are more dispersed than those observed for OO.t1 and OO.t3, aspect linked to the asymmetry of the rooms'plan. In Figure 6 are reported for each office the difference between averaged SPL variations in each standard receiver point with respect to the mean value of the room; the relationship, considering a cross section of the rooms, between extension of the fancoil-diffusers and room width is highlighted. The divergence between values in the standard receiver points, considered critical for their proximity to the sources, and the average value in the room is not related to the volume of the latter, but to the ratio of space occupied in width by the plant and width of the room. Evaluating the A-weighted noise levels in a receiver exactly below FC01 it is noted that for frequencies around 500 Hz, to which the human ear is more sensitive, the amplitude of the room has

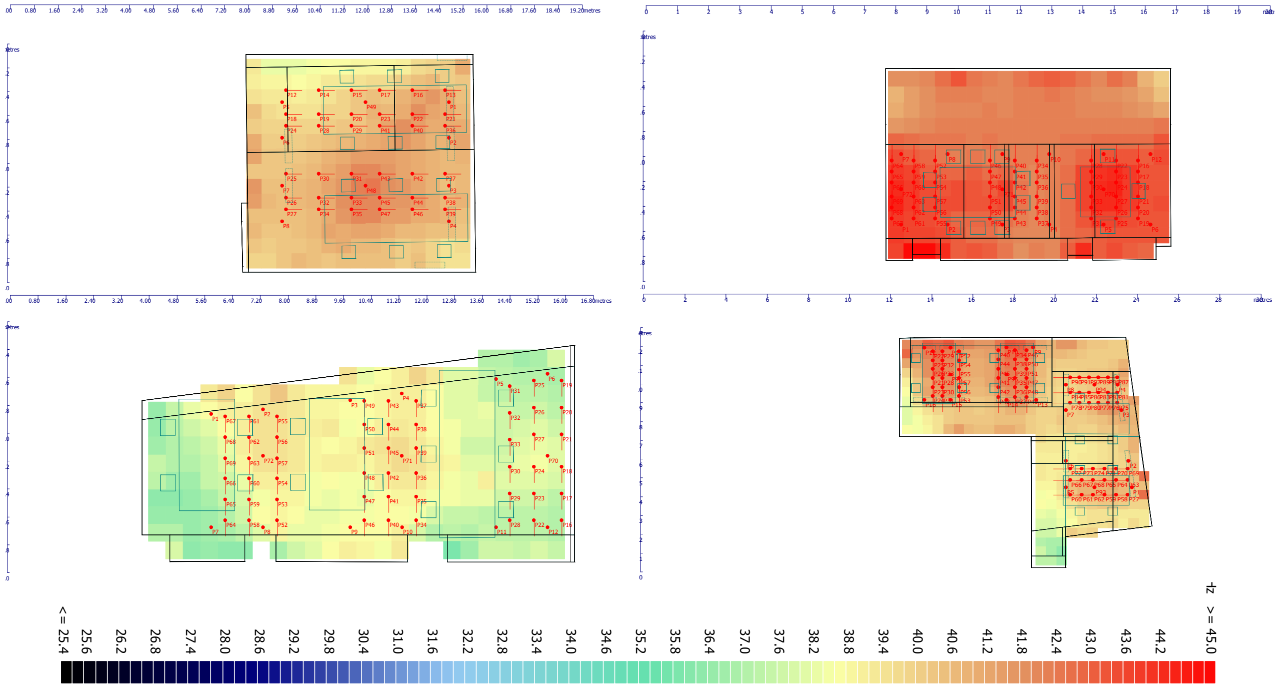

Figure 3. Maps of broadband background noise [dB(A)]: (a) OO.t1; (b) OO.t2; (c) OO.t3; (d) OO.t4.

significantly improving effects. At lower frequencies it is preferable to have rooms with sides of comparable lengths (OO.t1). A typical configuration of six workers sitting at a meeting table was identified. This group of receivers and speaker was placed in two different areas within the office OO.t12: exactly below a fancoil-terminal group, "STI-Under" configuration (Figure 2b) and in an area free of installations in the suspended ceiling above, "STI-Far" configuration (Figure 2c). First for both dispositions STI was evaluated considering the frequency data of the average background measured with turned off sources (Residual) in the validation office, LAeq=27 dB(A). The results were then compared considering the average background (Backg Avg) and the punctual background in the center of the table of each configuration (Backg Under, Backg Far), simulated in Odeon with sound sources turned on (Air+RadFC+RadDuct) (Table 8). In case of operating systems STI may be classified as "excellent" (for Backg Avg STI=0.86) although close to the lower limit. Compared to the off state (Residual) the configuration(Backg avg) has a mean deterioration of 0.01 point. Backg Under STI values are LOWER than Backg Avg values of maximum 0.02 in receiver 18, in the opposite half of the table near the fan coil body. Also in the case Backg Far the receiver in that position is the underdog, while others have STI in line with those calculated with Backg Avg.

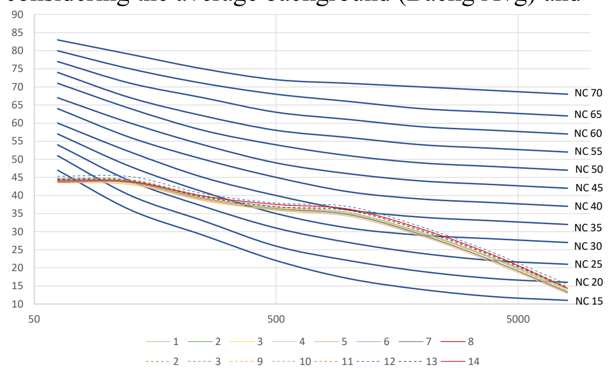

Figure 4. RC (16Hz-4000Hz).

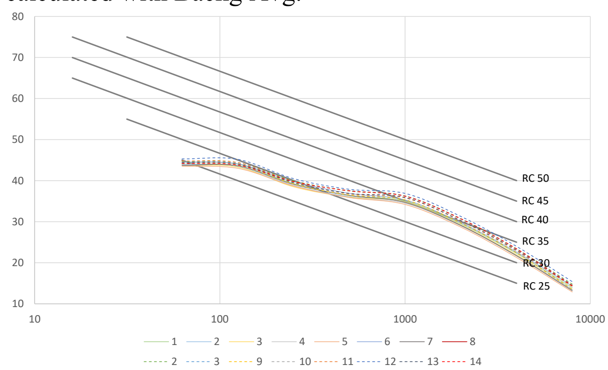

Figure 5. NC (63Hz-8000Hz).

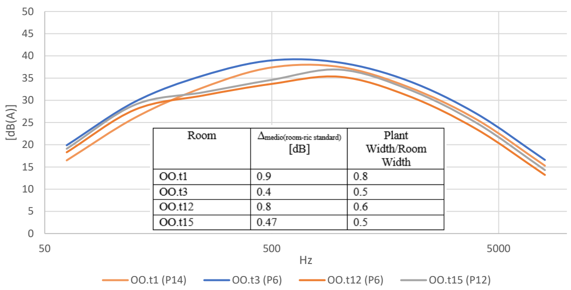

Figure 6. SPL at receiver under fancoil and spatial variation of SPL in each office.

Table 7. STI at each receiver point in OO.t12.

	Receiver	Residual	Backg Avg	Backg Under	Backg Far
Under	15	0.9	0.9	0.89	-
	16	0.86	0.84	0.83	-
	17	0.85	0.84	0.82	-
	18	0.87	0.86	0.84	-
	19	0.89	0.88	0.88	-
Far	20	0.9	0.89	-	0.89
	21	0.87	0.86	-	0.85
	22	0.84	0.83	-	0.83
	23	0.86	0.84	-	0.84
	24	0.88	0.87	-	0.87

5. CONCLUSIONS

All spaces present background noise from HVAC equipment, exterior sources (traffic, outdoor equipment, pedestrians) or other building services. When the sum of these noise exceeds comfortable levels, perception of spoken word can be impacted, reducing critical listening ability and task performance. Interior noise sources can be controlled selecting HVAC equipment with lower sound ratings and designing the system to reduce sound within ducts. These sound sources are easier to control at the earliest stages of design, for this reason is fundamental to know the accuracy of prediction methodologies related to spatial parameters distribution. The evaluation of four open offices with representative planimetric configurations showed that the difference between the mean analytical value and simulated value for SPL is less than 2 dB. Open plan offices acoustical conditions are largely influenced by BNL. The numerical method showed a variation range of SPL, in rooms of 150-250 m^3, around 3 dB. The reason is open offices tend to be large and flat with uneven distribution of the absorbing units, meaning sound field far from diffuse, and T60 not well defined. HVAC effect is poorly perceptible by STI: score decreases by 0.01 compared to the system-off condition. To assess the influence of aeraulic plants'noise on acoustic international requisites and on verbal communication, punctual NC and RC were observed. The not negligible spatial variation of SPL, inversely proportional to modal density, therefore greater in low frequencies systems, suggests the optimization of a corrective relation focused on HVAC prevailing frequencies, taking into account environmental absorption and geometry irregularity.

6. REFERENCES

[1] Saint-Gobain, Manuale Tecnico di Acustica Edilizia. Approccio teorico e normativo: le soluzioni Saint-Gobain Italia. (2021).

[2] I. Sharland, *Flakt Woods Practical Guide to Noise Control*. England: Flakt Woods Limited, 2005.

[3] WELL v2 Q4 2023 website, https://v2.wellcertified.com/en/wellv2/sound (accessed march 2023)

[4] American Society of Heating Refrigerating and Air-Conditioning Engineers. Chapter 48: Noise and Vibration Control. In: *ASHRAE Handbook: HVAC Applications.* 1-28 (2015).

[5] USGBC, LEED v4.1 Building Design And Construction, (2021).

[6] UNI 11367:2023 Acustica in edilizia - Classificazione acustica delle unità immobiliari - Procedura di valutazione e verifica in opera.

[7] F. Leccese, M. Rocca, G. Salvadori, Fast estimation of Speech Transmission Index using Reverberation Time: Comparison between predictive equations for educational rooms of different sizes, Appl. Acoust., Vol. 140, 2018, pp. 143-149.

[8] J.H. Rindel and C.L. Christensen, Acoustical simulation of open-plan offices according to ISO 3382-3, in *Proc. of Euronoise 2012*, (Prague, Czech Republic), 2012.

[9] M. S. Andargie, M. Touchie, W. O'Brien, M. Müller-Trapet, A field study of the relationship between sound insulation and noise annoyance, activity disturbance and wellbeing in multi-unit residences, Appl Acoust., Vol. 206, 2023, 109291.

[10] UNI EN 12354-5:2009. Acustica in edilizia - Valutazioni delle prestazioni acustiche di edifici a partire dalle prestazioni di prodotti - Parte 5: Livelli sonori dovuti agli impianti tecnici

[11] BS EN ISO 3382-3:2012. Acoustics — Measurement of room acoustic parameters Part 3: Open plan offices.

Funded by the European Union NextGenerationEU

The work of C.V. Fiorini was funded by the European Union-Next Generation EU under the PNRR (National Recovery and Resilience Plan – NEST "Network 4 Energy Sustainable Transition" – PE2 NEST Spoke 8 – B53C22004070006).