

# Results of the GVT of the Unmodified GARTEUR SM-AG19 Testbed in South America

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**This work presents the results of the modal analysis performed during the ground vibration testing of a testbed originally designed by the Group for Aeronautical Research and Technology in Europe (GARTEUR). The model testing brought challenges in determining modes with very close frequency values, which were detected independently of the excitation signal. A modal validation process was carried out in order to identify these close-spaced modes as well as their dynamic characteristics. The reliability of the experimental modal model was verified by modal assurance criterion calculations between the experimental data and validated by comparison with a finite element model.**

## Nomenclature

<i>CG</i>	=	center of gravity of the model
<i>FRF</i>	=	frequency response function
<i>GVT</i>	=	ground vibration testing
<i>ITA</i>	=	Instituto Tecnológico de Aeronáutica
<i>MAC</i>	=	modal assurance criterion
<i>MOV</i>	=	mode over complexity value
<i>MP</i>	=	mode participation

## I. Introduction

The design of an aircraft demands highly accurate and reliable analytical models for aeroelastic behavior predictions. One method largely used to generate data for the aeroelastic analysis is the ground vibration testing, which also plays an important role in the certification process of new or modified aircraft. Nevertheless, in order to achieve the required levels of accuracy and reliability, the GVT itself must be supported by methods and techniques, which are also heavily dependent on such criteria.

The constant improvement in computational resources has allowed the implementation of sophisticated software with pre-programmed routines and friendly interfaces, providing to the analyst plenty of useful and powerful tools. On the other hand, if the physical reasoning and the understanding of the basic features of a modal testing are lost on the way, those tools may become dangerous when dealing with non trivial results.

This paper discusses the GVT of a testbed, which consists of a simplified representation of an airplane, originally designed and built by ONERA<sup>1</sup> and used in a Round Robin work performed by the Group for Aeronautical Research and Technology in Europe (GARTEUR), in an effort to assess methodologies for the experimental determination of modal characteristics. Several companies and Universities produced, analyzed and compared a series of independent measurements.

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The aim of this work was to produce a reliable and accurate modal model only by Phase Resonance Method and Single Input Multiple Output (SIMO) testing techniques<sup>9</sup>. Tests were made without the ONERA's original viscoelastic treatment applied over the wing and without mass modifications<sup>1,2</sup>. The modal model was then validated by correlation with a finite element model.

## II. Review of previous works

The GVT of the GARTEUR's testbed has been addressed in several papers<sup>1,2</sup>. Discussions have been made on the effect of mass modification on the modal characteristics of the system, different excitation, measurement, and identification techniques, regarding the closely spaced modes and the effects of the viscoelastic treatment and mass modifications.

Balmès and Wright<sup>1</sup> compared different measurement and identification techniques applied to the original ONERA's testbed. The results from companies in France (ONERA, SOPEMEA, Aérospatiale, Interspace and CNAM), Germany (DLR), the Netherlands (NLR, Fokker), Sweden (Saab) and the UK (DRA, Manchester University, and Imperial College) were investigated to evaluate the efficiency and reliability of test methods. Load cells and electrical impedance were used to measure force, with proper mass compensation being done. A number of issues were discussed in Balmès and Wright's work, but the most relevant are the difficulty in distinguishing the modes with closely spaced frequencies and the probability of inappropriate mass compensation of the moving mass of the shaker. They also compared the estimated modal characteristics provided by the participants. Force appropriation and model identification methods were used and the results did not indicate any influence of the method used on the results. Deficiency in the mass compensation process may have been responsible for a high frequency estimate on some modes, while the inherently low damped structure and the influence of the instrumentation may have caused a wide spread in damping ratio estimative of up to 30%, thus the validity of the viscoelastic treatment. The influence of different lengths of cable connecting a plate in which the elastic suspension of the original model was clamped was also studied in their work, and no significant change in the value of natural frequency of the vertical rigid body mode shape (heave) was found.

Link and Friswell<sup>2</sup> analyzed the same structure in a benchmark study to compare different computational model updating procedures and to evaluate the validity of non-uniqueness of the results due to different measurements techniques, parameter settings, computational methods etc, considering some modifications to the original structure by mass addition. Results from the University of Wales, the University of Manchester, DLR and the Imperial College of Science, Technology and Medicine were compared. Firstly, comparisons of results from the unmodified structure were made in Link and Friswell's work and the agreement between the MAC values was very good (above 90%, on average). Modifications were introduced by adding masses to the tail and wing tips and these modifications significantly changed the dynamical behavior of the structure in terms of natural frequencies and mode order.

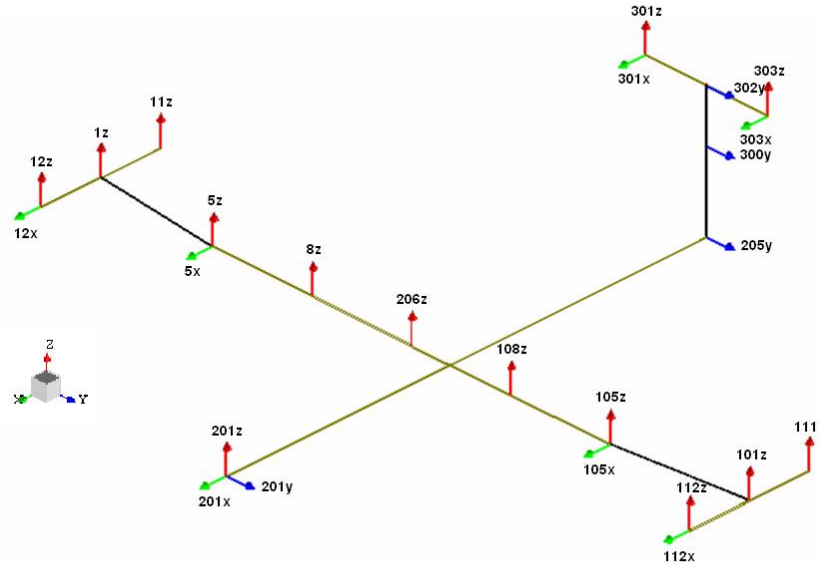
The participants produced different finite element models by using beams and plates and by changing the updating methodology. The aim was that the updated model could correctly predict the structural modifications of the physical structure when the masses were added to the wing tip or tail. MAC calculations and the value of the natural frequencies showed non-negligible differences, but they were taken as acceptable, since particular applications may not be significantly affected by a frequency deviation at high order modes. The general conclusion of Link and Friswell's work was that for a numerical model to be highly accurate and sensitive to updating, it is of paramount importance that sources of possible inconsistencies, such as modeling of the structural connections, need to be correctly described. In addition, the benchmark study paved the way for future works regarding practical applications for the updating process.

## III. TESTBED AND SETUP CHARACTERISTICS

The testbed used in the Round Robin work performed by the Structures and Materials Action Group of GARTEUR<sup>1,2</sup> was designed to have a group of three very closely spaced modes in order to bring some difficulty to the modal identification task. Due to the inherently low damping factor of the aluminum structure, a layer of a viscoelastic material was bonded to the upper surface of the wing in the original model. A shear treatment was also added by bonding a strip of aluminum of 1.1mm x 76.2mm x 1700.0mm on the top of the viscoelastic material.

The testbed for the present work was built following GARTEUR's model breakthrough, dimensions and general specifications, but neither the viscoelastic material nor the aluminum shear treatment were added, making the overall mass 41kg, instead of 44kg. The accelerometers and load cells were placed according to the locations specified in Fig. 1.

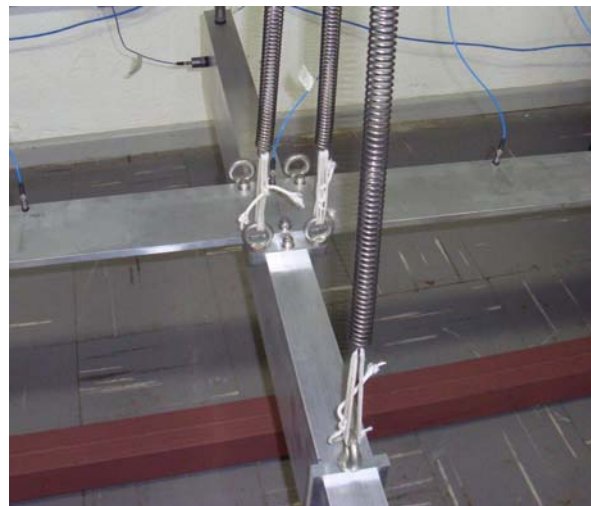
All measurements, data treatment, and analysis throughout this work were performed via SCADAS<sup>®</sup> III hardware and Test.Lab<sup>®</sup> software<sup>10</sup>. In particular, the Spectral Acquisition routine was extensively employed to provide a straightforward analysis of the modal parameters of the system.



**Figure 1. Location of the accelerometers and measurement directions.**

#### IV. GVT OF THE STRUCTURE

The structure was suspended by a set of three springs near to the CG, as shown in Fig. 2. The influence of rigid body modes on flexible modes was eliminated by the carefully choosing the suspension layout and dynamic characteristics. The excitation was made at points 12Z and 112Z for sweep sine and random signals via an electrodynamic shaker. The optimum amplitude of the excitation signal had to be determined in the test planning phase due to the low energy output of the shaker. Load cells and accelerometers were ICP<sup>®</sup> transducers from PCB<sup>®</sup> Piezotronics.



**Figure 2. Detail of the suspension system.**

The GVT included acquisition, post-processing and analysis of acquired data in the Test.Lab<sup>®</sup> environment. Each FRF was obtained after an average of 40 measurements in an attempt to reduce the presence of noise. Modes with closely spaced frequency values were detected independently of the excitation signal.

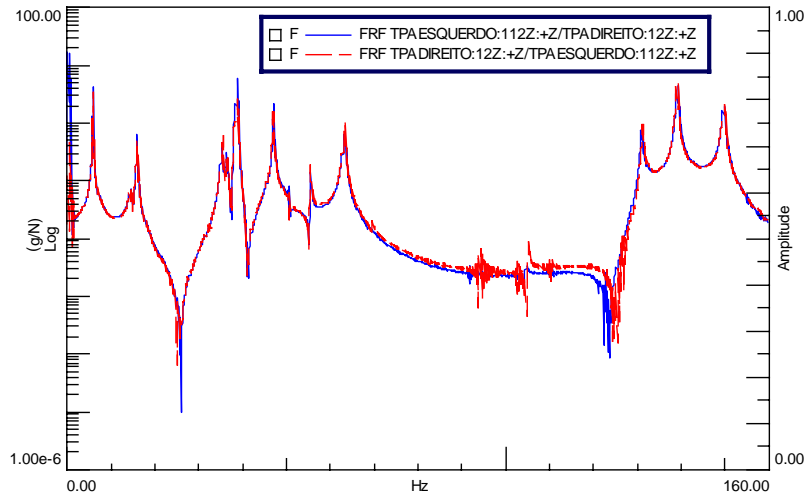
In order to achieve a modal model composed of modes as much orthogonal as possible, a modal validation process was carried on, leading to the deletion of some modes with small participation factors and notoriously non orthogonal. Figure 3 shows two cross FRF obtained after sweep sine and random excitation. Very good reciprocity was achieved between measurements.

Modal analysis was performed with the polyMAX<sup>®</sup> routine of the LMS Tes.Lab<sup>®</sup>, which in fact is an evolution of the least-squares complex frequency domain estimation method<sup>10</sup>. To construct the modal model, all possible modes indicated by a stabilized vector in the polyMAX<sup>®</sup> were first considered. Then, a MAC calculation was performed and cross-checked with the mode over complexity values (MOV) available at the Modal Validation routine of the software<sup>10</sup>. This quantity is defined as the weighted percentage of the response stations for which a mass addition decreases the natural frequency for a specific mode, and reads:

$$MOV_k = \frac{\sum_{i=0}^{N_0} w_i a_{ik}}{\sum_{i=0}^{N_0} w_i} \times 100\% \quad (1)$$

where:

$w_i$  is the weighting factor;  $w_i = 1$  for non-weighted calculations and  $w_i = |v_{ik}|^2$  for weighted calculations, with  $v_{ik}$  being the mode shape coefficient at response DOF  $i$  of mode  $k$ . If the  $k$  frequency sensitivity to a mass addition at point  $i$  is negative,  $a_{ik} = 1$ , otherwise  $a_{ik} = 0$ . For high quality modes the MOV is near 100%. If it is low, the considered mode shape vector is either computational or wrongly estimated, being called “over complex”. In this case, the phase angle of some modal coefficients exceeds a reasonable limit.

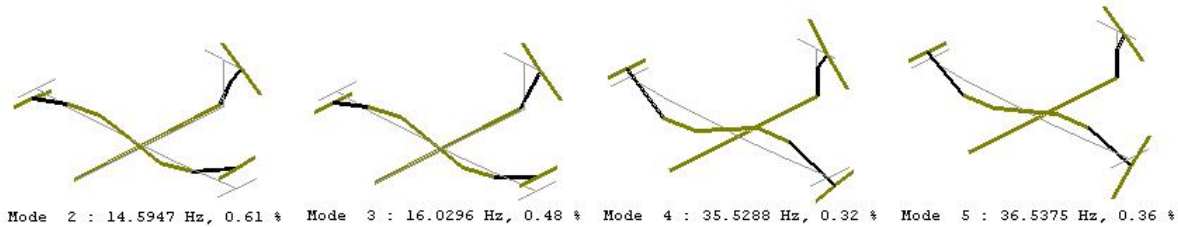


**Figure 3. Cross FRF. Blue line: sweep sine excitation at left drum (112Z). Red line: Random excitation at right drum (12Z).**

During the validation process for each modal analysis, some modes had MOV lower than 3%. Possible reasons for this can be poor excitation due to stinger misalignment or even the low energy output of the shaker. Following the validation process, it became clear that some modes given as almost purely mathematical were actually physical, a feature clearly seen when dealing with the pure torsion of the vertical tail, which MOV stayed below 2%.

The first set of measurements was made after sweep sine excitation at point 112Z. The groups of closely spaced frequencies occurred for modes 2 and 3, 4 and 5, and 6 and 7, as shown in Table 1. The MOV calculations suggested

that modes 2 and 7 could not be physical, so an auto-MAC check was performed for the 16 possible modes, revealing that modes 2 and 3, as well as modes 4 and 5 were highly linearly dependent, whereas modes 6 and 7 were orthogonal (Fig. 5). By observing the deformed shapes of those modes in Fig. 4, it becomes clear that mode 2 (at 14.49Hz) has the same deformed shape as mode 3 (at 16.02Hz), differing only in a slight bending of the vertical tail and in the amplitude of the deflection of the left wing drum. For modes 4 (at 35.53Hz) and 5 (at 36.54Hz) the difference between them is basically a torsion of the left wing drum.



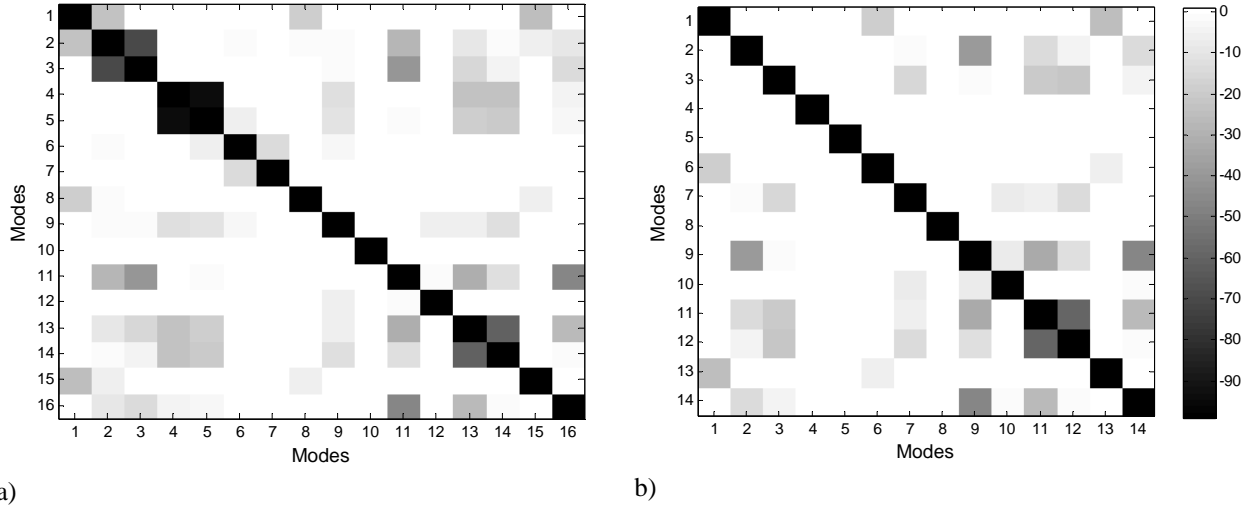
**Figure 4. Deformed shapes for modes 2-5 of Table 1**

Possible explanations for the deviations in the deformed shapes presented in Fig. 4 are improper compensation for the moving mass of the shaker or even the low energy output of the excitation, which may have caused the local torsion of the wing drum more important than the complete mode itself. Excitation by random signal with higher voltage levels caused modes 4 and 5 to have exactly the same deformed shapes and MAC values of 98.94%.

**Table 1. Comparison between modal indicators for sweep sine excitation at the left wing drum (112Z) before and after deletion of modes at 14,49Hz and 36,46Hz.**

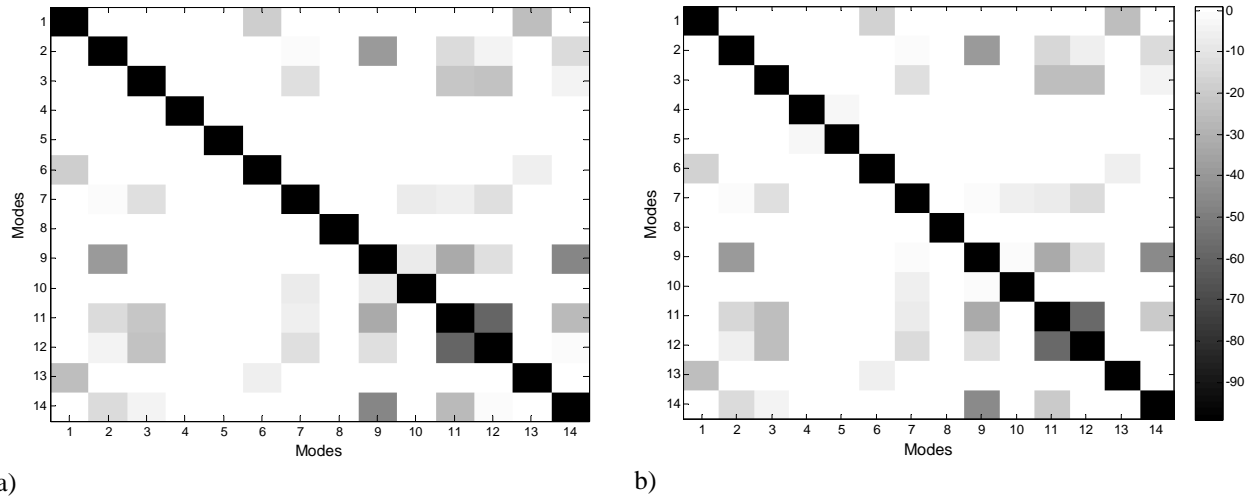
Mode number before deletion	Frequency (Hz)	Before deletion of modes 2 and 5		After deletion of modes 2 and 5		Mode number after deletion
		MOV (%)	MP (%)	MOV (%)	MP (%)	
1	6.00	99.61	53.47	99.36	54.41	1
2	14.49	46.44	1.39	-	-	
3	16.02	99.77	9.28	99.74	9.48	2
4	35.55	99.95	3.47	99.94	3.53	3
5	36.46	95.87	0.33	-	-	
6	38.26	100.00	20.90	100.00	21.25	4
7	38.95	28.75	1.62	31.41	1.65	5
8	47.09	99.97	3.30	99.98	3.35	6
9	50.82	92.19	0.20	92.14	0.20	7
10	55.69	99.97	0.49	99.96	0.50	8
11	63.35	100.00	2.05	100.00	2.08	9
12	68.97	1.32	0.10	1.39	0.10	10
13	105.18	13.38	0.07	13.08	0.07	11
14	131.44	99.99	0.88	99.99	0.90	12
15	139.20	99.97	1.27	99.96	1.29	13
16	149.94	100.00	1.15	100.00	1.17	14

If the mode participation factor is considered for modes 2 to 5 (left hand side of Table 1), along with what was discussed on mode shapes, deletion of modes 2 and 5 can be done. The right hand side of Table 1 shows the results for the modal validation of the new 14 modes model. The MOV indices have not changed significantly, as well as the MP factors, but auto-MAC values reveal very good orthogonality up to the mode at 55.69Hz (fore-and-aft bending of the wing) after deletion of modes at 14.94Hz and 36.46Hz.



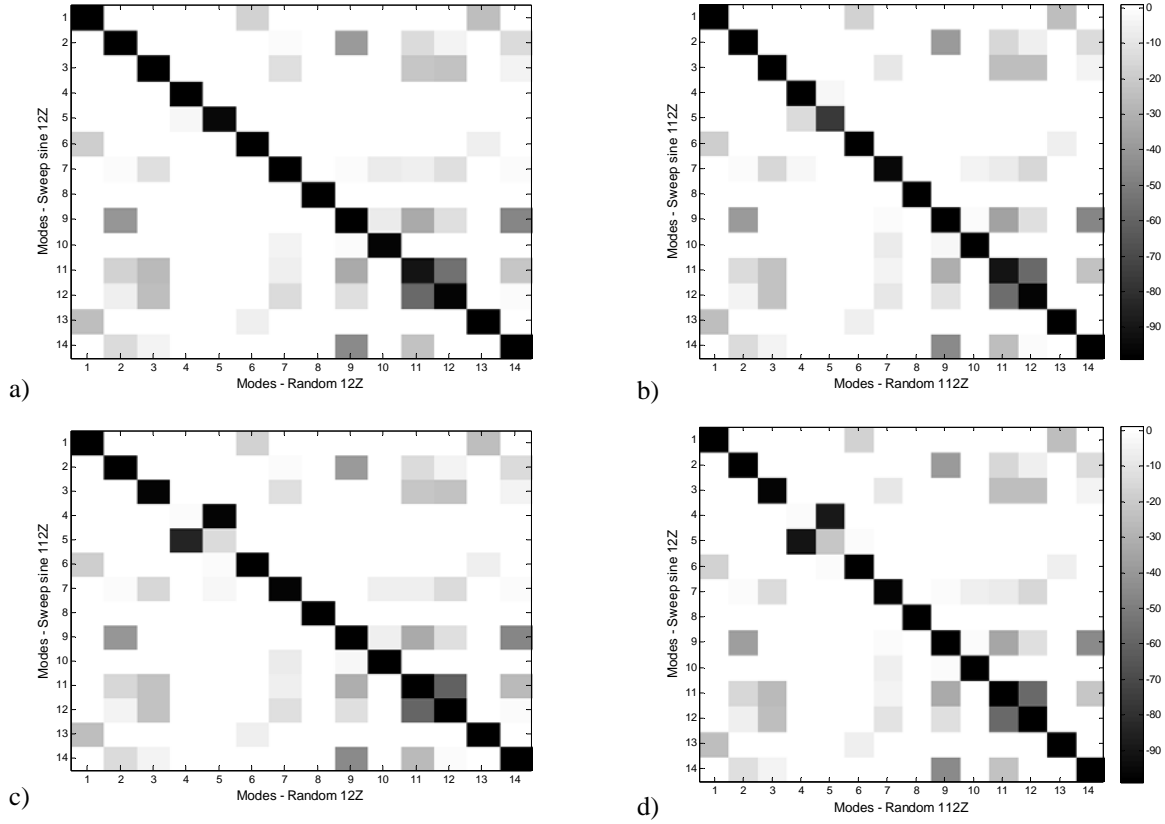
**Figure 5. Auto-MAC plot for the two modal models obtained by sweep sine excitation at 112Z: a) complete set of possible modes; b) modes at 14.49Hz and 36.46Hz deleted.**

The same procedure described above was used for all the remaining modal analysis. Figure 6 shows the auto-MAC matrices for the modal model obtained by excitation with random and sweep sine signals at point 12Z. Excitation with sweep sine produced better MAC values, as can be seen in Fig. 6 with emphasis on modes 9 to 12 (see right hand side of Table 1).



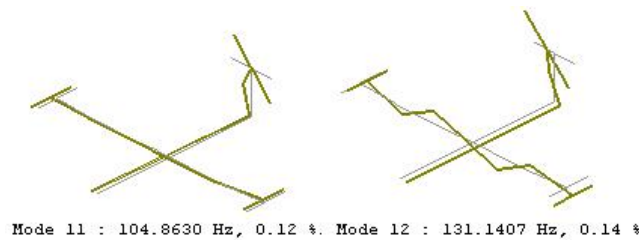
**Figure 6. Auto-MAC plot for the modal models obtained by random and sweep sine excitation at 12Z: a) random excitation; b) sweep sine excitation.**

In order to have a modal model as much accurate as possible, regarding a structure with very low damping ratios and using a shaker with low energy output, it became necessary to compare the results of the modal analysis obtained by input signals of different characteristics (random and sweep sine) applied at different reference points (left and right wing drum). Each modal analysis comprised its own modal validation to produce 14 normal modes for the final comparisons via MAC calculations.



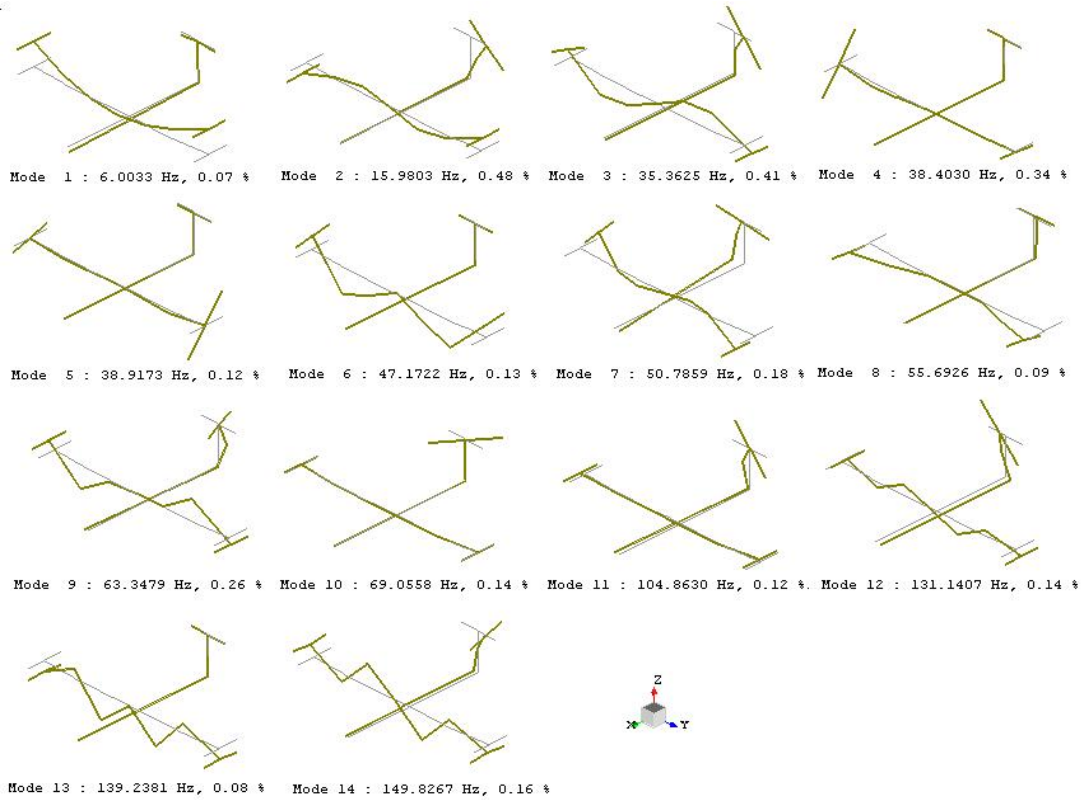
**Figure 7. MAC calculations between modal models: a) sweep sine 12Z x random 12z; b) sweep sine 112Z x random 112Z; c) sweep sine 112Z x random 12Z; d) sweep sine 12Z x random 112Z.**

Figure 7 shows the MAC matrices between all modal models. If models whose excitation was made at the same location are compared, the MAC matrix show very good correlation between them, remaining some linear dependency from mode 9 to mode 12, a feature already present in the auto-MAC matrices of Fig. 5 and Fig. 6. In fact, if the deformed shapes of modes 11 (at 105.86Hz) and 12 (at 131.14Hz) is studied, as shown in Fig. 8, it can be seen that they are distinct, although the MAC value between them is 35.04%, suggesting a high degree of linear dependency of the mode vectors. Link and Friswell<sup>2</sup> published similar results concerning those modes, but the mode orders and frequencies in their work were mode 11 at 102.9Hz and mode 14 at 151.3Hz, as presented in Table 1.



**Figure 8. Deformed mode shapes for modes 11 and 12.**

Another source of low MAC values is mode 9, at 63.35Hz. This combined wing-and-vertical-tail bending was also present in Link and Friswell's<sup>2</sup> work as mode 9 at 63.00Hz and in Balmès and Wright<sup>1</sup> as mode 9 at 63.04Hz. It has a participation factor of 2.08% for the sweep sine excitation at point 112Z and a MOV of 100%, but the MAC values of adjacent modes (10, 11, 12 and 14) lie between 9% and 48%.



**Figure 9. Mode shapes for the modal model.**

A comparison between the results obtained in this work with those from Balmès and Wright's<sup>1</sup>, and Link and Friswell's<sup>2</sup> can now be done. Mass modifications and viscoelastic treatment were not employed in this work, thus only mode number and natural frequencies were compared, as shown in Table 2, regarding similarity between mode shapes. Each line of Table 2 corresponds to equivalent modes in each work.

**Table 2. Comparison of mode numbers and frequencies between the present work, Balmès and Wright<sup>1</sup>, and Link and Friswell<sup>2</sup>.**

Rett et al.		Balmès and Wright <sup>1</sup>		Link and Friswell <sup>2</sup>	
Mode no.	Frequency (Hz)	Mode no.	Frequency (Hz)	Mode no.	Frequency (Hz)
1	6.00	1	6.38	1	6.40
2	16.02	2	16.10	2	16.10
3	35.55	5	35.65	5	35.70
4	38.26	3	33.13	3	33.10
5	38.95	4	33.53	4	33.50
6	47.09	6	48.38	6	48.40
7	50.82	7	49.43	7	49.40
8	55.69	8	55.08	8	55.10
9	63.35	9	63.04	9	63.00
10	68.97	-	-	10	66.50
11	105.18	-	-	11	102.90
12	131.44	-	-	12	130.50
13	139.20	-	-	13	141.40
14	149.94	-	-	14	151.30



## V. FINITE ELEMENT MODEL

A finite element analysis of the testbed was performed using the ABAQUS® software. The testbed structure was discretized using four-nodes shell elements (S4R) available in the finite element code. As a simplification method, the model was analyzed as a single piece, with perfect unions, ignoring the holes, joints and screws, so the mesh of the simplified model would need a smaller number of elements. The material used was aluminum, with density of 2780 kg/m<sup>3</sup> and modulus of elasticity 70 GPa. A structured mesh was created using 1586 elements, with variable element size depending on the section considered. In the central body of the structure, the mesh refinement is lower due to the low relevance of this component in the characterization of the frequencies and mode shapes, whereas critical sections such as wing and tail have a higher mesh refinement. Figure 10 shows the mode shapes and frequencies obtained from the finite element analysis.

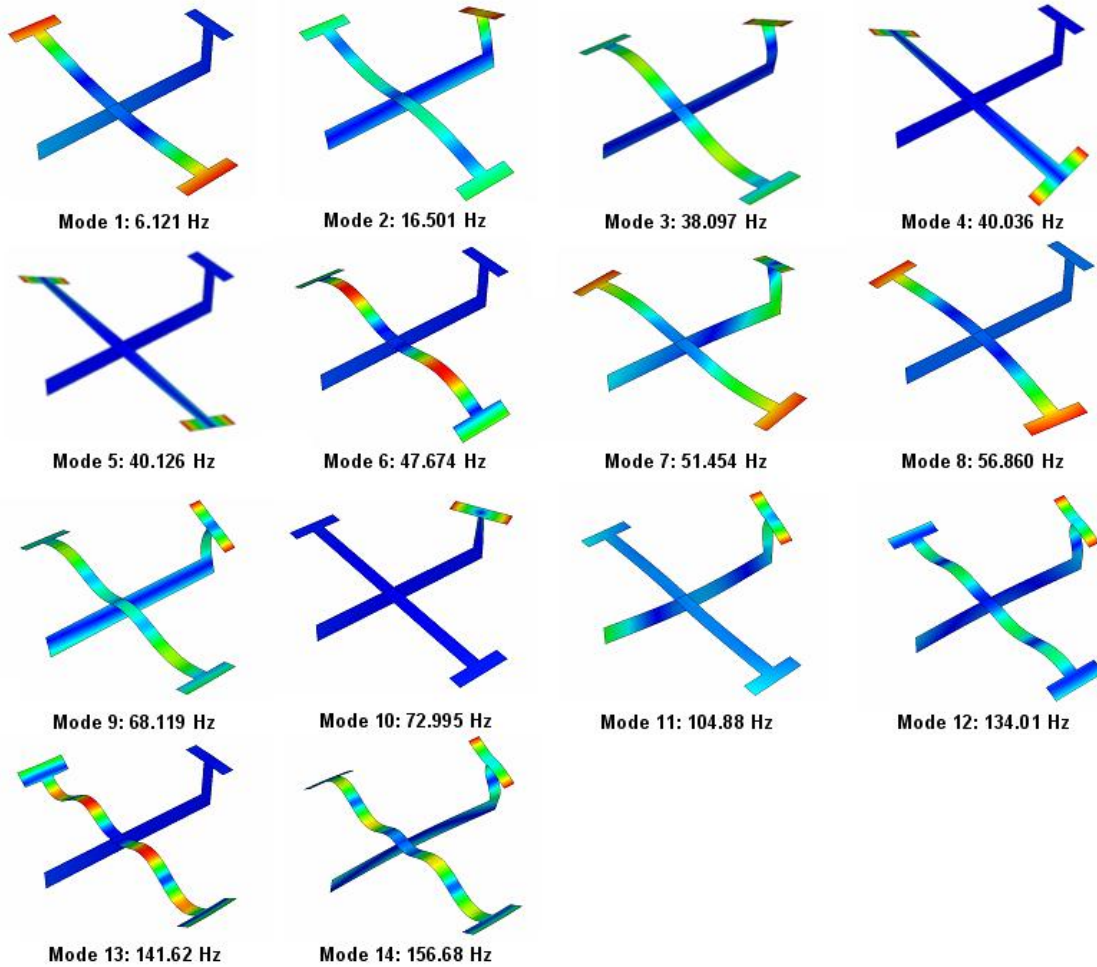


Figure 10. Mode shapes and frequencies for the GARTEUR SM-AG19 testbed. Finite element model, standard shell element S4R structured mesh, 1586 elements.

## VI. VALIDATION OF THE EXPERIMENTAL RESULTS

The results were correlated by considering the values of the natural frequencies and by comparing the mode shape produced by the finite element model (Fig. 10). Table 3 presents the results of the comparison. It can be seen that the differences in the values of the natural frequencies are lower than 10% and, in fact, only modes 3, 9 and 10 exceed 5%. Mode shapes are exactly the same.

**Table 3 – Comparison between natural frequencies of the experimental and numerical models.**

Mode	Experimental Frequency (Hz)	Numerical Frequency (Hz)	Error (%)
1	5,99	6,12	2,05
2	16,02	16,50	2,98
3	35,55	38,09	7,16
4	38,27	40,04	4,61
5	38,95	40,13	3,01
6	47,09	47,67	1,23
7	50,82	51,45	1,23
8	55,69	56,86	2,08
9	63,35	68,12	7,52
10	68,98	72,99	5,82
11	105,18	104,88	0,28
12	131,44	134,01	1,95
13	139,20	141,62	1,73
14	149,94	156,68	4,49

## VII. CONCLUSIONS

This paper presented an experimental investigation on the dynamic behavior of the unmodified testbed proposed by the GARTEUR group, validated by a simplified finite element model. In spite of being simple and straightforward, the methodology used proved to be capable of producing a modal model reliable and accurate, allowing for the identification of the closely spaced modes, independently of the excitation point or method.

From the numerical point of view, a finite element model was implemented using shell elements for the analysis of the testbed, giving satisfactory results.

Future works can take advantage of active constrained-layer damping treatments to control the amplitudes of modes of interest.

## ACKNOWLEDGMENTS

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