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Acoustic impedance measurements of aerospace resonators and high fidelity simulation of non-linear interaction of multiple tones

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Resumo

O ruído de turbinas do tipo turbofan apresenta característica tonal que causa penalização nas métricas de incomodidade utilizadas para avaliar áreas habitadas próximas aeroportos. Uma das maiores dificuldades para projetar materiais acústicos eficientes é causada pela interação não linear de múltiplos tons causados por altos níveis de pressão sonora nas turbinas. Os materiais de revestimento acústico utilizados para este fim são compostos por chapas perfuradas e cavidades que formam ressonadores instalados nos dutos das turbinas. Neste artigo são realizadas medições de impedância acústica de tais materiais chamados de liners, os quais foram modelados semi-empiricamente e através de técnica aeroacústica de alta fidelidade chamada de DNS (*Direct Numerical Simulation*). São apresentadas análises quanto a geometria dos furos nas chapas perfuradas e é proposta uma modelagem da impedância acústica não linear do material na presença de múltiplos tons em altos níveis de pressão sonora, sendo que um modelo semi-empírico de impedância acústica 1D para tons puros com correção multitonal foi desenvolvido a partir dos dados experimentais.

Palavras-chave: ruído aeronáutico, impedância acústica, não linearidade, múltiplos tons.

PACS: 43.28.-g.

Measurement of acoustic impedance of turbofan liners and high fidelity simulation of non-linear multiple tone interaction

Abstract

Aircraft noise is a problem near airports all around the world. As a result, aircraft certification requirements are becoming more stringent over time. The engine fan is one of the most prominent noise sources that produces high amplitude blade passing frequency tones, their harmonics, and engine order buzz tones. Acoustic panels, consisting of resistive layers backed by honeycomb cells and a reflective hard wall, are used to line the engine nacelle in order to attenuate both broadband and annoying tonal noise. The acoustic impedance of single layer perforate liners (SDOF), that are aerospace resonators, show greater sensitivity to high sound pressure levels and grazing flow in comparison with wire mesh liners. Two harmonically related tones with varying combinations of amplitude and frequency were used as excitation signals in experimental and numerical models. 2D DNS model, based on FEM, was developed and validated using the MATLAB & COMSOL Multiphysics 5.2a live link. The COMSOL code reproduces high SPL experimental conditions. These studies have aided the development of an improved semi-empirical 1D impedance model for pure tone excitation, and a resistance correction for multiple tone excitation. The new semi-empirical model for high level multiple tone excitation on liner surface is proposed and discussed, showing reasonable understanding about energy transference between tone components, and showing promising results in terms of acoustic resistance predictions of each frequency component.

Keywords: aircraft noise, liners, acoustic impedance, nonlinearity, multiple tones.



1. INTRODUCTION

Aircraft noise is a problem all over the world, causing community annoyance and restricting commercial flight operations at many airports. As air travel is becoming more popular, noise exposition near airports is increasing significantly.

Blade design parameters of aircraft fan such as sweep, thickness, blade count, blade shape and fan diameter influence the amplitude of fan noise [1]. The high amplitude multiple tones generated have levels well above 130 dB, and propagate non-linearly through the nacelle duct.

Acoustic liners are acoustic treatment materials usually made of aluminum, composite, or titanium. They act as Helmholtz resonators, and are typically tuned to give maximum absorption for a given source spectrum and for a specific flight condition. Liners are exposed to several conditions and the only parameter that summarizes the liner acoustical performance is the acoustic impedance in terms of resistance and reactance components.

High level multiple tone excitation, using two even harmonically related tones, were measured on a liner surface and the results were compared with a DNS model for a punched aluminum SDOF liner[2]. The model is 2D axi-symmetric and it solves the Navier-Stokes Equations using Finite Element Method from a commercial code using COMSOL and MATLAB integration. The experimental and numerical results are in good agreement considering the geometry uncertainties, and it permit impedance predictions using multiple tone excitation, that differ from pure tone excitation models. A new semi-empirical model for high level multiple tone excitation on liner surface is proposed and discussed, showing reasonable understanding about energy transference between tone components.

2. LINER IMPEDANCE

Impedance is defined for a time harmonic sound field as the ratio of complex acoustic pressure amplitude, $\tilde{p}(\omega)$, and complex acoustic velocity amplitude in a specific direction, $\tilde{u}(\omega)$, at a certain point in space. The symbol $\tilde{Z}$ means that it is a complex number. The general expression for

the acoustic impedance is given by,

$$\tilde{Z}(\omega) = Re(\tilde{Z}) + Im(\tilde{Z}) = R + iX = \frac{\tilde{p}(\omega)}{\tilde{u}(\omega)}, \quad (1)$$

The liner impedance is the value of this parameter calculated on the liner surface, where the velocity, $\tilde{u}$, is in the normal direction in respect to the surface, and it is calculated using the spatially averaged normal velocity at the liner surface. The resistance, R , is related to the acoustic losses of the liner, and the reactance, X , is related to the stiffness, which is non dissipative. When the reactance is zero, the liner is in resonance and the maximum absorption is experienced in the absence of grazing flow. Usually, the absolute values of impedance are normalized by the characteristic impedance of the air, $Z_0 = \rho c$ for a reference temperature and atmospheric pressure. The normalized impedance is described as,

$$\zeta = \frac{\tilde{Z}(\omega)}{Z_0} = \theta + i\chi = \frac{R}{Z_0} + i\frac{X}{Z_0}, \quad (2)$$

where θ is the real part of the normalized impedance, called specific resistance, and χ is the imaginary part, called specific reactance [3].

Liner impedance is determined by liner features, such as: orifice diameter, plate thickness, porosity, and the resonator cavity depth. Also signal characteristics change liner impedance, such as: multiple tones, positions of tones in the frequency domain (such as near each other on the frequency domain, or harmonically related), relative phase of each tone component, SPL of each tone component, and acoustic velocity produced by each tone component. No single parameter which controls the impedance for an arbitrary combination of tones has been proposed to date[4].

Several semi-empirical, numerical and experimental models have been developed to predict liner impedance. Frequency domain semi-empirical methods include these of Kooi & Sarin[5], Maa[6], Hersh [7], Elnady[8] and Murray[9]. Rienstra & Singh[10] developed an analytical frequency domain impedance model for a Helmholtz resonator that shows good agreement with the Montsinger & Kraft model[11]. The Rice [12] and the Cummings [13] time domain models were developed in order to predict the liner impedance

in the presence of any signal that potentially can cause non-linear interaction between frequency components. Other authors compared eduction and insitu techniques [14] and also the grazing flow influence on the liner impedance results [15][16].

For most practical geometries, $\lambda a \ll 1$, where λ is the characteristic wavelength and a is a characteristic dimension of the perforate, so this problem can be treated essentially as being one dimensional, by considering plane waves inside the resonator.

Although, micro fluid interaction is important for the case of sound incident upon a perforated sheet, it is computationally expensive if a 3D numerical simulation is used to solve the problem in time domain [17]. Analytical solutions are impractical for most cases because the time domain differential equations have non-linear terms and the nature of the incident pressure is complex. So, numerical solutions are needed to evaluate the influence of non-linear terms [13].

2.1 Proposed 1D semi-empirical model for pure tone excitation

Based on the Cummings model and Boden's end correction, it is hypothesized that the discharge coefficient C_D was mainly responsible for the lack of agreement between the existing models, because the C_D is frequency dependent, proportional to the end correction, and consequently the velocity too, as the SPL increases. An improved end correction ϵ_r is proposed [2] by using the expression,

$$\epsilon_r = \frac{C_D}{C_{Df0}} \left(1 + \frac{(\frac{\bar{v}_0}{2fd})^{1.2}}{24} \right)^{-1}, \quad (3)$$

where f is the frequency, d is the diameter of the perforated sheet hole, $\bar{v}_0$ is the averaged velocity at the liner surface, C_{Df0} is the discharge coefficient at the resonant frequency. Usually this value is assumed to be 0.76 and allows the calculation of the POA_{eff} if a range of resistances is available at different SPL. After defining the POA_{eff} a frequency dependent discharge coefficient can be calculated, based on the non-linearity of the sample [18].

The proposed model for a pure tone excitation by

using a frequency dependent discharge coefficient can be obtained by the equation,

$$\begin{aligned} & \rho L \frac{d^2 x_0}{dt^2} + \\ & + \left[\rho |v_0| + Q + \frac{32\mu l}{C_D d^2} + \rho \sqrt{8v\omega} (1 + l/d) \right] \frac{dx_0}{dt} + \\ & + \frac{\rho c^2 S}{V} x_0 = P \sin(\omega t), \end{aligned} \quad (4)$$

where $L = l_0 + (l + l_0)\epsilon_r$ is the effective length for a frequency dependent end correction and $Q = 0.3M(3.15 \times 10^{(-2)}/POA + 1.77 * POA + 0.3)$. Observe that the velocity v_0 is the a solution of the equation. If the grazing flow is dominant in respect to the bias flow, the component Q is higher than the nonlinear term $\rho |v_0|$ which is vanished. So, the model is non-linear and it need to be iterated until the ϵ_r and the velocity both v_0 converge. The discharge coefficient is also frequency dependent, which means that each pure tone excitation considering $P = P \sin(2\pi ft)$ will be solved for f by using the correspondent value of the discharge coefficient for that frequency component $C_D(f)$. The jet length was defined by Cummings and Boden [4] as

$$L_1 = \frac{\int_0^{T/2} v_0 dt}{d}, \quad (5)$$

where half period corresponds to $\frac{T}{2} = \frac{1}{2f}$ if a single tone is considered. The Strouhal number can be written as

$$St = \bar{v}_0 / 2\pi f d. \quad (6)$$

So (6) can be included into (3) obtaining

$$\epsilon_r = \frac{C_D}{C_{Df0}} \left(1 + \frac{(\frac{\pi}{St})^{1.2}}{24} \right)^{-1}. \quad (7)$$

Observe that the plate effective thickness, and consequently the end correction, is linearly proportional to the $1/St$.

3. COMSOL DNS 2D AXI-SYMMETRIC FEM MODEL

The 2D axisymmetric numerical model proposed is based on the fundamental equations of Navier Stokes [19]. The geometry is based on a single hole resonator in an virtual impedance tube. The real experimental and virtual impedance tube lengths are 208.2 mm. However, the diameter

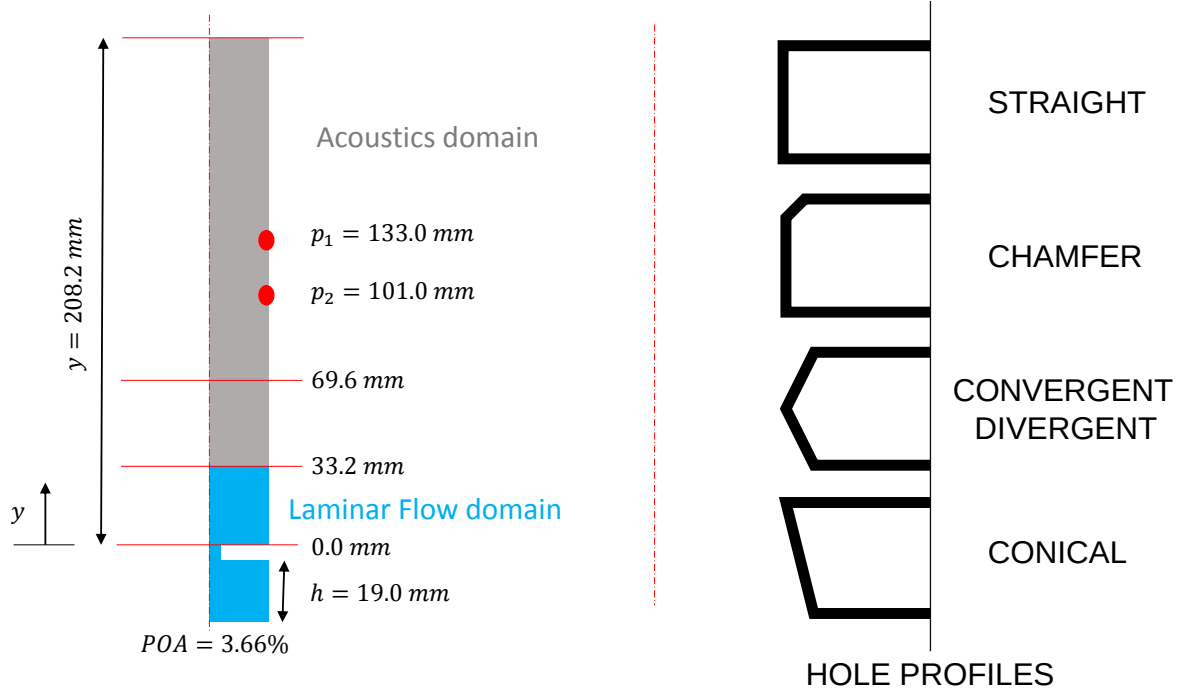


Figura 1: 2D liner multiphysics model geometry and hole profiles.

of the model is adjusted in order to maintain the same effective POA as the experimental sample measured.

Four different geometries were simulated, in order to evaluate the impact of the resonator hole shape. A straight hole model was compared to models with three differing hole profiles (small chamfer, convergent-divergent and conical). The hole profiles are shown on the right hand side of Figure 1. The chamfer model contains a chamfer of 1/10th the facing sheet thickness with the same minimum diameter as the straight model hole. The convergent divergent model has the minimum diameter equals the straight model and 45° angle of aperture, the same as the conical model. The straight hole model geometry is shown on the left hand side of Figure 1.

The modeled hole diameter, d , facing sheet thickness, l , and cavity depth, h , are the nominal values of the punched aluminum SDOF liner sample. The virtual pressure taps positions, p_1 and p_2 , are located at the same positions as in the portable impedance meter[2] in order to replicate the ASTM E1050 impedance calculation method[20]. The solid red lines show the positions of the control planes. The dash-dotted line represents the axisymmetric axis.

The POA used in the model is based on the calculated POA_{eff} [18] which was obtained by using pure tone experimental data and Two-Microphone Method[21] on the punched aluminum liner in the sample holder. The computational domain was reduced by defining a circular hole shape and cylindrical domain. This reduced the computational time by around one order of magnitude.

Other authors have studied Helmholtz resonators also exploring the effect of the hole shape on liner impedance [22][23]. The COMSOL model is an idealized version of the actual liner geometry. Figure 2 contains four pictures of cross-sections of punched aluminum SDOF liners. The liners show the facing sheet on the left and the honeycomb on the right of each picture.

It is clear that the holes are not perfectly straight. Some holes look conical and some seem to have small chamfers or rounded edges. Also, the picture on the right hand side of the Figure 2 shows an alternative punched aluminum liner, not used in the experiments, which has a honeycomb wall in front of a facing sheet hole. This is expected to cause higher resistance due to a decrease of the POA and also by viscous dissipation at the honeycomb walls. Furthermore, alternative manufacturing procedures use adhesive which blocks

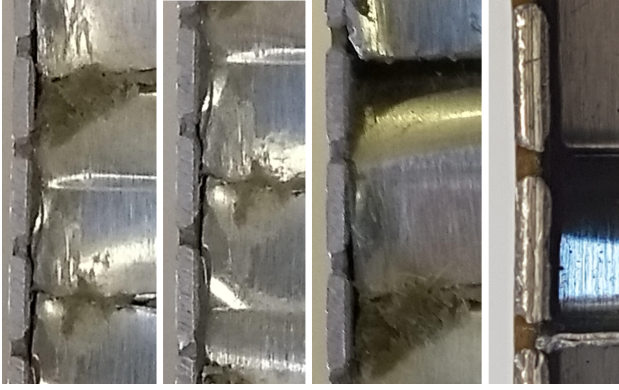


Figura 2: Hole shapes of a punched aluminum liner facing sheet cross section having $POA=5.18\%$, $d=0.99$ mm, $l=0.635$ mm.

some of the holes, thereby reducing the effective open area. So, it is possible to have two liners with the same POA, but with completely different effective percentage of open areas POA_{eff} . Hence, different geometries were modeled in the axi-symmetric model in order to address the effect of such features on the impedance.

3.1 Mesh design

The FEM unstructured mesh is composed of triangular elements in the Navier-Stokes domain (Laminar Flow consideration), in blue on the Figure 1, which are smaller than $24\Delta x_{Stokes}$. A variable element Growth Ratio (GR), which corresponds to the multiplication factor of the element size in respect to the adjacent element, was optimized. The mesh design for a pure tone at 2100 Hz is shown in Figure 3 as an example for a GR=1.5 in the laminar flow domain. Further refinement by using GR=1.05 was used in order to obtain an improved representation of the vortex near the liner hole. At the rectangular and triangular domains in the hole region, which in fact are axisymmetric volumes, the maximum element size set is Δx_{Stokes} . On the other adjacent domains the element size grows until $24\Delta x_{Stokes}$ by using the GR as a scale factor until it reaches the acoustics domain.

Rectangular elements are used on the acoustics domain, where 20 elements per wavelength are used for pure tone and multiple tone excitation. This means that the mesh is frequency dependent and its resolution is determined by the highest tone frequency component of the excitation signal.

Figure 4 shows a comparison of four different

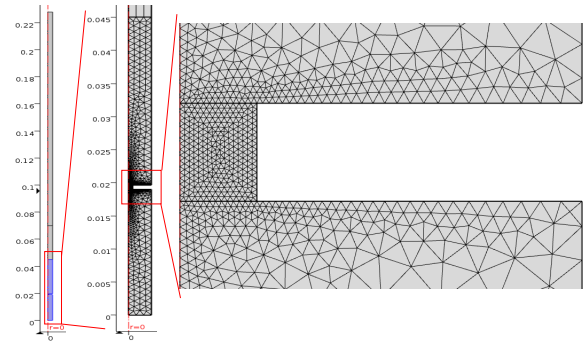


Figura 3: Single resonator 2D multiphysics mesh design. Straight model for a single tone excitation at 2100 Hz.

meshes created to perform simulations at 2100 Hz for pure tone excitation. All of the geometries used the GR=1.05 criteria and the dimensions are in millimeters.

Impedance results for pure tones at 155 dB were used to evaluate the mesh convergence. A range of 21 frequencies from 600-5100Hz were calculated one at a time. The simulations were performed on a Dell Precision M4700, 2.80 GHz i7 CPU with 8GB of RAM. The most expensive calculation is in the situation with highest non-linearity and most refined mesh. The time taken to perform the straight hole model calculation for each GR was:

- GR=1.05 -> total solution time = 9h27min,
- GR=1.10 -> total solution time = 5h56min,
- GR=1.30 -> total solution time = 3h55min,
- GR=1.70 -> total solution time = 2h44min.

3.2 Model validation for pure tone excitation

Several analysis were performed in order to validate the numerical models against experimental data for pure tone (PT) excitations. Different hole geometries (straight hole, 1 side chamfer, convergent-divergent, and conical) were included in the study. The COMSOL numerical predictions were evaluated by comparing them to the punched aluminum SDOF liner in-tube TMM measurements ($POA_{eff} = 3.5\%$). The iterative process was used to match the experimental pressure measured at the liner surface with the spatially averaged pressure calculated by the COMSOL models. The comparisons were made using the liner surface impedance. The facing sheet mass reactance

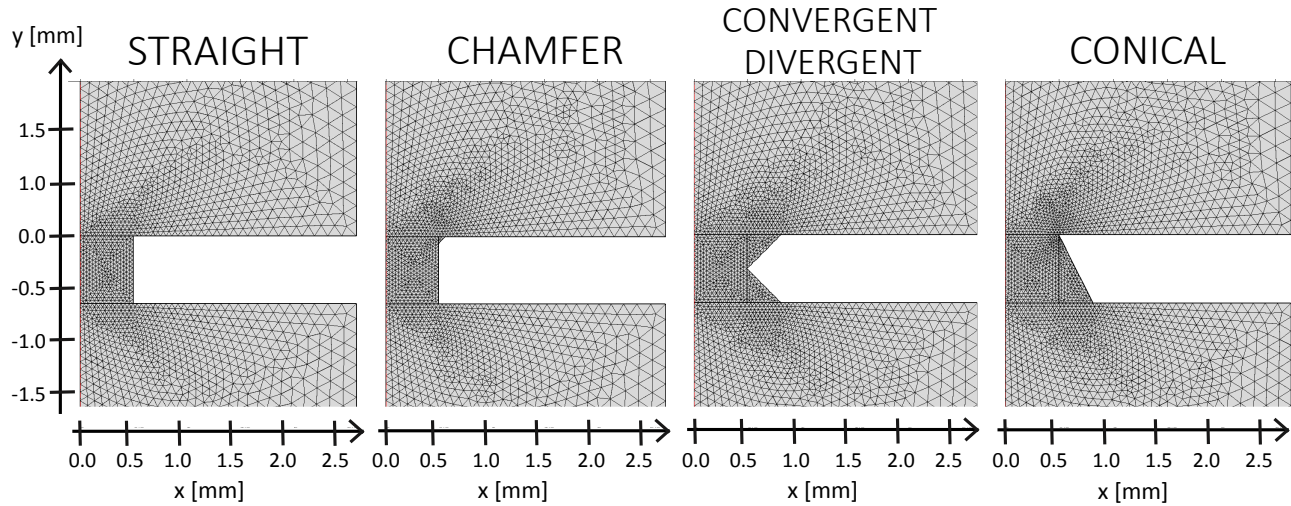


Figura 4: Mesh comparison near the hole for different hole geometries and GR=1.05 at 2100 Hz.

was also extracted by using $m_f = \chi + \cot(kh)$, where k is the wave number and h is the cavity depth.

Figure 5 shows a comparison of normalized resistances from the experimental data and from four different COMSOL numerical models implemented for pure tone (PT) excitations. Four different geometries were simulated in order to evaluate the effect of the hole shape on the impedance. Three different Over All Sound Pressure Levels (OASPLs) are presented, where the OASPL measured at the liner surface was compared to the numerical spatial average OASPL in each model and calibrated by using the algorithm[2] to ensure an error of less than 0.5 dB. The resistances of the experimental data for the punched aluminum SDOF sample in the sample holder were obtained by the TMM. The FFT impedance calculation method was used to extract the impedances, based on the spatially averaged pressure and velocity at the liner surface of the numerical models.

The COMSOL resistance predictions follow the trends of the experimental data. In general, the resistance non-linearity is underestimated at low SPL, and over-estimated at high SPL, for each of the modeled hole shapes. However, the COMSOL results clearly demonstrate the importance of hole shape.

At 130 dB OASPL the numerical models show relatively poor agreement with the experimental data. At 140 dB OASPL the straight hole model shows excellent agreement with experimental

data. At 150 dB OASPL the convergent-divergent model, shows good agreement with experimental data from 1.0 kHz until 2.0 kHz. In general, the model with a Chamfer on one side shows reasonable agreement with experimental data for different OASPLs.

The experimental resistances from 3.0 kHz to 5.0 kHz at 130 dB and 140 dB gives values that does not correspond to any of the models, which suggests that TMM is not providing reliable results. The relative increase in resistance at high frequencies reduces as SPL increases. This may suggest some gaps in the sample which close off acoustically as the resistance of the gap increases as observed by Murray[24].

Figure 6 shows the reactance predictions for four different hole geometries simulated in COMSOL for pure tones at 150 dB OASPL. The chamfered model shows excellent agreement with the experimental data up to 3.0 kHz. The straight model shows excellent agreement up to 4.0 kHz. The convergent-divergent and conical models again diverge from the experimental data.

Three main types of flows within and near the hole were observed in the numerical results. The first type occurs for $\omega_0 < 1$ and is characterized by a jet during both inflow and outflow phases of the cycle. The second type is characterized by a pulsation that causes strong vortex shedding for excitations near the resonant frequency $\omega_0 \approx 1$. The third type occurs at higher frequencies $\omega_0 > 1$ where the velocity magnitudes are smaller but

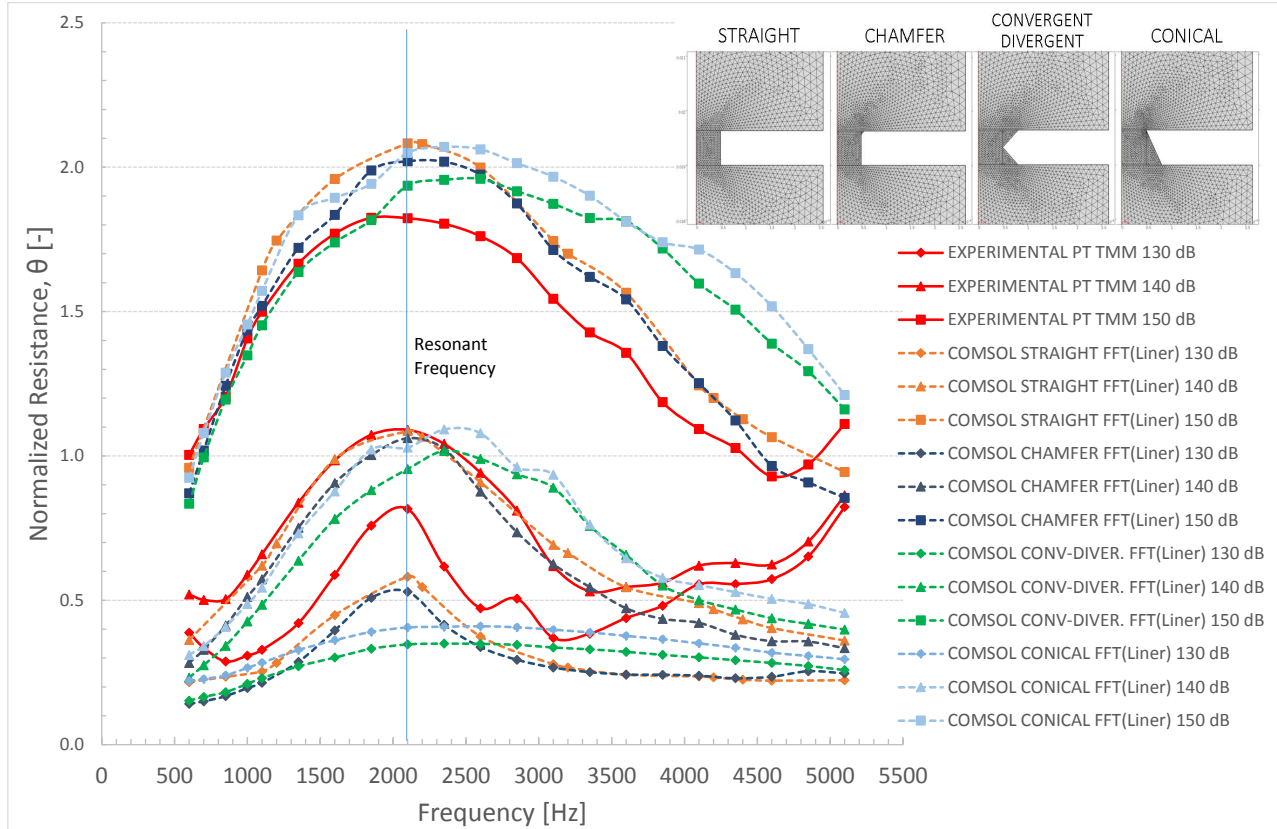


Figure 5: Normalized resistance comparison for different geometries of the COMSOL model against experimental data for pure tones at 130 dB, 140 dB and 150 dB OASPL.

interact more with the walls of the hole creating increased localized shear stresses.

When the pressure outside the resonator is higher than the pressure inside the cavity, the sound wave accelerates the fluid particles through the hole from outside into the liner cavity. If the geometry of the hole is symmetric, the jet changes direction during the other half cycle when the pressure difference reverses, showing the same behavior outside the resonator. The particle displacement in the hole also depends on this pressure difference, and on the excitation frequency.

A jet with high velocity in the core is observed when the excitation frequency approaches the resonant frequency of the resonator, $w \approx 1$, as shown in Figure 7. Five stream lines are used to identify the trajectory of the fluid particles on the entrance plane of the liner hole. The velocity profile on the hole creates a shear layer that extends beyond the hole edges and creates a vortex ring that is convected away from the resonator hole. The flow field near the resonator aperture contains

flow reversal during both the inflow and outflow periods, in order to maintain mass conservation. The conical model shows a narrow jet with more transverse velocity components, that cause larger vortices that are convected more efficiently than for the other hole shapes. The sharp edge model shows a *vena contracta* and higher vorticity magnitudes near the hole walls, suggesting that the shear stresses due to the boundary layer and flow reversal are stronger in that region.

4. PROPOSED MULTIPLE TONE IMPEDANCE CORRECTION

The numerical results for multiple tone excitation were used to correlate the resonant frequency of the liner ω_0 and the amplitude difference between tones, ΔSPL , with the resistance ratio between the multiple tone and pure tone excitations $\frac{\theta_{MT}}{\theta_{PT}}$. These normalized factors were chosen in order to clearly represent the relative differences between multiple tone and pure tone resistances. An expression that adjusts the impedance calculated using a pure tone prediction model was developed

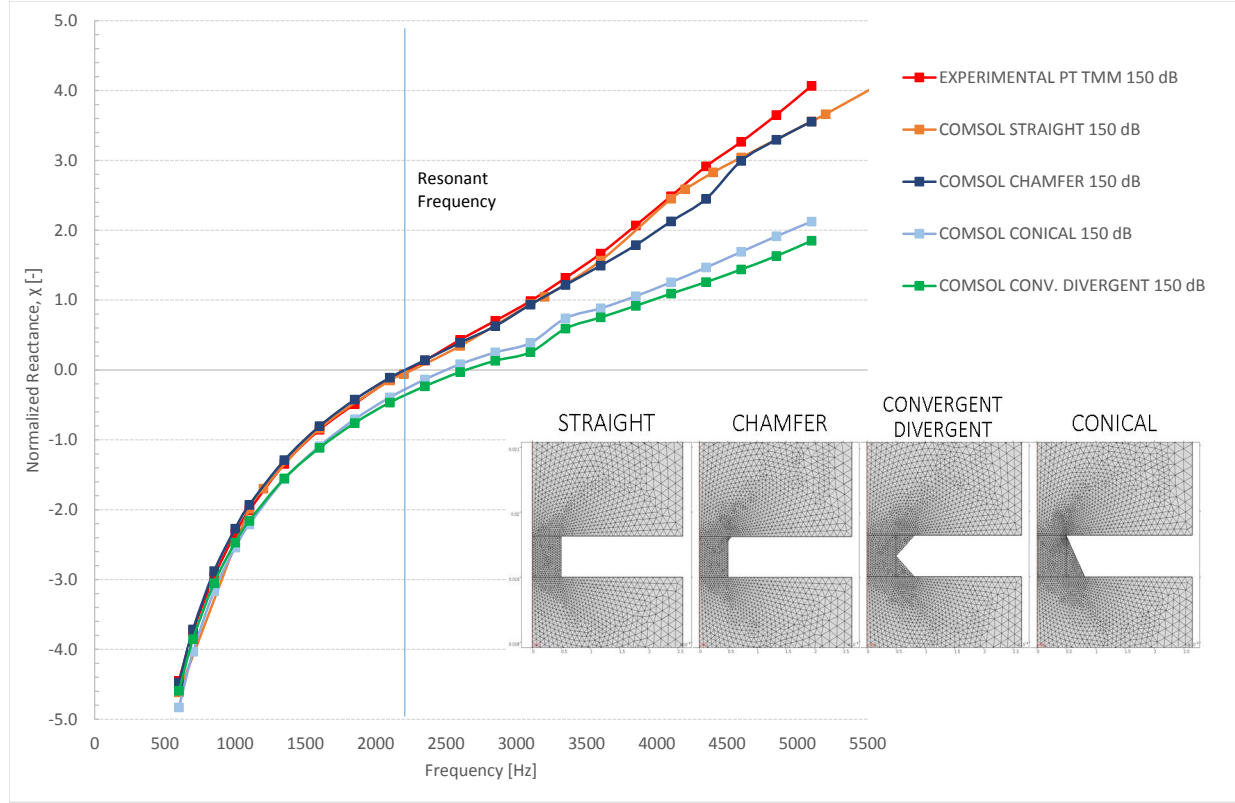


Figure 6: Normalized reactance comparison for different geometries of the COMSOL model against experimental data calculated by Two Microphone Method and measured on a punched aluminium SDOF liner, POA=3.5%, $d=0.990$ mm, $l=0.635$ mm, $h=19.1$ mm, for pure tones at 150dB OASPL.

in order to account for the presence of significant multiple tone excitation.

Figure 8 shows the resistance ratios between multiple tone and pure tone excitations at a given frequency relative to the normalized frequency $\omega_0 = 2200$ Hz, where ω_0 approximates the resonance frequency. It is recognized that the resonance frequency changes with SPL and tonal content, but it may be considered second order for the purposes of this exercise. A semi-empirical fit was applied in order to provide a correction to the pure tone impedance predicted by traditional liner impedance models. The fit is plotted in Figure 8 with the correspondent equations and correlation coefficients.

The resistance of a frequency component is strongly influenced by the pressure amplitude of other harmonically related tone. Considering the SPL difference between the fundamental and harmonic tones as $\Delta SPL = SPL_1 - SPL_2$, if this difference is higher than 10 dB $|\Delta SPL| > 10$, the nonlinear interaction between tones will be small, as the response of the liner at a given frequency is domi-

nated by the SPL at that frequency. Consequently, the resistance of the highest tone will be similar to the resistance of a pure tone excitation, so that $\theta_{MT}/\theta_{PT} \approx 1$. On the other hand, if the SPL difference is below 10dB ($|\Delta SPL| < 10$), then the non-linear interaction is strong and it will affect the impedance in both tones.

Considering a multiple tone excitation so that SPL_1 is the spatial average sound pressure level of the fundamental tone at the liner surface, and SPL_2 is the spatial average sound pressure level of the first even harmonic at the liner surface, it is possible to predict the resistance of each frequency component using the following proposed equation

$$\frac{\theta_{MT}}{\theta_{PT}} = \pm \frac{\Delta SPL - 10}{20} \ln(\omega_0) + C_1, \quad (8)$$

where the sign $\pm$ is positive for $\omega_0 < 1$, and negative for $\omega_0 > 1$. $\Delta SPL = SPL_1 - SPL_2$ when $\omega_0 < 1$, or $\Delta SPL = SPL_2 - SPL_1$ when $\omega_0 > 1$. The coefficient C_1 is approximately equal to 1.

Equation 8 can be written in terms of two fre-

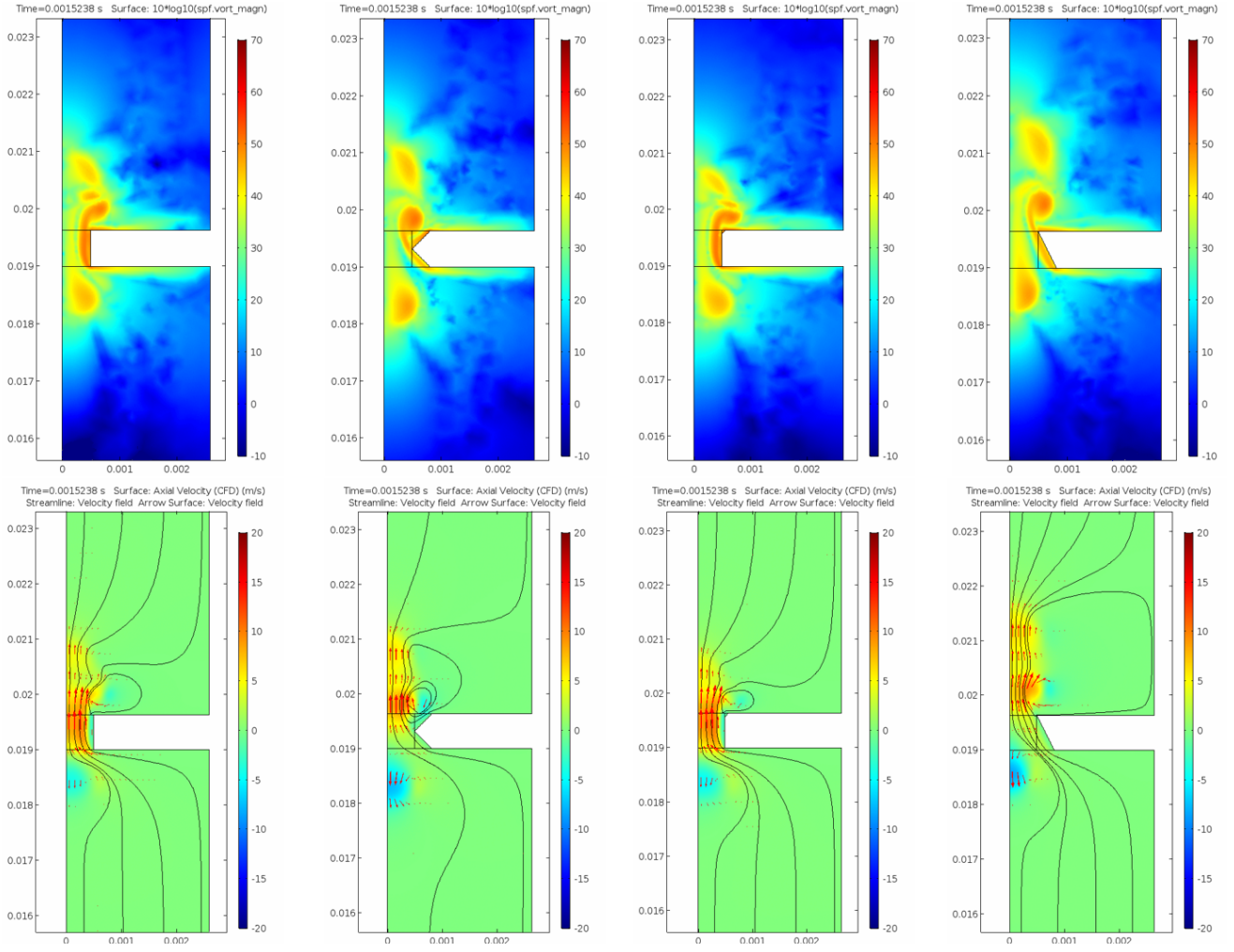


Figura 7: Vorticity (top) and velocity (bottom) comparisons among different model geometries for a pure tone at 2100 Hz and 130 dB OASPL. Arrows represent the velocity directions and relative magnitudes.

quency components with root mean square pressures P_1 and P_2 as

$$\begin{aligned} \frac{\theta_{MT}}{\theta_{PT}} &= \pm \frac{\Delta SPL - 10}{20} \ln(\omega_0) + C_1 \\ &= \pm \frac{20 \log\left(\frac{\langle P_1 \rangle}{\langle P_2 \rangle}\right) - 10}{20} \ln(\omega_0) + C_1 \\ \frac{\theta_{MT}}{\theta_{PT}} &= \pm \left(\log\left(\frac{\langle P_1 \rangle}{\langle P_2 \rangle}\right) - \frac{1}{2} \right) \ln(\omega_0) + C_1 \quad (9) \end{aligned}$$

The logarithmic regressions produce different slopes depending on the sound pressure level difference between frequency components of the signal. The curves are also frequency dependent. The resistance at the fundamental tone will be higher than the PT resistance for $\omega_0 < 1$ in a presence of an even harmonic with significant energy ($H > F$). On the other hand, the resistance at the fundamental tone will be lower than the PT resistance for $\omega_0 > 1$ when the harmonic has more

energy ($H > F$). The opposite behaviour occurs at the harmonic, where the resistance will be higher than the PT resistance when $\omega_0 > 1$ and the fundamental has more energy ($F > H$), and the resistance of the harmonic will be lower when $\omega_0 < 1$ using the same signal ($F > H$).

This fit provides a reasonable first order correction to the pure tone resistance predicted at either tone for a given combination of multiple tone amplitudes at high SPLs. The application of the proposed correction for experimental data collected for the punched aluminium SDOF liner by using the TMM, is in Figure 9. It is possible to observe quite good agreement that may be improved by using a quadratic fit rather than a linear fit to the measured data. Also, it is likely that the resonance frequency changes with the frequency content of the signal.

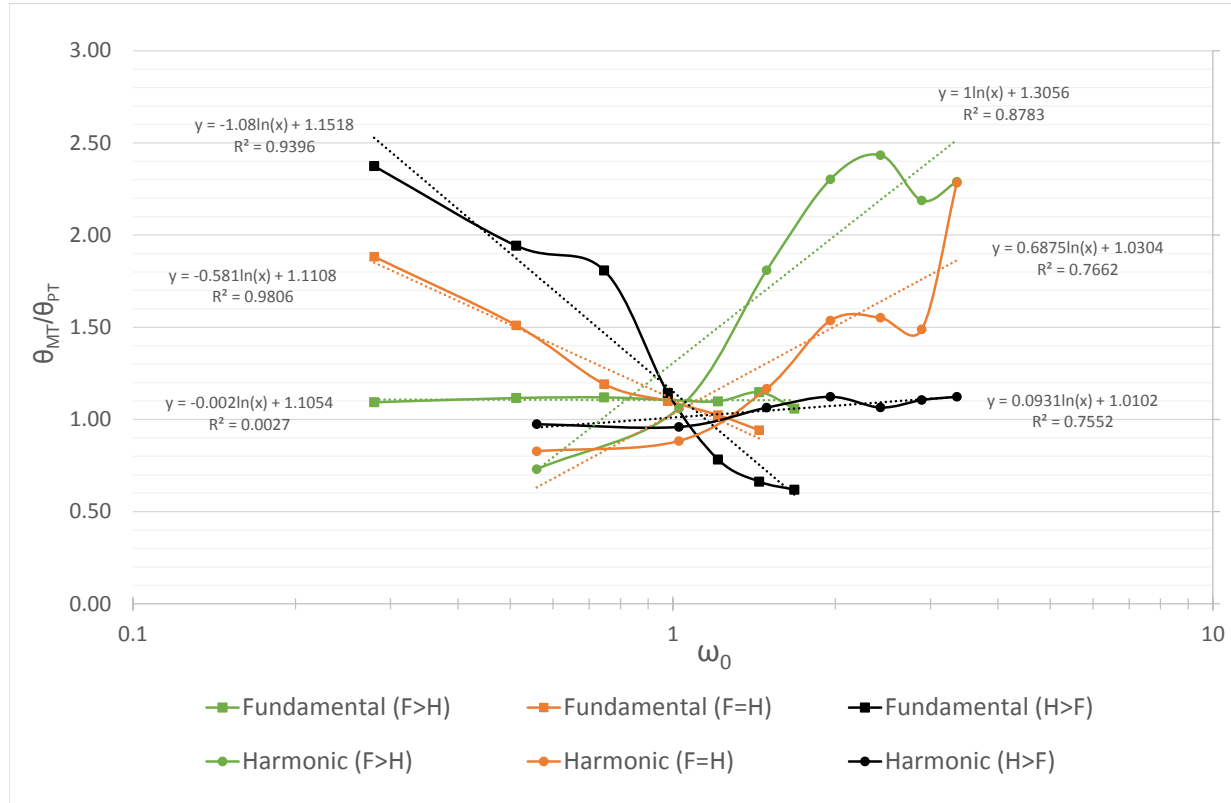


Figura 8: Resistance ratio of a COMSOL numerical result between MT and PT excitations for a liner with POA=3.5 %, $d=0.990$ mm, $l=0.635$ mm, $h=19.100$ mm. Green solid lines - F>H, Orange solid lines - F=H, Black solid lines - H>F. Squares are the fundamental resistance ratios, and circles are the harmonic resistance ratios. Dotted lines are the curves fitted to perform a semi-empirical analysis.

5. CONCLUSION

Acoustic liners are used in aircraft engine nacelles to attenuate fore and aft propagating noise in the inlet, bypass, and core ducts. Liner impedance models are needed to allow the definition of liner constructions which most closely match the optimum impedance for maximum attenuation in each duct. While previous experimental and theoretical work has led to the development of good quality impedance models for pure tone, and fully random sources, the non-linear response of these liners to combinations of tones containing high SPLs at the frequencies seen in aero engine ducts is not well understood.

This paper has reported on theoretical work to model the response of SDOF perforate liners to high amplitude multiple tones. Initial studies confirmed the highly non-linear response of these liners to pure tone signals. A time domain 1D semi-empirical impedance model was proposed.

A DNS time domain model was constructed using

a 2D axi-symmetric FEM in COMSOL, which was validated against the experimental data [2], along with other DNS data in the literature (Zhang & Bodony [17]) and a liner semi-empirical prediction model [9] for pure tone signals. The model was refined to run significantly faster than 3D DNS models, while retaining excellent accuracy and agreement with other prediction models proposed. It was also used to evaluate the significant influence of hole shape on non-linear impedance, showing the high sensitivity of the liner impedance to the hole profile.

The COMSOL model was used to visualize the complex flow patterns within perforate sheets for pure and multiple tone signals in order to evaluate the physical phenomena occurring during the sound propagation inside the liner orifice. The model predictions using DNS were again validated against experimental data using pure tones. The modelling showed that the large variation in acoustic resistance of each tone is a function of the relative tone amplitude and their frequencies.

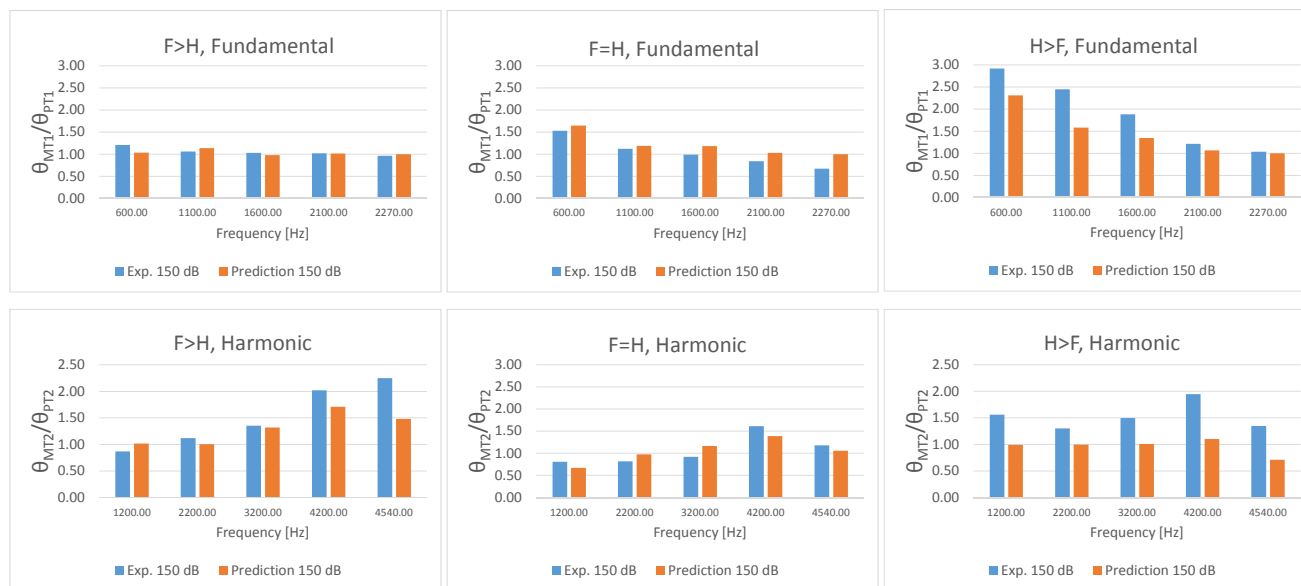


Figura 9: Validation of the proposed semi-empirical correction for multiple tone resistance based on pure tone experimental data, using three signals (F>H, F=H, H>F) with different ΔSPL at 150 dB OASPL.

Finally, the COMSOL results were used to develop a semi-empirical correction to tone resistance in order to account for the presence of multiple tones. The results of such new semi-empirical model were validated against the experimental data acquired on a portable impedance meter from 130 dB to 150 dB, showing good agreement for a first order approximation.

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