MEASUREMENTS IN BUILDING ACOUSTICS
Speech Intelligibility in a System of Many Coupled Rooms

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The internal space within a building may sometimes be divided into several rooms. The rooms may be considered to be acoustically coupled via areas which connect the rooms. In the method of coupled rooms an acoustic energy balance equation, applied to a diffuse field, is derived for each room and takes into account, apart from the energy absorbed in the room, the sound energy exchange between adjacent rooms. This approach has been applied to a large public building and used to estimate the modulation transfer function and hence the speech intelligibility.

INTRODUCTION

Rooms which are coupled together by a small open area are known to exhibit some interesting acoustical phenomena [1]. The most dramatic evidence of coupling occurs when one of the two rooms contains a large amount of acoustic absorption and the second room is more reverberant. If only one source exists, and that source is in the room with large absorption, the curve of the sound decay for the source room exhibits a "sagging" appearance. The curve consists of two distinctive parts which refer to the early and late stages of the sound decay. The first part of the curve is characterised by a steep slope which changes in time, while the second part has a constant but less steep slope. During the early stage of the process the sound decay in the source room is rapid, as energy is absorbed in the room as well as transferred to the adjacent non-source room. The decay rate gradually becomes less, since the adjacent room gives back to the source room some of the stored energy. Finally, in the late stage the sound decay rate achieves a constant value. The sound decay curve for the non-source room is said to have a "ballooned" appearance [1]. Its initial (t=0) slope is zero. During the early stage the sound decay rate in the non-source room is small, since the sound energy is supplied to the room from the source room. The decay rate increases gradually when the sound energy is transferred back to the source room and becomes constant during the late stage of the sound decay.

THE BUILDING

The present paper is concerned with the acoustic properties of a space which is the interior of a large public building. The interior of the building consists of a repetitive system of rooms [2]. There is a pattern of large rooms, separated by arches and columns. The larger rooms, of which there are 15, are referred to as "central" rooms. The central rooms exchange sound energy with four rooms called the "corridors" which in turn exchange energy with central rooms and "corner" rooms. Each corner room is in contact with four corridors. There are 38 corridors and 24 corner rooms. The rooms form a regular pattern with an unit which consists of a central room surrounded by four corridors and four corner rooms. Each central room is 15 m by 15 m in plan, the corridors are 15 m long by 5 m wide and the corner rooms are 5 m by 5 m. The height of the rooms is 9.64 m. Thus there is a total of 77 rooms, 336 arches and 248 columns. The materials used for the interior of the building are mostly marble, stone and plaster. The coupling area between the central room and a corridor is equal to 84.6 m², whilst the coupling area between the corridor and a corner room is 32.2 m², respectively. The volume of a central room is 2060 m³, the volume of a corridor is 683 m³ and the volume of a corner room is 224 m³. The total volume of the 77 rooms is 62,230 m³.

SPEECH INTELLIGIBILITY

As the interior space may be divided into 77 rooms, the method of coupled rooms may be applied. In this method an energy balance equation is established for each room and a system of 77 equations is formulated with appropriate initial conditions which may be solved as the eigenvalue problem [2,3]. The method allows the sound energy density and its decay with time to be estimated in different locations in the interior space. Not only the energy density level but also the early reverberation time and impulse response may be estimated in each room. In particular the spatial distribution of the impulse responses can be obtained and, consequently, the modulation transfer
function (MTF) and the speech transmission index (STI) may be determined at different locations within the building. The speech intelligibility may then be estimated \([4,5]\). The modulation transfer function method is the basis of an experimental technique, usually applied at 500 and 2000 Hz (RASTI); it has been applied previously to a large building \([6]\). The MTF method has been applied to the building described in this article and the results for STI at 500 and 2000 Hz are shown in Figure 1 for the case when no people are present. RASTI values may be obtained from an arithmetic average of the results at 500 and 2000 Hz. In this building the speech intelligibility is poor. It is a large space of considerable height. The reverberation time estimated from the late stage of the sound decay for the empty interior are 3.0 and 2.4 s at 500 and 200 Hz, respectively, although it should be noted that the reverberation time, referring to the early stage of the sound decay, varies from position to position \([2]\).

Figure 1 illustrates the influence of the reverberation time for the early stage of the sound decay on the STI values. Since the early reverberation times depends upon the distribution of the sound sources as well as the distribution of equivalent absorption in the interior the conclusion can be formulated that there is an optimal configuration of sound sources in the space in order to achieve the best speech intelligibility.

In large spaces which can be divided into several coupled rooms with a high difusivity of the sound field, for example large churches, the method described herein may be applied successfully.

REFERENCES


Environmental-Acoustic Impact on Optimum Sonar Search

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The Genetic Range-dependent Algorithm for Search Planning (GRASP) contains a genetic algorithm approach using Monte Carlo simulation and Bayesian detection statistics to optimize sonar search tracks in non-homogeneous environments. The optimization metric is maximum Cumulative Detection Probability (CDP) for a specified sonar (passive or active) against a target with specified characteristics (acoustic and tactical) during a fixed time period. GRASP solutions to simple and realistic ocean environments indicate that it produces intuitive search tracks that correlate highly with acoustic signal excess, as expected.

The essence of good search planning is to maximize the cumulative detection probability (CDP) within the search time available, but maximum possible CDP cannot be determined analytically except in special cases. In a homogeneous, isotropic environment, the standard Navy-doctrine ladder search path is near-optimal against stationary targets [1]. Further, maximizing CDP in real environments is an NP-hard problem, so the true optimum CDP achievable will never be known with certainty.

The Genetic Range-dependent Algorithm for Search Planning (GRASP) is an application of genetic algorithms which evolves search tracks to maximize some measure of effectiveness (MOE), typically CDP. Since the true optimum cannot be calculated, GRASP has been benchmarked against ladder patterns, where it often outperforms Navy doctrine by 20-50%. To assess the GRASP results, simulations in simplistic, non-homogeneous, anisotropic environments, where the “best answer” can be determined intuitively, were performed. If GRASP were to consistently attain CDPs close to those solutions, it would increase confidence in using the genetic approach in more complex and realistic environments.

Initial GRASP results for a sensor modelled with simple definite-range performance reproduced intuitive optimum solutions in a modest number of generations, proving its computational efficiency [2]. This second round of benchmarking is more realistic, assuming a range-dependent transmission loss in an environment with flattened coarse sand ridges 10-nm wide and 200m below the surface (high detection range), rising over a silty clay basin 1700m deep (low detection range). The ridges are laid out in various configurations: a single ridge, a double ridge, a square annulus, and a circular annulus. An example configuration with both intuitive and GRASP paths is shown in Figure 1, along with a CDP comparison.

FIGURE 1. The square annulus test case. The intuitive solution (dotted line) goes straight down the center-line of the ridge. The GRASP solution (dashed line) correlates well, but meanders and outperforms the intuitive solution at all search times.
When an intuitive solution is obvious, GRASP tracks do spatially correlate with intuitive solutions. They also usually out-perform intuition and standard tactics (in terms of CDP), in spite of the fact that GRASP paths are rarely the perfectly straight paths that intuition favors. This in turn improves intuition about what sorts of search paths are most efficient. In particular, sometimes the shortest path to success is not a straight line. Analysis of these and the previous benchmark trials reveals four competing principles, which are only strictly true for stationary targets and cookie-cutter detection functions.

1) *Straight lines are not uniquely optimal in a homogeneous, isotropic environment.* It can be easily shown that as long as the path’s radius of curvature is everywhere greater than the sweep radius, all non-redundant paths sweep the same area. Since CDP is generally proportional to sweep area, GRASP has no incentive to straighten the paths any further.

2) *Sharp turns, however, do introduce losses.* Sharp turns always introduce an area of overlapping sweep area. See Figure 2. This overlap maximizes (to 100%) for a U-turn, but soft changes in direction are low-cost.

3) *Anisotropy implies a preferred direction of travel.* In a homogeneous but anisotropic region (i.e., sweep width in a given direction is constant from point to point, but sweep widths in different directions vary at any given point), the preferred travel direction is perpendicular to the direction of maximum sweep width. See Figure 3. This obviously maximizes sweep area, and therefore CDP. This means a straight line is optimal in any such region if search time will be exhausted before a boundary is encountered.

4) *Inhomogeneities and boundaries often favor wiggling.* A single, straight search path is no longer possible if the preferred-direction dimension of the search region is small compared to the search time available, and may no longer be optimal if the search area is inhomogeneous. The optimal path will negotiate the best possible trade-off between the first three principles, which will usually involve some wiggling. Such will typically be the case on a ridge: detection range will be greatest when the searcher is on the ridge and looking along it, and so the preferred direction of travel will be across the ridge (and therefore quite short). Wiggling is the inevitable result, but wiggling with the most gentle serpentine motion possible. It is precisely such paths that GRASP tends toward as the number of generations increases, even in the simple environments studied during these benchmarking trials. In real environments, such cases are quite common, and this appears to be a major reason GRASP usually outperforms intuition.

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Acoustical Absorption and Critical Thickness

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This paper presents a theoretical and experimental study that shows the presence of a horizontal asymptote of the absorption coefficient function \(a(f)\) in relation with the thickness of the sample. This asymptote conduces to the existence of a critical depth, characterized by a certain attenuation of the sound wave, that limits the absorbing properties of the material. Above this thickness, \(a(f)\) does not change noticeably and a progressive increase of the layer thickness only means an unnecessary waste of acoustic material.

INTRODUCTION

Up to the moment, the existing bibliography does not cover in detail this topic whose economical consequences can be very relevant. From a general point of view, engineering texts [1] treat this problem showing only general concepts or basic design recommendations. In order to make a profound theoretical study of the behaviour of absorbing devices based on porous acoustic materials (fibrous or granular), it is necessary to count on mathematical algorithms that describe accurately the response of these materials under an acoustic excitation.

During the last decades, an important number of works have tried to describe analytically the acoustical properties of porous materials [2]. In this paper, mineral and glass wool will be treated being necessary to know only one microstructural parameter, the flow resistance, \(\sigma\), to describe completely the acoustic behaviour. Some other parameters, porosity, \(\Omega\), and tortuosity, \(T\), take values close to 1. In this case, the well-known empirical model by Delany and Bazley [3] has proved to show very good agreement with the experimental results, like the more recent model developed by Allard and Champoux [4].

Nevertheless, this study can be perfectly extended to porous granular materials, taking into account that in this case, the acoustical absorption is very sensitive to the parameters \(\Omega\) y \(T\), which depend on the internal structure of the material. Usually, \(\Omega\) varies between 0.3 and 0.7, and \(T\) lies in the range between 1.2 and 2.7. Granular materials have been treated by several authors, employing in this text the proposals of Stinson and Champoux [5], whose formulations also add one or two adjustment parameters related to internal shape factors.

To simplify the treatment of the problem, local reaction materials have been considered, so the dependence with \(\cos \theta\) (angle of incidence) is neglected. In addition, normal incident has been also considered in order to compare the theoretical study with the experimental tests carried out in a standing wave tube. In so doing, we are considering materials with a plane surface and backed by a rigid wall.

CRITICAL THICKNESS

Theoretically, the propagation of a sound wave must be extended to a seminfinite layer so that the surface impedance \(Z_s\) is the same as the characteristic impedance \(Z_c\) of the material, namely, the relation between the pressure and the velocity is the same at any point of the propagation line. In case of having a seminfinite layer and according to the well known expression of the reflection coefficient,

\[
R(f) = \frac{p_r(f)}{p_i(f)} = \frac{Z_c - Z_0}{Z_c + Z_0} \tag{1}
\]

the difference between that characteristic impedance of the material and the impedance of the fluid \(Z_0\) will determine the degree of absorption.

On the other hand, along the propagation of the sound wave, the acoustical pressure amplitude decays exponentially inside the material; in fact, the dependence of the amplitude of the sound pressure (plane wave) with the distance \(l\) is given by:

\[
\frac{|p(l)|}{|p(0)|} = e^{-\beta l} \tag{2}
\]

being the propagation constant, inside the material, \(k_c = \alpha + j\beta\). The corresponding attenuation, in dB, from \(l = 0\) to \(l = L\) is then:

\[
Att = 20\log e^{-\beta l} = -8.68\beta L \text{ dB} \tag{3}
\]

However, although theoretically the sound wave can propagate to infinite, an incident sound wave penetrates into the material only to a certain depth, which depends on both the wavelength (frequency) and the flow resistivity. It always exists a finite thickness where the value of \(Z_s\) approximates \(Z_c\), meaning that in practice, an infinite attenuation inside the material has been reached. In this way, a “critical thickness”, \(L_c\), can be defined:
Introducing the value of $L_c$ in the expression that relates the surface impedance of an acoustic layer backed by a rigid surface:

$$Z_s = -jZ_c \cot k_c \frac{-At_{t_c}}{8.68 \text{Im}(k_c)}$$

and applying the condition $Z_s = Z_c$, it results:

$$-j \cot k_c \frac{-At_{t_c}}{8.68 \text{Im}(k_c)} = 1$$

In figure 1, the real and imaginary parts of eq. (6) are represented in function of the attenuation. In order to “universalise” the curves, the parameter proposed by Delany $C = \rho_0 f / \sigma$ has been used. From the figure, it follows that a wave attenuation of 20 dB is sufficient to characterise the “critical depth”. The error existing between the absorption coefficient obtained with the critical depth and that obtained with an infinite layer is not larger than $\pm 0.8\%$. If we considered an attenuation of 8.7 dB, equivalent to a pressure loss of $1/e$, the error could reach the value of $\pm 6\%$. Although, the difference is considerable, the fact of accepting an error of 6% would represent to reduce the critical depth in a factor 2.3, with the corresponding economic and constructive advantages.

**Experimental results**

The experimental tests have been carried out in a standing wave tube (normal incidence) with both glass wool and granular samples of 9.9 cm diameter and 22.3, 18.6, 14.8, 11.2, 7.5 and 3.8 cm long. The acoustic absorption coefficient at the central frequencies of the 1/3 octave bands between 100 and 2000 HZ has been measured, and compared with the theoretical absorption curves. Figure 2 shows the agreement between three of them. It can be observed the evolution of the absorption coefficient in function of the thickness of the sample and frequency. After showing an increasing degree of absorption during the first centimetres of the material, it reaches a maximum (frequency dependent) followed by a plateau, characterised by an horizontal asymptote of the function $a(f, \sigma, l)$, that serves to define the “critical thickness”.

Analogous results were obtained for granular materials.

**Conclusions**

For absorptive acoustic materials, it has been shown the existence of a critical depth, based on the asymptotic behaviour of the function $a(f, \sigma, l)$ with the thickness, above which the absorption saturates, being unnecessary to increase the volume of material. This aspect is relevant in the correct use of this type of materials from a practical and economical point of view.

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Measurement of the Vibration Reduction Index $K_{ij}$ on Masonry Walls in the Laboratory

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The vibration reduction index $K_{ij}$ is related to the vibrational power transmission over a junction and is needed to calculate the airborne sound insulation between rooms according to EN 12354-1. Measurements to determine $K_{ij}$ are based upon the draft of the European Standard pr EN 10848. The work on this European standard has not finished yet and a research project at the University of Applied Sciences Stuttgart was being carried out to investigate the acoustic behavior of different masonry junctions typical for German housing constructions. Parallel one of these junctions is built up in a second laboratory, where airborne sound excitation is being used to determine $K_{ij}$ by measuring the flanking sound reduction index $R_{ij}$ and the sound reduction index $R_i$ of both flanking walls.

INTRODUCTION

The total sound transmission between two rooms can be calculated by the summation of the transmission paths from every element in the source room to the adjacent elements in the receiving room. The flanking sound reduction index $R_{ij}$ from element $i$ to element $j$ via one path can be calculated from the sound reduction $R$ of the elements $i$ and $j$, the direction-averaged level difference $D_{v,ij}$ and the areas of the separating element $S_i$ and the corresponding areas $S_j$ after [1] by:

$$ R_{ij} = \frac{R_i + R_j}{2} + D_{v,ij} + 10 \log \left( \frac{S_i}{\sqrt{S_i S_j}} \right) $$

(1)

The vibrational level difference $D_{v,ij}$ between two structures separated by a junction depends on the junction itself (transmission coefficient $\gamma_{ij}$, transversal wavespeed $c_B$, and the mass $m'$ of the structures) and on the characteristics of the structures connected to the junction (area $S$, the junction length and the damping). The direction-averaged level difference over a junction therefore is in [1] expressed in terms of the vibration reduction index $K_{ij}$ and a correction term for the in-situ situation. With the use of the equivalent absorption length $a$, the relation between velocity level difference and vibration reduction index is given by:

$$ D_{v,\text{equ}} = K_{ij} - 10 \log \left( \frac{l_s}{\sqrt{a_{\text{equ}} a_{\text{equ}}}} \right) dB $$

(2)

$$ a_{\text{equ}} = \frac{2 \pi^2 S}{T_{\text{equ}} c_n} \sqrt{\frac{f_{\text{ref}}}{f}} $$

(3)

Measurements to determine $K_{ij}$ are based upon the draft of the European Standard prEN 10848 [2]. In this research project the use of this draft on masonry walls is investigated. Therefore three different masonry junctions typical for German housing constructions were build up in the laboratory of the University of Applied Sciences, Stuttgart. Each T-junction consists of a separating wall joined with mortar to a flanking wall. The following results were obtained from measurements on a T-junction consisting of a separating wall ($l = 3.7$ m, $h = 2.7$ m, $t = 0.24$ m: CaSi $m' = 418$ kg/m²) and two flanking walls ($l_1 = 5.8$ m, $l_2 = 5.0$ m, $h = 2.7$ m, AAC, $t = 0.3$ m, $m' = 126$ kg/m²). All the walls are mounted on an elastic interlayer for decoupling from the ground.

EXPERIMENTAL RESULTS

Velocity Level Distribution on Masonry Walls

The first measurements carried out were to investigate the velocity level distribution on the masonry walls. Therefore the walls were excited by a stationary source and the velocity levels were measured on equally spaced positions along a line away from the source. As expected at low frequencies a big variation of the velocity level due to single modes was measured. At mid frequencies the level on the walls were nearly constant. At high frequencies the vibrational level on the walls decreases strongly with increasing distance to the source. This level decrease at high frequencies where the wavelength is in order of the dimension of the masonry stones is due to the reflection of transversal waves at the unmortared vertical joint between the masonry stones.

Measurement uncertainty

The vibration reduction index $K_{ij}$ was measured by several groups of students at the same junction. The standard deviation of the measured velocity level difference and the structural reverberation time were calculated. The standard deviation $\sigma$ of the velocity level difference rises at low frequencies due to low...
mode count and at high frequencies due to the level decrease over distance. In the mid frequency range from 400 Hz to 1600 Hz the standard deviation is approximately 1 dB. The standard deviation for the structural reverberation time is much smaller. Therefore the measurement uncertainty of $K_{ij}$ measurements is determined by the measurement of velocity level difference.

**Influence of plaster and wall size on $K_{ij}$**

Only in the high frequency region a significant influence of plaster on $K_{ij}$ values was observed. The influence on the measured $K_{ij}$ values by the length of a flanking AAC-wall which was decreased from 5.0 m to 3.5 m and 2.5 m is shown in figure 1.

![Graph showing influence of plaster and wall size on $K_{ij}$](image)

Fig. 1: $K_{ij}$ on path Fd for 3 lengths of the flanking AAC-wall

Only in the high frequency range a significant change in $K_{ij}$ values can be observed which is due to the level decrease over distance when the velocity level is measured in the area closer to the junction.

**Different measurement methods**

Two different methods to determine the vibration reduction index $K_{ij}$ are described in prEN 10848 [2]. The direct method is to measure the velocity level difference by exciting the structures with airborne or impact sound. The indirect one is to calculate the velocity level difference according to eq. (1) when the flanking sound reduction index $R_{ij}$ and the direct sound reduction index $R_{i}$ of both walls are measured.

In both cases the structural reverberation time $T_s$ has to be measured as well. Figure 2 shows the results obtained from the 3 different measurement methods for the same T-junction. Good agreement exists between the three different measurement procedures at least in the frequency range between 250 Hz and 2000 Hz. For very low frequencies $f < 100$ Hz the indirect measured data of $K_{ij}$ differ significantly. Modal differences in the airborne sound field of the different rooms are probably the reason for the bad matching. The dotted lines in the figures 2 and 3 show the value for $K_{ij}$ calculated from the mass per unit area according to 12354-1, Annex E /1/. Measured and calculated data differ significantly.

**Different laboratory situations**

The previously described AAC-CaSi junction was build up identically and in the same dimensions in three different laboratories but with different connections to the laboratory itself. Figure 3 shows the measured data on path Ff obtained by the direct method with structure borne excitation:

![Graph showing different laboratory situations](image)

Fig. 3: $K_{ij}$ on path Ff for 3 different laboratory situations:

The results do not agree expected. The $K_{ij}$ values in the laboratories differ approximately 5 dB in the mid-frequency range. The cause of the difference has not yet been found but it is expected that the connection between test object and laboratory may cause the difference. It is therefore suggested to measure $K_{ij}$ values on masonry walls in laboratories where the walls have a rigid connection to the laboratory to ensure a certain power flow to the coupled structures.

**REFERENCES**


Angular characterization of the urban frontages diffusivity factor

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With taking part of an interdisciplinary CNRS research program, we intend to constitute a simulation tool for complex urban soundscapes. For this purpose, we aim to quantify the urban diffusivity factor, related to the roughness distribution of the urban frontages, taking into account its surface constitution. In order to provide the angle dependent scattering characterization of those surfaces, we attempt to define a new method of measurement, applied to a 1/5th scale model of a French example of the 19\textsuperscript{th} frontage morphology. Scattering experimental results are successfully compared to the geometrical angular estimation of diffusivity, which involves fractal computing of the urban geometry. This will allow the generalization of this diffusive characterization method to any complex urban configuration, structurally considered as an acoustic diffusive system.

\section*{PROBLEMATIC AND PURPOSE}

The current work, within our laboratory, aims to establish a geometrical characterization of urban frontage reflection properties, exploiting the technique of mathematical morphology.

The essential purpose of this action is to develop an acoustic measurement method of scattering, both on scale model and on site, in order to confirm the acoustical pertinence of this methodology for the urban frontage characterization. Irregular surfaces, like urban frontages, produce an anomalous back-scattered field. For upper frequencies (small wavelengths), multiple reflection and diffusion phenomenon causes acoustic interference in the neighborhood of the building. Thus, in order to be able to detect that scattered energy’s minima and maxima we have to take into account number of angle incidence both for acoustical measurement and geometrical characterization.

\section*{DIFFUSIVITY CHARACTERIZATION}

Models

This angular description of the urban frontages leads us to describe both metrological and geometrical approaches, with taking into account three surface configurations. The first one, a single flat surface, will provide a reference point for both methods. The second one takes into account the windows distribution into this surface, placed in the depth of the stonework. The third configuration is analogous to a real neoclassical frontage in Nantes, with windows, doors, and freestone casting off.

\section*{Methodology}

As our geometrical approach uses a 3-D numerical model of those frontage configurations, acoustical measurements have been done on a 1/5th scale model for the flat and windowed configurations by the Laboratoire d’Acoustique de l’Université du Maine and the Laboratoire Central des Ponts et Chaussées. In a near future, in situ measurements will be done for the completed third configuration, a neoclassical frontage in Nantes. Both measurements use the same technique based on impulse response, which provides, for each incidence angle of reception, the frontage’s Frequency Transfer Function through the frequency domain analyze by time windowing.

The geometrical characterization considers frontages as tridimensional objects situated in an ortho-normed space. In order to evaluate the vertexes multiscale densitometry distribution at each incidence angle of the object, we apply a fractal Minkowsky operator, called Minkowsky sausage [1]. Practically, we replace each point of the vertexes of the urban geometry with a sphere: this corresponds to the \textit{dilation} operation in the morpho-mathematical context.

\section*{COMPARATIVE RESULTS}

Results of the geometrical characterization describes the polar response of the urban frontage, with an increment of 5 degrees, and for localization lengths of 0.5 m and 0.1 m. Global polar responses shows a decreasing diffusivity for growing localization lengths, confirming both the acoustical “c / 4 e” law and the Raleigh principle [2].
Comparison between acoustical measurements on the 1/5th scale model and the corresponding geometrical characterization provides a good angular agreement for the frequencies 1 kHz and 5 kHz, which corresponds to localization lengths of 0.5 m and 0.1 m (figures 2 and 3).

Figure 3 shows us three diffusivity peaks at 90°, [120°, 60°], and [145°, 35°], due to interreflections between the corners of the three windows of the lateral active diffraction zone, at 1 kHz. The two last peaks are readable on the geometrical analysis of the frontages too (Figure 2), with a setup of 10°, which giving the vertexes distribution density peaks at [150, 30°], [125, 55°] and [105°, 75°]. Those two last azimuthal densitometries increase at 5 kHz, conserving their peaks till localization length 0.05-m (10 kHz), with the apparition of a new 90° tip.

CONCLUSIVE DISCUSSION

With reducing the results setup by matching the windows dimensioning between the numerical and the scale models and with accurating the image extraction threshold at small incidence angles, those angular interference modes can consequently be geometrically detected with fractal analysis of the urban frontages [3]. This will be confirmed with the near at hand results of our last in situ experimental measures of the neoclassical frontage in the history center of Nantes.

Developed in the aim of architectural design tools for urban acoustics, the frontal angular diffusivity factor allows a good evaluation of the acoustical local reaction of an urban surface. With discerning the morphological attributes of main types of architectures, we will be able to compute their specific acoustical behavior at every frequency, from the numerical 3-D model of their geometry.

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Parameters for Reverberation Curve Evaluation

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Fundamental theory of reverberation time is based on the fact that after the sound source is turned off, sound pressure amplitude decreases exponentially e.g. its level decreases linearly. However, due to stochastic character of excitation and multiple reflections of sound ray, the decreasing line is not smooth but it has random fluctuation. On the other hand, because of wave character and various distribution of damping constants, decreasing line is not linear but deformed.

There are some characteristic deformations of shapes that mainly have the reliable theoretic explanations. Two deformation of a decay curve is specially characteristic: uniformly convexity curved (viewed from the time axis) and broken curve.

The paper deals with mathematical describing and recognizing the first deformation for detail analyses of measured RT. The results should also be used in automatic instrumentations for RT measuring. The paper is based on the results got by using sound level meter RION NL-18 and on processed results got by software package AK2.

THEORETICAL BASE FOR DECAY CURVE EVALUATION

If the room is excited by a quasistationary random signal \( s(t) \) which gets equal to zero for \( t<0 \), the room response \( h(t) \) can be expressed as convolution of the driving signal and impulse response \( g(t) \):

\[
h(t) = \int_{-\infty}^{0} s(t-\tau) g(\tau) d\tau = \sum_{n} c_n e^{-\delta_n t} \cos(\omega_n t - \varphi_n)
\]

The coefficients \( c_n \) and \( \varphi_n \) depend on the exciting signal, \( \omega_n \) is eigenfrequency, \( \delta_n \) damping constant.

Energy density is proportional to \( [h(t)]^2 \) and the average energy density can be expressed as:

\[
\bar{w}(t) = \sum_{n} c_n^2 e^{-2\delta_n t}.
\]

According to Kuttruff [1] coefficients \( c_n^2 = c_n^2(\delta) \) can be substitute with probability density of damping constants \( H(\delta) e^{-2\delta t} \) and energy density is expressed as:

\[
\bar{w}(t) = \int_{0}^{\infty} H(\delta) e^{-2\delta t} d\delta.
\]

The sound pressure level is determined as:

\[
L_r = 10\log \bar{w}(t) = 4.34 \ln \bar{w}(t) \quad [\text{dB}]
\]

and the decay rate can be calculate as:

\[
\frac{dL_r}{dt} = \frac{4.34}{w} \frac{d}{dt} \int_{0}^{\infty} H(\delta) e^{-2\delta t} d\delta = \frac{8.69}{w} \int_{0}^{\infty} \left[H(\delta) e^{-2\delta t} \right] d\delta
\]

The slope of the reverberation curve at time \( t \) is given by the mean damping constant, weighted with the time dependent distribution \( H(\delta) e^{-2\delta t} \). Since the values for small \( \delta \) become more and more pronounced, with increasing time, the slope of the curve decreases and the curve becomes flatter i.e. RT rises. Therefore, reverberation curves are generally convex viewed from the time axis. As a limiting case, they can be straight lines only if all damping constants that are involved in the decay process are equal.

Estimation of random fluctuation

Because of stochastic character of the excitation (white noise) and normal room modes decay curve is not smooth line but it has random fluctuation. The estimation of the influence of these variations is based on modal coefficient \( MC \) defined as standard deviation during the steady state sound field (before switching off the source):

\[
MC = \sqrt{\frac{1}{N-1} \left[ \sum_{j=1}^{N} L_j^2(t) - \left( \sum_{j=1}^{N} L_j(t) \right)^2 \right]^2}
\]
ANALYZING METHODS AND RESULTS

Measuring is done according [5], by using interrupted noise method. The analyses is based on the decay record with exponentially averaging with successive discrete sample points from the continuous average as output.

For the purpuses of these analyses moving reverberation time $T_{MOVING}$ is defined on the following way. During the measuring decay curve is obtained as a set of $N$ discrete values of reverberation level. It is beeing choosen a sequence $S$ including first $m$ ($m<N$) samples. On the basis of those data the RT is calculated. Appling least -squares fit (LSF) line sequence $S$ is moved for one sample right and next RT values is beeing obtained. Repetition of the procedure of overall samples data the series of $N-m+1$ values for moving reverberation time $T_{MOVING}$ is beeing obtained. This series $T_{MOVING}$ is plotted as relative sequence position in time. Series $T_{MOVING} = T_{MOVING}(n)$ can be fitted with LSF line. The slope of that LSF straigth line is measure of convexity. This kind of the procedure recognizes both decay curve deformation: convexity and broken line. For the slope coefficient the confidence interval with probability of 95% was calculated and correction of the modal coefficient has been taken in account.

For the purpose of these analyses user friendly program for PC is made that enables analticily and graphically analysis of RT, $T_{MOVING}$, EDT, in many different way. During the developing and testing of this methods one is committed RT analyses for more different rooms and different shapes, volume from $V = 40\times250$ m$^3$. The third octave RT value was from $T = 1\times15$ s. Independant on RT duration one is used time constant $\tau = 10$ ms for time averiging and sampling time $t_S = 10$ ms by using sound level meter RION NL-18.

Characteristic results $T_{MOVING}$ vs. relative sequence position and for different values of parametar $m$ are ploted on Fig. 1. The measurements are refered to the room dimensions 410 cm x 450 cm x 380 cm for 1/3 octave band with $f = 100$ Hz.

CONCLUSION

Decreasing line during the reverberation process is not a straight line. It is a complex curve, which can be mathematically represented as a convex smooth line and random variation.

It is not convenient to evaluate stochastic variations and modal components at decreasing part of the curve. The estimation of these influences could be good enough evaluated during the steady -state period, before sound source is switched off. For this reason standard deviation of sound pressure level during this steady -state period should be calculated.

The slope coefficient of least -squares fit lines for moving reverberation time is good measure to estimate convexity. Specified procedures could be effectively used on measuring equipment for direct reverberation time measurements as well as rejection criteria when graphic plot is available.

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Measurements of sound propagation inside air conditioning ducts

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Sound generated by the fan of the air conditioning system propagates along air ducts up to the conditioned rooms. The impossibility of lining the ducts from the inside with sound absorbing materials facilitates sound propagation up to great distances from the generation point. Moreover, particular conformations of the system, as small bending radius, fire barriers, ventilation grilles, may generate new noise also near the expulsion points. Measurements have been carried out in a number of hospitals HVAC systems, where inspection openings have been realised at various distances from the fan. Results show sound spectra typical of the systems analysed and point out the necessity of particular insulating devices where there is noise generation inside the ducts. The comparison with limiting values defined in Italian regulation for hospitals points out some critical situations.

INSTRUMENTATION AND MEASUREMENT PROCEDURE

Measurements have been carried out by means of a Sound Level Meter Bruel & Kjær 2230 equipped wit a microphone B&K 4155, a foam ball (diameter 90 mm), a preamplifier and a cable extension. The signal has then been recorded with a Digital Audio Tape and post processed with frequency analysis in laboratory.

According to ISO 5136 [1], the air velocity inside the ducts was lower than 15 m/s, as it is necessary for measurements with foam ball of 90 mm diameter. The microphone was inserted inside the duct through an inspection opening. The position of the microphone inside the duct was parallel to the air flow (with the air pressure perpendicular to the microphone membrane) and at a radial distance, r, from the duct axis so that

\[ 2r/\phi \approx 0.5 \] (\( \phi \) is the diameter of the duct).

All sound analysis was performed from 80 Hz to 10 kHz and with global values of Equivalent SPL in dBA.

Inspection openings were realised near the fan and at various distances from it, up to the receiving rooms.

MEASUREMENTS RESULTS

Figure 1 shows three sound spectra measured near the fan and at various distances from it. The upper line refers to the measurement point placed 2 meters after the fan. A silencer realized with sound absorbing material was present between the fan and the measurement point.

From the graph, the decay of the sound level of 5 dB/oct, typical of centrifugal fans [2], is evident.

A peak in the SPL, probably a resonance frequency, is also evident at 250 Hz. The intermediate line refers to the measurement point located between the fan and the receiving room.

The sound attenuation due to distance and to bends is approximately 20 dB at all frequencies. At high frequency, a sound resonance due to fire barrier reduces the attenuation.

According to Allen [2], in circular unlined metal ducts, the attenuation due to distance is of about 0.1 dB/meter for frequencies below 1000 Hz and of about 0.3 dB/meter for higher frequencies. The attenuation provided by round or square bends with turning vanes may be estimated in the range from 0 to 3 dB, being greater for higher frequencies and for larger ducts diameters.

![Figure 1: Sound spectra measured at various distances from the fan, inside the duct: first case study. (I_{eq} = 39 dBA in the receiving room).](image-url)
In the lower line of figure 1, referred to the measurement point located inside one receiving room, at 1 meter distance from the grille, the two sound resonance at low and high frequency are also evident.

**FIGURE 2**: Sound spectra measured at various distances from the fan, inside the duct: second case study. (L eq = 40 dBA in the receiving room).

Figure 2 shows measurements results for a second case study in the same hospital and for the same centrifugal fan. The first line refers to the same measurement point as in figure 1.

**COMPARISON WITH LIMITING VALUES**

Table 1 shows the limiting values for noise produced by ventilation systems, according to the Italian norm [3] and legislation [4].

Limiting values are given as global values of equivalent SPL in dBA, measured in the receiving room.

In the two case observed, equivalent SPL (40 and 39 dBA), measured inside the receiving rooms at 1 meter of distance from the diffuser, exceeds the limiting value of Italian legislation of about 15 dB.

<table>
<thead>
<tr>
<th>Kind of interior</th>
<th>Limiting value (dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital rooms</td>
<td>30</td>
</tr>
<tr>
<td>Wards</td>
<td>40</td>
</tr>
<tr>
<td>Surgical rooms</td>
<td>35</td>
</tr>
<tr>
<td>Corridors</td>
<td>40</td>
</tr>
<tr>
<td>Public spaces</td>
<td>40</td>
</tr>
<tr>
<td>Services</td>
<td>40</td>
</tr>
<tr>
<td>All interiors (Italian law)</td>
<td>25</td>
</tr>
</tbody>
</table>

It is necessary to underline that the limiting values defined by Italian legislation refer to mean values measured inside the receiving room. For this reason, values measured in front of the diffuser have to be corrected to keep in count the absorption of the receiving room.

**CONCLUSIONS**

The global values of sound pressure level have been measured in eight different conditioned rooms, in the same conditions as described above. Results are quite always in the range from 40 to 50 dBA.

These values are from 10 to 20 dB higher than the limiting values of the Italian norm and from 20 to 30 dB higher than the value of the Italian law.

It is therefore necessary to use devices to reduce these levels below the limiting values.

With reference to figure 1, the sound resonance at 4000 Hz, probably due to the fire barrier, produce an increase of about 1 dB in the global sound pressure level in dBA measured in the room.

In all cases, sound spectra are particularly high at low frequencies. A reduction of 10 dB in sound pressure levels from 100 to 500 Hz can result in a reduction of the global level in dBA of more than 5 dB.

**ACKNOWLEDGEMENTS**

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Changes in Frequency Deviation of FM Signals
Propagating in a Room

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Changes in frequency deviation of FM signals were measured using a dummy head in rooms with different acoustic parameters. Measurements were made for different values of carrier and modulation frequencies. It was found that the influence of room parameters on a signal with variable frequency was quite complex in many aspects. The complexity was manifested as changes in both the frequency deviation and the amplitude envelope of the signal tested. The experimental data show that the frequency deviation of a signal transmitted to a room is subject to different changes. The range of changes depends on the character of the acoustic field, carrier and modulation frequencies and time-amplitude relations of superimposing waves.

INTRODUCTION
Changes in the acoustic signal propagating in a room are manifested in the amplitude and phase domain. The amplitude changes have been a subject of study by many authors. The outcome of the studies was an objective method for measurement of speech intelligibility in a room [1]. It has been also found that interaural difference in modulation depth does not have a significant influence on perception of speech [3, 4]. Moreover, the role of phase changes of a sound in a room has been found insignificant from the point of view of the sound perception [2]. With regard to these findings it seems interesting to study the changes in the frequency structure of sound in a room. A study was undertaken to determine the interaural differences in frequency deviation of FM signals propagating in rooms of different acoustic parameters.

CHANGES IN THE FREQUENCY STRUCTURE OF SOUND
It has been shown [5, 6] that the sounds of variable frequency propagating in a room can get different nonsteady forms leading to changes in the frequency structure. The author of [6] determined the so-called amplitude weighted frequency modulation transfer function (WFMTF). Exemplary courses of this function are shown in Fig. 1. As follows from [6], the frequency changes in a room can be described by the expression:

\[ \omega_x(t) = \omega_0 + \Delta \omega_x \cdot WFMTF \cdot M(t) = \omega_0 + \Delta \omega_x \cdot M(t) \]

where: \( \omega_0 \) - the carrier signal frequency, \( \Delta \omega_x \) - frequency deviation of the signal transmitted to a room, WFMTF – the amplitude weighted frequency modulation transfer function, \( M(t) \) - the modulating function, \( \Delta \omega_y \) - the frequency deviation of the signal received in a room.

As shown in Fig. 1, only for the lowest modulation frequency and short reverberation time, the modulation transfer function does not change. This means that only then the frequency changes of the signal are relatively well transferred by a room.

EXPERIMENT AND RESULTS
The FM signals of carrier frequency ranging from 250 - 4000 Hz were transmitted to rooms with determined acoustic parameters. The frequency deviation was equal to 50 Hz. The signals were recorded by a dummy head (KU100) at chosen measurement points in the room. The signals from each ear were converted into the digital form and
decomposed to obtain the modulating signals whose
RMS values were related to the effective frequency
deviation. The measured parameter was the relative
frequency deviation determined by \( R = 20 \log (\frac{\text{dev}_{\text{out}}}{\text{dev}_{\text{in}}}) \), where \( \text{dev}_{\text{in}} \) and \( \text{dev}_{\text{out}} \) stand for the
effective frequency deviation of the signals transmitted
and received in the room. The dependence of \( R \) on
modulation frequency \( (f_m) \) is related to the frequency
modulated transfer function (FMTF). Fig. 2a presents
the courses of the FMTF measured in the left and the
right ear of the dummy head located in the diffuse
field, for carrier frequency of 500 Hz. Fig. 2b presents
a similar dependence for the carrier frequency of 2000
Hz.

The room reverberation time was 0.6 s for 500 Hz and
0.55 s for 2000 Hz. As follows from Fig. 2a, the
courses of FMTF versus the modulation frequency are
not the same for the two ears. The interaural
differences in the frequency deviation reach values
close to 8 Hz for \( f_m \) ranging from 8 to 32 Hz. However,
for frequency of 2000 Hz, the differences are very
small (cf. Fig. 2b). Fig. 2c shows the FMTF measured
at a few selected points (1 - 4) in the room. These
points were located along the diagonal of a room a
nearly square shape (side length of 6 m) at the
distances of 2, 4, 6 and 8 m from the source of the
sound placed in the corner. The modulation frequency
was 4 Hz and the carrier frequency 1000 Hz. It has
been found that the greatest differences in the
interaural frequency deviation occur when the dummy
head is placed close to the reflecting planes (the room
walls). Independently of the physical study, a
preliminary subjective assessment of the recorded
differences in frequency deviation was performed. The
assessment was aimed at comparison the measured
values of the frequency deviation in a room with the
binaural detection thresholds of the interaural
difference in frequency deviation. It was found that in
many cases the interaural difference in frequency
deviation clearly exceed the respective perception
threshold. One can expect that these differences are
perceived by the subjects. This fact may have some
consequences for the subjective evaluation of the
acoustic quality of a room.

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