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Optimisation of passive/active systems for reduced interior aircraft noise

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ABSTRACT

Aircraft (both fixed wing and helicopters) as well as launch vehicles are subjected to high external dynamic loads, both from mechanical and acoustic origins.

Since structural design very rarely includes the reduction of interior noise into account in its specifications, manufacturers often resort to add-on systems to reach acceptable interior noise levels. Passive and active systems are implemented, but the physical complexity of the interior acoustic field leads to a very large number of such systems. On the other hand, the constraint of minimum weight penalty calls for an efficient optimization of the global treatment.

Airbus Group Innovations is developing a numerical tool to ease the designer task. Optivib_kit® is a Matlab® based software that allows the spatial as well as the parametric optimisation of a set of active or passive structural control devices.

This optimisation can be carried out from a mechanical model of the structure of interest, or from a large set of measurements made on a real test article.

In the paper, several examples will illustrate the use and the performance of this piece of software, targeted to the minimisation of vibration levels or that of interior noise.

Keywords: Vibration Control, Optimisation

INCE Classification of Subjects Number(s): 38, 46

1. INTRODUCTION

The necessity for noise and vibration reduction does not need to be emphasized again in this paper: what would be the reason for developing experimental methods and numerical prediction tools, if not to help the design of anti-vibration and noise reduction devices…

Passive and active noise control systems are being implemented in a number of aeronautical products at the present time, should they be passenger aircraft (especially propeller driven A/C, SAAB 2000, ATR72, Do 328 propeller), military transport aircraft (C130 cockpit…), rotorcraft (several Airbus Helicopter products), launch vehicles, etc. This paper is however limited to passive and active devices implemented at structural level. So, pure active noise control, involving acoustic secondary sources, is not considered. The framework is therefore that of a vibrating structure, with:

- initial (primary) excitation of acoustical or mechanical origin,
- potential structural devices for vibration reduction (secondary active forces, discrete vibration absorbers, stiffeners/dampers mechanically connecting two locations…),
- a vibrational or acoustical measure of performance: acoustical pressure, acceleration/displacement at given locations, transmitted force, etc. This quantity has to be minimised in a sense to be defined more precisely.

The implementation of such devices is a challenge, especially in the vibro-acoustic domain, where primary excitation is spread over rather large surfaces, and where anti-vibration devices can be placed at a large number of potential locations.

The acoustic/dynamic engineer is faced with the difficulty of choosing optimal locations and optimal parameters at the same time for a set of vibration reduction devices. This mixed nature of the

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optimisation task calls for quite different numerical approaches: parametric optimisation is done using continuous methods, whereas topological optimisation is based on discrete variables (the locations on the structure where devices can be practically implemented) and is therefore carried out using dedicated approaches (heuristic methods, simulated annealing…).

The performance requirement is very often stated in the frequency domain: devices are generally supposed to be designed and optimised for discrete frequencies or for a given frequency range. The software is therefore designed to carry out optimisation in the frequency domain, under the strong assumption of linearity: primary excitations and the effect of the secondary devices (local structural modifications or external secondary forces) add up when calculating output variable of interest.

This assumption of linearity is used a second time when choosing to represent the initial structure with a set of transfer functions between the points of interest: transfer paths between primary excitation, device implementation locations, performance indicators… are recorded and build up the database on which all optimisation steps are performed.

2. Software features

2.1 Software architecture

As mentioned earlier, the software bases its computation on the representation of the primary structure as a set of transfer functions. This set can be obtained from a numerical model of the structure (e.g. from a finite element model), but also from experimental data. The following flowchart illustrates the different steps of the preparation/optimisation/post-processing process.

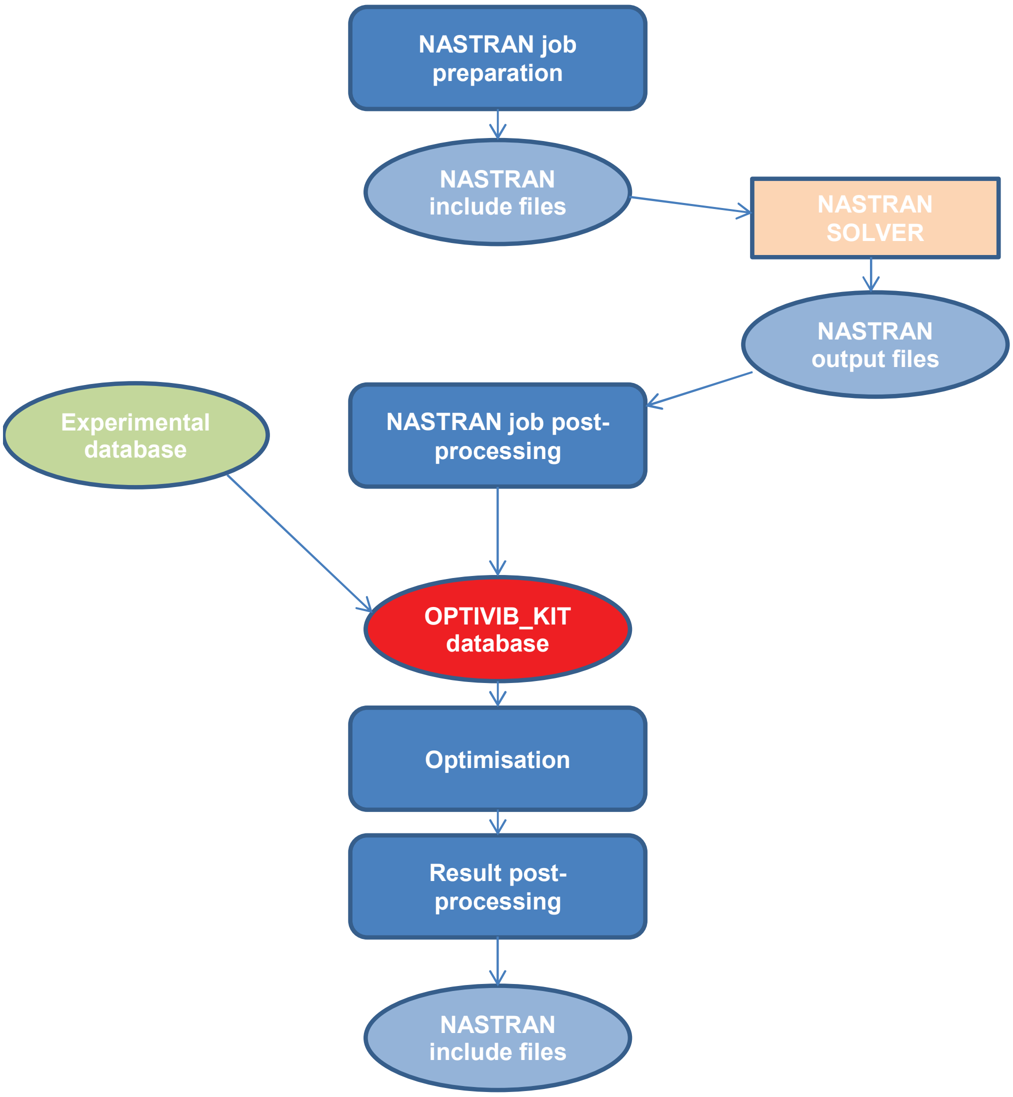

Figure 1: Optivib_kit ® flowchart

All those steps are monitored by the user relatively seamlessly. The user defines all initial data in an Excel file (type of secondary devices, cost function to be minimized, potential secondary device implementation locations and directions, etc.). After this preparation step, a Matlab supervisor guides the user through the various steps of the calculation, including the NASTRAN runs to prepare the database made up of transfer functions.

The software can be provided as a standalone executable file, so that no access to Matlab and its toolboxes is necessary.

During the whole optimisation process, the structure is known only through the transfer function database, therefore as a “black box”. It is therefore quite important to double check the optimisation results back into the physical domain. To this end, the characteristics of the optimized vibration reduction devices are output as Nastran elements (forces in the case of active control, mass/spring/dampers in the case of discrete vibration absorbers, stiffeners/dampers in the case of point-to-point devices), included in the Nastran input file. A Nastran run in then performed, completely independently of the Matlab prediction, to verify the result.

2.2 Numerical ingredients

Optimisation methods, as mentioned earlier, belong to two different categories:

- For parametric optimisation, least square methods are used, together with constrained optimisation methods available in Matlab Optimisation Toolbox (such as “fmincon”)
- For topological optimisation, various methods were developed: heuristic approaches, but also the better-known “Simulated annealing” algorithm

Both optimisation processes are carried out simultaneously, in such a way that a new set of location is optimised in the parametric sense (best damping, best stiffness, etc.), before going to the next configuration.

Parametric optimisation for DVAs can sound strange for vibration reduction practitioners, because in many cases there is no real flexibility in the mechanical characteristics of practical DVAs. First of all, Optivib_kit® allows the user to carry out the topological optimisation alone, without parameter changes. Then, to allow some flexibility in the tuning of DVAs gives rise to very interesting vibration reduction results, as was demonstrated in reference 1.

The key to successful optimisation is to be able to compute the performance of one given configuration (realisation) in a very short time. In the case where numerical transfer function are used (calculated initially by a finite element package such as Nastran), going back to the FE package to introduce the new devices and computing the modified transfer function is not an option, it would be much too costly. Therefore, using the transfer function database presented earlier, new devices are “implemented” using the impedance coupling method, which allows calculating their effect on the initial transfer functions (see references 2. and 3.).

3. Optimisation of passive devices in the rear part of an aircraft fuselage

The software characteristics and potential are now illustrated in the vibroacoustic application case. A finite element model representative of the fuselage rear part was prepared to predict the impact of the turbulent boundary layer excitation on the cabin acoustics. The investigation consists in analysing the placement of Discrete Vibration Absorbers (i.e. tuned-mass dampers) on the fuselage to reduce the interior sound level. The software topological optimization option is exploited to determine a set of very good locations (if not the optimal ones) in a very limited time.

3.1 Finite Element Model

The structural finite element model is displayed in Figure 2; the acoustic one in Figure 3. The fuselage is made of a tail section and a rear cabin part, with a main deck and a vertical wall between the tail and cabin, and the tail and cargo area. The skin model is constituted of shell elements, while the frame and stringers are modelled with beam elements. More details about the model characteristics are given in Table 1.

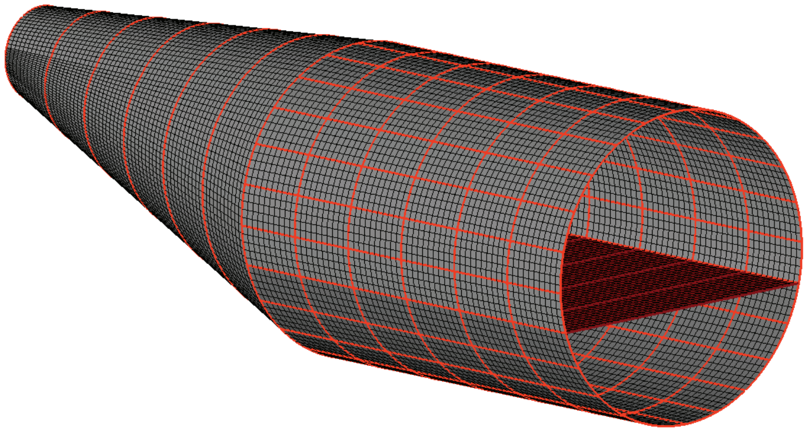

Figure 2: A/C rear fuselage structural model (shell and beams)

The excitation is a turbulent boundary layer loading spreading over the entire fuselage outer skin. The loading spectrum varies spatially and is different for each single point: the loading specification files are thus substantially large. They may stem from either computer-intensive aeroacoustics simulations or from a complete wind channel test campaign.

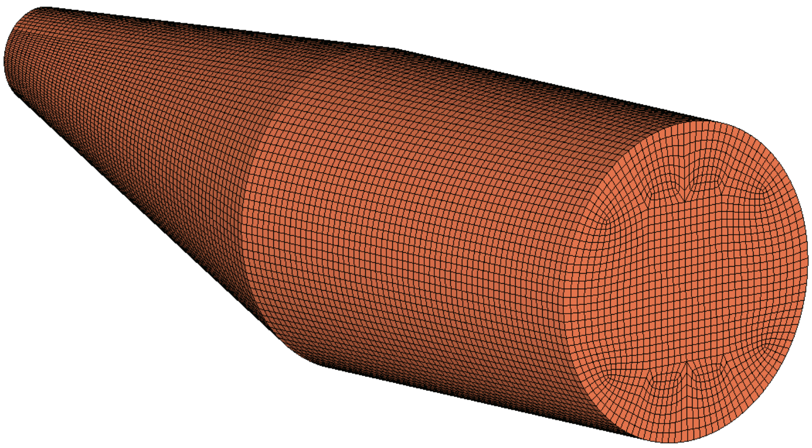

Figure 3: A/C rear fuselage acoustic model

Table 1 – A/C rear fuselage model characteristics

Finite element entity	Number
Structural nodes	18790
Cavity nodes	183243
Shell elements	18752
Beam elements	3266
Fluid elements	170352

The model was written using Nastran syntax and solved using the coupled vibroacoustics direct solution algorithm (SOL 108) to obtain a reference acoustic pressure at three arbitrary points (Figure 4). Ten minutes were approximately necessary to compute a single point of the frequency axis. Figure 4 and its 70 points frequency axis was obtained after 12 hours of computation.

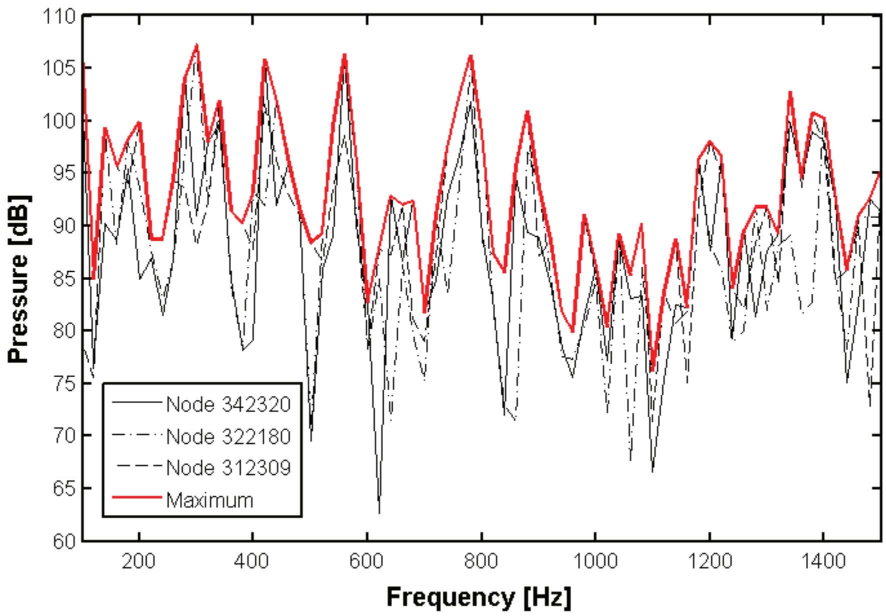

Figure 4: A/C cabin pressure in three illustrative points, as well as the associated pressure envelope

3.2 Optimization analysis set-up and objectives

The background idea for this numeric illustration consisted in endorsing the role of an acoustics engineer subject to practical limitations:

- Discrete Vibration Absorbers (i.e. tuned mass dampers) are “off the shelf” devices, the characteristics of which are fixed in advance
- DVA must be located on stiffeners (frames or stringers)
- The added mass must remain under a strict limit

These practical rules restrict the use of the implemented software since:

- Other types of devices such as dampers, elastomer mounts, rods, or any type of one-dimensional device with any frequency-dependent characteristics could have been introduced instead of DVA, as long as it is linear and rotation free
- The program includes an option to optimize both topology (i.e. the placement itself) and the DVA characteristics (mass and structural damping)

The set of possible locations, although restricted, remains quite large (168 points), as illustrated by Figure 5. Other DVA and optimization problem characteristics are displayed in Table 2.

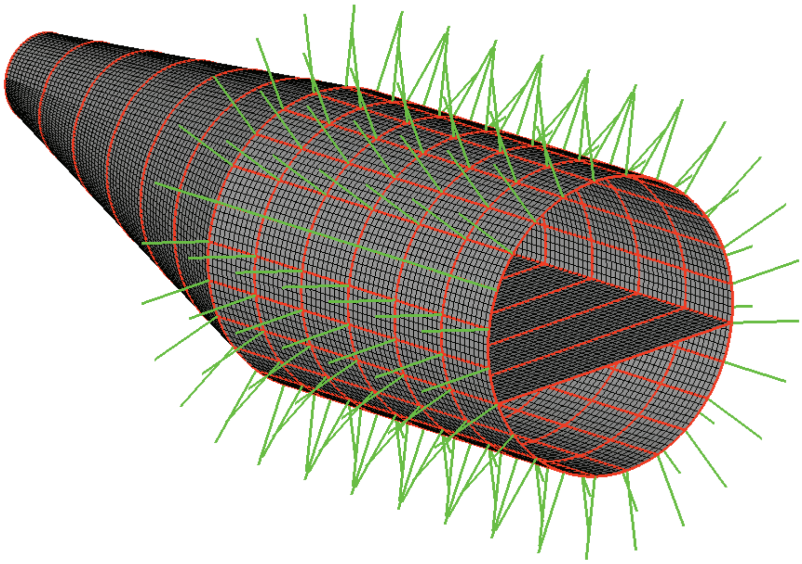

Figure 5: DVA set of possible locations with their orientation (in green)

Table 2 – DVA and optimization problem characteristics

DVA mass [kg]	0.2
DVA damping loss factor	0.1
Maximum added weight	1 kg
Number of possible DVA locations	168
Total number of configurations	~10^{11}
Equivalent CPU time [years]	1900000
Optimization target frequency	780 Hz
Minimization	Pressure (3 points)

Given the maximum added mass of 1 kg and a fixed DVA mass of 0.2 kg, and assuming that the best performance can be obtained with 5 DVA, more than a hundred of billions of configurations are possible. Using Nastran as a simple “black-box” program called by a global optimization code would thus result in weeks of computation. If the optimization problem were to be solved parametrically, assuming a cost of 10 minutes per realization, the overall computation time would reach a few millions of years.

3.3 Optimization results

In the 5 DVA case, the global optimization algorithm tends to converge with approximately 10000 iterations of the simulated annealing algorithm in about 10 minutes. The employed convergence criterion is not a strict academic criterion but rather a pragmatic one, based on the reduction obtained at the target frequency. The convergence is thus reached if the number of iterations can be varied without a significant impact on the reduction figure. The obtained solution is not optimal in a strict sense but may be satisfactory enough from an industrial point of view.

A brief overview of the optimization main features is given in Table 3. Despite the practical limitations, a significant noise reduction of 12.8 dB is obtained at the target frequency of 780 Hz.

Table 3 – Optimization main features

Number of iterations	10000

CPU [min]	10
CPU per realization [s]	0.06
Overall gain at 780 Hz	4.36
Overall noise reduction at 780 Hz [dB]	12.8
Added mas [kg]	1

The optimization software yields a number of outputs, among which:

- The "best" configuration as a set of nodes IDs (i.e. the most relevant DVA locations), as presented in
- The pressure envelope response (used as functional) with and without DVAs: this result is a broadband result, and can be used to check if no amplification due to the presence of DVAs is predicted outside of the close vicinity of the target frequency. See Figure 8.
- The gain at each single functional evaluation node, at the target frequency (780 Hz, Figure 6). This can be very useful to understand possible convergence issues.
- The actual local displacement of the moving DVA mass. This information can be critical, because it would be foolish to accept an optimal configuration where the moving masses would reach displacement levels incompatible with the DVA design or with the allowable space around it.

We note in Figure 6 that the optimization does not only apply to the functional itself, but also to each functional node separately. For sure, in practice, one would certainly choose many more than only three points to double check whether the acoustic pressure is still contained in most parts of the cavity volume (see second application case in Section 4).

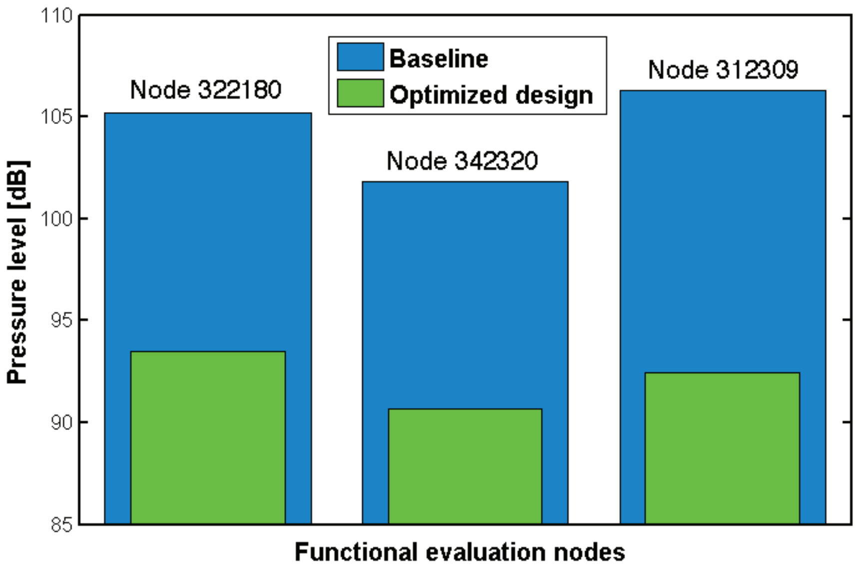

Figure 6: Comparison of the pressure level at the functional nodes at 780 Hz between the baseline and the optimized solution (5 DVA, total added mass 1 kg)

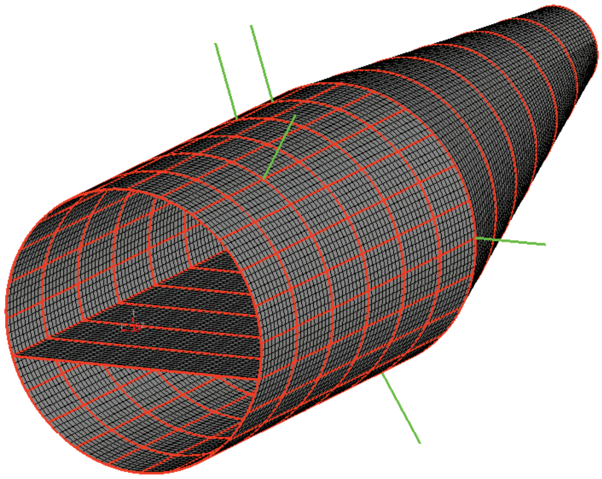

Figure 7: Optimal DVA locations for the target frequency of 780 Hz

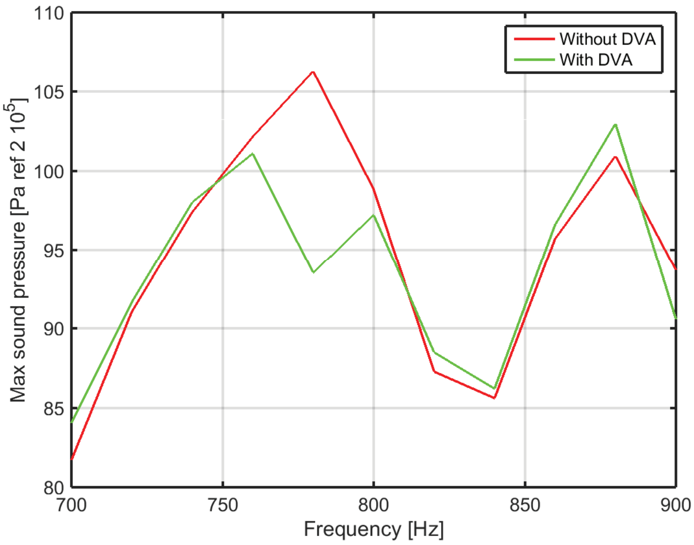

Figure 8: Narrowband effect of optimally positioned DVAs (sound pressure envelope over three nodes)

3.4 CONCLUSIONS

The vibroacoustic analysis of the rear of a fuselage under a turbulent boundary loading has been taken as illustration. The optimization has focused on a standard NVH task consisting in placing a few “off the shelf” DVAs to lower the pressure envelope of 3 arbitrary points at the target frequency of 780 Hz, where a resonance occurs. The design has been driven by practical rules: the possible locations were restricted to the connection points between stringers and frames (168 positions in total) and a maximum mass constraint of 1 kg has been enforced.

The 5-DVA “best” configuration has been determined after roughly 20 minutes: 10 minutes to compute the database of transfer functions needed to apply the impedance coupling method (Nastran); and 10 minutes to solve the optimization problem with a simulated annealing algorithm. By contrast, Nastran solving time is roughly 10 minutes per frequency axis point.

The overall pressure level reduction is almost 13 dB and could be observed not only at the functional level, but also at each functional node separately. The implemented toolbox is thus a powerful means of achieving significant noise reductions on very large models excited by complex vibroacoustic loadings, under strict industrial constraints.

4. Introduction of discrete vibration absorbers in a propeller aircraft airframe

Another illustration is provided here of the use of the Optivib_kit® software for the optimisation of a large set of DVAs to be placed into the fuselage of a generic propeller aircraft. The main differences compared to the previous case are:

- lower target frequency (close to 100 Hz)
- larger potential DVA number

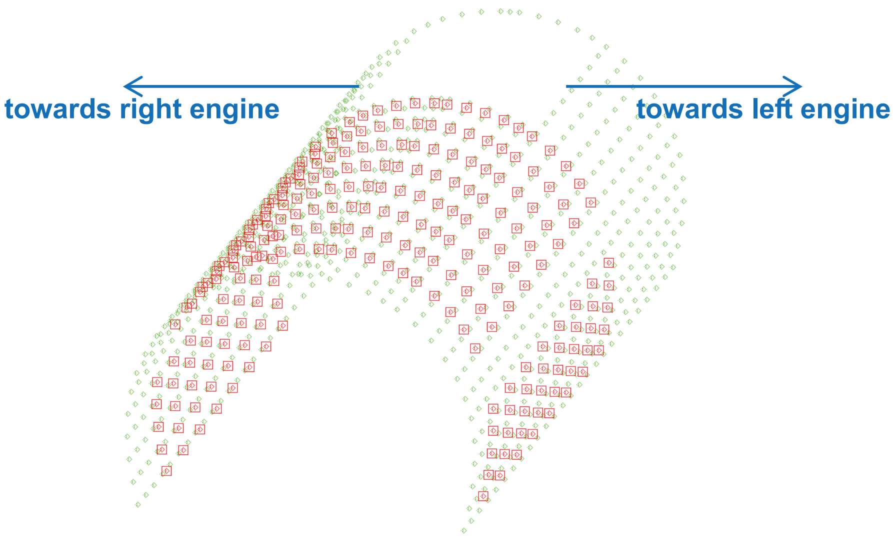

Figure 9: Sketch showing a partial view of the aircraft fuselage with (in red) the potential implementation locations for DVAs

4.1 Mechanical configuration

As in most aircraft airframe structures, this section is made up of a skin, stiffened by a mesh of stringers (in the longitudinal direction) and frames (circumferential stiffeners). DVAs are very often placed on the frames (allowing more space around them, and locally stiffer than the stringers). In the present case, the best implementation scheme was the following:

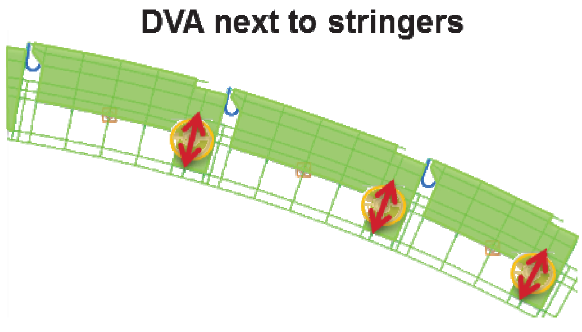

Figure 10: DVA implementation on frame

240 potential implementation positions were considered. Those DVAs had a fixed natural frequency (BPF (Blade Passing Frequency), close to 100 Hz), but a flexible mass, between 100 and 800 g (in fact two off-the-shelf designs, with two different masses).

The objective here is to minimize the total number of DVAs in order to have a good performance on the acoustic cost function, i.e. the sound pressure level (SPL) measured in a specification plane with 2431 nodes.

4.2 Optimisation run

In this search for the minimal number of DVAs (therefore minimal mass penalty), a specific feature of Optivib_kit® is employed, which consists of reducing step by step the number of DVAs used in the computation of the cost function, removing at each step the least efficient one.

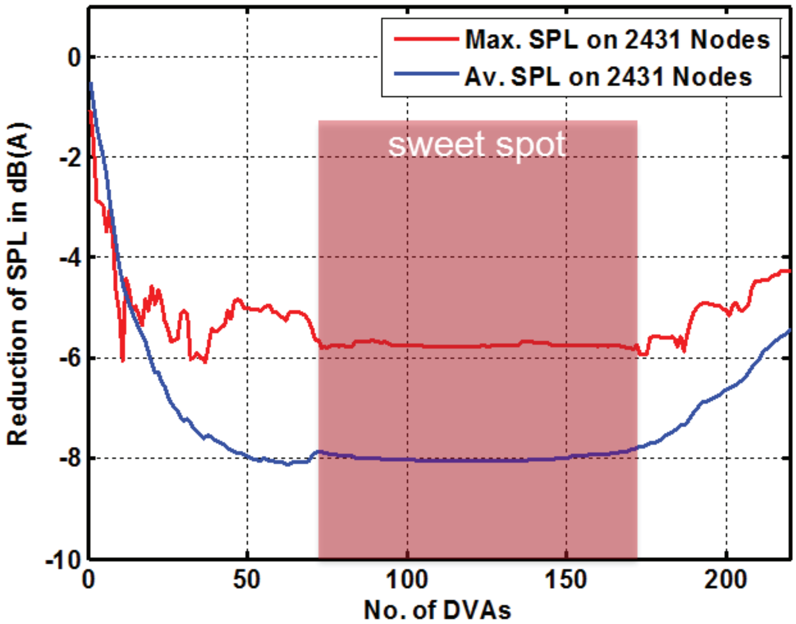

Figure 11: Study of the effect of step by step DVA removal from a large initial set (SPL: Sound Pressure Level)

The result of this first run is that the acoustical performance is significant (given the compromises involved with the minimization of a very large set of pressure measurement points), and that there exists a "sweet spot", i.e. a range of DVA numbers where the noise reduction performance does not depend highly on the DVA number.

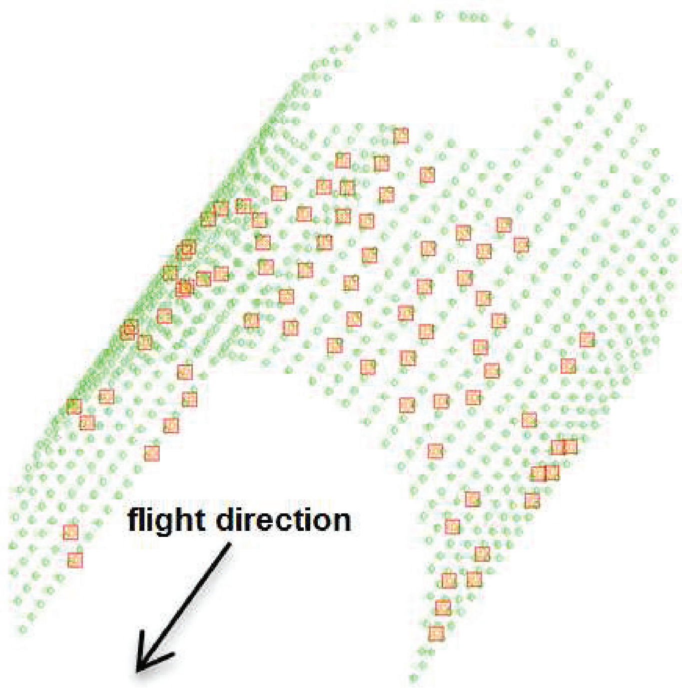

Figure 12: Example of DVA distribution (nDVA = 80)

This illustration of DVA distribution shows that optimal positions are located on the fuselage upper section where the engine noise levels and vibration levels are the highest.

5. CONCLUSIONS

In the practice of vibration control, should it be for vibrational or acoustical cost function minimisation, the selection of the best devices at the optimal location is a necessary step. Here the help of numerical tools is very valuable, because the number of possible configurations is very fast impractical for a brute force approach (checking all configurations in turn).

Airbus Group Innovations has developed a software tool Optivib_kit® available to its business units to assist the engineer in this task. In this paper, two applications were described in detail.

Other applications were performed for more structure-oriented cases, such as engine mount optimisation, which show more the interest of parametric optimisation. Indeed, in these cases, the possible implementation locations are not numerous.

In the vibro-acoustic example shown here, the focus is on the topological optimisation and the two examples give a demonstration that such a tool can at least drive the choice of best configurations.

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