ANALYSIS OF THE GEOMETRIC PARAMETERS OF ACOUSTIC FILTERS OF THE EXPANSION CHAMBER TYPE

Maria dos Reis Santos Borges[1]
Maria Alzira de Araújo Nunes[2]
Universidade de Brasilia, Faculdade do Gama, Graduate in Engineering Materials Integrity – Integrity PPG
Área Especial de Indústria e Projeção – Setor Leste - Gama-DF, 72444-240.
[1]borgesmaria9421@gmail.com; [2]maanunes@unb.br

Abstract. *Reactive acoustic filters are elements that use geometry as a tool for reflection of incident acoustic energy. These include those of the expansion chamber type, which consists of one or more chambers acting as resonators, where part of the incident sound energy returns directly to the source. Such elements are widely used in automotive, industrial and air conditioning systems. In this paper, the acoustic performance of three simple expansion chambers: with circular, square and rectangular cross section, was analyzed by the Transmission Loss (TL) curve. The analysis was performed in two steps: analytical and numerical. Firstly, through the analytical formulations, which are based on the one-dimensional plane waves theory, the TLs were calculated for the three different geometric forms, obtaining results only up to the highest cut-off frequency among the three models. To visualize the influence of such geometries at high frequencies, we used the Finite Element Method, since analytically the TL curve does not express the influence that different acoustic filter forms can exert. Comparison between the results showed that changing the cross-sectional area of this type of element can modify the TL of the acoustic filter at high frequencies.*

Keywords: *expansion chamber, acoustic filters, transmission loss, geometric parameters, finite element method.*

1. INTRODUCTION

Widely used for noise attenuation and sound absorption, acoustic filters are devices that, through their composition or geometry, effect reflection, suppression or dissipation of noise. Such devices are used in combustion engine ducts, exhaust systems, air conditioning systems, pumps, compressors, among others. Due to their composition, geometry and noise attenuation capacity, the silencers are divided into dissipative, reactive or dissipative-reactive. The dissipative silencers are those in which the mechanism of sound attenuation is given by the dissipation of this, and for that, it uses acoustic absorbent materials in its composition, that absorb the acoustic energy transforming it into heat. This type of acoustic filter is efficient over wide frequency ranges (Beranek and Ver, 2006).

The reactive acoustic filter, object of study since this work, consists of tubular elements with different area discontinuities that directly influence the acoustic energy reflection process, through the impedance incompatibilities caused by the waves that propagate inside this system. The most common types of reactive silencers are Helmholtz resonators, expansion chambers (single, double or mixed), quarter-wave tubes, perforated elements, among others (Bies and Hansen, 2003). Among the most commonly used for noise control in air circulation ducts is the expansion chamber type, whose main mechanism is its discontinuities of area, which is made up of sudden contractions and expansions of area (Cartaxo, 2007). One of the geometrical variables of this type of acoustic filter is the geometric format that the camera can present, which can be determined according to its application.

To evaluate the performance of this type of acoustic filter, it is calculated the Transmission Loss (TL) that calculates the difference between the levels of incident and transmitted sound power, considering for this an anechoic termination, which guarantees non-reflective properties in that region. Thus, the geometric aspects of the expansion chamber, as well as the TL, must designed to meet the stipulated design requirements (Silva, 2016; Barron, 2003).

The analytical estimation of TL in acoustic filters is based on the one-dimensional theory of plane waves propagation, which is limited by the cut-off frequency, i.e. the frequency below which only plane waves propagate. One-dimensional theory is, therefore, efficient in expressing the TL of expansion chamber type acoustic filters in the low frequency region, which is not valid for frequency ranges above the cut-off frequency, where both multidimensional wave propagation occurs as well as the excitation of higher order modes. Another limitation of analytical formulations is that it considers only the geometric parameters of the acoustic filter, regardless of the shape they may assume. These facts highlight the need to employ numerical approaches capable of expressing both TL at high frequencies, as well as the influence that format can have on the TL curve. In order to analyze the influence that such formats exert on the acoustic performance of expansion chambers, this work will use the Finite Element Method - MEF, with the aid of Ansys® software. (Wu and Wang, 2011; Yu and Cheng, 2015).

At the end of the 1990s, the work of Selamet and Radavich (1997) and Selamet and Ji (1999), analyzed and emphasized this particularity presented by the one-dimensional theory. This studies showing the efficiency of the numerical methods when compared to the experimental, in the representation of the influence of the geometric complexities existing in simple expansion chamber type silencers and with extended inlet/outlet ducts. Since then, recent studies have attempted to develop several alternatives capable of replacing one-dimensional theory, as well as improvements in existing numerical methods.

Wu and Wang (2011) studied the behavior of a simple expansion chamber with right angle entry through the numerical Boundary Element Method-BEM and experimentally. The authors note that this change in the input duct influences the TL values at high frequencies (3500 ~ 5000 Hz) when compared to the performance of the straight input chamber. This performance could be verified only by numerical methods, given the difficulties that analytical theories present in expressing such effects.

In order to analyze the influence of internal configurations on expansion chambers, Yu and Cheng (2015) used an alternative computational approach, called the Patch Transfer Function Method (PTF), a sub-structuration technique, allowing a better analysis of the acoustic behavior of complex geometries within this type of acoustic filter. In the comparisons made, the PTF showed good agreement with the FEM and with the experimental, differently from the comparison with one-dimensional theory, as expected.

In order to verify the influence that chamber format has on TL values and to know the behavior of such chamber types, in this article, it will be shown through comparisons between the two main ways of obtaining the TL (analytical and numerical), the influence that different cross-sectional formats of reactive expansion chamber type acoustic filters. The results were obtained for three different cross-sectional formats: circular, rectangular and square, and to better understand the influence of such formats, the distribution of the sound pressure field in the values where the maximum TL was obtained was obtained.

2. TRANSMISSION LOSS OF A EXPANSION CHAMBER

2.1 Transfer Matrix Method

The Transfer Matrix Method (TMM) or 4-pole method is the analytical method most used in the design of reactive silencers, considering that it is able to take into account both the geometric effects of the element and the effects of flow velocity, pipeline properties, among others (Beranek and Ver, 2006). The method is based on the relationship between each element of the acoustic filter and its transfer function, so that the complete element is described as a multiplication of the transfer functions that constitute the element, considering propagation of plane waves, and in this case considering absence flow. Figure 1 shows the longitudinal section of the one-dimensional model of an expansion chamber-type reactive silencer with circular cross-section, where the four poles (p_1, v_1 e p_2, v_2) are represented which are sound pressure and velocity values at the inlet and outlet (Chitale-Patil, Patil, and Patil, 2014; Fang, Ji, and Liu, 2017).

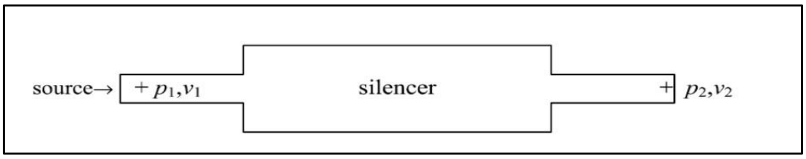


Figure 1 - Longitudinal section of the cylindrical expansion chamber (Chitale-Patil, Patil, and Patil, 2014).

By the four-pole method, the silencer of Fig. 1 can be described according to Eq. (1), where $T_{11}, T_{12}, T_{21}, T_{22}$, are the parameters describing the inlet duct, sudden expansion, sudden contraction, and the outlet duct of the acoustic filter, respectively. p_i is the acoustic pressure at point i throughout the system, v_i is the mass flow velocity, which is given by the relationship between the density of the middle (ρ_0), the cross-sectional area of the duct at point i (S_i), and the acoustic velocity particle (u_i).

$$\begin{bmatrix} p_2 \\ v_2 \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p_1 \\ v_1 \end{bmatrix} \tag{1}$$

For ducts with uniform cross-section, such as the inlet and outlet ducts of the silencer, the transfer matrix is given by Eq. (2), where $k = \omega/c_0$ is the wave number, $\omega = 2\pi f$ is the is the angular frequency in rad/s, f is the frequency of excitation, c_0 is the velocity of the sound in the middle, in m/s, and L is the length of the duct (Beranek and Ver, 2006; Cazzolato and Howard, 2015).

$$T_{pipe} = \begin{bmatrix} \cos(kL) & j\frac{c_0}{S}\sin(kL) \\ j\frac{S}{c_0}\sin(kL) & \cos(kL) \end{bmatrix} \tag{2}$$

Thus, the total transfer matrix of the silencer is given by the product between the transfer matrices of the three ducts $(\mathbf{T_1}, \mathbf{T_2}, \mathbf{T_3})$ given by Eq. (2.1) and Eq. (3), depending on its configurations, in this case all and also considering the impedances of the source at the input of the ducts (Z_s), and the impedance of the output of the acoustic filter (Z_T).

$$\begin{bmatrix} p_s \\ v_s \end{bmatrix} = \begin{bmatrix} 1 & Z_s \\ 0 & 1 \end{bmatrix} \boldsymbol{T_1 T_2 T_3} \begin{bmatrix} 1 & Z_T \\ 0 & 1 \end{bmatrix} \begin{bmatrix} 0 \\ v_2 \end{bmatrix} \tag{2.1}$$

$$\begin{bmatrix} p_s \\ v_s \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} 0 \\ v_2 \end{bmatrix} \tag{3}$$

The TMM is based on the one-dimensional wave propagation theory, where the cut-off frequency (f_c) is established, that is, where below this frequency, there is only the propagation of only plane waves within the duct. This frequency is calculated according to Eq. (4) for a circular cross-sectional area, where d_c is the chamber diameter in meters, and according to Eq. (5), when the duct has a rectangular cross-sectional area, with H being the largest in meters (Munjal, 1987; Cazzolato and Howard, 2015).

$$f_c < \frac{1{,}84 c_0}{\pi d_c} \tag{4}$$

$$f_c > \frac{c_0}{2H} \tag{5}$$

The main property used to analyze the behavior of reactive acoustic filters is Transmission Loss (TL), which differs from other parameters that are also used to analyze the performance of these types of elements, such as Loss of Insertion (Insertion Loss-IL) and Noise Reduction (NR), because the TL depends only on the characteristics of the silencer, independent of the source modeling. The TL is defined as the difference between the levels of incident sound power (L_{wi}) and transmitted (L_{wt}), according to Eq. (6) (Munjal, 1987; Barron, 2001).

$$TL = L_{wi} - L_{wt} \tag{6}$$

Using the Transfer Matrix Method, TL can be written according to Eq. (7), S is the cross-sectional area of the main duct, in m²; and the parameters $T_{11}, T_{12}, T_{21}, T_{22}$, are obtained from the transfer matrix of the expansion chamber, according to Eq. (3).

$$\text{TL} = 20\log_{10}\left|\frac{T_{11} + \left(\frac{S}{c_0}\right)T_{12} + \left(\frac{c_0}{S}\right)T_{21} + T_{22}}{2}\right| \tag{7}$$

By replacing the parameters found in the transfer matrix and writing in a simpler way, the TL can be described by Eq. (8), assuming no flow and anechoic termination (non-reflective properties) in the silencer (Munjal, 1987; Beranek and Ver, 2006).

$$\text{TL} = 10\log_{10}\left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 sen^2 \frac{2\pi L_e}{\lambda}\right] \tag{8}$$

In the determination of TL, the geometric parameters of the expansion chamber are considered through the variable$m = \frac{S_2}{S_1}$, which is the ratio between the cross-sectional areas of the chamber, in m² and the ducts external to it. The variable L_e representing the length of the expansion chamber in m, and through the wavelength λ, which is the ratio of the velocity of the sound in the middle (c_0), in m/s, and the analysis frequency (f) in Hertz (Munjal, 1987).

Equation 9 is obtained through the analysis of Eq. (8), and describes the behavior of the TL curve with respect to the maximum and minimum points. In the frequencies of minimum (TL = 0), the sound power incident on the acoustic filter is the same as that transmitted to the anechoic termination, and therefore there are no reflective effects. In this way, it is possible to obtain both the frequency range in which maximum and minimum TL occur, as well as the peak values in which they occur (Pasqual, 2005).

$$f_n = \frac{n c_0}{4 l_e} \tag{9}$$

2.2 The Decomposition Method

Among the experimental methods to obtain TL of a silencer, one of the most well known and used is the technique of the two microphones, which is based on the decomposition of the incident sonorous spectra transmitted between the source and the acoustic element. Thus, the analytical method used by this experimental form and also in numerical simulations like the one used in this article, is called the Decomposition Method, which experimentally has a configuration as shown in Figure (2). The method is based on the theory of plane waves, but is able to express the behavior after this region, and is valid also for propagation of non-plane waves, considering its experimental applicability (Tao and Seybert, 2003).

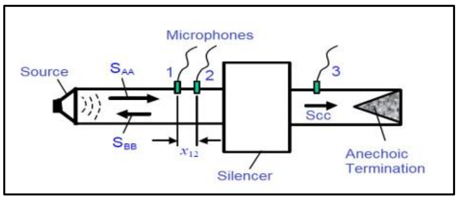


Figure 2 – The decomposition method (Tao and Seybert, 2003).

In the decomposition method, as can be seen from the Fig. 2, the sound wave S_{AA} propagates along the duct, finding the area discontinuities in the silencer. This wave becomes stationary, and decomposed into two parts: S_{CC} which directed to the end of the duct, where it finds an anechoic termination, and; S_{BB} which reflected back to the source, due to the impedance incompatibilities encountered during the passage of this wave by the acoustic filter.

Through the method of the two microphones, we can describe the spectrum of the incident wave S_{AA} and according to Eq. (10), where S_{11} e S_{12} are the acoustic pressures measured by the microphones 1 and 2 respectively. C_{12} e Q_{12} are related to the imaginary parts existing at the intersection between the incident spectrum and points 1 and 2; k is the wave number, and finally, x_{12} which is the distance between the two microphones.

$$S_{AA}=\frac{S_{11}+S_{12}-2C_{12}\cos kx_{12}+2Q_{12}\sin kx_{12}}{4\sin^2 kx_{12}} \tag{10}$$

From the equation describing the incident wave spectrum, we can write the effective pressure (RMS) for this wave as: $p_i=\sqrt{S_{AA}}$, and we have the equation of TL, Eq. (11), by the decomposition method:

$$TL=20\log_{10}\frac{p_i}{p_t}+10\log_{10}\frac{S_i}{S_0} \tag{11}$$

Where p_t is the sound pressure level transmitted by the acoustic filter, S_i is the area at the duct entrance, and S_0 is the area at the duct outlet. Eq. (11) is also valid only considering anechoic termination, and due to the fact that it does not have the relevant limitation to plane wave theory, it will be used here to express the behavior of the acoustic filter outside of the plane wave region through the FEM.

3. METHODOLOGY

The methodology used was divided into three steps: obtaining the analytical TL, obtaining the numerical TL and obtaining the sound pressure field at the TL peak values found. In the first part, formulations based on the theory of plane wave propagation were used, through the TMM, as explained in section 2.1, where it is valid only until the cut-off frequency, according to Eqs. (4) and (5) , as shown in the last column of Table 2.

Table 2. Dimensions of the models used and their respective cut-off frequencies

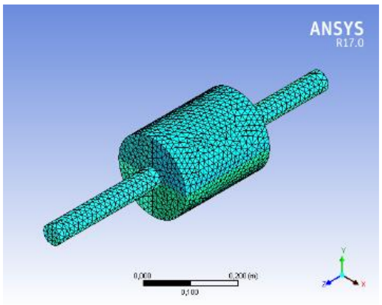
Model	Geometry of chamber (Length maintained in 0,25m)	Dimensions used [m]	Cut-off frequency [Hz]
1	Circular	Radius = 0,100	1004,45
2	Rectangular	Length = Width = 0,25 Height = 0,1256	686
3	Square	Length of edge = 0,1772	967,83

The types of cross-sectional areas used were: circular, rectangular and square, with the area values being maintained at S_c= 0,0314 m² and the acoustic chamber length at 0,250m , the other dimensions of each model can be visualized in the third column of Tab. 2. The analytical TL was obtained in a *Matlab*® software.

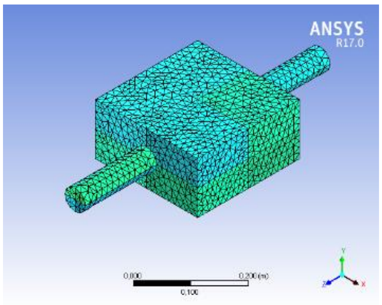
The second step of the methodology consists in calculating the TL using numerical methods, in this case, the Finite Element Method (FEM), which, as previously mentioned, is one of the ways to obtain the acoustic behavior of different silencer geometries. The method is based on the representation of subdivided geometry in a number of finite elements, called mesh, where each link between elements is called by node (Silva, 2016). Similar to the analytical obtainment, the three types of cross-sectional area were used, as can be seen, where the constant cross-sectional area values and acoustic chamber lengths were maintained. Obtaining using the Finite Element Method occurs through the Decomposition Method, which does not have the limitation in the analytical formulations with respect to cut-off frequencies.

Thus, using the software of Finite Elements *Ansys® Workbench* v.17, with the aid of the *ACT Acoustics* extension, the numerical TL was obtained. First, the type of analysis was chosen as *Harmonic Response*, and then the geometric characteristics of the model were inserted in the *Design Modeler* section. With the geometric models ready, in the *Mechanical* section, begins the obtainment of the mesh, where was inserted the type and size of element for the composition of the mesh. The type of element chosen was FLUID30, which is a 3D linear element that can assume hexahedral, prismatic, tetrahedral and pyramidal configurations. This element has 8 nodes where each node has a degree of freedom of pressure and three translational degrees of freedom along the x, y and z axes (Cazzolato and Howard, 2015).

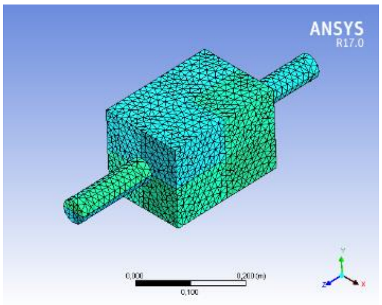
Based on this, and adopting the air density as $\rho = 1{,}21$ kg/m³ and sound velocity c_0 = 343 m/s, the chosen frequency range was 0 to 3 kHz, with intervals of 10Hz. According to Cazzolato and Howard (2015), meshes formed by acoustic elements must contain a density based on the number of elements per wavelength (EPW). Thus, the software recommends that when using acoustic elements, a mesh with density of at least 6 EPW should be used. With a wavelentgth of $\lambda = 0{,}114m$, so the numerical models (meshes) for each format are shown in Fig. (3), where tetrahedral elements of 0.0191m in size were used to better visualize the behavior of non-plane waves inside the acoustic filter. In both forms of obtained TL, the external ducts remained with a circular cross-sectional area, with a diameter of 0,05 m and a length of 0,30 m.



Model 1: Circular



Model 2: Rectangular



Model 3: Square

Figure 3 – Numerical models in FE of the acoustic filters of simple expansion chamber type.

With the numerical models, also in the *Mechanical* section, the boundary conditions that relate to the acoustic characteristics that enable the TL were inserted. The TL is a parameter independent of the sound source or excitation, to represent this characteristic numerically, a unit velocity or pressure is normally inserted in the inlet duct. Thus, with the *ACT Acoustics* extension menu bar, in the *Excitation* section, was selected *Acoustic Mass Source* type excitation, which is a way of representing the sound source as a mass flow rate per unit volume. Thus, selected the input faces of the duct, and in the row *Amplitude of Mass Source* was entered unit value. In the same section, but selecting *Radiation Boundary* were inserted unit impedance and admittance, corresponding to the anechoic termination, in the inlet and outlet ducts.

To obtain TL, two external ports were also defined, one at the inlet and the other at the outlet duct, also on the *Boundary Conditons* section. In the *Results* tab, the *Acoustic Power Result Plot* was selected, so that the software already provided both the TL curve as a function of the frequency, calculated by the Decomposition Method, and the values of sound pressure levels in each defined port and in addition, the software also provides the reflection and absoption coefficients of the acoustic element. At the end of the analysis, TL data for the three models were carried out in *Matlab*® software, where, as previously mentioned, the analytical TL was also calculated.

The third part of the methodology consisted of investigating the sound pressure field in the cut-off frequencies and in the maximum TL frequencies in order to verify the behavior of the sound pressure in the different types of acoustic chamber. For this, in the *Mechanical* section, *Results* tab, *Acoustic Pressure* was selected, and in the *Frequency* row, each of the frequencies was inserted. The cut-off frequencies were calculated according to Eqs. (4) e (5). Already the frequencies obtained by the numerical method, were those in which, there was maximum TL.

4. RESULTS AND DISCUSSIONS

In Figure 4 are plotted the three analytical TL curves to the highest cut-off frequency among the models used, in this case, 1004.45 Hz. The curves were obtained through the TMM, as shown in Section 2.1.1, which is only valid up to the cut-off frequency, where below it only plane waves propagate. Due to this limitation, it is not possible to perceive through the analytical curve the influence of different types of cross sections, since the curves are equal to their maximum cut-off frequencies. The small difference between the analytical and numerical results of approximately 1dB is due to the accuracy of the numerical model. This behavior, according to Selamet and Ji (1999), is justified by the fact that after the cut-off frequency, the non-plane waves propagation begins and, consequently, the excitation of high order modes. Such features, one-dimensional theory is not able to express, resulting in changes in the TL curve viewed only through numerical models.

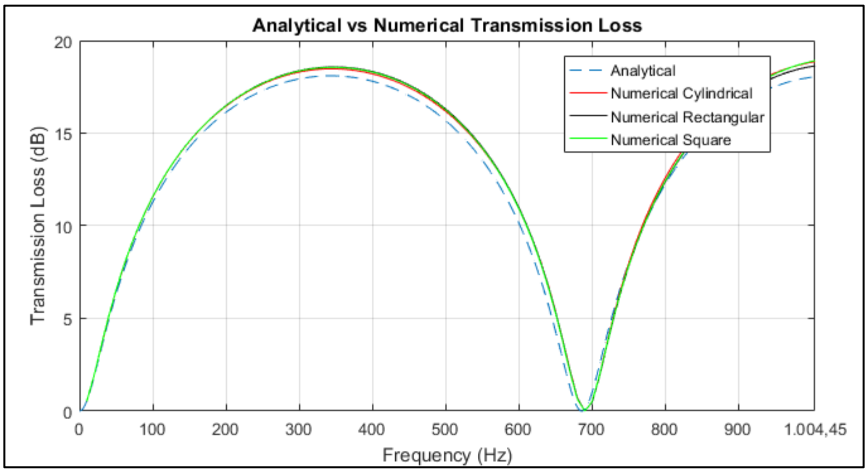


Figure 4 – Comparison between the Analytical vs Numerical Transmission Loss for the three different cross-sectional area up to the high cut-off frequency among the three models.

Therefore, as mentioned in the methodology, the numerical TL curves, illustrated by Fig. (5), were plotted up to the frequency of 3000 Hz to visualize the influence that the different cross-sectional areas have outside the plane wave region. It is noticed that initially, in the low frequency region, both curves follow a pattern similar to the analytical curves, due to the fact that the Decomposition Method can also express the acoustic behavior in the plane wave region. Finishing up this region, it is noticed that there is a difference in the behavior of TL mainly after their respective cut-off frequencies. This is due to the type of cross-sectional area of the filter, i.e. the influence of this feature can only be initially seen around 1300 Hz, where the behavior of the curves is characterized by peaks where the maximum TL occurs and the excitation of the high-order modes, a feature that is also very important in acoustic filter design.

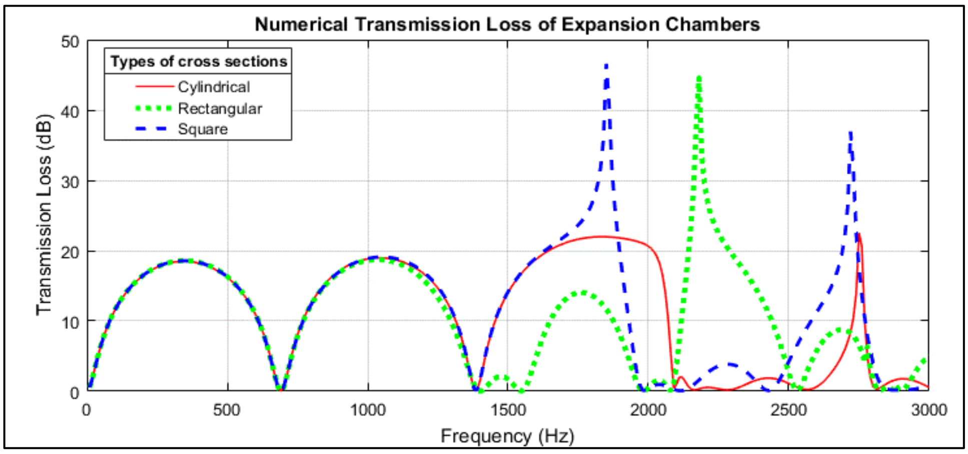


Figure 5 - Numerical Transmission Loss for the three different cross-sectional area.

Given the numerical curves of TL and continuing the methodology used here, one can obtain the maximum attenuation values and their respective frequency of occurrence. In Table 3, on the second column, the values of frequency where the maximum peak of TL obtained analytically, calculated according to Eq. (9), are listed, in the third column are listed the values in which there was maximum TL for such frequencies. The fourth column shows the numerical frequency values where there was maximum TL and finally the fifth column with the TL values corresponding to those frequencies.

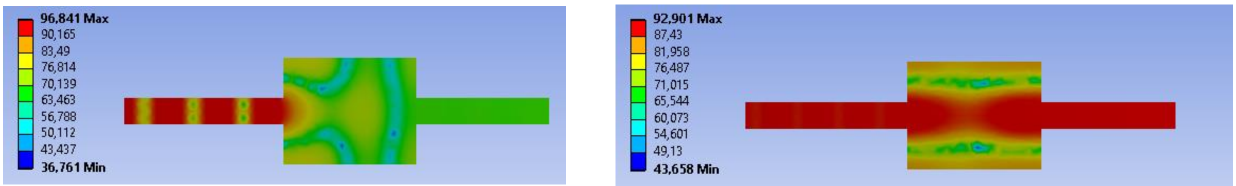
Table 3. Frequencies of maximum TL values obtained analytically and numerically

Model	Frequency of maximum analytical TL [Hz]	Maximum analytical TL (dB)	Frequency of maximum numerical TL [Hz]	Maximum numerical TL (dB)
Circular	343	18,10	2750	22,5
Rectangular	343	18,10	2180	44,63
Square	343	18,10	1850	46,58

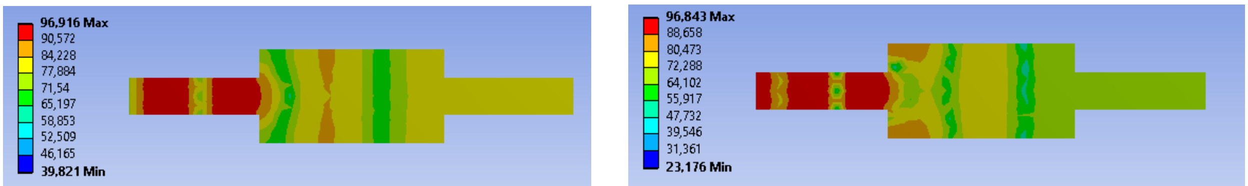
As already predicted, in the region below the cut-off frequency, obtained through the one-dimensional models, the same TL results were obtained, since the only parameter that differentiates each model is its cut-off frequency. Numerically, it is seen that not only the frequencies, but also the peak values, differ from those obtained analytically. According to Pasqual, this is due to the two-dimensional effects caused by the area discontinuities within the acoustic chamber (Pasqual, 2005).

In order to know the sound pressure distribution in each silencer, the sound pressure field at the maximum TL peaks of the square model at 1850 Hz and at the TL peak of the rectangular model at 2250 Hz were plotted in Fig. 6. .

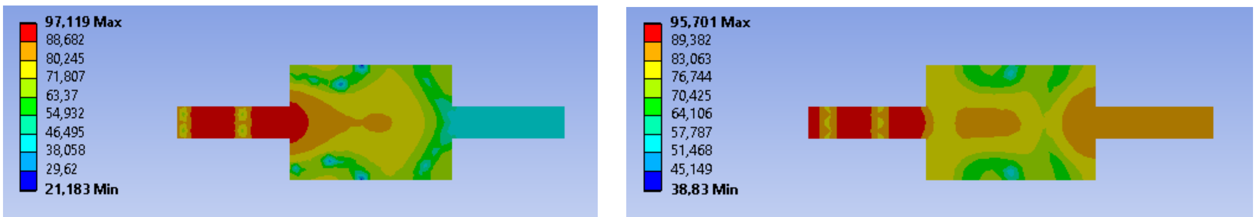
f=1850 Hz f=2250 Hz



a) Circular Model



b) Rectangular Model



c) Square Model

Figure 6 - Distribution of the sound pressure along the expansion chamber along the TL curve.

At 1850 Hz occurs is a significant change between the three models used. The maximum TL of the square model occurs, which is around 45dB, but the TL values differ between each model, which is justified by observing the sound pressure field through Figure 6. It is noticed that the distribution of sound pressure is directly influenced by the chamber geometry, where at this frequency, in the circular model, is where the formation of the third axial mode begins, which explains the behavior of the curve at this frequency. In the square model, it can be seen in figure 6.c, that in this frequency region is where the chamber shape most influences the sound pressure distribution, justifying the peak value. At the frequency of 2250 Hz, it can be seen in figure 5 that the circular model has almost zero attenuation, where the pressure distribution, as can be seen in figure 6.a, is concentrated in the central part of the silencer, justifying the low PT values. The rectangular model has a pressure distribution indicating the formation of the third axial mode. In the square model, it is found that at this frequency begins the formation of the fifth axial mode.

5. CONCLUSIONS

In this work, in order to know the influence that the geometry of the expansion chamber in the parameter most used to describe the acoustic performance of reactive filters, the Transmission Loss (TL). For this, two calculation methods were used: analytical, using TMM, and numerical, through MEF, where the cross-sectional area formats were altered and the lengths of the acoustic chamber were maintained. It was also investigated the behavior of the sound pressure in the frequencies of maximum attenuation, in order to understand its behavior within the different formats studied here.

When observing the TL curve analytically, it is observed that there is a behavior limited to the frequency of propagation of plane waves (the cut-off frequencies), where after the same, the theory has no validity since it is assumed that after this, non-plane waves. This fact demonstrates the need to use numerical and experimental methods to obtain the TL. Trough MEF, it was observed that the change in the geometry of this type of acoustic filter is of significant order with respect to both the frequency band and in the maximum values of TL. Which is only possible viewed at high frequencies, i.e. outside the region of plane waves.

Although the analytical theory is a fast and efficient method to obtain TL in the region low frequencies, but it is importance known the acoustic behavior of high frequencies, which is where the excitation of the high order modes takes place that in this were expressed using numerical analyzes using numerical methods. Thus, due to the differences between the values found in both methods, and because they occurred in frequency bands outside the region of plane waves, it became interesting to know the behavior of the sound pressure inside the acoustic filter.

One of the alternatives to improve the design of reactive silencers of the type expansion chamber, would be the use of optimization techniques, as proposed by Lee and Jang (2012), the use of sub-structuration techniques, such as PTF, showed good agreement with MEF in the studies of Yu and Cheng (2015). Another alternative would be to perform experimental bench tests, using established methods such as two-microphone method or two-load method, as described by Tao and Seybert (2003).

6. REFERENCES

Barron, Randall F., 2001. *Industrial Noise Control and Acoustics,* Marcel Dekker Inc, New York.

Beranek, L. L. and Ver, I. L. 2006, *Noise and Vibration Control Engineering: Principles and Applications*, Jonh Wiley & Sons Inc, New York, 2nd edition.

Bies, D.A. and Hansen, C.H., 2003. *Engineering Noise Control: Theory and Practice*, E & FN Spon, London, 2nd edition.

Cartaxo, Antonio José Ferreira Mourão, 2007. *Caracterização de Filtros Acústicos Baseada no Método dos Elementos Finitos*. Master Thesis, Universidade Técnica de Lisboa, Lisboa, Portugal.

Chitale-patil, Varsha; Patil, Amol R; and Patil, R B, 2014. “Design Analysis and Performance Evaluation of Reactive Silencer by SYSNOISE.” *International Journal of Innovative Research in Science, Engineering and Technology*.

Fang, Z.; Z. L. Ji; and C. Y. Liu, 2017. “Acoustic Attenuation Analysis of Silencers with Multi-Chamber by Using Coupling Method Based on Subdomain Division Technique.” *Applied Acoustics*.

Howard, Carl Q., and Benjamin S. Cazzolato. 2015. *Acoustic Analyses Using MATLAB Ans ANSYS*. Vol. 32. New York.

Lee, Jin Woo; Jang, Gang-Won, 2012. Topology design of reactive mufflers for enhancing their acoustic attenuation performance and flow characteristics simultaneously. *International Journal for Numerical Methods in Engineering.*

Munjal, Manchar Lal, 1987. *Acoustics of ducts and mufflers with application to exhaust and ventilation system design*, John Wiley & Sons.

Selamet, A., and Z. L. Ji, 1999. “Acoustic Attenuation Performance of Circular Expansion Chambers with Extended Inlet/Outlet.” *Journal of Sound and Vibration*.

Silva, Gabriela Cristina Cândido, 2016. “*Estudo Analítico e Numérico Do Desempenho Acústico de Silenciadores Reativos Na Presença de Escoamento Médio Incompressível*.” Master thesis. Universidade de Brasília, Brasília, Distrito Federal, Brasil.

Pasqual, Alexander Mattioli, 2005. “*Análise de sistemas reativos para controle de ruído em dutos pelo método dos elementos finitos”.* Master Thesis. Universidade Federal de Minas Gerais, Belo Horizonte, Brasil.

Z. Tao, A.F. Seybert, 2003. “A review of current techniques for measuring muffler transmission loss”. SAE Transactions.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper