DYNAMIC MECHANICAL ANALYSIS OF AN AIRCRAFT WING WITH EMPHASIS ON VIBRATION MODES CHANGE WITH LOADING

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Abstract: *This paper presents the results of a study of the dynamical behavior on an aircraft wing structure. The study has consisted in a set of dynamical tests in which the vibration parameters of the structure have been determined. The aim of this research is dual: to study the viability of performing vibration testing with strain gages instead of accelerometers and to determine the change in the dynamic behavior of the structure when loading is applied. The report presents the conclusions derived from the analysis of the generated data correlating the results with the available structural theory.*

Keywords: *dynamic testing, strain gage, ground vibration test, modal analysis, jump phenomena*

1. INTRODUCTION

Dynamic testing is an extended practice in aviation certification whenever any changes are performed on an aircraft that change its structural behavior. These tests are usually referred to as Ground Vibration Testing or GVT.

The purpose of GVT is to validate and improve the structural dynamic models. This is performed through the experimental determination of the low-frequency vibration modes. These models are later used to predict dynamic aero-elastic behavior and plan the safety-critical flight testing phase. [2, 7]

Basically, two techniques are used to determine the vibration modes of a structure. A first technique is the phase-resonance method; in this approach a sine force is applied through one or multiple shakers at a single frequency in order to excite the structure at each of its natural frequencies at a time. When

such condition is met, the structure acts like a single degree of freedom (DOF) system. [8]

Another, newer, approach is to use phase-separation techniques; these methods allow for the extraction of the different frequency response functions (FRF) of a structure simultaneously while using random excitation in one or more shakers. [10]

Because of the extra complexity that the phase-separation techniques present and provided that the sufficient testing time has been available, the method used to study the structure has been the phase-resonance method using one shaker. This allowed for increased precision in the results with lower post-processing complexity.

It is one of the main focuses of the article to determine if the results obtained through GVT can be obtained with the use of strain gages instead of accelerometers. This presents an advantage provided that the use of gages is

more economical. Such application of strain gages has been considered in [4].

Moreover, the aim of the study is to determine the change in the dynamic behavior of the structure as loading is applied, intending to simulate the real operating conditions of the aircraft. This analysis could be relevant in further study of aero-elastic phenomena. The available bibliography predicts an increase in frequency in the vertical vibration modes and a decrease in the torsion modes. [1] The study of the lagging modes is not in the scope of this research because they do not significantly influence the aero-elastic effects.

2. EQUIPMENT USED AND TEST CONFIGURATION

During the experiments performed on the studied wing, two acquisition systems have been used to monitor the dynamic behavior of the structure. The systems used have been the Prodera installation and the HBM installation.

2.1 Prodera installation. The Prodera installation is the modal analysis equipment that uses 16 unidirectional accelerometers simultaneously to determine the vibration parameters of a structure. Its software allows modeling the experiment configuration in order to provide graphical representation during or after each test. This system features also post-processing tools that allows calculating the parameters of a vibration mode using the phase-resonance method.

2.2 HBM installation. Parallel to the Prodera installation, another acquisition system has been used in order to provide redundancy in the measures to validate the results obtained with Prodera; and also to monitor the dynamic tests using the strain gages embedded in the structure. The HBM installation has been used with 3 accelerometers of type B&K model 4507B005 and 34 strain gauges.

2.3 Test specimen. The studied structure is an IAR-99 aircraft wing structure. The specimen has been tested without control surfaces installed and with the lower surface inspection hatches removed. No pods for external equipment have been mounted at the time of the test either. The structure has been tested vertically and excited with a 200N shaker mounted in the horizontal direction (vertical in relation to the wing).

Despite the position of the wing during testing, the vertical direction for the wing is considered as if the wing was mounted on an aircraft horizontally. The vertical modes refer to the flapping modes of the wing.

The wing has been tested by clamping the wing-fuselage attachments to a steel base on the ground that can be considered rigid.

The static loading has been applied using a spring. This allowed for the loading of the structure with the minimum influence on the vibration modes because the mechanism is flexible. A drawing of the tested model is shown in Figure 1.
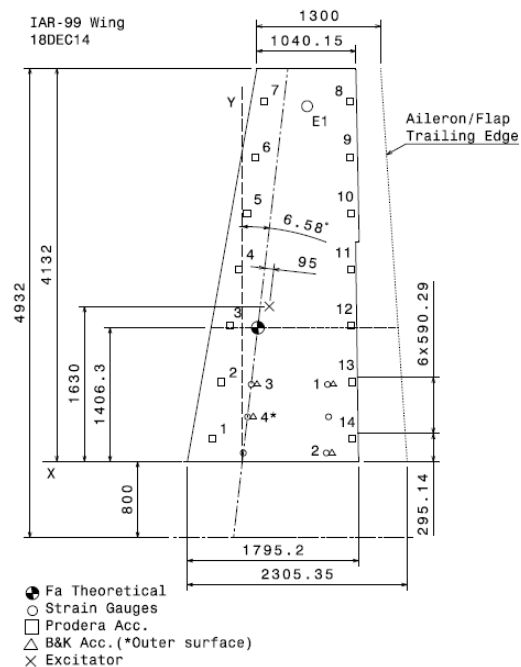

Figure 1. Design of the tested wing and location of the analyzed sensors.

2.4 Sensor locations. The distribution of the accelerometers is shown in Figure 1. The 16 accelerometers in the Prodera installation are located on the inner surface of the wing at equal distance from the wing root to determine the different mode shapes of the structure. The accelerometers linked to the spider acquisition system (HBM) are located both in the inner and outer surfaces; the gages are installed in the outer and inner surfaces and also in some internal structural elements.

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3. EXPERIMENTAL RESULTS

Different dynamic testing has been performed on the wing structure in order to determine the vibration parameters for the different modes. This testing is presented divided according to the loading of the wing at each test. Two test cases have been considered to establish a correlation between the modal frequencies and the loading of the wing.

The results have been post-processed using respectively Prodera and HBM Catman software [5] in order to determine mode shape, frequency, damping factor and generalised mass for each vibration mode. In order to calculate these parameters the complex power method has been used in Prodera [6].

3.1 Tests without loading. A first set of tests has been performed without any loading on the wing in order to determine the vibration modes of the structure as a reference for further analysis. This has been the most extensive loading case in terms of number of tests because the different vibration modes have been analyzed. The viability of using strain gages in dynamic analysis is studied with no loading. The average results for the tests without loading are presented in Table 1.

Table 1 Average testing results obtained without loading.

	1st mode	2nd mode	3rd mode	4th mode
Type	Bending	Bending	Torsion	n/a
Freq.	9.61Hz	47.11Hz	50.27Hz	98.72Hz
γ	0.0233	0.0072	0.010	n/a
$\tilde{\mu}$	746kgm^2	431kgm^2	288kgm^2	n/a

3.2 Tests at 1.5kN loading. The maximum loading that has been applied to the structure has been the operational limit of the spring used. The obtained results are shown in Table 2.

Table 2 Average testing results for the loading case at 1.5kN.

	1st mode	2nd mode	3rd mode	4th mode
Type	Bending	Bending	Torsion	n/a
Freq.	11.25Hz	49.49Hz	50.81Hz	n/a
γ	0.0169	0.0065	n/a	n/a
$\tilde{\mu}$	4819kgm^2	380kgm^2	n/a	n/a

During the post-process of the results with Prodera it has been possible to detect errors in the calculation of parameters due to the wrong interpolation of the results. In order to correct the modal parameters an software has been developed that interpolates the results and recalculates the modal parameters using the formulation used in [9] shown below:

$$\gamma_i = -\left.\frac{dS}{d\omega}\right|_{\omega=\omega_i} \tag{1}$$

$$\mu_i = \frac{\gamma_i}{\omega_i^2} \tag{2}$$

$$\zeta_i = \frac{R(\omega_i)}{\omega_i \gamma_i} \tag{3}$$

Where γ is the generalized stiffness, μ the generalized mass, ζ the damping factor, ω the pulsation, $R(\omega)$ and $S(\omega)$ the real and imaginary parts of the complex power and ω_i the ith resonance frequency. The normalized generalized parameters can then be obtained with the amplitude of the reference transducer A_n (located close to the excitation point).

$$\tilde{\mu} = \frac{\mu}{A_n^2} \tag{4}$$

$$\tilde{\gamma} = \frac{\gamma}{A_n^2} \tag{5}$$

These results have been later compared with Prodera to analyze the dependence of the calculated modal parameters with the interpolation method used in the complex power method.

4. CRITICAL ANALYSIS

The analysis of the results according to the focus of this article is presented in this section. The emphasis is put in the determination of the modal parameters of the different vibration modes for the studied wing. These parameters are modal frequency, damping factor, mode shape and generalized mass. The analysis is performed using the results presented before for the different load cases.

4.1 Vibration modes. It has been possible to identify 4 vibration modes in the range of frequencies from 0 to 100Hz. These modes have been 2 flapping (or vertical) modes at 9.61 and 47.11Hz; and 1 torsion mode at 50.27Hz. A 4th mode has been detected at 98.72Hz but has not been properly characterized. This mode is outside of the frequency range of interest. Such results have been compared with the analysis performed in [3] for the same tests.

It has been possible to determine the modal shape through two analysis methods. The first method has used the HBM system to compare the signals of the available accelerometers; the phase analysis of the accelerometer response provided information on the way the structure vibrated. Figure 2 shows a signal analysis for a torsion mode characterized.

The second method used the code implemented in the Prodera installation to perform the same process through the 14 accelerometers connected to the system.

4.2 Calculated modal parameters. An estimation of the modal parameters has been possible through the Prodera installation. The system integrates post-processing software that calculates the modal parameters using the representation of the complex power in the vicinity of the studied natural frequency. This representation allows for the calculation of the slope of the imaginary power and the value of the real power at the resonance point. Through these values it is then possible to calculate the damping factor, the generalized mass and the generalized stiffness. The obtained results for these values are presented in Table 1 and in Table 2.
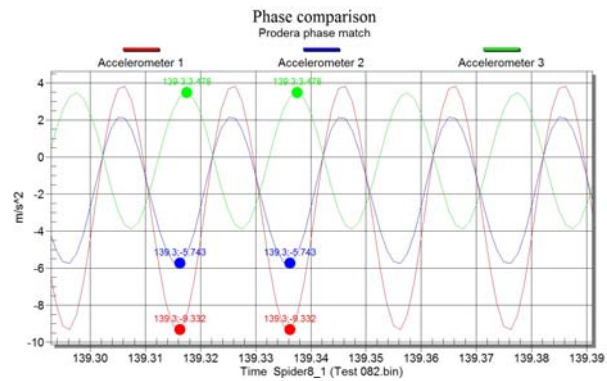

Figure 2. Signal analysis for a torsion mode in HBM Catman. Note the front accelerometer in anti-phase with the two accelerometers located in the trailing edge part of the wing.

The analysis of these values leads to the conclusion that the results obtained are coherent. The damping factor is a function that increases with frequency while the generalized mass decreases. The experimental values obtained follow this trend apart from specific anomalies.

4.3 Correlation between acceleration and deformation. The study of the acceleration deformation correlation has been performed using HBM software to compare the power spectrum of a representative accelerometer with a strain gage mounted close to the position of the accelerometer.
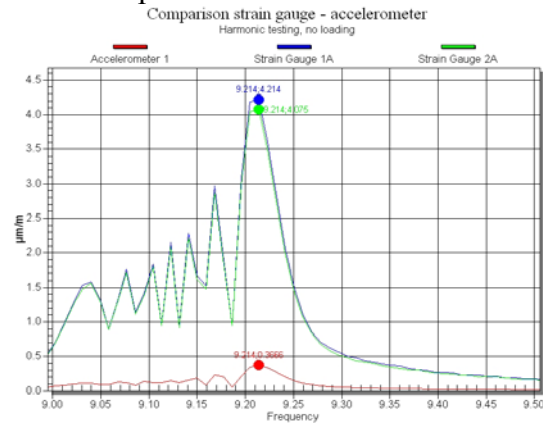

Figure 3. Frequency spectrum comparison for an accelerometer and 2 strain gages (No amplitude scale is provided for the accelerometer).

The obtained results, exemplified in Figure 3, show that there is a good correlation between the strain gage and the accelerometer results. A major drawback has been detected in the amplitude of the strains sensed. The average strains sensed have a value around 4 micro-strains for the performed tests. These

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values might not be big enough for a precise characterization of the structure vibration.

Table 3 Comparison between accelerometer and strain gage natural frequencies [Hz].

Mode	1st	2nd	3rd	4th
	No loading			
Acc.	9.61	47.11	50.27	98.72
Gage	10.11	47.77	50.75	n/a
Difference	5.2%	1.4%	1.0%	
	1.5kN loading			
Acc.	11.25	49.49	50.81	n/a
Gage	11.14	49.53	50.85	n/a
Difference	-1.0%	0.1%	0.1%	

Despite the precision problem for the strain gages it has been possible to determine the natural frequencies of the structure with the gages. The comparative results with the accelerations are presented in Table 3.

4.4 Effect of loading. The effect of loading in the structure has had the expected result. When a structure is loaded, the stress increases the stiffness of the structure; therefore, the expected vibration modes are expected to be found at a higher frequency. This has been corroborated in the experiments, as can be seen in Table 3, for all modes except for the torsion one. In the torsion mode, the variation of frequency is too small to determine whether the recorded difference is caused by loading or by the precision of the instruments.

4.5 Structural non-linarites detected. During the testing phase it has been possible to identify instability in the first bending mode that led to believe that a non-linearity was involved. The effect manifested as a sudden change of vibration amplitude close to the mode frequency during harmonic testing. This sudden change occurred without noticeable external cause and it changed its behavior depending on whether the test was performed increasing frequency or decreasing it.
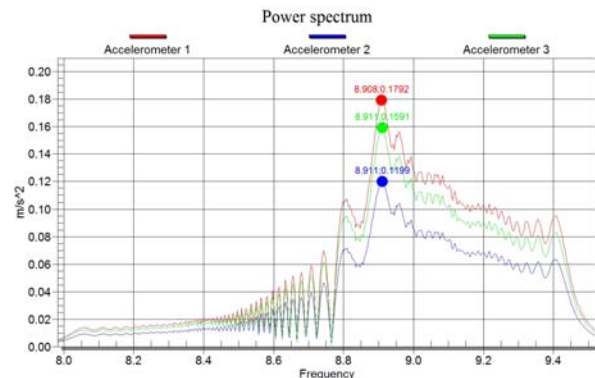

Figure 4. Behavior of the detected non-linearity for the first bending mode during a harmonic test with increasing frequency.

Figure 4 illustrates how the acceleration recorded during the test suddenly steps up when passing close to the calculated resonance frequency (around 8,9Hz). This behavior is known as jump phenomenon and it occurs in softening or hardening springs. In these cases, the amplitude - frequency diagram of the system presents a region of instability in which two possible dynamic states coexist; depending on whether the analysis is conducted increasing or decreasing frequency, the amplitude will be different up to a point in which the system brusquely changes its vibration characteristics. This jump usually occurs at different frequencies depending on the direction of the test; these two frequencies define the instability region. An example of the jump phenomena detected during testing is presented in Figure 5 for the first bending mode.
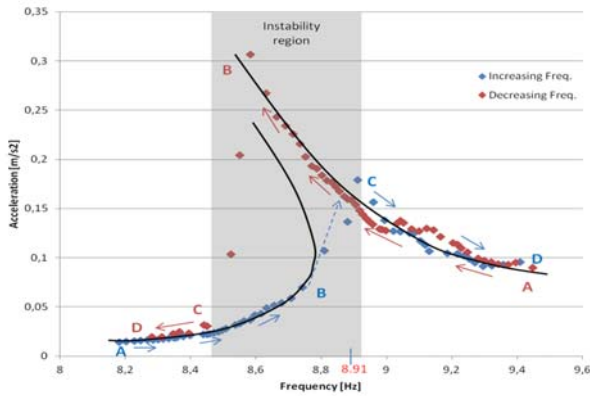
Figure 5. Jump phenomena depicted for the first bending mode using data from accelerometer 1.

According to the definitions found in the bibliography, the jump phenomenon found in the first bending mode is of softening spring type.

5. CONCLUSIONS & ACKNOWLEDGEMENTS

Some conclusions can be drawn from the analysis presented. Firstly, the results obtained using the frequency response function in the Prodera installation have been slightly lower than the results obtained with the power spectrum performed with the HBM system. This difference is around 4%.

The calculations performed parallel to Prodera to determine the modal parameters led to the following results. The difference obtained for the natural frequencies are up to 5%; for the damping factor the maximum difference is around 5%; the normalized generalized mass presents a maximum difference of 23%.

These results have been calculated with an accelerometer close to the excitation point but not in the exact point of excitation. This fact could explain the difference in the normalized generalized mass, very influenced by the amplitude of the excitation point. The utilization of an accelerometer in the excitation position should improve the results.

The use of strain gages is enough to study the vibration analysis if the precision of the gage is good enough and if the natural frequency is the only required parameter.

The recorded evolution with the loading of the damping factor and the natural frequency has been coherent with the expected behavior, with a variation of around 16%.

Non-linearity can be observed in harmonic testing if the frequency is slowly changed in the affected modes (0,004Hz/s).

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