THE PERFORMANCE OF REFLECTIVE AND ABSORPTIVE NOISE BARRIER IN CITROSUCO'S INDUSTRY IN MATÃO (SP) - BRAZIL

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Abstract *The purpose of present work is to realize an analysis of noise barrier based on boundary element method, direct formulation, which has showed good performance mainly in cases of noise propagation in open field. The boundary integral equation is formulated in terms of half-space Green's function which satisfy the impedance boundary condition on the barrier surface. Introducing an absorbent material, fiberglass wool, on side of the barrier some improvement in insertion loss was observed only 315 Hz and 4000 Hz.*

Keywords*: noise barrier, boundary element method, numeric method, surface admittance.*

1. Introduction

The Boundary Element Method (BEM) has been used extensively for the numerical solution of noise barrier problem in an infinite domain. The most important feature of the BEM in acoustics is that only the surface has to be modeled. The Sommerfeld radiation condition at the far field is automatically satisfied in formulation. The half-space problems can be constructed by using the method of images, Li et al. (1992).

In general, the boundary condition on the noise barrier surface can be a Neumann, Dirichlet or mixed type boundary condition.

A noise rigid barrier was built in Citrosuco's industry (Matão - SP) in 2004 to protect the residential community against the main noise source, the evaporator 7A10. The barrier, 5 m height and 100 m length, is next to the community. A layout of the site is shown in Fig. 2.

The study of absorption in the face of the noise barrier was made together with Isobrasil Company after the construction of noise barrier to verify the efficiency of the absorption for the frequency of 315 Hz.

The spectrum of sound field measured in point 12 without the noise barrier was not supplied by the company, only the global sound pressure level. The value measured was 63 dB(A) in point 12. In this way, the no-barrier situation was simulated for a prediction of the sound field in point 12.

The absorbent and reflective barrier performance was simulated and was compared with each other and with the no-barrier situation by studying the difference in the sound levels in the residential area behind the barrier.

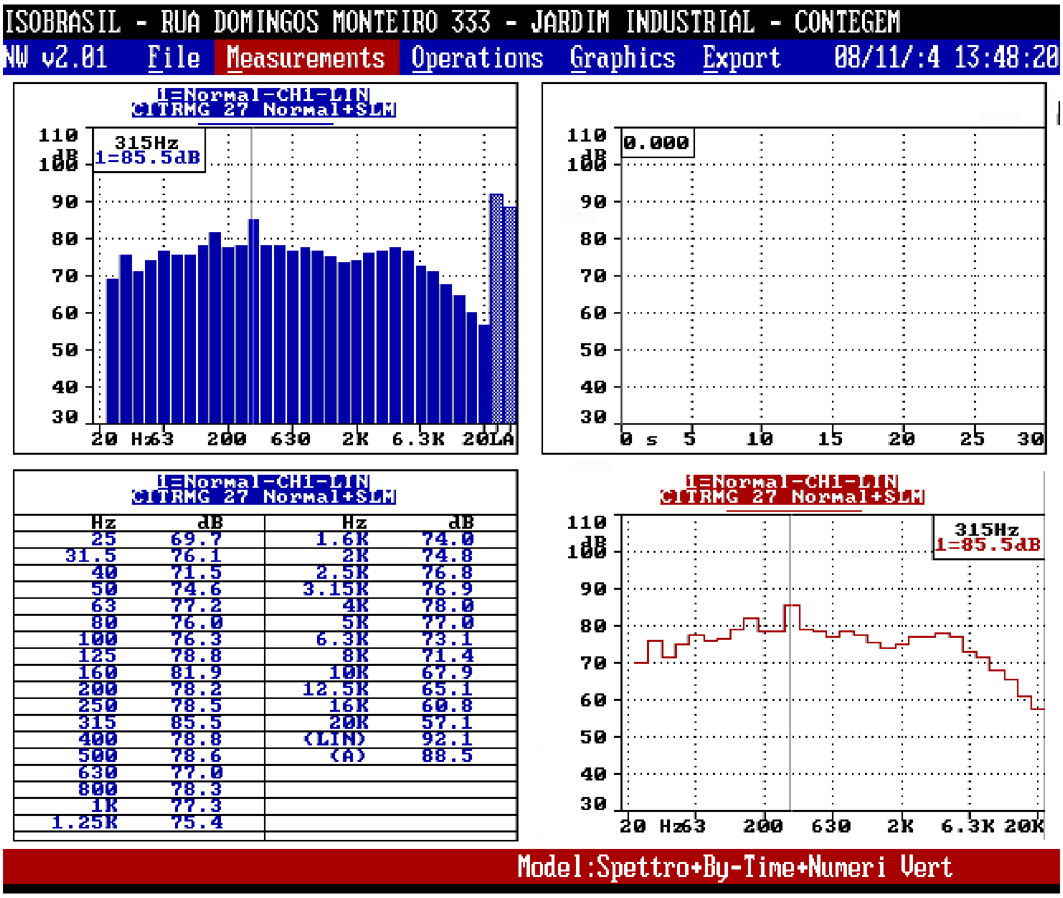
The absorbent surface was tested because of prediction, May et al. (1979), that it might increase the insertion loss behind the barrier.

The objective in this present work is to realize an acoustics barrier analysis with normal surface admittance, for the frequency of 315 Hz, based in the two-dimensional Boundary Elements Formulation. This method is efficient, specially in acoustics environment with sound propagation. The insertion loss can be calculated when a barrier is inserted between the noise source and the receptor using this proceeding. By using the method, broadband insertion loss results for an A-weighted industry noise spectrum have been calculated. The relative performance of the barrier with absorption is discussed.

2. Layout of barrier site and measurements supplied for the Isobrasil Company

The Isobrasil Company measured the spectrum of sound field in front of evaporator 7A10 showed in Fig. 1. This is the main noise source to point 12 in Fig. 2. This figure shows the layout of barrier site. The one-third-octave spectrum has showed the highest sound pressure level to 315Hz. This is the most important frequency to be treated. Table 1 shows the sound pressure level in dB, SPL dB, for octave center frequencies and the global sound pressure level in dB(A), SPL dB(A), measured in point 12.

Before the construction of noise barrier the global sound pressure level weighting-A was measured by CETESB - Basic Sanitation Technological Center of São Paulo State. The value measured was 63 dB(A) in point 12.



Hz	dB	Hz	dB
25	69.7	1.6K	74.0
31.5	76.1	2K	74.8
40	71.5	2.5K	76.8
50	74.6	3.15K	76.9
63	77.2	4K	78.0
80	76.0	5K	77.0
100	76.3	6.3K	73.1
125	78.8	8K	71.4
160	81.9	10K	67.9
200	78.2	12.5K	65.1
250	78.5	16K	60.8
315	85.5	20K	57.1
400	78.8	(LIN)	92.1
500	78.6	(A)	88.5
630	77.0		
800	78.3		
1K	77.3		
1.25K	75.4		

Figure 1. One-third-octave spectrum for the sound field measurement next to the evaporator 7A10

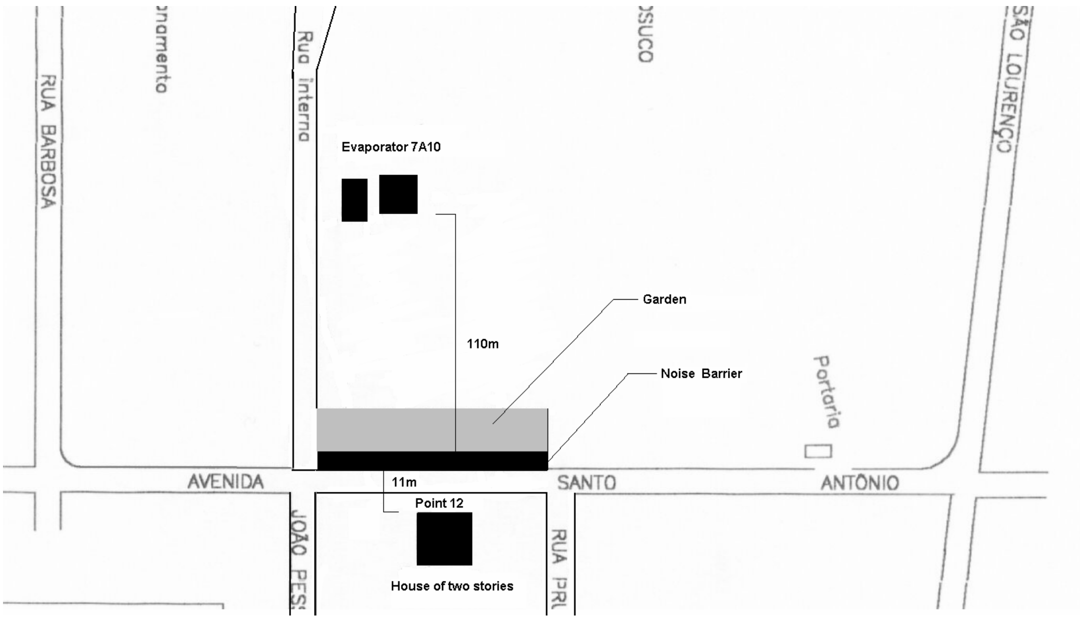


Figure 2. Layout of barrier site

Table 1. Value of global sound pressure level in dB(A) measured in point 12 by Isobrasil with noise barrier

Freq. (Hz)	P1	P2	P3	P4	P5	P7	P10	P11	P12
63	70	65	68	65	61	60	60	64	62
125	61	57	61	61	58	55	55	58	59
250	54	50	54	55	50	49	49	53	51
315	56	52	52	54	51	49	50	53	53
500	54	49	52	55	50	48	46	52	49
1k	47	44	45	46	43	44	43	47	44
2k	45	40	40	42	40	40	41	45	42
4k	51	43	48	49	43	41	44	47	42
SPL dB(A) - measured by Isobrasil with rigid noise barrier									53

3. Boundary element formulation for the Helmholtz Equation

3.1. Matrix formulation

The boundary integral equation generally used as a starting point for boundary elements has been deduced. The boundary element method and evaluation of integrals for a noise barrier problem have been showed by Ciskowski *et al.* (1991), Hothersall *et al.* (1991), Piacentini *et al.* (1996), Duhamel (1996), Fyfe *et al.* (1995), Muradali (1996), Li *et al.* (1992) and Papini *et al.* (1999).

The matrix formulation of boundary integral equation is shown:

$$\frac{1}{2}\phi\left(\vec{r}_0\right)+\sum_{j=1}^{N}\phi_j\int_{\Gamma_j}\frac{\partial u^*}{\partial\vec{n}}d\Gamma=\sum_{j=1}^{N}\frac{\partial\phi_j}{\partial\vec{n}}\int_{\Gamma_j}u^*d\Gamma+\phi^I\left(\vec{r}_0\right) \tag{1}$$

where $\phi^I\left(\vec{r}_0\right)$ is the incident velocity potential from the source, u^* is the fundamental solution of Helmholtz Equation, $\frac{\partial u^*}{\partial\vec{n}}$ is the derivative of the fundamental solution in relation to the normal vector in the boundary, ϕ_j and $\frac{\partial\phi_j}{\partial\vec{n}}$ are the boundary conditions. These integrals of boundary Γ_j relate the node *i* where the fundamental solution is acting on any other node *j*. Because of this their resulting values are called influence coefficients. They will be called:

$$G_{ij}=\int_{\Gamma_j}u^*d\Gamma\ ;\ H_{ij}=\int_{\Gamma_j}\frac{\partial u^*}{\partial\vec{n}}d\Gamma. \tag{2}$$

The fundamental solution is applied at a particular node *i*, which is not explicitly indicated in the u^* and $\frac{\partial u^*}{\partial\vec{n}}$ notation, Ciskowski et al. (1991).

Now assuming that the position of node *i* also varies from 1 to N, one assumes that the fundamental solution is applied at each node successively. In this way one obtains a set of equations that can be expressed in matrix form as:

$$-\boldsymbol{G}\frac{\partial\vec{\phi}}{\partial\vec{n}}=\boldsymbol{H}\vec{\phi}+\vec{\phi}_I \tag{3}$$

where $\boldsymbol{H}$ and $\boldsymbol{G}$ are two NxN matrices and $\vec{\phi}$ and $\frac{\partial\vec{\phi}}{\partial\vec{n}}$ are vectors of length N.

Given that, for each node on the Γ boundary, either ϕ ou $q=\frac{\partial\phi}{\partial\vec{n}}$ is known, this reduces to

$$\boldsymbol{A}\vec{x}=\boldsymbol{B} \tag{4}$$

where $\vec{x}$ contain the unknown values of ϕ e q.

Robin's or mixed boundary conditions are implemented in matrix form. This type of boundary condition implies that there exists a relationship between velocity potential ϕ and particle velocity q in the form

$$q = i\,k\,\beta\,\phi \tag{5}$$

where k is the wave number, and β is the normal surface admittance, Chandler-Wilde *et al.* (1985). In matrix form

$$\vec{q} = i\,k\,\beta\,\vec{\phi} \tag{6}$$

where β is a diagonal matrix containing the known values of the node under consideration.

Substituting Eq. (6) into Eq. (3) produces

$$\left(-\boldsymbol{H} - \beta\boldsymbol{G}\right)\vec{\phi} = \vec{\phi}_{\boldsymbol{I}} \tag{7}$$

which can be easily reduced to the form in Eq. (4). Note that the above operation should only be used at nodes where mixed boundary conditions are applied, Ciskowski *et al.* (1991).

3.2. Half-space Green's function

The solution of the wave equation is simplified by introduction the acoustics point source (or monopole source) and the corresponding Green's function. The pressure field $p(\vec{r}\,)$ is similar to the solution obtained for a uniformly pulsating sphere of finite radius. The half-space Green's function *Gr* for a point source is defined as the solution for a point source with source strength unity. For two dimensional problems with semi-infinite flat plane, perfectly rigid, the Green's function is:

$$Gr\left(\vec{r}\right) = -\frac{i}{4}Ha_0^2\left(k\vec{r}\right) + \frac{i}{4}Ha_0^2\left(k\vec{r}_1\right) \tag{8}$$

where Ha_0^2 is the Hankel function of type 0 and order 2 and $\vec{r}_1$ is the distance from the image of source.

4. Results

The results are showed below. Three distinct meshes had been used. The first mesh was used for the frequency of 500 Hz with 103 elements of 0,100 m. The second mesh was used for the frequency of 2000 Hz with 511 elements of 0,05 m. The third mesh was used for the frequency of 4000 Hz with 1021 elements of 0,01 m. The results were calculated making use of 6 nodes per wavelength rule. Table 2 shows the application of half-space Green's function to simulate the sound pressure level in decibel, SPL dB, on point 12. The simulated results gave evidence of correct distribution of coherent line sources on the surface of evaporator 7A10. Table 3 shows insertion loss calculated and the SPL dB(A) for point 12, considering a rigid noise barrier. Table 4 shows the insertion loss calculated and SPL dB(A) at point 12, which include the effects of absorption on one side of the barrier. On the further assumption that the absorbent material is normally reacting and has thickness T and is backed by rigid barrier. In Table 4 the normal surface admittance β was given by Chandler-Wilde et al. (1985),

$$\beta = -\tan(T\,k_G)\,\beta_G \tag{9}$$

$$\beta_G = \left[1 + 9.08\left(1000\,f/\sigma\right)^{-0.75} + i\,11.9\left(1000\,f/\sigma\right)^{-0.73}\right]^{-1} \tag{10}$$

$$k_G = k\left[1 + 10.8\left(1000\,f/\sigma\right)^{-0.70} + i\,10.3\left(1000\,f/\sigma\right)^{-0.59}\right] \tag{11}$$

where k_G is the wave number in the soft material, f and σ are the frequency and flow resistance in SI units. The flow resistance is calculated in function of the absorption coefficient. The coefficient of absorption of the fiberglass wool was defined for each octave band by Gerges (1992). The thickness T = 0,05 m.

Table 2. Sound pressure level measured and simulated to point 12 without noise barrier.

Simulation on point 12 without noise barrier			
Freq. (Hz)	P12	Freq. (Hz)	P12
(Hz)	(dB)	(Hz)	(dB)
20	75	315	60
25	73	400	56
31,5	70	500	58
40	66	630	53
50	62	800	53
63	61	1k	46
80	65	1,25k	50
100	65	1,6k	47
125	60	2k	47
160	47	2,5k	50
200	52	3,15k	43
250	57	4k	24
SPL (dB) - Simulated			79
SPL dB (A) - simulated			62
SPL dB (A) - measured by CETESB - SP			63

Table 3. Insertion loss spectra with $\beta = 0$ and SPL dB(A) with barrier.

Freq. (Hz)	Simulation - P12	IL (dB) - P12 without absorption	SPL dB(A) with barrier
63	61	4,7	30,3
125	60	4,0	40,4
250	57	3,7	44,0
315	60	5,5	47,4
500	58	3,1	51,6
1000	46	0,0	44,1
2000	47	10,7	35,5
4000	24	6,6	18,6
SPL dB(A) - without absorption			54

Table 4. Insertion loss spectra with $\beta \neq 0$ and SPL dB(A) with barrier.

Freq. (Hz)	Simulation - P12	IL (dB) - P12 with absorption	SPL dB(A) with barrier
63	61	4,6	30,4
125	60	3,6	40,8
250	57	3,4	44,3
315	60	9,6	43,4
500	58	3,8	50,9
1000	46	0,0	44,1
2000	47	11,0	35,2
4000	24	8,8	16,4
SPL dB(A) - with absorption			53

Figure 3 shows the values presented by Table 3 and Table 4, which compare the IL (dB) x frequency considering the effects of absorptive surface and rigid surface condition.

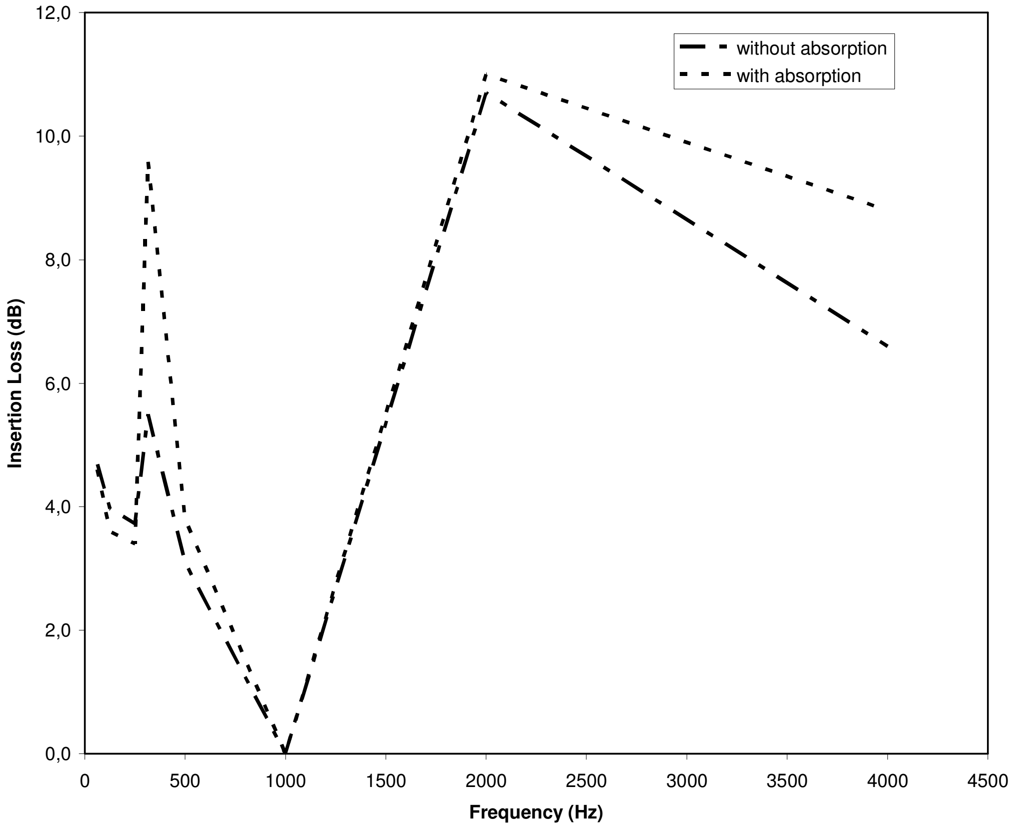


Figure 3 - Insertion loss spectra

5. Conclusions

The attenuation of sound by single noise barrier, 5 m height and 100 m length, has been investigated by Boundary Element Formulation. The model is two-dimensional. This model provides useful predictions of the performance of the long barrier. The comparison of the numerical SPL dB(A) with the SPL dB(A) measured shows a good agreement for the acoustic field in point 12, considering a rigid noise barrier. Tables 3 and 1 show these results.

Introducing an absorbent material, fiberglass wool (T = 0,05 m), on side of the barrier some improvement in insertion loss was observed only 315 Hz and 4000 Hz.. Application of absorbent material is not efficient at low frequencies. It did not have a significant improvement in the global sound pressure level in dB(A) in point 12 using absorption in the noise barrier. Tables 3 and 4 show these results. The noise barrier, 5 m height, was not efficient for the frequency of 1000 Hz, for both analyzed conditions.

6. Acknowledgements

The authors would like to thank the Isobrasil Company.

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