EXPERIENCES WITH THE EVALUATION OF HIGH MODAL DENSITY STRUCTURAL DYNAMICS

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Introduction

This paper reports the numerical and experimental experiences carried out at the Department of Aeronautical Engineering of the University of Naples concerning the structural dynamics and the interior noise evaluations. These activities have been performed during the EEC BRITE/EURAM Project called R.H.I.NO.: Reduction of Helicopter Interior NOise. The analysis of the structural, acoustic, and structural-acoustic predictive methodologies in frequency ranges at low, medium, high and very high modal densities has been a real challenge involving the major european partners: manufacturers, universities and research centres. The R.H.I.NO. concluding remarks of projects have demonstrated the feasibility of the S.E.A. and A.S.E.A. (Statistical Energy Analysis and Advanced Statistical Energy Analysis) approaches together with the possibility to adopt the F.E.M. also in a frequency range where the modal base of the model is not more suitable.(1)(2)(3)(4)

In this work the whole new set of the numerical and experimental activities concerning the specific test-article, will be presented and discussed.

The Test Article and a Summary of the 1991 Results

The test-article is an 'irregular box', assembled using six plates: with no couple of planes parallel one to each other. It was called ASANCA Box (using the name of the project when developing the pilot phase).

Tab.1: The Eight Vertices of the ASANCA Box

Points	x [mm]	y[mm]	z[mm]
A	0	0	0
B	0	1161	'
C	600	800	0
D	600	0	0
E	0	0	1300
F	0	900	1500
G	500	600	1433
H	500	0	1300

The plates have different shapes, thicknesses, and dimensions, and they are assembled by using L-shaped alluminum beams; these latters increase the stiffness of the single plate but are not connected one each other. For this reason the beams do not constitute a frame on which the plates can be considered as supported. The internal sides of the plates have been partially covered of damping material ISODAMP C-2003™, in order to get a damping loss factor between the 1% and 4%.

Tab.2: The Plates of the ASANCA Box

Plate	Itemised	Thickn. mm	Area m^2	Damping Material m^2	Added mass Kp
1	Base	4.	0.588	0.35	0.87
2	Top	4.	0.384	0.23	0.57
3	Door	3.	0.941	0.36	0.89
4	Side	2.	0.715	0.26	0.64
5	Front	2.	0.979	0.25	0.61
6	Back	3.	1.455	0.87	2.16

The analysis carried out in 1991 gave a set of results that can be so summarised: (1)

✓ *the F.E.M. is not suitable when the structural mode-shape becomes enough complex for a twofold reason: the computational cost and the possibility to identify the mode-shape correctly;*

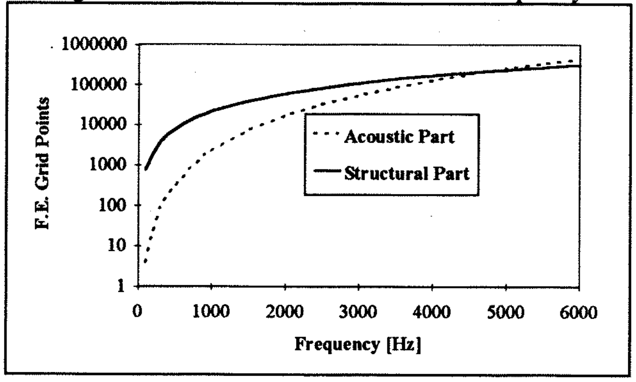
- ✓ *the finite element method can give detailed results when a correct modal representation is achievable;*
- ✓ *the structural-structural and acoustic-structural transfer functions obatined with a "blind" test are qualitatively acceptable but they cannot be used for an analysis of the absolute levels; they could be efficiently used for the parametric trends;*
- ✓ *SEA applications can give acceptable results when, increasing the excitation frequencies, the modal densities and the modal overlap factors reach the proper values (in particular a sufficient number of modes must be involved or the modal overlap factors must be greater than unity);*
- ✓ *the cost of the F.E. flexural waves simulations could reach unacceptable levels, if compared with the quality of results; the finite elements could be used in average sense, although further investigations are needed.*

Numerical Activities during 1995

The principal goal of R.H.I.NO.-Task 9 was the validity assessment of the Finite Element approach in studying the prediction of structural vibrations and interior noise of irregularly shaped flexible cavities. During this time the experimental support has been extensively used: actually a Finite Element model is prepared by structural analysts but it should be verified by proper experimental tests in order to get the required confidence level with the real dynamic behaviour of the structure under examination. The need of a new numerical phase has been strictly envisaged for testing the new acoustic capabilities offered by the MSC/NASTRAN Version 68. These latters are linked to the definition of new modules in which the acoustic part is properly defined. It is no more necessary to use a structural analogy (NAVIER Equation Set transformed in HELMHOLTZ Equation) since it is now possible among the new capabilities: (i) to use fluid finite elements that adopt special shape functions, (ii) to use the proper material definition, in term of acoustic speed and material density, (iii) to define the structural-acoustic coupling directly.

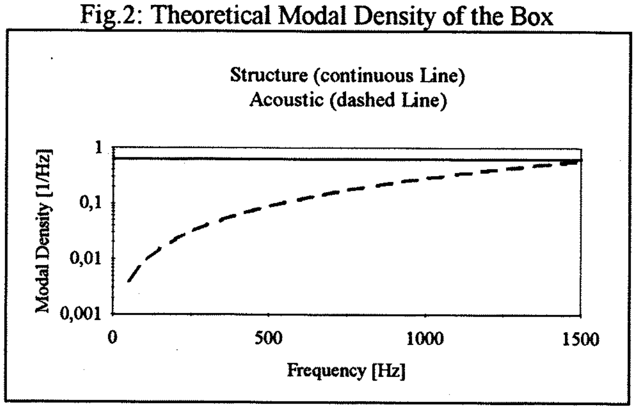
First of all it is not worth to recall how many grid points are needed for the structural and acoustic part of the box.

Fig.1: The F.E. Grid Points Needed vs. Frequency



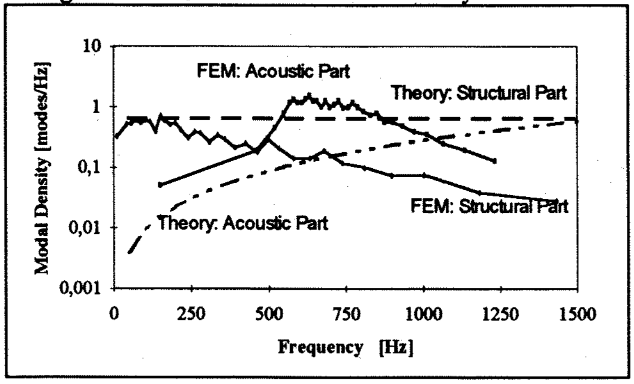
So it is clear that the authors would like to verify if, for the acoustic part, the new capabilities would allow an extension of the frequency applicability of the finite element model. The situation of the old numerical model can be summarised using a classical SEA parameters as the modal density (fig. 2, 3).

Fig.2: Theoretical Modal Density of the Box



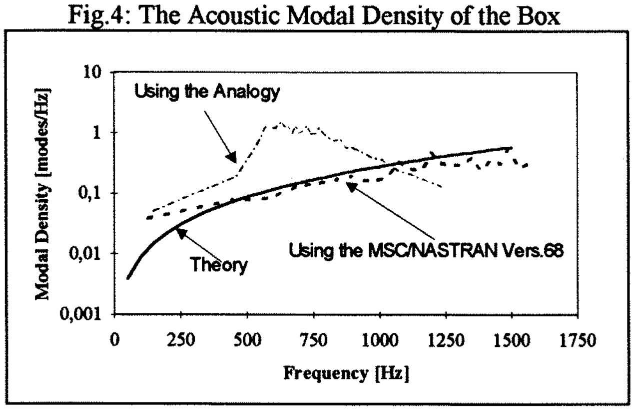
It is evident from fig.3 that the strong limitation is due to the plate simulations that could be carried out no more than 150 Hz using a finite element mesh 5x9x11 (with reference to the acoustic domain). The effect of new capabilities of the acoustic finite element developed inside the MSC/NASTRAN Vers.68 could be easily demonstrated looking at the contents of the fig.4.

Fig.3: Theoret. and Num. Modal Density of the Box



The simple run of the old finite element model in connection with the new modules and instructions has generated a quite different situation in term of acoustic modal density, fig.4. The result can be read from two different points of view: the new solutions allow to get with the same mesh natural frequencies and mode shapes more accurate than in the past, there is the possibility to use the acoustic finite element model for higher values of the frequencies range.

Fig.4: The Acoustic Modal Density of the Box



However in the coupled solution, the structural part will still rule the total behaviour as shown in fig.5, where the structural behaviour is dominating up to 250 Hz.

The effect of increasing the degrees of freedom has been also evaluated: it is clear that increasing the computational costs also the structural domain can be adequately represented, as clearly shown by the following figure. For sake of conciseness only the analysis of the plate #1 has been reported since analysis of the other plates show the same trends.

Fig.5: Modal Density of the Coupled Model

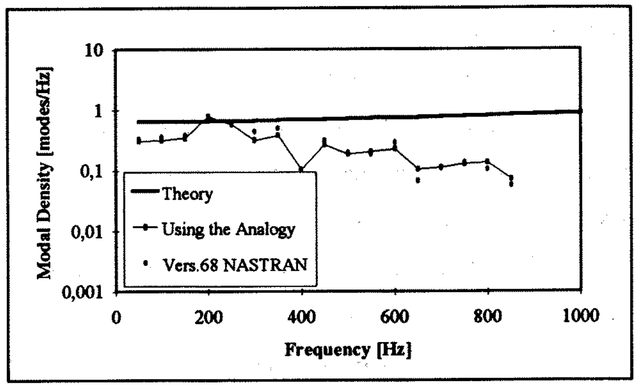
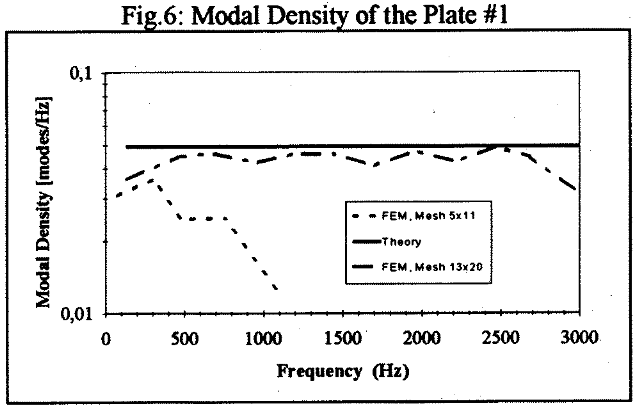


Fig.6: Modal Density of the Plate #1



The acoustic enhancements will be also reported in the next paragraph when discussing about the experimental activities. It is useful to recall that in these new numerical activities the experimental support has been fundamental: the "blind" test were used to demonstrate the feasibility of F.E.M. as pure predictive technique, while the aim of the present phase was to check the limits of enhancement of this latter. It is not worth to report the structural modal analysis result.

Tab.3: Comparison of the Struct. Nat. Freq. and Ident.

#	Natural Freq. [Hz] Num.	Natural Freq. [Hz] Exp.	Diff. %	Mode Identification
1	19.85	21.20	6.4	excellent
2	27.60	30.02	8.0	good
3	41.48	41.88	0.9	good
4	57.87	57.65	0.4	acceptable
5	62.30	61.26	1.7	acceptable
6	83.21	85.37	2.5	good
7	101.21	100.15	1.1	good
8	113.08	115.38	2.0	acceptable
9	118.04	118.83	0.7	acceptable
10	152.94	152.87	0.05	acceptable

It is evident that the F.E.M. allows the mode identification only when these latters are very clean and smooth. At growing frequencies the identification is not more possible and probably suitable, due to the fact that the plate dynamics is not more dominated by the modal behaviour. The increase of the acquired experimental points could unacceptably enlarge the test execution time. For all these reasons it has been decided to work with the transfer functions to evaluate the predicitive capabilities.

The Experimental Set-Up and Activites

The experimental set-up used for the laboratory test is based upon the Single Input Single Output acquisition logic and it has been composed by:

- ✓ Dual Channel Analyser FFT ONO-SOKKI CF350
- ✓ Piezoelectric Impedance Head
- ✓ Bruel & Kjaer (B&K) Microphones
- ✓ Endevco Σ Accelereometer
- ✓ In-Home Built Signal Amplifier AVIORADIO
- ✓ Excitation Signal Amplifier B&K
- ✓ Response Signal Amplifier B&K
- ✓ Analog-Digital Interface IEEE488
- ✓ Personal Computer

The box has been suspended at a steel frame through four springs to simulate the free condition in the space. An electrodynamic shaker has been used for the mechanical excitation, while a small loudspeaker has been used for the acoustic internal excitation. the pressure responses have been acquired using two microphones: the first one was fixed, the other movable inside the internal volume. For the acquisition phase an automatic 3-axis positioning system has been designed, assembled and used. Each axis allow to move the moving microphone in the desired position: this latter can be selected by a Personal Computer that drives the motor driven units. The fixed microphone has been positioned close to the center of plate #6.

Tab.4: Excitation & Acquisition Locations (mm.)

Mechanical Excitation				
Shaker				
	Grid	x	y	z
	633	0.	300.	850.
Microphones				
	Grid	x	y	z
Mic. #1	1113	275.	173.	666.67
Mic. #2	1173	256.25	233.64	1181.2
Mic. #3	1263	256.25	389.39	1210.4
Mic. #4	1248	275.	432.58	691.66
Mic. #5	1205	0.	412.12	690.
Acoustic Excitation				
Speaker				
Grid	x	y	z	
1054		104.	165.	
Microphones				
N°	Grid Point	x	y	z
Mic. #4	1248	275.	432.58	691.66
Mic. #5	1205	0.	412.12	690.

Comparison between Num. and Experimental Results

The comparison between the experimental and the numerical results has been carried out firstly analysing the natural frequencies and mode shapes. This is reported in the fig.7 and fig.8 that shows the obtained confidence level.

Fig.7: Correlation of the Structural Modes

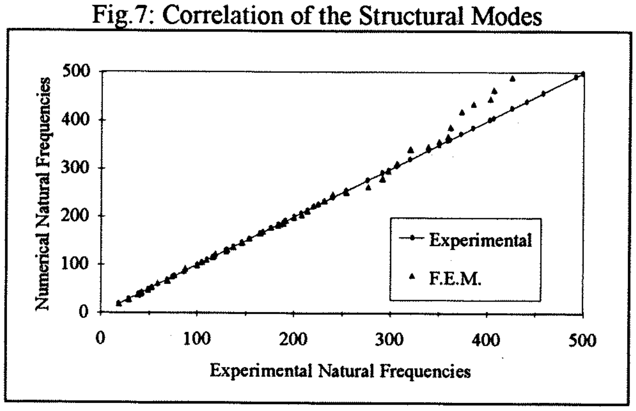
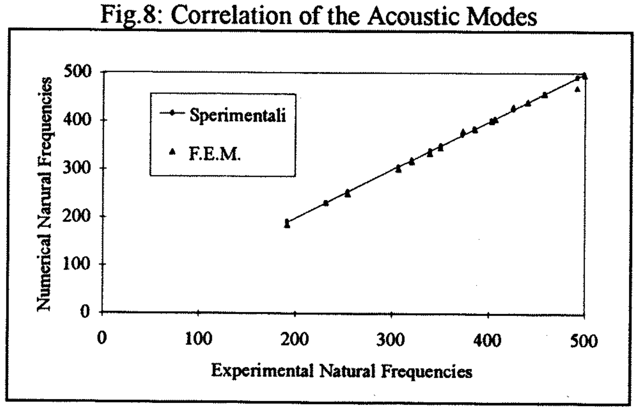
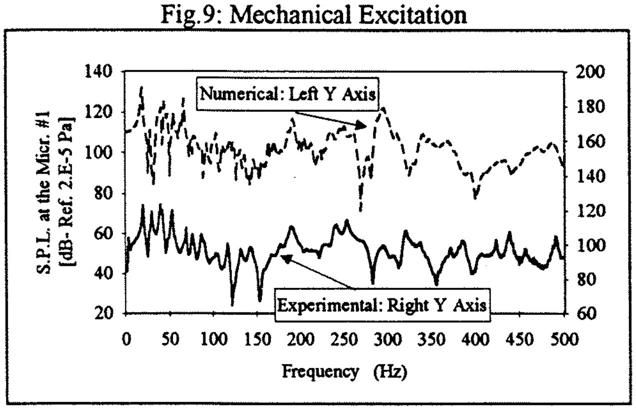


Fig.8: Correlation of the Acoustic Modes



The most interesting comparison is due to the transfer function between the (structural or acoustic) excitation and the microphone acquisition. In fact the authors have already clarified the difficulty to identify correctly the mode shapes for decreasing wavelenghts, so that it was decided to work directly with the following dynamic representation.

Fig.9: Mechanical Excitation



The fig.9 reports one of the classical result obtained. It is easy to verify the quality of the finite element representation in comparison with the experimental results. The enhancements of the acoustic module of the MSC/NASTRAN allowed an extension of the applicability of the finite element model, even if the structural domain is not enough represented: *fortunately at growing frequencies the acoustic modal density is able to cover the corresponding structural so to explain the dynamic behaviour reported in the fig.9.*

The Energy Based Similitude

During the research the authors tried also to define an enhanced use of the finite element method. Some preliminary papers about the subject have been also given, but all concerning the possibility to average the information coming from a finite element program[6], and to extend the finite element applicability in higher ferquency domain by using a scaling procedure based on an energy similitude[7]. In order to outline the ongoing activities on vibration prediction here some results, obtained with the above mentioned approach, are reported. In particular in fig.10, it is shown the driving point admittance for an irregular flexural plate. The basic finite element model was designed to represent the modal behaviour up to 200 Hz. The figure shows that the energy based scaled finite element model can reach frequency ranges well over the modal domain.The fig. 11 reports the mean square velocities of the same plate compared with the S.E.A. evaluation. We would like to stress that while the S.E.A. values gives only mean velocities without any information on the real velocity distribution over the plate, the F.E. may give additional information on the vibrational field. Moreover these approaches, whose analysis is still in progress should allow to easily calculate the coupling parameters among subsystems: these latters are the classical and fundamental input information within the S.E.A. approach.

Fig.10: Real Part of the Driving Point Admitt. vs. Freq.

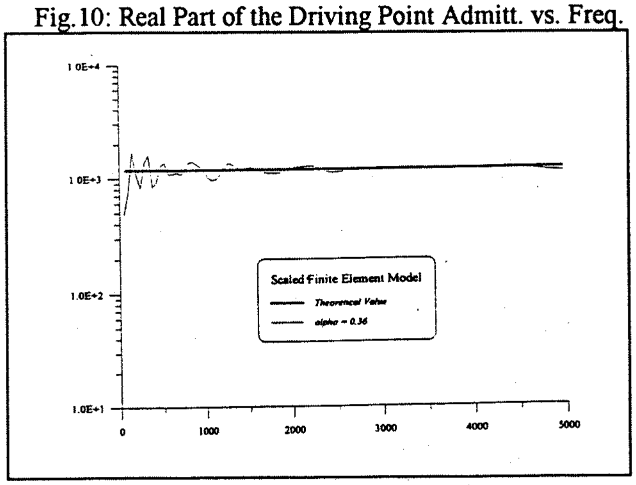
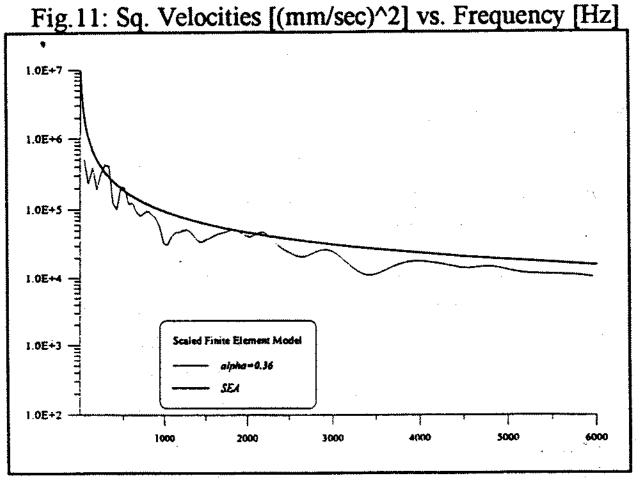


Fig.11: Sq. Velocities [(mm/sec)^2] vs. Frequency [Hz]



Final Remarks

The contents of the present report can be considered as the release of a milestone report inside a large research about the possibility to use the finite element as predictive tool for the structural dynamicss and the interior noise. The information collected inside the EEC research program are here briefly reported and presented toghether with the new possibilities that could be analysed in the near future concerning the use of finite element models for calculating the fundamental SEA parameters as the coupling loss factors.

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