Numerical Vibroacoustic Analysis of a Fuselage Section: Low-Frequency Noise Reduction

Giosuè Carducci* and Salvatore Carducci †
University of Naples "Federico II", 80122 Naples, Italy

In this paper the application of a structural-acoustic analogy within the NX Nastran finite element (FE) program for the prediction of aircraft interior noise, at the low frequencies associated with propeller tonal loads, is presented. The reliability of the procedure is assessed through an analysis of some simple models involved in the fluid-structure coupling, and is compared to known theoretical results. Within the context of noise reduction techniques, it is proposed to investigate the effects of introducing dynamic vibrations absorbers (DVA) instrumented on the fuselage structure and additional airgap that separates the outer skin from the interior trim panel. Vibroacoustic (VA) FE modeling and a coupled frequency response analysis of the fuselage section is carried out to predict the sound pressure level (SPL) response for each position of passengers. Matlab scripts for semi-automatic generation of structural and the acoustic meshes and production of Nastran cards are developed. The propeller noise is introduced as external load on the structure. A Matlab program is developed in order to read data from a text file containing pressure loads on the external acoustic mesh. The output of this program is an interpolated pressure distribution on the assigned structural mesh.

Keywords: aircraft interior noise, finite element method, fluid structure interaction

I. Introduction

In recent years the stringent norms on noise emission levels and the customer demand in the comfort level in the aircraft have made the vibroacoustic (VA) behaviour into an important criterion in many design problems. Noise and vibration inside an aircraft cabin cause increasing risks in health and performance of flight and cabin crews besides the discomfort to the passengers. Many aircraft industries are therefore striving hard in achieving a higher comfortable level. Aircraft noise contains the following main components: engine noise, propeller noise, airframe noise (turbulent boundary layer) and structure borne noise [1]. Interior noise is combination of all mentioned components that, with various degrees, penetrate into the aircraft cabin. The aim of this work is to validate a numerical method based on finite elements (FE) analysis to obtain the prediction and the reduction of aircraft interior noise at the low frequencies associated with propeller component. Propeller noise is composed of tonal and broadband components. The tonal component contains basic frequency and harmonics. The basic frequency or blade-passage frequency (BPF) is the product of propeller rotation speed and number of propeller blades. The harmonic components are integral multiples of the basic one. Sound pressure level (SPL) outside the fuselage of regional turboprop aircraft is typically in the order of 130 dB at the BPF. In order to reduce the acoustic noise level inside the aircraft cabin some design variables have to be considered. In particular, the effects of dynamic vibrations absorber (DVA) and additional airgap, that separates the outer skin from the interior trim panel, are investigated. FE analysis are made by using NX Nastran as the solver, and Femap as the pre and post-processor. The paper is organized as follows: Section II presents the theoretical background. In section III acoustic performances of simple models are presented in order to demonstrate the application of NX Nastran software to the solution of fluid structure interaction (FSI) problems and to understand how to achieve the noise reduction. VA modeling and analysis of a fuselage section are presented in sections IV and V, respectively. The model will be first introduced and then a coupled frequency response analysis is carried on the fuselage section with inner cabin air to predict the SPL response for each position of passengers. Matlab scripts for semi-automatic generation of Nastran input files of structural and the acoustic FE models are developed. The propeller noise is introduced as external load on the structure. A Matlab program is developed in order to read data from a text file containing pressure loads computed by CIRA (Italian Aerospace Research Centre).

*MSc in Aerospace Engineering, Department of Industrial Engineering, g.carducci@studenti.unina.it
†MSc in Aerospace Engineering, Department of Industrial Engineering, sa.carducci@studenti.unina.it

II. Theoretical Background

A. Structural-Acoustic Analogy

It is possible to solve acoustic problems using structural code which already exists in Finite Element Method (FEM). The technique is based on a structural-acoustic analogy which relates structural displacement to acoustic pressure. Specific problems have been solved using this approach and the theoretical development has been well documented [3, 4]. In this paper the fundamental steps are included for the sake of clarity. The scalar acoustic wave equation in terms of the variation of pressure from the equilibrium pressure, in Cartesian coordinates is

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{1}$$

The equation governing the equilibrium of stresses in a material in a particular fixed direction (x, for example) is

$$\frac{\partial^2 \sigma_{xx}}{\partial x^2} + \frac{\partial^2 \tau_{xy}}{\partial y^2} + \frac{\partial^2 \tau xz}{\partial z^2} = \rho_s \frac{\partial^2 u_x}{\partial t^2} \tag{2}$$

Equations (1) and (2) are mathematically similar, and an "analogy" can be obtained if

$$\sigma_{xx} = \frac{\partial p}{\partial x}; \qquad \tau_{xy} = \frac{\partial p}{\partial y}; \qquad \tau_{xz} = \frac{\partial p}{\partial z}; \qquad \rho_s = \frac{1}{c^2}; \qquad u_x = p; \tag{3}$$

Where σ is the axial stress, τ is the shear stress, ρ_s is the structural mass density and u_x is the displacement in the x-direction. Thus, it is possible to solve acoustic problems using existing structural analysis codes based on the displacement formulation of the FEM, in particular NX Nastran.

B. Fluid Structure Interaction

A complete description of the FSI problem, in terms of FE models of the structure and the enclosed acoustic volume, is given by the following coupled equation of motion

$$\begin{bmatrix} \boldsymbol{M_s} & 0 \\ -\boldsymbol{A}^T & \boldsymbol{M_f} \end{bmatrix} \begin{Bmatrix} \ddot{\boldsymbol{u}}_s \\ \ddot{\boldsymbol{p}} \end{Bmatrix} + \begin{bmatrix} \boldsymbol{D_s} & 0 \\ 0 & \boldsymbol{D_f} \end{bmatrix} \begin{Bmatrix} \dot{\boldsymbol{u}}_s \\ \dot{\boldsymbol{p}} \end{Bmatrix} + \begin{bmatrix} \boldsymbol{K_s} & \boldsymbol{A} \\ 0 & \boldsymbol{K_f} \end{bmatrix} \begin{Bmatrix} \boldsymbol{u_s} \\ \boldsymbol{p} \end{Bmatrix} = \begin{Bmatrix} \boldsymbol{L_s}^e \\ \boldsymbol{L_f}^e \end{Bmatrix} \tag{4}$$

where the matrix, $\left[A\right]$, ensures the proper coupling between structural and acoustic models. Consider the boundary of the acoustic medium which is in contact with an elastic structure. The structure satisfies the dynamic equilibrium equations, and the fluid satisfies the wave equation. In order to establish the interaction between fluid and structure, two conditions are required at the boundaries. This conditions are fixed by the coupling terms $\{L_s\} = \left[A^T\right]\left\{\ddot{u}_s\right\}$ and $\{L_f\} = -\left[A\right]\left\{p\right\}$. $\{L_f\}$ is a function of the fluid pressure and $\{L_s\}$ is a function of the structural displacement. All other vectors on the right-hand side of the structural and acoustic equations are true load vectors. The general equation of motion, therefore, can be written as

$$\left[M_s\right]\left\{\ddot{u}_s\right\} + \left[D_s\right]\left\{\dot{u}_s\right\} + \left[K_s\right]\left\{u_s\right\} = \left\{L_s{}^e\right\} - \left[A\right]\left\{p\right\} \tag{5}$$

for the structural model and

$$\left[M_f\right]\left\{\ddot{p}\right\} + \left[D_f\right]\left\{\dot{p}\right\} + \left[K_f\right]\left\{p\right\} + \left[A_f\right]\left\{\ddot{u}_{f_n}\right\} = \left\{L_f{}^e\right\} + \left[A^T\right]\left\{\ddot{u}_s\right\} \tag{6}$$

for the acoustic model. $\left\{u_s\right\}$ is the structural displacement vector and $\left\{p\right\}$ is the vector of pressure values at the grid points. $\left[M\right]$, $\left[K\right]$ and $\left[D\right]$ are mass matrix, stiffness matrix and damping matrix, respectively.

III. Development and Verification

A. Free vibration analysis of a 3D plate/acoustic cavity system with damping interface

The first computed example is a $3D$ rectangular acoustic cavity of size $A = 0.6\ m$, $B = 0.5\ m$ and $C = 0.4\ m$, completely filled with air (mass density $\rho_F = 1\ \frac{kg}{m^3}$ and speed of sound $c = 340\ \frac{m}{s}$). One wall of the cavity is a flexible plate of thickness 6 mm clamped by its whole boundary and covered with a thin layer of absorbing material. The other walls are considered perfectly rigid. The mechanical parameters of the plate are: mass density $\rho_s = 7700\ \dfrac{kg}{m^3}$, Young's modulus $E = 1.44 \times 10^{11}\ Pa$ and Poisson's ratio $\nu = 0.35$. The absorbing material, which is considered mass-less in this example, has two parameters: $k^I = 5x10^6\ \frac{Pa}{m}$ and $d^I = 50\ \frac{Pa}{m}s$. These parameters are average frequency dependent impedance coefficients from experimental data corresponding to a thin layer of a typical insulating fabric (a Johns Manville glass wool of thickness 1 inch) in the frequency range 50–500 Hz. In order to check the quality of FE

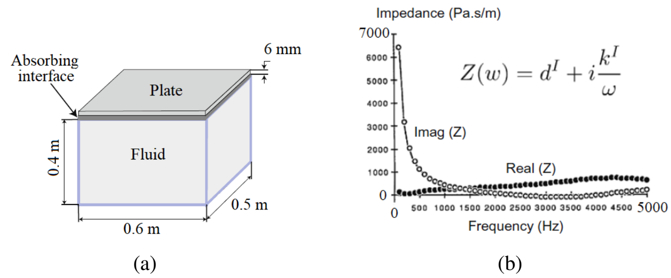


Fig. 1 Plate/acoustic cavity system: (a) geometric data and (b) acoustic impedance [5].

model the wavelength of waves in acoustic volume is calculated by applying Nyquist–Shannon sampling theorem. The maximum frequency of this study is $f_{max} = 500\ Hz$. Sampling frequency is therefore $f_s = 1000\ Hz$ and the shortest wavelength is $\lambda = 340\ mm$. At least 4 elements are required per wavelength, which means the minimum length for each element edge should be about 85 mm. The size elements in this study is less than or equal to 40 mm, the quality of the mesh is therefore acceptable.

Firstly, we present the results obtained for the 3D acoustic cavity with and without damping interface. Table 1 gives the first four eigenfrequencies with uniform meshes (CHEXA 3D element) and with increasing number of degrees of freedom. The first and second columns present the frequencies of the rigid cavity, without absorbing material, computed with NX Nastran code and third column presents those obtained with the exact solution [7]. The three other columns correspond to the first four eigenfrequencies of the damping cavity computed from NX Nastran code and compared to exact solution (last column) given in [6]. A good agreement between exact and computed values can be observed. A modal analysis (SOL 103) is performed for the undamped system to solve a linear eigenvalue problem. In NX Nastran one can use elements CAABSF to define properties of thin layer of a viscoelastic material. The acoustic absorber element CAABSF defines frequency-dependent impedance boundary conditions. This type of boundary is meaningful only in the frequency domain and it leads to a nonlinear eigenvalue problem [6]. Therefore, in order to determine the natural frequencies and mode shapes of the damped system a direct frequency response analysis (SOL 108) is performed.

Table 1 Natural frequencies (Hz) of a 3D acoustic rigid cavity with damping interface

Undamped			Damped		
2464 DOF	16926 DOF	Exact [7]	2464 DOF	16926 DOF	Exact [6]
283.85	283.46	f_{100} =283.33	276.2	275.9	f_{100} =274.85
340.82	340.22	f_{010} =340.00	330.9	329.7	f_{010} =329.46
426.75	425.44	f_{001} =425.00	404.0	403.0	f_{001} =402.00
443.55	442.83	f_{110} =442.58	429.5	427.7	f_{110} =427.71

Secondly, we consider the plate/acoustic cavity coupled system with damping interface. Before the coupled system is analyzed, it is important to understand the behaviour of the structural and fluid models separately. Note that the natural frequencies for both systems occur in the same range, which indicates that the structural modes will interact with the acoustic cavity resonances. The interface between the fluid and structure can be modeled with coincident grids in order to have a correspondence "one to one". The coupled natural frequencies must be obtained from an unsymmetrical matrix equation which requires a Complex Eigenvalue method even in the undamped case (SOL 110). The coupled FSI can be specified using the ACMODL bulk entry. Results are interpreted by examining the frequency shifts from the uncoupled system. Table 2 gives the eigenfrequencies in four cases: (i) 3D rigid acoustic cavity; (ii) clamped plate; (iii) plate/acoustic cavity coupled system without damping interface; and (iv) plate/acoustic cavity coupled system with damping interface. In the third and forth case, the results are in good agreement with exact solution obtained in [6].

Table 2 Computed frequencies (Hz) of the coupled system with mesh size element 40 mm (3472 DOF)

Mode	Undamped				Damped	
	F(i)	S(ii)	FSI(iii)	FSI[6]	FSI(iv)	FSI[6]
A	-	158.68	158.72	156.61	158.7	156.91
B	283.85	-	282.43	280.90	274.1	273.43
C	-	288.22	290.40	294.37	289.2	294.07
D	340.82	-	340.09	338.01	329.7	326.64
E	-	361.96	362.37	375.80	362.2	375.97
F	426.75	-	426.85	422.97	403.4	394.04
G	443.55	-	443.15	441.91	428.3	417.79

B. Analysis of 3D plate/acoustic cavity system with dynamic vibration absorbers

In this section the plate/acoustic cavity system, instrumented with DVA, is analyzed. DVA is simulated in NX Nastran with a spring-mass system. CONM2 element defines a concentrated mass at a grid point that is coincident but unconnected with a structural grid of the plate. This mass element is connected with the main structure by using a spring element CELAS2. Mass value is assumed as 10 % of the plate mass, 14.091 kg. In order to make easier to understand this study, firstly it is useful to analyze DVA effects on a flat rectangular plate. The device is installed on the central grid point of the plate. A direct frequency response analysis is performed to determine eigenfrequencies of the system. Stiffness of the scalar spring is computed in order to neutralize the plate vibration response when the excitation frequency is the first natural frequency of the plate (Table 2). Therefore, stiffness is $14.0 \times 10^5\ kg/s^2$. DVA also introduces two new resonating frequencies that are kept sufficiently away from the expected excitation frequency. Figure **??** shows the results for the plate with and without DVA. Secondly, a trade off-study of the DVA number and position has been performed for the complete system in order to determine the cavity interior noise reduction. These parameters have a significant influence on the solution. The FLSTCNT case control command is used to specify the reference pressure to compute the SPL in decibels. The number and position of devices placed on the plate depend on the modes that are representative of the structural dynamics at the frequency band investigated in the study. At different frequencies, the flat rectangular plate does appear to have a different number of lobe modes, de-phased of 90 deg, respect to each other. In this section two cases are presented: (a) at first resonant frequency of the plate f_{11}; (b) at second resonant frequency of the plate f_{12} (Table 2). In the first case 1 DVA is installed on the central grid point of the plate and it is designed to perform more efficiently at f_{11} because the structure does appear to have 1-lobe mode. Stiffness can be evaluated directly for assigned resonant frequency and it is the same of previous case. The average noise reduction in the cavity is 62 dB. In the second case 2 DVA are installed on the plate and they are designed to perform more efficiently at f_{12} because the structure does appear to have 2 coexistent 1-lobe modes, de-phased of 90 deg, respect to each other. Stiffness is $46.0 \times 10^5\ kg/s^2$ and the average noise reduction in the cavity is 34 dB.

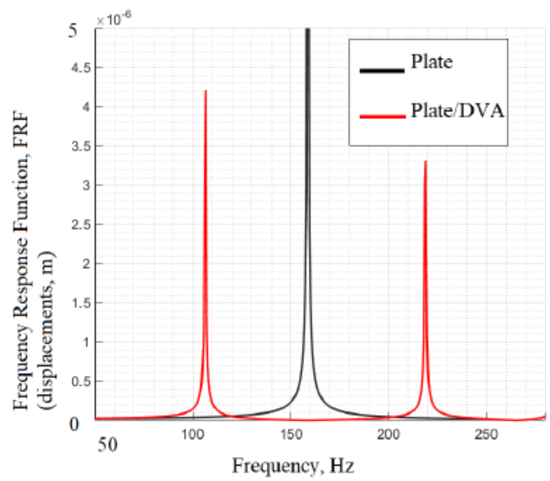


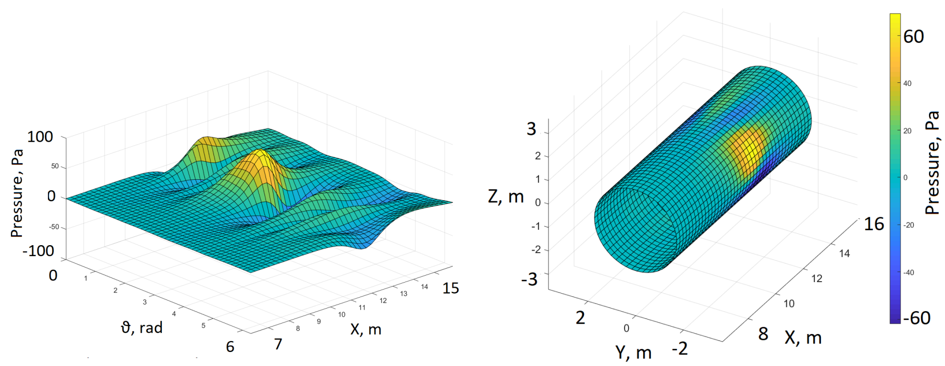
Fig. 2 Effect of DVA on the flat rectangular plate.

IV. Numerical Vibroacoustic Model

In the present study a numerical VA analysis of a fuselage section is presented. The model will be first introduced and then a frequency response analysis is carried out. After that, based on the results of previous studies some changes are introduced, according to the optimization process.

A. Propeller pressure loads

The response of the entire fuselage to the pressure field generated at BPF by the propeller is achieved through a direct frequency response analysis (SOL 108). The propeller noise is introduced as external load on the structure and a performance aerodynamic analysis of a propeller has been computed with a Blade Element Momentum Theory (BEMT) by CIRA. In this section a Matlab script for checking the input file of the propeller loads is presented. This Matlab program reads data from a text file containing pressure loads (real and imaginary) on each single node of a cylinder. The cylinder data, coordinates of the nodes and panel meshes, are also included in the files. Thus, reading the pressure



(a) Interpolated values of a function of two variables, $P = P(\theta, X)$.

(b) Pressure distribution on the cylindrical skin

Fig. 3 Pressure distribution, values computed by CIRA and re-organized by a Matlab program.

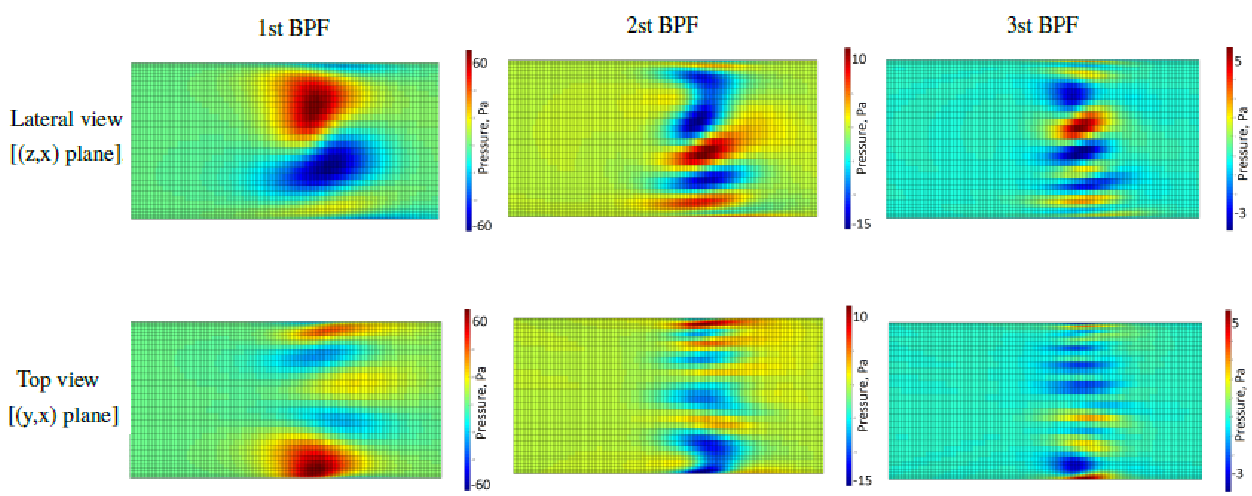


Fig. 4 Pressure real component distribution over fuselage structural mesh.

loads on the external acoustic mesh, the output of this program is an interpolated pressure distribution on the assigned structural mesh of the cylindrical skin. The interpolation is computed transforming Cartesian coordinates to cylindrical ones, therefore interpolated values of a function of two variables $P = P(\theta, X)$ are returned (refer to Fig.3). In this study a 8-blades propeller configuration is considered and the harmonics occur at 100, 200 and 300 Hz, respectively. Figure 4 shows the re-organized Real part of the acoustic pressure distribution deriving from propeller BPFs and this pressure loads distribution has to be used in the NX Nastran code formulation to compute the dynamic response of the structure.

B. Structural finite element generation

The structural FE model of the fuselage section, whose geometric features are reported in Table 3, is generated by a Matlab program. In order to generate the structural mesh some inputs have to be assigned. In Table 4 this inputs are shown. Firstly, this Matlab program generates the Nastran cards for the fuselage section. The low frequency cabin noise reduction in commercial aircraft is studied and it is showed that structural response and noise transmission at low frequencies are influenced by the stiffeners in different frequency bandwidths [8]. The following work is based on this consideration and no complicating effects are considered, such as anisotropy, variable thickness, initial stress or shear deformation. Figure 5 illustrates the FE structural mesh of the fuselage section with skin segment, stringers, frames, floor and stanchions. In the following model the material used is Aluminum (refer to Table 9). Reinforcement components are modelled through one-dimensional CBAR elements. Frame and stringer cross sections used are Z and C sections, respectively. Stringers are built along the $X - axis$, whereas frames are built along the circular direction. Both axial stringers and circumferential frames are assumed as equally spaced.

Table 3 Geometric data of the fuselage section

Geometric parameters	
Section radius	3450 *mm*
Section length	9450 *mm*
Skin thickness	2.2 *mm*
Number of stringers	36
Number of frames	22
Floor height	1135 *mm*

Table 4 Input data for the Matlab Script that generates the .bdf file of the fuselage section

Input Data			
lc	X-coordinates along the length of the cylinder [mm]	ns	Number of first node
cc	Polar meshing of the cylinder cross-section - angle [deg]	dn	Delta node number between two x-coordinates
dcy	Diameter of the cylinder [mm]	ne	Number of first quad element
tc	Thickness of the cylinder [mm]	de	Delta element number for quad between two bay to the next one
fp	Frame positioning along the x-coordinates (≥0 no frame; <0 yes frame)	sp	Stringer positioning at specific angles (≤0 no stringer; <0 yes stringer)
nt	Number of first bar element used for modelling frames		
dt	Delta element number for bar (frame model) from the first to the next one	dc	Delta element number for bar (stringer model) from the first to the next one
fl	Definition of the floor and stanchions position and their meshing	nc	Number of first bar element used for modelling stringers

Finally, the external skin and floor are defined as quadrilateral plate element CQUAD4 with a thickness of 2.2 *mm* and 30 *mm*, respectively. Calling the Matlab function of the previous section, the re-organized pressure loads distribution on the structural mesh is obtained. Reading the input pressure $Real$ and $Imag$ and computing areas for each structural grid point, the local forces are computed. The forces can be defined writing the input cards for the dynamic loads. The output of this program is a .bdf file which re-organizes the data to be used by the Nastran program. Thus to check the quality of FE models, one can calculate the wavelength of bending waves in plates and compare it with the element size. Figures in section IV.B show that the pressure loads are more uniformly distributed along the fuselage and higher especially at 100 Hz. Therefore the investigated frequency of this study is 100 Hz. The highest frequency leads to the shortest wavelength. Applying Nyquist–Shannon theorem, the wavelength is 326 *mm*. At least 4 elements are required per wavelength, which means the maximal length for each edge should be about 82 *mm* and most elements in this model satisfy this requirement.

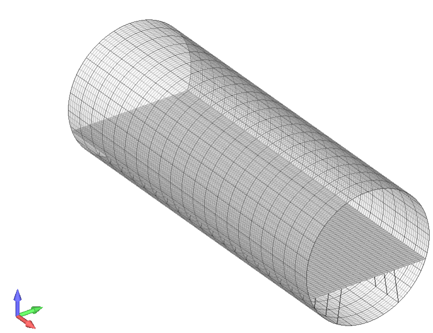
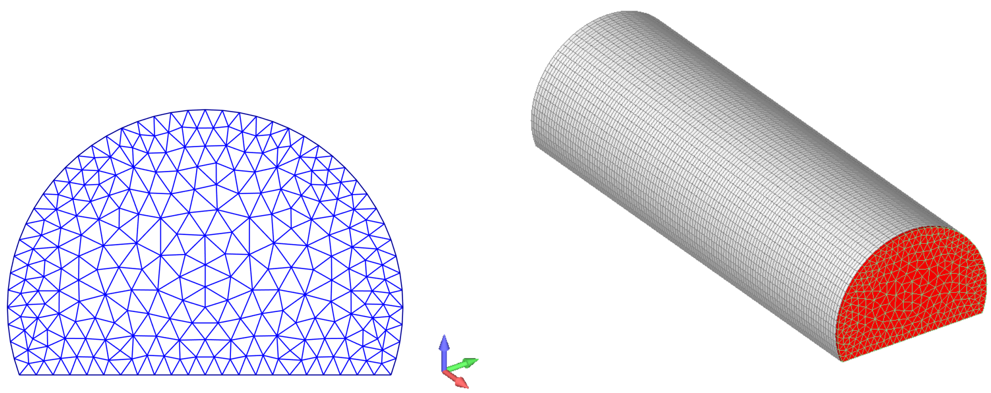


Fig. 5 Fuselage section FE model.

C. Acoustic finite element generation

In this study we mainly focus on the passenger cabin, which is located above the floor. The FE model of the fluid body is generated by a Matlab program. This program generates the NX Nastran cards for a cylinder volume. It uses a group of routines for 2D meshing. Firstly, a 2D unstructured triangular mesh is generated based on a piecewise-linear geometry input, which is linked to the structural mesh. Thus starting from the structural mesh, the internal acoustic mesh is defined on a triangular basis (refer to Fig. 6a). Accordingly, the basic assumption is the correspondence "one to one" between internal acoustic and structural mesh of the cylindrical skin. The interior fluid volume is modelled in Nastran by a five-sided solid element with six grid points, CPENTA and the material properties are defined using MAT10 bulk entries. For fluid elements are used the material properties of air. The wavelength is far larger than the plate dimensions, the quality of the mesh is therefore acceptable. This fluid model is only one section of the whole cabin, this means when the sound wave reaches the borders of the fluid model, it should not be reflected by the boundary but propagate forward. By using the absorbing boundary conditions, two layers of CAABSF elements are created on the selected surfaces (red selection in Fig. 6b) to eliminate the unwanted reflections. In this way the sound propagation of a full-sized cabin is simulated.



(a) (b)

Fig. 6 (a) 2D triangular mesh based on the structural mesh. (b) FEM Fluid cabin model, pre-processor FEMAP software.

D. Vibroacoustic FE model

The fuselage section is built and the inner air cavity of the fuselage is also modelled, therefore the data of this two separate files can be "easily" joined to create the VA model of the fuselage. The number of DOF is 108020. CPENTA elements, namely fluid elements contribute about 30% DOF of the whole model. And the remaining DOF mostly belong to the shell element CQUAD.

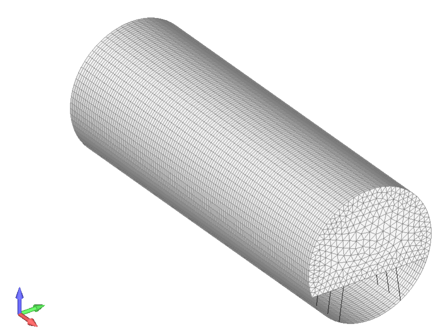
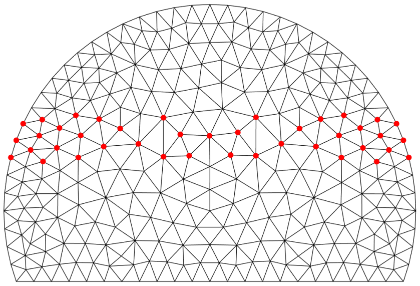


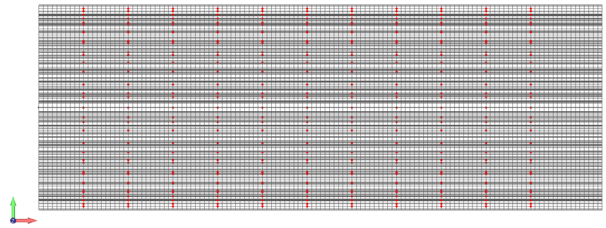
Fig. 7 FE VA fuselage section model.

E. Loads and measuring points

In order to perform the frequency response analysis a dynamic load is needed. To simulate the actual loading condition, a harmonic forces distribution is computed, on the exterior of the cylindrical skin. This forces are computed by multiplying the nodal pressure by nodal area for each structural grid point of the cylindrical skin. On the other side, the measuring points are at the locations where passengers are likely to perceive it. The allocation of the ear positions of passengers are illustrated in Fig. 8a. There is a problem for picking these points because the fluid model is not specially meshed for the ear position, it is impossible to locate the desired place precisely. To solve this problem, a short Matlab program is applied to search for the nearest node from the FE model.



(a)



(b)

Fig. 8 (a) Sketch of ear positions in the cabin. Chosen nodes for ear position: (b) front view; (c) top view.

V. Results and optimization

After the calculation is done, Nastran will output the results to a .pch file, from which it can be read directly by Matlab. The results from the original model are presented in the following sections and provide a starting point for further optimization.

A. Tonal components FEM analyses

In this section a coupled frequency response analysis is carried on the fuselage section with inner cabin air to predict the SPL response. The results, due to the first engine-propeller tonal loads (100 Hz), are presented. It is feasible to analyze the response for each ear position (1.2 m from the cabin floor). For simplicity, the global results which are obtained by averaging all the local results for each section are mainly concerned. This averaged SPL values are listed in terms of dB/dBA in Table 5. In this Table, the maximum averaged SPL values can be observed, which are near the engine position. The baseline configuration provides an averaged SPL of 90.85 dBA and a maximum SPL of 99.8 dBA. Figure 10 shows the pressure map for some sections. In this case there is not any experimental and analytical comparison because the parameters of the considered model have been selected in a generalized way.

Table 5 Averaged SPL in dB/dBA results of the baseline configuration, for 1st BPF

Averaged SPL for each section		
Section x (m)	Averaged SPL (dB)	Averaged SPL (dBA)
0.75	101.9	82.8
1.50	102.6	83.5
2.25	108.4	89.3
3.00	111.5	92.4
3.75	111.7	92.6
4.50	110.4	91.3
5.25	111.1	92.0
6.00	112.3	93.2
6.75	112.2	93.1
7.50	109.4	90.3
8.25	102.9	83.8

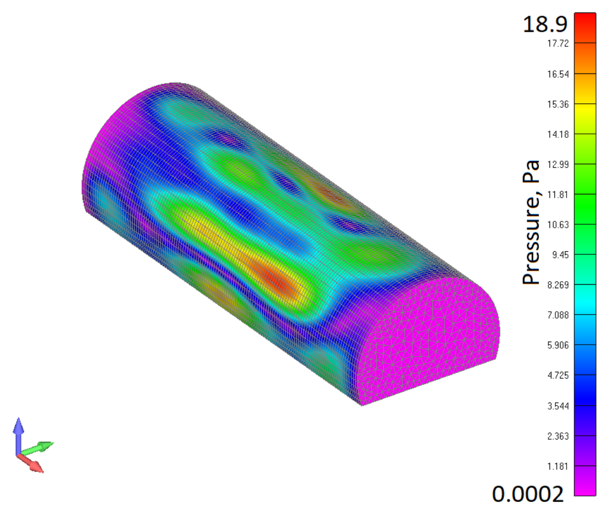
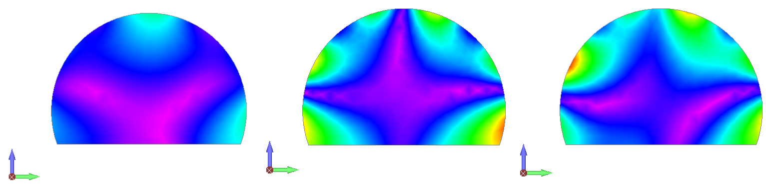


Fig. 9 SPL inside cabin, 1st BPF (Max 99.8 dBA).

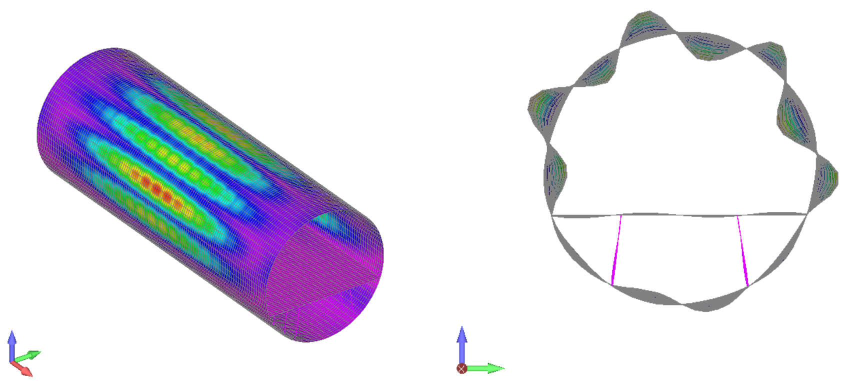


(a) (b) (c)

Fig. 10 Pressure map, 1st BPF, section: (a) x = 1.5 m (Max 83.5 dBA); (b) x = 4.5 m (Max 91.3 dBA); (c) x = 6 m (Max 93.2 dBA).

B. Passive technologies models at cabin level: DVA

DVA will be designed to have a peak frequency that is related to the engine-propeller tonal loads. A complete parametrization of number and position of DVA is complex. The optimization of the DVA will address their number and positioning. The first BPF can be drastically reduced by the introduction of DVA. Such devices were placed within the fuselage at the mode that is representative of the structural dynamics at the frequency band investigated in this study (refer to Fig.11). At that frequency the structure does appear to have 2 coexistent 4-lobe modes, de-phased of 90 deg, respect to each other. Then, a trade off-study of the DVA position and number has been performed. In the best configuration each section, near the propeller plane, is instrumented with 8 DVA, for each frame up to a total of 80 DVA. Table shows the parameters used for the spring and concentrated mass elements. The response is computed once again and the obtained results are compared with the baseline configuration. Using tuned mass dampers in these locations the SPL is 8 dBA reduction, as shown in Table .



(a) Dimetric view. (b) Front view.

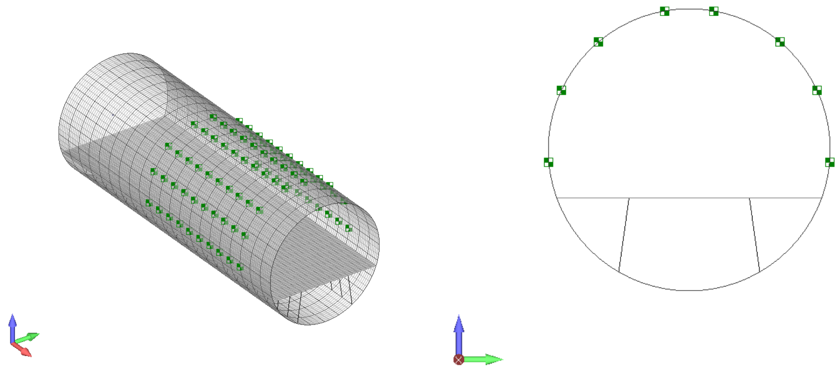
Fig. 11 Displacement contours of the fuselage section, natural mode at Freq. 103.01 Hz.

Table 6 SPL computed on the nodes plane at passenger seated ear height (1.2m from the cabin floor)

n. of DVA	added mass [kg]	SPL average
0	0.0	90.80 dBA
48	28.61	82.72 dBA
64	38.14	82.67 dBA
80	47.68	**82.58 dBA**
112	66.75	83.25 dBA
136	81.06	83.99 dBA
168	100.12	85.6 dBA

Table 7 Parameters for DVA

Mass [kg]	0.596
Stiffness K_x [Kg/s^2]	1.18E+05
Stiffness K_y [Kg/s^2]	1.18E+05
Natural frequency [Hz]	100
n.	176

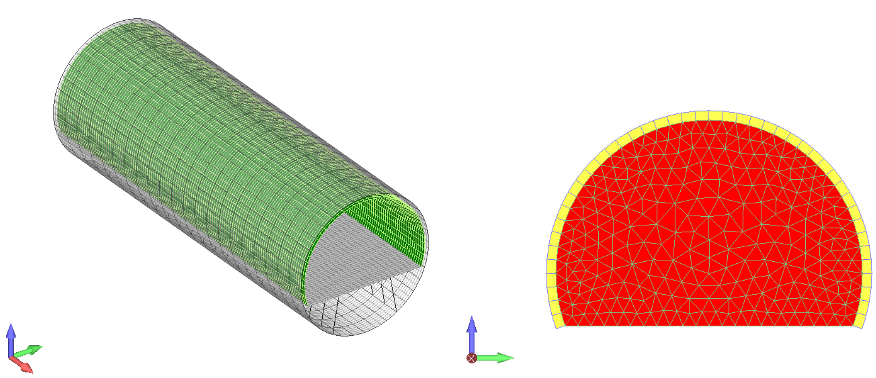


(a) Dimetric view. (b) Front view.

Fig. 12 Best configuration with 8 DVA for frame for 10 cabin fuselage sections (80 DVA).

C. Noise reduction with doublewall system

The term "doublewall system" is used to identify the system composed of the external skin, acoustic insulation and interior trim. The typical fuselage doublewall system provides numerous paths for acoustic and vibration transmission. A detailed analysis of all possible paths is outside the scope of the present study. Thus a simplified model is proposed for the doublewall system. The model will include the fuselage skin, stringer, frames, interior trim and airgap, but will exclude the complicating effects of windows and doors, like in the previous sections. There are two transmission paths, one being structural via the frames and the other acoustical through the cavity between the skin and trim panels. The connections between the airframe and the trim were assumed as rigid links for sake of simplicity, using rigid bar elements. This implies a conservative solution is obtained in term of SPL, because of the high transmissibility of the structural path compared to real conditions [10]. The fuselage FE model has been completed by modelling also the trim panel with full 2D elements. Firstly, the Matlab programs are modified in order to introduce the trim panel for the structural system and the airgap for the acoustic system. The interior trim panel is separated from the outer skin by a layer of airgap, whose the thickness is 10 cm. The acoustic models generated to fill the fuselage interior cavities are reported in Fig. 13b. The structural components of the aircraft require high stiffness with minimum weight in order to maximize the aircraft's performance and ensure the safety of the passengers. To assess the VA performance of fuselage doublewall system, different materials like aluminum and fiber metal laminate (FML, Ref. [11]) are chosen. The material properties are tabulated in Table 9 and the thickness is 1,3 *mm*. Table 8 shows averaged and maximum SPL in dBA scale on a surface plane at passenger seated ear height (1.2 *m*) for different materials, the values have been computed considering the 1st BPF. From the previous average SPL results, a difference of 4 dBA and 5 dBA is seen in Aluminum and FML cases compared to baseline configuration, respectively.



(a) (b)

Fig. 13 (a) Fuselage section FE model with trim panel. (b) Acoustic cavities: cabin (red) and airgap (yellow).

Table 8 SPL computed on the nodes plane at passenger seated ear height for different materials

	Baseline configuration	Trim panel configuration	
	-	Al	FML
SPL Max	99.8 dBA	95.3 dBA	98.4 dBA
SPL average	90.8 dBA	86.6 dBA	**85.3 dBA**

Table 9 Material properties for trim panels

Material	E (Gpa)	ν	ρ (kg/m^3)
Al	71	0.33	2795
FML	65	0.6	2470

VI. Conclusions

A NX Nastran FE application has been presented which can predict the interior noise of an aircraft fuselage. The principal theoretical steps have been presented and comparisons between the numerical predictions and exact theoretical results for simple models have shown the method's practicality, especially in the low modal density region. Thus, validation of the FE structural and acoustic models has been obtained. To obtain the SPL inside fuselage cabin for various configurations, a frequency response analysis is performed with propeller noise as load. The coupled FSI problem, implemented with NX Nastran, was then described and preliminary results show good agreement with the available literature. In this work, the VA performance of a general fuselage is analyzed and no complicating effects are considered. It is worth noting that structural response and noise transmission at low frequencies are influenced by the stiffeners, therefore simplification of the model should not be considered as a limiting factor. Three cases are studied: baseline configuration, DVA configuration and doublewall configuration. The SPL in terms of dBA are extracted from the frequency response analysis, performed on the structure with propeller loading. Firstly, the response is computed for the baseline configuration that provides an averaged SPL of 90.85 dBA. Then, the response is computed once again and the obtained results are compared with the baseline configuration ones. Using tuned mass dampers in particular locations, the averaged SPL is 82.58 dBA, i.e. a 8 dBA reduction. In the last step, the VA performance of airgap and various trim panels materials, i.e. Aluminum and FML, is analyzed. From the analysis it is found that, out of all the configurations that have been assessed for VA performance, FML configuration performs better, the averaged SPL is 85.3 dBA, i.e. a 5 dBA reduction. Ongoing analysis of the FE model will study the effects of various parameters on the interior noise level, such as acoustic absorbing materials, damping, structural modifications, etc. These are important aspects which can be studied easily due to the great flexibility of the FE model. It is worth noting that these Matlab programs can be applied for a generic fuselage section and they, generating the Nastran cards, make VA modeling easier.

References

[1] D. Miljkovic, M. Maletic, M. Obad, *"Comparative investigation of aircraft interior noise properties"*, 3rd Congress of the Alps Adria Acoustics Association, 2007.

[2] Marulo F. and Todd B. Beyer, *"Nastran application for the prediction of aircraft interior noise"*. NASA Langley Research Center Hampton, VA 23665-5225.

[3] Everstine, G. C., *"Structural Analogies for Scalar Field Problems"*, Int. Journal of Numerical Methods In Engineering, vol. 17, no. 3, March 1981, pp. 471-476.

[4] G. SenGupta, Landmann A. E., Mera A. and Yantis T. F., *"Prediction of Structure-Borne Noise, Based on the Finite Element Method"*, AIAA Paper, 86-1861. AIAA 10th Aeroacoustics Conference, July 9-11, 1986, Seattle, WA.

[5] Jean-Francois Deu, Walid Larbi, Roger Ohayon, *"Vibration and transient response of structural-acoustic interior coupled systems with dissipative interface"*. Computer Methods in Applied Mechanics and Engineering, Elsevier, 2008, 197 (51-52), pp.4894-4905 3021-3038.

[6] A. Bermùdes, L. Hervella-Nieto, R. Rodriguez, *"Finite element computation of the vibrations of a plate-fluid system with interface damping"*. Computer Methods in Applied Mechanics and Engineering, 190(24-25) (2001) 3021-3038.

[7] A.Rona, *"The acoustic resonance of rectangular and cylindrical cavities"*. Department of Engineering, University of Leicester, Leicester LE1 7RH, UK.

[8] G. SenGupta, *"Reduction of cabin noise during cruise conditions by stringer and frame damping"*, AIAA Journal, Vol.17, No. 3, March 1979.

[9] Mladen Chargin and Otto Gartmeier, (1990). *"A Finite Element Procedure for Calculating Fluid-Structure Interaction Using MSC/NASTRAN"*. NASA TM-102857, Ames Research Center Moffett Field, CA 94035-1000.

[10] J. F.Wilby and D. T. Sharton, *"Acoustic transmission through a fuselage sidewall"*. NASA-CR-132602, Bolt, Beranek, and Newman, Inc. Cambridge.

[11] R. Alderliesten, *"Fatigue and Fracture of Fibre Metal Laminates"*, Springer International Publishing AG, 2017.

[12] Siemens *"Acoustic User's Guide"*.