Sound flanking transmission by curtain wall mullions

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Abstract
This paper presents a numerical model and method to simulate the acoustic Transmission Loss for the flanking transmission through curtain-walls mullion.
In practice, measurements are conducted using two reverberant rooms and it is well-known that the modal behavior of theses rooms will affect the results on the low frequency band.
In this work, a numerical study of the flanking sound transmissions will be made. A numerical vibro-acoustic model will be presented. The results of a parametric study on a mullion of Hydro will be made and a method to isolate the internal flanking paths will be proposed.
Finally, a model reduction method will be proposed to reduce the mullions into a super-element usable to calculate the Transmission Loss.

Keywords: Acoustic; Flanking Transmission; Curtain-Wall; Finite Element Method

1 Introduction

Nowadays, the use of aluminum curtain-walls is increasing, their material properties (weight, corrosion resistance, rigidity) , easy method of production and installation makes them a prime actor in the sector of building façade.
However, when coupled with glazing, the vibroacoustic behavior of the curtain-wall needs specific attention as the dependency to exterior factor, such as room geometry and internal factor, such as assembly uncertainties, increase especially in the low frequency band (inferior to 500Hz in this study).
When studying direct transmission, the impact of the bond between glazing and mullion is not preponderant thus it is not much studied but in the case of flanking transmission, this bond is one of the main paths to transmit the sound.
Research on sound flanking paths in curtain-wall systems has been carried out with statistical energy analysis (SEA) models **[1-3]**, this method present hypotheses valid only at high frequencies of study when the modal density of the system is high enough. One example of this method is on the European Standard EN 12354 **[4]**.
Other researches have studied the vibration reduction index using the Finite Element Method (FEM) **[5]**.
Extensive measurements in laboratory have been made at the NRC/IRC Laboratory **[6]** on wooden structure and in **[7]** to determine the mechanisms of transmissions through a curtain-wall.
In some instance **[8-9]** study using FEM has been made. As this method is highly suitable to low frequency studies where the SEA hypotheses are not verified.
In this paper, a numerical analysis of the phenomenon will be made using a Finite Element Model. A strongly coupled vibroacoustic formulation is used to calculate the Transmission Loss (or TL).

A parametric study will be conducted to investigate the impact of diverse factors (such as mullion dimensions, boundary conditions) on the TL. A condensation method to reduce the size of the systems and the computation times will be proposed.

2 Formulation of the finite element Vibro-acoustic problem

To define the vibro-acoustic coupling between an acoustic field and a structure, we will consider the vibration of an elastic structure coupled with an inviscid and compressible fluid.
The structure will be modelized by using Euler-Bernoulli beams.
In this section we will recall the equations of the coupled problem (Figure.1) and validate our development by an analytical solution.

2.1 Analytical Solution for the vibro-acoustic coupling

Let us consider the following vibro-acoustic system with a simply supported beam.:

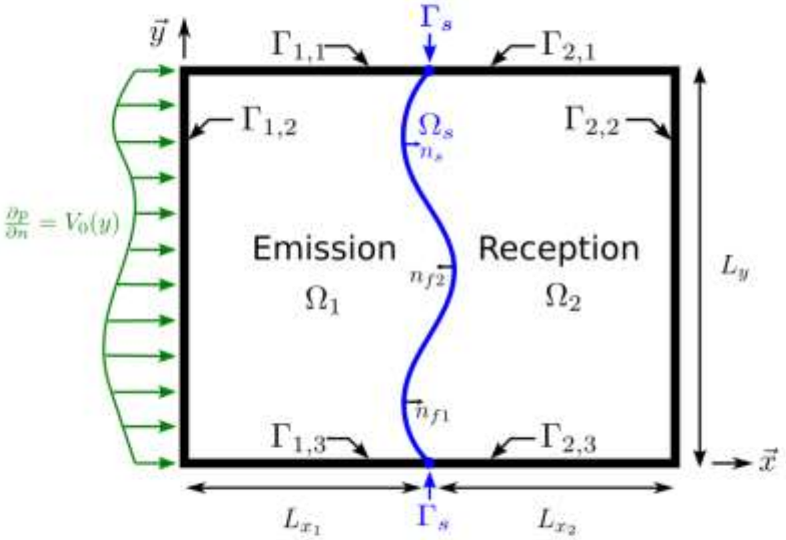

Figure 1.Schematic of a vibro-acoustic model

The equation of the fluid-structure coupled problem described in Figure 1 are :

$$\begin{cases} (\Delta + k^2)p_i = f_{a_i} & \text{On } \Omega_i \quad (1) \\ E_s I_s \nabla^4 w - \rho_s A_s \omega^2 w = p_1 - p_2 + f_s & \text{On } \Omega_s \quad (3) \\ \dfrac{\partial p_i}{\partial n_{fi}} = \rho_a \omega^2 w & \text{On } \Omega_s \quad (4) \\ p_i \; or \; \dfrac{\partial p_i}{\partial n_{fi}} \;\; conditions & \text{On } \Gamma_{1,k}, \Gamma_{2,k} \quad (5) \\ w = w_{imposed} & \text{On } \Gamma_s \quad (6) \end{cases}$$

For the structure note $\boldsymbol{E_s}$ the Young Modulus of the material, $\boldsymbol{I_s} = \frac{bh^3}{12}$ the inertia moment, $\boldsymbol{\rho_s}$ the density of the beam and $\boldsymbol{A_s}$ the section of the beam. As for the acoustic domain, $\boldsymbol{k} = \frac{\omega}{c}$ is the wavenumber, $\boldsymbol{\rho_a}$ is the density of the fluid, $\boldsymbol{\Delta}$ is the Laplacian operator, $\boldsymbol{\nabla}$ is the gradient operator.

2.2 Variational formulation of the problem

In the finite element context, the equations and boundary conditions can be expressed in the following weak forms :

$$
\begin{cases}
\int_{\Omega_s} E_s I_s \nabla^2 \Psi_s \nabla^2 w \, d\Omega_s - \omega^2 \int_{\Omega_s} \rho_s A \, \Psi_s w \, d\Omega_s - \int_{\Omega_s} \Psi_s \, n^f p \, d\Gamma_c = \int_{\Omega_s} \Psi_s \, f_s \, d\Omega_s & (7) \\
\frac{1}{\rho_a} \int_{\Omega_a} \nabla \Psi_a \nabla \mathrm{p} \, d\Omega_a - \frac{\omega^2}{\rho_a c^2} \int_{\Omega_a} \Psi_a p \, d\Omega_a - \omega^2 \int_{\Omega_s} \Psi_a \, n^f \, w \, d\Gamma_c = \frac{1}{\rho_a} \int_{\Omega_a} \Psi_a \, f_a \, d\Gamma_a & (8)
\end{cases}
$$

With $\boldsymbol{\Psi_s}$ and $\boldsymbol{\Psi_a}$ the weighting functions respectively for the structure and the acoustics, $\boldsymbol{w}$ is the flexural displacement of the beam and $\boldsymbol{p}$ is the pressure in the acoustic domain. Theses variational formulation lead to the following matrix system:

$$
\left(\begin{bmatrix} K_s & -C \\ 0 & K_a \end{bmatrix} - \omega^2 \begin{bmatrix} M_s & 0 \\ C^t & M_a \end{bmatrix} \right) \begin{Bmatrix} w \\ p \end{Bmatrix} = \begin{bmatrix} f_s \\ f_a \end{bmatrix} \quad (9)
$$

With $\boldsymbol{K_s}$, $\boldsymbol{M_s}$ the structure matrixes. $\boldsymbol{K_a}, \boldsymbol{M_a}$ the fluid matrixes. $\boldsymbol{C}$ is the coupling matrix that induce the transmission.

2.3 Definition of the Transmission Loss

In this paper, to evaluate the influence of the studied parameters, the Transmission Loss (TL) is defined as :

$$
TL = L_e - L_r \quad (10)
$$

Where L_e is the averaged Sound Pressure Level (SPL) on the emission room and L_r is the averaged SPL on the reception room. The SPL is given by :

$$
L_i = 10 \log_{10} \left(\frac{\int_{\Omega_i} p_\mathrm{i}^2 \, d\Omega_i}{P_{reff}^2} \right) \quad (11)
$$

Where P_i the pressure inside of the rooms.

2.4 Comparison with an analytical solution

An Analytical solution can be obtained by applying boundary conditions on the system presented in Figure 1:

$$
\begin{cases}
f_s = f_{a_1} = f_{a_2} = 0 & \text{On } \Omega_1, \Omega_2, \Omega_s & (12) \\
w = w'' = 0 & \text{On } \Gamma_s & (13) \\
\frac{\partial p_2}{\partial n_f} = 0 & \text{On } \Gamma_{2,2} & (14) \\
p_i = 0 & \text{On } \Gamma_{1,1}, \Gamma_{2,1}, \Gamma_{1,3}, \Gamma_{2,3} & (15) \\
\frac{\partial p_1}{\partial n_f} = V_0(y) = V_n \sin(k_y y) & \text{On } \Gamma_{1,2} & (16)
\end{cases}
$$

The pressure and displacements fields can be expressed as :

$$\begin{cases} p_1(x,y) = \left[\frac{V_n}{kx}\left(j\, e^{jk_x x} - \frac{e^{jk_x Lx1}}{sin(k_x Lx1)}\cos(k_x x)\right) - \frac{\rho_a \omega^2 w_n}{k_x sin\,(kx\; L_{x1})}\right] sin(k_y y) & (17) \\ p_2(x,y) = j\frac{\rho_a\; \omega^2 w_n}{k_x(e^{jk_x L_{x1}} - \; e^{2jk_x(Lx1+Lx2)}\;\; e^{-jk_x L_{x1}})}\left[e^{jk_x x} + \; e^{jk_x(Lx1+Lx2)}\;\; e^{-jk_x x}\right] sin(k_y y) & (18) \\ w(y) = w_n\, sin(k_y y) = \dfrac{\frac{V_n\; e^{jk_x x}}{k_x}\left(j - \frac{1}{ta\,n(k_x L_{x1})}\right)}{(E_s I_s\; k_y^4 - \rho_s A_s \omega^2) + \frac{\rho_a \omega^2}{k_x}\left(\frac{1}{tan(k_x L_{x1})} - \frac{j(e^{jk_x L_{x1}} + e^{2jk_x(L_{x1}+L_{x2})}\;\; e^{-jk_x L_{x1}})}{(e^{jk_x L_{x1}} - \; e^{2jk_x(L_{x1}+L_{x2})}\;\; e^{-jk_x L_{x1}})}\right)} sin(k_y y) & (19) \end{cases}$$

With $\boldsymbol{k_y} = \frac{n\pi}{L_y}$, n being the number of the sent mode on $V_0(y)$, $\boldsymbol{k_x} = \sqrt{k^2 - k_y^2}$.
The numerical data used for the verification are for the acoustic $\boldsymbol{\rho_a} = 1.2$ kg/m^3, $\boldsymbol{c} = 340$ m/s, $\boldsymbol{L_{x1}} = \boldsymbol{L_{x2}} = 1\,m, \boldsymbol{L_y} = 2\,m$ and the beam $\boldsymbol{E_s} = 6.95\, e^{10}$ Pa, $\boldsymbol{\rho_s} = 2700$ kg/m^3,with a thickness $\boldsymbol{h} = 0.005\,m$ and $\boldsymbol{b} = 1\,m$.

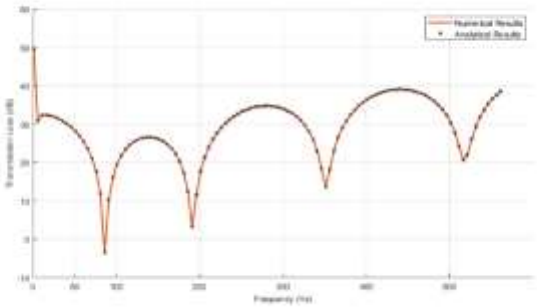

Figure 2. Comparison of the Transmission Loss (in dB)

Figure 2. shows that the TL results between analytical and numerical solution are in good agreement. A maximum error of 0.35% was obtained over the frequency range of the study.

3 Parametric study of the flanking acoustic transmissions on a curtain wall

3.1 Description of the problem

When an acoustic source generates a sound field in a building. The acoustic power is transmitted by the boundary wall between two adjacent rooms. But , more commonly, a fraction of this power is also transmitted by other junction elements, those are called flanking transmission. In the case of elements placed on a building facade, the curtain wall consists of those paths. In this study, only the Flanking-flanking (F-f) path will be studied. As it is only controlled by the flanking element. To isolate this path, the demising partition will be considered perfectly rigid and will act as a wall in the simulation.

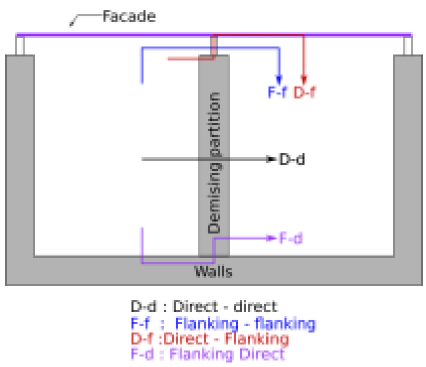

Figure 3. Schematic of the flanking paths between two rooms

3.2 Details of the numerical model

The model is composed of two rooms, one for the emission and the other for the reception. And the mullion is placed in between the rooms. The glazing can be considered by adding structural domains on the top part of the emission and reception rooms.

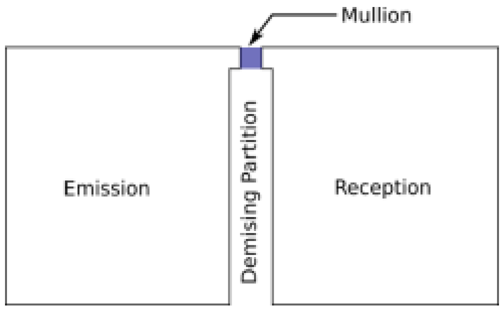

Figure 4. Schematic of the system of the emission/reception rooms

Monopolar sources will be placed inside the emission room. The demising partition will be supposed perfectly rigid and thus will not be considered in the transmission. The rooms will be fixed with an area of 16 m² (4x4 m), the rigid demising partition will be of 0.15m wide and the mullion placed in the middle. The mullion used is a WICTEC 50 from WICONA, with a width of 0.05 m and a length of study of 0.09 m to 0.15 m. The material of the mullion is an Aluminum AW 6060.

3.3 Transmission Loss considering the mullion

In this part the results of a mullion alone between two rooms will be presented using three parametric studies: the first one for the impact of the size of the mullion, the second for the impact of the boundary condition between the mullion and the demising partition, the third one to show the impact of the room geometry on the TL.

3.3.1 Impact of the size of the mullion on the transmission loss

For this study, three values of length of the mullion will be used: 0.09 m, 0.11 m and 0.15 m.
The mullion will be considered free from the demising partition. And the length of the mullion corresponds to the length of the space between the demising partition and the top wall of the rooms.

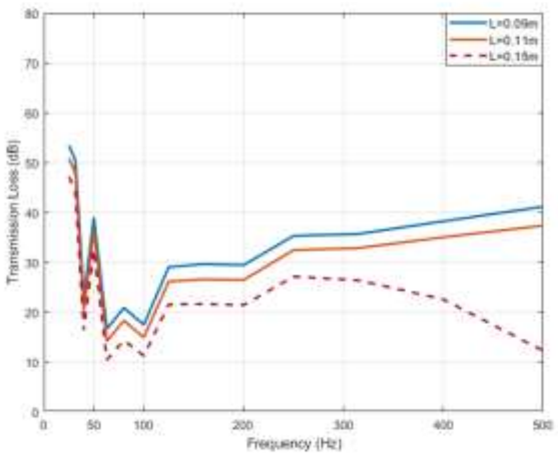

Figure 5. Impact of the size of the mullion on the Transmission Loss

As can be seen from the results, the lengthier the mullion is, the lower the TL results are.

The 0.15 m mullion present an important decrease in performance on the 500 Hz third-octave band. This phenomenon is explained by the presence of the first respiratory mode of the mullion at 469.8 Hz.
In the lower frequency ranges (inferior to 250 Hz), the behavior of the TL is the same for the three configurations, inducing TL controlled by the modal behavior of the rooms.

3.3.2 Impact of the boundary condition of the mullion on the transmission loss

In this part the impact of an elastic connection between the mullion and the demising partition is studied. The mullion is connected to the demising partition via an elastic link.

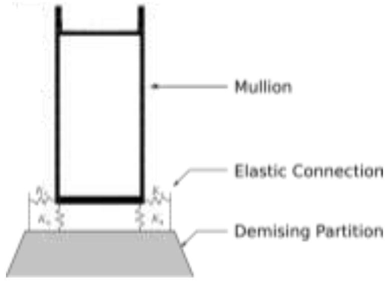

Figure 6. Schematic of the elastic boundary condition between the mullion and the demising partition

The rigidity used for this study are the same as the one for the study of the link between the mullion and glazing are $K_x = K_y = K$ with : $K = 0$ N/m , $K = 1.031e^6$ N/m and $K = (1 - 0.4j)1.031e^6$N/m.

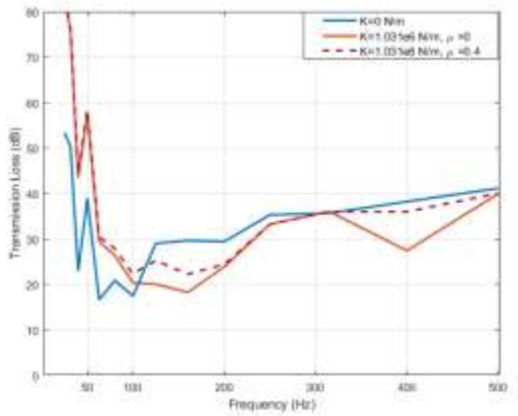

Figure 7. Impact of the fixation with the demising partition on the Transmission Loss

The results show important variation of the Transmission Loss when an elastic junction is considered.
The TL decreases until the 150 Hz frequency band, in this band two modes added by the fixation at 136.7 Hz and 157.5 Hz are present. The decrease at the 400 Hz frequency is explained by the presence of a mode of the mullion at 405.1 Hz.
This decrease in performances in the mentioned frequency bands is reduced by adding damping to the connection, mitigating the effect of the modal behavior of the mullion.
As the frequency considered for study go higher, the impact of the boundary condition is lessened when damping is considered as the values of the transmissions converge.

3.3.3 Impact of the emission and reception rooms geometry on the transmission loss

In the case of low frequency studies, another important factor is the heavy contribution of the room geometries on the general behavior of the rooms. Having a varying geometry will impact the form and frequency of the modes of the room. For this study four configuration of rooms were tested, the reference is the one used for the results presented in **3.3.1** and **3.3.2,** To obtain the modified rooms, the bottom left corner point (on the emission) and bottom right corner point (on the reception) have been moved by 0.3 m on the inside of the rooms.

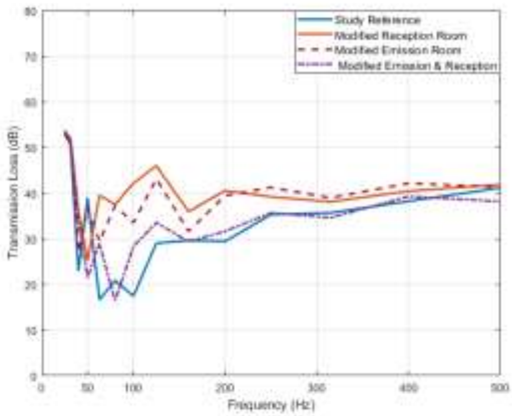

Figure 8. Impact of the room geometry on the Transmission Loss

The main difference between the Reference and modified rooms can be found between the 50 Hz and 300 Hz band where the TL obtained is higher when the rooms are modified than on the reference model.
In the case where the two rooms are modified and are symmetric in regards of the mullion, the Transmission has the same behavior as the results from the reference rooms indicating that having a modal similarity between emission and reception rooms will have a significant impact on the low frequency results of these calculations.

3.4 Transmission Loss considering the mullion/glazing system

In this part, the interaction between the glazing and the mullion will be studied. The junction between the glazing and the mullion will be simulated by the means of an elastic connection composed of a spring-mass-damper system added between the components.

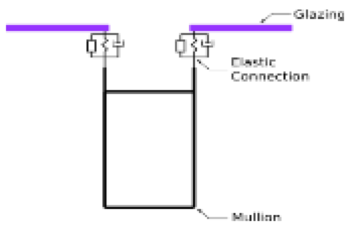

Figure 9. Schematic of the mullion/glazing connection

In this study, the gasket will be modelized by a spring with a complex young modulus to induce damping:

$$E_g = E(\,1 - \mu * j) \tag{20}$$

The gasket used with this model of mullion is made of EPDM, it has a young modulus $E_g = 1.375e^6$ Pa and a loss factor $\mu = 0.4$ the rigidity of connection is calculated by making the approximation:

$$K_g = \frac{E_g A_g}{L_g} = 1.031e^6 \text{ N/m}$$

With $A_g = 0.009\, m^2$, $L_g = 0.012\, m$.
For this study, the considered mullion will be of length $L = 0.09\, m$, the glazing is a unified simple glazing with a thickness of 0.006 m it is considered clamped on the edge of the rooms. Three values will be used for the rigidity of the gasket: $K = 0$ N/m , $K = 1.031e^6$ N/m and $K = (1 - 0.4j)1.031e^6$N/m

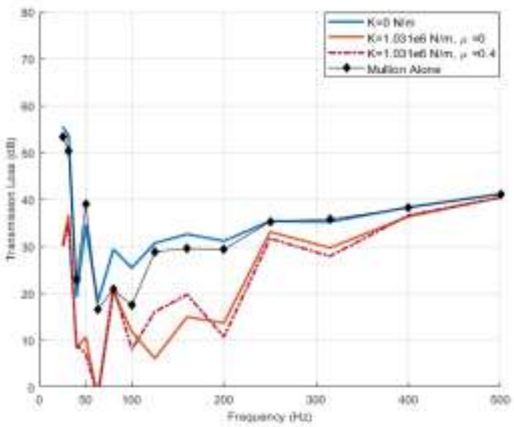

Figure 10. Transmission Loss for the Mullion/Glazing system

Compared to the previously obtained results with a mullion alone between the two rooms, the connection between the glazing and the mullion does have an impact on the transmission Loss on the band with a frequency inferior to 200 Hz as the modelized performances are lower when the connection is considered.
The difference between the mullion alone and the $K = 0$ N/m rigidity is due to the modification of the modal behavior of the system considering the vibro-acoustic coupling with the glazing.
When a coupling rigidity is present, the glazing actively transmits the acoustic power to the reception room. Which is the cause for the lower performances of the system in this case.

4 Model Reduction

In this section a condensation method will be proposed, and its usage discussed. A condensation method akin of the works of Guyan **[10]** will be used. Its usage will reduce the size of the systems for the calculation and accelerate the computations times.

4.1 Reduction by a condensation method

The goal of this method of reduction is to keep only the degrees of freedom (DOFs) who are to be coupled with the acoustic in future calculations.

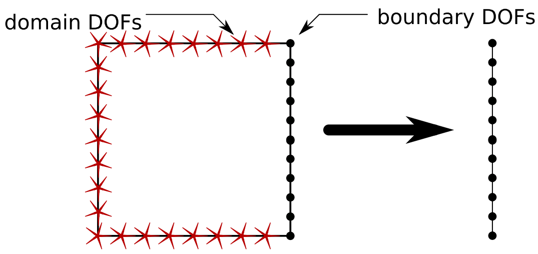

Figure 11. Schematic of the condensation

To do so, the dynamic stiffness matrix $\widetilde{K}$ is defined as:

$$\widetilde{K} = K - j\omega D - \omega^2 M \tag{21}$$

With K,M,D respectively the Rigidity, Mass and Damping Matrixes of the FEM model. The domain will consider internal source terms on the second term.
This matrix can be expressed in a way to the keep specific DOFs (b) and express them in function of the DOFs that are to be condensed (d) :

$$\widetilde{K}u = f \iff \begin{bmatrix} \widetilde{K}_b & \widetilde{K}_{bd} \\ \widetilde{K}_{db} & \widetilde{K}_d \end{bmatrix} \begin{Bmatrix} u_b \\ u_d \end{Bmatrix} = \begin{Bmatrix} f_b \\ f_d \end{Bmatrix} \quad (22)$$

The domain term can then be written :

$$u_d = \widetilde{K}_d^{-1} f_d - \widetilde{K}_d^{-1} \widetilde{K}_{db}\, u_b \quad (23)$$

By inserting the Eq. 23 into Eq. 22 the resulting system is :

$$\left(\widetilde{K}_b - \widetilde{K}_{bd}\, \widetilde{K}_d^{-1} \widetilde{K}_{db}\right) u_b = f_b - \widetilde{K}_{bd}\, \widetilde{K}_d^{-1} f_d \quad (24)$$

The benefit of this kind of condensation is that it permits the suppression of certain degrees of freedom even though they possess an acoustic or structural source term. And by doing the reduction on the coupling interface makes it independent of external vibro-acoustic coupling terms creating the possibility to reduce the mullion on his own and use pre-calculated element in the computation of the flanking transmission loss.

4.2 Comparison with numerical results obtained with a full model

In this part a comparison of the results obtained by the reduced model will be made with those obtained by the full model. The mullion is considered alone and free from the demising partition. Its length is of L = 0.09 m. For the reduction, the mullion is reduced to its coupling interface with the emission and reception rooms presented in **3.3.1**.

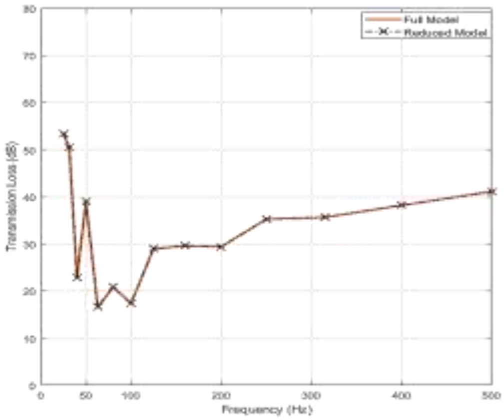

Figure 12. Comparison of the transmission between a reduced mullion element and the full model

The results obtained are in good agreements in third octave band. As for the reduction, the size of the system went from : 23.544 DOFs to 17.737 DOFs which is a 24 % of reduction of the size.
The computation time went from 367 seconds to 297 seconds which is a 20% of reduction of the time.
The reduction method proposed in this paper can be used for faster computation of the transmission loss in the study of flanking transmission loss of the curtain-wall mullions.

5 Conclusions

In this paper, we presented a numerical model able to give quantitative results on the flanking transmission loss of curtain-wall mullions.
A study of different parameters and their influences on the performance of the mullion have been made.
Finally, we presented a reduction method able to reduce the mullion and gives results like those of a full model.

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