Design of A Ground Vibration Test Certification System For Unmanned Air Vehicles

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Acknowledgments

Writing is said to reflect one’s dedication on the words on the paper. In this case it reflects my journey and hard work in this phase of my life and studies. But I could not have done this alone for I have to say that I am lucky to have by my side my family and friends who have supported me along the way.

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Abstract

The characterization of the structural dynamics of flexible aircraft is an important aspect in the aircraft design process. The experimental Ground Vibration Tests (GVT) are an integral part of the evaluation and validation step in order to ensure structural integrity, enable structural certification and promote flight safety. Experimentation also plays a vital role in design, especially when it is properly integrated with analytical and computational design processes. Experimentation serves two important functions in such design activities. The first is to obtain measured data with which the accuracy of theoretical predictions is checked and the second is to check for the completeness of the dynamic model.

Towards this end, an experimental GVT equipment was setup and modal analysis has been performed to obtain system frequencies and mode shapes for validation, evaluation and update of the computational models developed for an Unmanned Air Vehicle Aeroservoelastic Demonstrator. Impact testing and shaker testing on two different sets of wings (rigid and flexible) were carried out. The results obtained were compared with the computational results and good agreement was achieved.

Keywords

1. Experimental Modal Analysis
2. Ground Vibration Testing
3. Finite Element Model Updating
4. Modal Testing
5. Modal Parameter Estimation
Resumo

Nas últimas décadas, surgiu uma tendência de definir cada aspecto das estruturas que nos circun- dam. A previsão do seu comportamento em cada situação operacional é necessária para que possamos ter um ambiente circundante mais seguro e no qual possam confiar. Um aspecto importante sempre presente é a integridade estrutural do que fabricamos. Numa perspectiva mais específica, podemos analisar os modos de vibração de estruturas.

Em aeronaves este problema tem um impacto ainda maior. É preciso ter aeronaves de confiança que consigam suportar todos os tipos de vibração impostos e há um estudo contínuo desses aspectos.

Para fazer isso, a análise modal experimental tem sido um aspecto muito importante no processo de certificação de aeronaves. Permite validar e actualizar os modelos computacionais necessários para ter permissão de voo e para poderem ser feitas as análises estruturais e aeroelásticas finais.

Nesta tese, foram feitos testes de vibração em solo num UAV de pequenas dimensões de maneira a adquirir dados modais críticos necessários para aactualização de modelos de elementos finitos. Foram feitos testes de impacto e com o uso de vibradores em duas configurações de asas diferentes, para permissão de voo nos dois casos. É o objectivo deste trabalho apresentar todo o processo e resultados dos testes feitos e avaliar ao mesmo tempo se os procedimentos usados podem ser ou não aplicados não só a modos lineares mas também a não lineares.

Palavras Chave

1. Análise Modal Experimental
2. Testes de Vibração em Solo
3. Actualização de Modelos de Elementos Finitos
4. Testes Modais
5. Estimacão de Parâmetros Modais
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Abbreviations

DFT  Discrete Fourier Transform
DOF  Degree of Freedom
EMA  Experimental Modal Analysis
FEA  Finite Element Analysis
FEM  Finite Element Model
FFT  Fast Fourier Transform
FRF  Frequency Response Function
GUI  Graphical User Interface
GVT  Ground Vibration Testing
IRF  Impulse Response Function
LDV  Laser Doppler Vibrometer
MAC  Modal Assurance Criterion
MDOF Multiple Degree of Freedom or multiple mode method
MMU  Multi Model Updating
MPC  Modal Phase Collinearity
ODS  Operating Deflection Shape
PSM  Phase Separation Methods
PRM  Phase Resonance Method
PRH  Peak Reference Hold
SDOF Single Degree of Freedom or single mode method
SLDV Scanning Laser Doppler Vibrometer
List of Symbols

\( \Delta f \) Frequency bandwidth
\( \xi_r \) Viscous Damping Ratio of Mode \( r \)
\( \zeta_r \) Damping Ratio
\( \zeta_i \) Viscous Damping Ratio
\( \omega \) Natural Frequency Of the Undamped SDOF System
\( \omega_r \) Natural Frequency of Mode \( r \)
\( \omega_d \) Damped Natural Frequency Of the Undamped SDOF System
\( c_i \) Damping
\( c_{cr} \) Critical Damping
\( [C] \) Damping Matrix (force/unit of velocity) (n by n)
\( f_r \) Resonance Frequency
\( f(t) \) Excitation Force n-vector.
\( F[x(t)] = \) Fourier transform of the excitation force n-vector
\( [H(\omega)] \) Frequency Response Function (FRF) matrix (n by n)
\( [K] \) Stiffness Matrix (force/unit of displacement) (n by n)
n Number of Measured DOFs
\( m \) Number of Modes
\( m_i \) Modal Mass
\( [M] \) Mass matrix (force/unit of acceleration) (n by n)
\( r A_{jk} \) Residue of mode \( r \)
\( t \) Time (seconds)
\( x(t) \) Displacement Response n-vector.
\( \dot{x}(t) \) Velocity Response n-vector
\( \ddot{x}(t) \) Acceleration Response n-vector
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1.1 Introduction

Over time, methods and techniques have evolved to determine and evaluate structural response of complex engineering systems in order to ensure structural integrity and operational safety.

Structures present resonant frequencies, which means that, under the proper conditions and excitation, a structure can be made to vibrate with excessive, sustained, oscillatory motion. The resonance phenomenon is a vibration caused by an interaction between the inertial and elastic properties of the materials within a structure and an external force [1] and is one of the important design considerations in aircraft design, especially in aeroelastic and aeroservoelastic evaluation of flexible aircraft.

When analysing vibration problems, there are two types of vibrations: forced and free vibration. Every vibration phenomenon is a combination of the two. Forced vibration can be caused by:

- Internally generated forces
- Unbalances
- External loads
- Ambient excitation

On the other hand, resonant vibration occurs when one or more of the resonances or natural modes of vibration (of the structure in cause) is excited by an external force. What makes resonance dangerous is that when it happens it can amplify the vibration response to higher levels, causing consequently higher levels of deflection, stress and strain than the ones caused by static loading. In other words, at a specific resonance frequency, a very small input of force is amplified causing a very large response. Another characteristic of dynamic systems is anti-resonances, where it works in the opposite way of the resonance: an input, even if it is large at an anti-resonance frequency, it will have a very weak response.

Resonances, often called modes, are properties that exist for every structural system. Vibration problems in structures, in particular in many types of machinery and equipment, are often associated with resonance related issues during operation. Therefore, there is a need to study these cases and know how to prevent them. When analysing these phenomenon, the resonances are determined by:

- The material proprieties:
  - Mass
  - Stiffness
  - Damping properties

- Boundary conditions of the structure

Each mode is defined by its modal parameters:

- Modal frequency
- Modal damping
• Mode shape

It should be noted that if any of the properties of the structure change, i.e., the material properties and boundary conditions, so will the modes. Structural modes can be further classified as rigid body and free body modes. Each structure can show up to six rigid body modes, considering that it can have three translation modes and three rotational modes, i.e., one for each axis movement, either along it or rotating with that axis. Mode shapes of flexible structures often are a combination of axial, bending and torsion modes. The names are self-explanatory and modes are commonly enumerated as they show bending, torsion, axial behaviour or a combination of them.

Modal analyses is very important in the process of designing a new aircraft for structural certification. Regarding the design process, there are some fundamental steps that need to be performed for evaluation and validation of flexible aerostructures in order to have a reliable and safe aircraft:

• Modelling the Finite Element Model;

• Run a First Modal Analysis;

• Perform an Experimental Modal Analysis;

• Validate the Results;

• Update the Model (if necessary, in order to be able to have flight clearance of the aircraft).

The first step in this design process involves modelling the new aircraft using numerical techniques such as finite element methods. This model describes the dynamic properties of the aircraft, as it defines the mass and stiffness of the parts of the structure and some considerations on the damping properties. Upon completion of the modelling phase, the model can be used to predict not only the vibration properties of the structure, by determining its modal properties, but also aeroelastic behaviour.

Following this stage, the prototype of the aircraft is built, and Ground Vibration Tests (GVT) can be performed on the wings in order to experimentally determine the structural dynamic characteristics of the aircraft. Experimental Modal Analysis (EMA) is necessary when performing structural analysis because it allows to verify and validate the dynamic results obtained numerically by the Finite Element Analysis (FEA). By means of impact testing or shaker testing, it allows to determine the real modes of vibration of a structure and its modal shapes. When performing EMA there are some critical points that need to be considered: how and where should the structure be excited and how the responses should be measured.

EMA is an integral aspect of GVT which is a complex and time consuming procedure, and necessary in the certification processes.

When the aircraft is built, the testing is done and it is carried out for different configurations in order to evaluate and determine all the structural dynamic characteristics of the aircraft. This process starts when all the Ground Vibration Testing (GVT) results are available and it runs in parallel to the model updating. This process can be quite long because it has to be guaranteed that the validated and updated model is as accurate as possible.
The measurement of the critical modes present in the aircraft, yielding the modal parameters (modal frequencies and mode shapes and damping), are used to validate the computational results from the FE-model and to determine if an update of the model is needed.

When performing the model update, if it happens, along with the modal information, there is a need to know which parameters should be updated. To this end, a sensitivity study is performed in order to know the most critical structural parameters that have impact on the structure, e.g. the Young’s modulus and the density of the structural components.

With those parameters selected and with the experimental data obtained, the updating process can be completed. A correlation between the analytical and experimental results is done, matching the modes and their frequencies. This completes the model updating process and validates the computational model.

1.2 Motivation

Nowadays, the established concept of designing an aircraft is based upon intensive computational analysis, relying in complex computer models to predict and evaluate the performance of an aircraft, including structural analysis.

The use of experimental testing is seen has a very important step in this process, and this thesis aims to validate and evaluate a computational aeroservoelastic model of an Unmanned Air Vehicle in the framework of a collaborative project with a major Original Equipment Manufacturer (OEM).

Before the start of the flight testing phase of the Aeroservoelastic UAV Demonstrator, it is necessary to evaluate the structural dynamics response of the structure to confirm that it will be able to sustain flight dynamic loads. This work is an important part of a larger collaborative project with an OEM and performed in the Center of Aerospace Research at the University of Victoria. The main purpose of the project is to quantify and evaluate controllability of the Unmanned Air Vehicle with flexible wings and determine the coupling between structural modes in the wing and the flight dynamics modes (short period mode) in an UAV-QT1.

This thesis has contributed to the successful flight test campaign of the flexible UAV by developing and evaluating the structural dynamics model of the aircraft by determining experimentally the modal frequencies, shapes and damping of the flexible aircraft and completing the updating of the finite element models for flight clearance purposes. The flexible aircraft has successfully completed the flight test campaign and very good correlation between the predicted computational results and experimental flight test results were obtained.

1.3 Objectives

All the stages of Ground Vibration Testing (GVT) and associated finite model updating of the flexible aircraft model with application to a UAV are required to be completed. It is presented the process of choosing the right test set-up, with all its options, how to run the test itself and the underlying difficulties. Towards this end, the two main objectives of this thesis are as follows:
• Design and perform a Ground Vibration Test and collect the data for modal analysis;

• Perform modal analysis of the data and update the finite element models with the information gathered.

It is a secondary objective to evaluate if the methods and techniques used in this work can be applied to nonlinear models.

1.4 Main Contributions

The main contribution of this work includes the implementation of a new GVT facility at the Center for Aerospace Research at the University of Victoria and dynamic characterization of a novel aeroservoelastic flight test demonstrator in order to ensure structural integrity and flight test safety. The aeroservoelastic project is a collaborative work with a major OEM in the pursuit of next generation regional aircraft with high aspect ratio wings. The results obtained have ensured a safe flight test campaign and the agreement between the computational and experimental modal results is excellent.

1.5 Thesis Outline

This thesis follows the natural order the tasks for a typical Ground Vibration Testing (GVT) process and the associated model updating procedures. Each chapter presents the background information, chronological steps and considerations needed for each phase of the experimental evaluation and validation process.

Firstly, in Chapter 2, is presented the background for an experimental GVT operation with the associated instrumentation used, the type of installation, tools, description of impact and shaker testing, types of shaker controls and the description of how the modal parameters are extracted from the data taken from these tests.

In Chapter 3, a state of the art review is introduced in order to synthesize the current and future technologies and techniques in the world of experimental modal testing. Phase Separation Methods and Phase Resonance Methods are mentioned, as are the latest spatial-optical approaches, including photogrammetry, videogrammetry and laser measurements. Test planning is also discussed and presented.

In Chapter 4, the finite element models to be updated are shown and discussed.

In Chapter 5, the results obtained in the Ground Vibration Testing are exhibited along with the experimental procedures and issues encountered. Three different cases are analyzed: a small cantilevered beam, that is used as an introduction to Experimental Modal Analysis; and the two wing configurations of the QT1, the conventional rigid wings and the flexible wings.

In Chapter 6, the FE models are updated. The model updating parameters and responses are explained along with the steps and considerations needed in order to obtain good results. A sensitivity analysis was made before the updating process that determined the critical criteria that should be used
in the whole procedure. The criteria used for deciding the convergence parameters and a comment on the results are presented as well.

Finally, in Chapter 7, conclusions resulting from the current work and future work are synthesized and debated.
Ground Vibration Testing

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2.1 Introduction

Generally a schedule with advanced notice is required to accommodate the complicated set-up for data gathering. This includes sensor installation and acquisition for potentially multiple configurations in order to fully meet testing requirements. Often this requires an aircraft to be reserved for several weeks to gather enough data for FE model validation, where all the different configurations have to be tested, hundreds of sensors placed and everything set-up. Consequently, the aircraft can be reserved for GVT work for several weeks.

2.2 Instrumentation

GVT, as any other experimental work, has its own unique set of instruments and tools for data acquisition. There are plenty of options in the market for GVT testing, but in the end, the final choice takes into consideration the object under study and the budget of the project itself. For the instrumentation of the testing we can have it classified into different groups:

- Excitation Source
- Response Measurement System
- Data Collection System

2.2.1 Excitation Source

Here it is being discussed the instrumentation that will input energy into the test object or system. In the tests presented this was done either by a very short energy transfer, i.e. by a very short timed impact, or through a continuous excitation for a bigger length of time. These two cases are more commonly known as, respectively, Impact Testing and Shaker Testing. The two methods imply two different sets of instruments and, with them, multiple advantages and disadvantages.

2.2.1.A Impact Testing

Impact testing is a very simple procedure that consists on hitting the test object with an impact hammer. This impact hammer resembles an ordinary hammer with the slight difference that, on its tip, it has an impedance head that will measure the force input into the test object, that will later be sent to the Data Collection System. The impacts are done by a technician and therefore, the results may vary from technician to technician and by his/her technique. The hammer should be impacted in a 90°angle in relation to the objects surface. The force input should be the roughly same for each impact, even tough the impacts are normalized (provided you do not excite the structure non-linearly), and its value is usually decided by the technician taking into consideration the test object and its material. In order to be able to acquire good results for different objects and materials there are different tips available from where one can be selected that better suits a particular case, from super-soft to hard tips. This available choice is important because, depending on the tip, the results can differ since ideally, with a Dirac input,
the bandwidth of the signal is infinite. The softer the tip the smaller the bandwidth. However, it should be noted that, the softer the tip the longer the time for which energy is transferred to the test piece, allowing more energy to be input without as likely a chance of non-linear deformation.

All the impact testing performed was done using the hammer that was available in the laboratory, the PCB Piezotronics model 086C03 with the specifications detailed in Table A.1.

2.2.1.B Shaker Testing

Shaker testing implies the use of a shaker to excite the test object or system. This method is more versatile than the impact testing because the excitation from the shaker can be modelled electronically by the user and thus, guaranteeing that the excitation input into the object is always the same throughout the test, which cannot be said for the impact testing, as it is humanly operated. In this work, the term shaker is used to refer to the whole excitation source in which the shaker is included, but the shaker system is not only composed by the shaker. This system is composed firstly, and obviously, by the shaker; the stinger, a long connecting rod, that will attach the shaker to the structure; cabling and lastly the impedance head, or force sensor.

This impedance head, in this case a PCB Shock ICP © model 288D01, is a very sensitive device that allows to measure the force input into the system or the acceleration at the point of installation. The second feature is that, since it can also work as an accelerometer, it is used as checking system while doing the installation. Reciprocity tests have to be made before the beginning of the testing to guarantee that the shaker is well connected. Otherwise, errors could be included into the measurements and all the work made while testing will be lost.

In these tests, the shaker results are compared to the results from impacting the measuring point in the shaker condition and measuring it at the shaker’s installation point. The reciprocity feature of the measurements guarantees that in theory these tests should yield the same results. If the comparison of the data shows similar results it can be concluded that the shaker is properly installed.
Shaker testing in modal testing usually is done using one of the two types: electromagnetic, often called electrodynamic and the electro hydraulic, or hydraulic. [3]

This type of excitation, unfortunately does not come without its unique set of problems. The first one is regarding to the mounting of the shaker.

Setting up the shaker is a very tricky task as the shaker, and more specifically, the stinger has to be perfectly aligned for the test. The function of the stinger is to separate the effects of the shaker from the structure. It allows to have an interface of the test piece with the shaker such that the excitation forces are imparted orthogonally to the structure and is designed to diminish the twisting and shearing forces from slight misalignments.

A slight deviation from the stinger and the test object can induce parasitic vibrations and even damage the components involved. In order to prevent this situation, usually a lot of time is spent trying to have everything perfectly aligned, which, from personal experience, can take more than 25% of the time spent testing. This, obviously, is a major disadvantage for this type of testing. Since mounting requires an actual physical connection to the test object, there is the possibility of altering the dynamics of the structure itself. This is particularly important with lightweight structures where the attached transducers weight causes a significant change in the structures measured dynamics.
Table 2.1: Comparison between Impact and Shaker Testing

<table>
<thead>
<tr>
<th></th>
<th>Impact Testing</th>
<th>Shaker Testing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Setup Time</td>
<td>Fast</td>
<td>Long to Very Long</td>
</tr>
<tr>
<td>Setup Difficulty</td>
<td>Easy</td>
<td>Medium</td>
</tr>
<tr>
<td>Input Control</td>
<td>Depends on technician handling the hammer</td>
<td>Electronically controlled</td>
</tr>
</tbody>
</table>

2.2.1.C Impact Versus Shaker Testing

In theory, there is no difference between getting the results either from impact testing or shaker testing. The data is theoretically the same. This is because, it does not matter how the data is obtained, as long as there are the same input-output characteristics.

In theory forces can be applied to structures without having any kind of extra interaction between the said structure and the applied force, measure the response with a massless transducer, without this one too, having effect on the results. But since this is not possible, there is a real difference between impact and shaker testing.

Consequently, the effects that each method has on the structure should be considered before making the choice between one of them.

It is important to remember that, unfortunately, when testing, it is not just the structure being measured. There are many sources of noise that can affect the data that can range from the structure itself to everything else involved in the recording of the data and test environment: the suspension, the transducers, the possible stiffening and mass loading effects of the shaker/stinger attachment.

How the exciting methods correlate to these factors will be shown in differences in the data collected.

One of the most notorious cases will be when roving accelerometers while shaker testing. The constant repositioning of the accelerometers can cause shifts in the attachment position of the shaker and it can alter the neutral position of the structure, along with the change of location of possible mass loading associated with them, since their weight may be quite large relative to the effective weight of different parts of the structure.

2.2.1.D Response Measurement System

Throughout the testing were used the PCB \(^\text{\textregistered}\) miniature single axis piezoelectric accelerometers model 352A24. At the same time as the test object is being excited its response must be measured. This can be done with a set of well placed accelerometers throughout the object. One could think that, in order to have good measurements, a high number of accelerometers could be spread out throughout the object and measure every possible point. But, there are some reasons why this is not feasible:

- The cost of an extensive response system cannot always be supported, which means that simplified models have to be used
- Every object that is placed on the structure being tested, in this case an accelerometer, can cause a phenomenon called mass loading. The mass loading phenomenon where the accelerometer mass affects the dynamic stiffness at the its point of placement, thus affecting the vibrations of
the test object [4] and consequently the characteristics that are trying to be determined. That is why an ideal accelerometer should be as light as possible in order not to affect the vibrations the sensor is intended to measure;

- From the data-processing point of view, the increase of parameters to take into consideration (in this case it is the response data) can lead up to higher computation times.

2.2.1.E Data Collection System

The data collection system is often only composed by:

- Digital acquisition system, where all the information routs to;
- A computer to store the data and check the data recording in real time, if possible.

With this Data Collection System, all the physical material is listed and explained, so now, the data processing method can be explained.

2.2.2 Tools

When everything is ready in the laboratory, i.e. all the instrumentation is set-up and the data can be recorded, then the testing can start. But this can only happen with the help of tools, more specifically, commercial computer programs and user made ones and that will be present in every step of the testing and processing. Following the normal order of this genre of experiments we will have programs to:

1. Record the data
2. Pre-process the data
3. Analyse the results
4. Post-process the results for other applications or further analysis

2.3 Testing Framework

Before the recording of the data, as a safety feature for a good working environment in the lab, all the cables should be taped and carefully organized to not only protect the people testing but also the equipment that is, in many, if not all cases, expensive.

In order to have reliable results, the test object should be placed in a location where no vibration disturbances can occur, if possible a table built for vibration testing, which damps very well the vibrations felt through the floor. Also when choosing a test location another parameter to take into consideration is the surrounding noise. Even if it doesn’t seem apparent at first, the noise, as it is spread trough the air by the vibration of the air particles, can be sensed by the best accelerometers causing an unwanted source of error that can be easily avoided. As a curiosity, with the accelerometers used, a normal conversation between two people could be measured by the accelerometers placed on an aircraft from over 4 meters away!
One of the first aspects that need to be defined in the beginning of the test set-up is related to the excitation and response points, even before the definition of the excitation sources. These two locations can greatly define the whole process, as such they are of great importance and so there is a question of how the testing is going to be made. There are 2 options:

1. **Roving Excitation**: If the excitation is roving and the response fixed, how many fixed response Degree of Freedoms (DOFs) are needed, and where should they be located.

2. **Roving Response**: If the response is roving and the excitation fixed, how many fixed excitation DOFs are needed, and where should they be located.

When a single fixed (reference) excitation or response DOF is used, this is called a **single reference test**. If more than one reference excitation or response DOF is used, this is called a **multiple reference test**.

A reference is the fixed excitation or response DOF used to acquire data during a modal test. In a single reference roving impact test, a single accelerometer is fixed to a DOF of the structure and then the structure is impacted at two or more different (roving) DOFs using the hammer. In a single reference shaker test, a shaker is fixed to a DOF of the structure, and responses are measured at two or more different (roving) DOFs. In a multiple reference roving impact test, two or more accelerometers are fixed to different DOFs while in a multiple reference shaker test, two or more shakers are fixed to different DOFs.

One question that comes after is where to position the excitation or the response DOF. A very important thing is to **avoid Nodal Points**, which are DOFs where the mode shape magnitude is zero.

In a single reference test, having only one reference, it’s very important that the reference excitation or response DOF is chosen where no mode shape is at a nodal point. By the way the Frequency Response Functions (FRFs) are calculated, the roving DOFs define the DOFs of the mode shapes, thus, the reference DOF can be chosen anywhere on the structure without affecting the mode shapes. It can be concluded then that the **optimum single reference DOF is the one where all of the modes of interest are not at nodal points**. If a Finite Element Model (FEM) of the test structure is available prior to testing, the FEM can be used to calculate a set of analytical mode shapes for the structure. These mode shapes can be used to locate one or more optimum references, where none of the mode shapes is at or near a nodal point, but since these tests are made for FEM validation, a second check after the first test results should be made.

A deeper analysis of FE models is going to be made in the Chapter 4.

In most modal testing, the structure is assumed to be “symmetric”, obeying Maxwell’s reciprocity (it will be shown in 2.3.2 that this means that the \([H(w)]\) is symmetric). When this assumption is valid, a roving excitation test and a roving response test will yield equivalent modal parameter estimates and it corresponds to having only one row or one column of the transfer matrix, explained in Subsection 2.3.2, needs be measured since all other rows and columns contain redundant information. In other words, the reciprocity applied to modal testing yields that it does not matter, from the results point of view, if we have a fixed reference and roving excitation. This means that the FRF can be obtained.
by impacting a point 'i' and measuring the response at a point 'j' and have exactly the same FRF as impacting point 'j' and measuring the response at point 'i'[5].

In the case of impact testing where the hammer keeps impacting different points with an accelerometer always at the same place, or in the other hand, if we have a fixed excitation and a roving response, like in the case of shaker testing where the shaker remains fixed at the same point and it is the accelerometer that shifts positions.

The choice of input to be used to excite a system depends upon the characteristics of the system itself, the parameter estimation and upon the expected utilization of the data. The characterization of the system is primarily concerned with the linearity of the system and if the system is linear, all input forms should give the same expected value, but, as it is real systems that are being discussed, they all show some degree of non-linearity.

Regarding impact testing, there isn’t a very specific way of preparing for the testing. As long as the accelerometers are well placed, in which case, attached with petrol wax, the testing can proceed normally.

On the other hand, shaker testing comes with various problems, how and where to attach it, as well as verifying regularly while testing if the initial conditions maintain. This is because, in one of the testing series done, halfway through a test using a shaker it was found that it had shifted position by a considerable amount, rendering all the testing done so far, useless. So, the shaker installation is a very important topic when testing.

2.3.1 Shaker Installation

The first task when handling the shaker installation is to define how to attach it to the structure. The structure, being already a final construction of the aircraft, couldn’t be tampered with, specially because it would alter the flight dynamics of it. The normal attachment procedures consisting in having the impedance head screwed or bolted into the surface was out of the question.

Fortunately, the shaker provided had two connecting magnets that allowed to have the shaker installed without damaging the structure. In order to provide extra adherence, petrol wax, the same used to place the accelerometers, was placed in the magnetic attachment to prevent sideways movement caused by the shaker excitation.

When testing with vertical excitation, tranversal to the wing, a single shaker, the only one available, the Modal Shop’s Mini SmartShaker model K2007E01, was attached to below the right wing, close to the root. Further location details are going to be given in Section .

As for the conventional wing one of the magnets was placed inside the wings skin to provide the attachment and for the flexible wing, it was placed inside the aluminium spar. Both of them were aligned with a previously chosen DOF in the pre-test analysis.

With one of the magnets placed inside the structure, then the other magnet is glued to provide extra stiffness to the attachment and is to that magnet that the impedance head is connected and consequently to the stinger and the shaker as well.

The attachment method is very important because it defines the interaction between the shaker and
the structure during testing. A flexible or more stiff attachment can affect the results, specially when it comes to damp out the vibrations.

Depending on the shaker control specifications either the force is kept at zero, in the case of a current-controlled shaker, or the velocity of the shaker is kept at zero, in the case of a voltage-controlled shaker.

In the first case, no interaction happens between structure and shaker when the excitation signal is turned off, and consequently the shaker does not affect the structure. In the second case, it happens the opposite, the shaker does add damping to the structure, making it reach a resting situation quicker. But, fortunately, in this case, this does not influence the measured FRF because the force being applied by the shaker is still measured by the impedance head, and consequently the relation between force and acceleration maintains itself valid.

Other types of shaker placements were experimented, but the results were not satisfactory. This placements included placing the shaker exciting the aircraft whilst in a horizontal position, connected to the fuselage. Also, while trying to get results from the flexible wing that was being proven harder to get results from, the shaker was positioned just like in the first case under the wing, but since the wing was mounted on the table proper for the vibration testing, no proper stand was found for the shaker. Foam and wood stands were tried, but without success. From these unsuccessful experiments, the shaker was hanged from the ceiling with bungee chord, alike the structure, as it is going to be shown in Section 2.6. But the results once again were proven to be non satisfactory.

2.3.2 Theoretical Background for Frequency Response Functions and Fast Fourier Transform

In modal testing, the results that are obtained are called FRFs. They are fundamental measurement that isolates the inherent dynamic properties of a mechanical structure from which experimental modal parameters are estimated by curve fitting. An FRF is a 2-channel measurement which is calculated between two signals: the excitation DOF and a response DOF signal. But how exactly are these FRFs calculated?

In the vast majority of engineering problems, with the systems at hand, entities which have one or more inputs causing one or more outputs are continuous. The analysis of these systems without any approximation would be extremely difficult, and very time consuming, with the need to solve complicate systems. And for most cases, those solutions do not exist. A dynamic system is often defined as a linear system if it can be described by linear differential equations. If it is not linear, it is called a non-linear system.

Therefore, if the system is not linear to begin with, a linear approximation is used and the solution becomes relatively simple. That is for many analysis, continuous systems are often simplified.

The equation of motion for vibrating structures is derived from Newton’s Second Law for a multi-degree-of-freedom system. This results in a system of equations represented by the Equation 2.1:

\[
[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\}
\] (2.1)
Where, in Equation 2.1 [6] [7] \([M], [C], [K]\) are called the mass, damping, and stiffness matrices, respectively, \(x, \dot{x}, \ddot{x}\) and \(f\) are the displacement, velocity, acceleration, and force. It should be noted that the excitation forces and responses are functions of time \((t)\), and that the coefficient matrices \([M], [C], [K]\) are constants, and so this represents a dynamic model for the vibration of a linear, time invariant structure.

We can, from this point use what is called the Fourier Transform, that will serve as a basis for frequency analysis of aperiodic signals, or in other words, random and transient signals. The Fourier transform is very useful because it relates the abstract transfer function with the frequency response that can actually be used for modal analysis. A short introduction to this subject will be made, but more information can be found in [6] and [8]. The Fourier Transform is defined for continuous signals and the Fourier transform, \(X(f)\) of a time signal, \(x(t)\) is defined by the Fourier transform integral:

\[
F[x(t)] = X(f) = \int_{-\infty}^{\infty} x(t) e^{-2\pi j ft} dt
\]  

When disregarding the initial conditions of the structure, the equivalent frequency domain form for its dynamic model can be represented in terms of a Discrete Fourier Transform (DFT). As the Fourier transform is defined for continuous signals, the DFT works for discrete, sampled, signals. In steady-state conditions, and from the harmonic excitation case, it is possible to establish the relation between the complex amplitudes of response and the amplitudes of the applied forces, through the frequency response function matrix \([H(w)]\). The dynamic model can then be represented as the ratio of the Fourier transform of an output response \(X(w)\) divided by the Fourier transform of the input force \(F(w)\) that caused the output:

\[
\{X(w)\} = [H(w)]\{F(w)\}
\]  

With Equation 2.3 we used the Fourier transform to transform the impulse response that we record into the frequency domain. Doing that we obtain the frequency response function, FRF, \(H(w)\). It should be noted that this equation is valid for all discrete frequency values for which the DFTs are computed, for it complies with all the precious conditions.

Each element of \([H]\) is called a receptance frequency response function, which relates the response at coordinate \(j\) to a force at coordinate \(k\) and it is given by [9]:

\[
H_{jk}(\omega) = \sum_{r=1}^{N} \left( \frac{rA_{jk}}{\omega_r \xi_r + i(\omega - \omega_r \sqrt{1-\xi_r^2})} + \frac{rA^*_{jk}}{\omega_r \xi_r + i(\omega + \omega_r \sqrt{1-\xi_r^2})} \right)
\]  

where \(rA_{jk}, \omega_r,\) and \(\xi_r\) are the residue, the natural frequency and viscous damping ratio of mode \(r\), respectively.

Equation 2.4 represents the behaviour of the structure among the selected points, and one can work in the time domain or the frequency domain. Moving from one domain to the other is a matter of applying Fourier transforms.

The most intuitive interpretation of the frequency response is, that it is the ratio of a sinusoidal output, and a corresponding sinusoidal input. \([H(w)]\) at each frequency is a complex number and the magnitude of it is the ratio of the two amplitudes, and the phase of \([H(w)]\) is the phase difference \(\angle(Y(f)) - \angle(X(f))\).
While the transfer function is a mathematical, abstract entity which we can use as a tool for solving differential equations, the frequency response is an entity which we can measure experimentally.

The Fourier transform of the forced response is made up of a summation of the transforms of all of the excitation forces times the columns of the FRF corresponding to the excitation DOFs.

Now that is known how a FRF is calculated, how can it be used?

FRFs can be measured in various ways, but one thing remains true throughout testing: The roving DOFs of the FRFs in a set of FRF measurements define the DOFs of the mode shapes [8].

An important notion that should be retained about FRFs is one about driving point FRFs. A driving point FRF is any FRF where the excitation DOF is the same as the response DOF and each one is obtained by attaching the accelerometer at a DOF and impacting the structure at (or near) the accelerometer location.

But, there are problems with driving point FRFs: a driving point FRF at or near a nodal point of a mode shape will have a small resonance peak or no peak at the frequency of the mode.

When, in a FRF, there is a large resonance peak, that means that one mode was excited in a positive way at the driving point and no nodal point is present in the vicinities. If that is not the case, a small resonance peak is an indication that the measurement is near a nodal point. When this happens, it is obvious that that mode is not going to be represented as good as it should and not as good as it could be. Which means that the reference used is not a good reference for a single reference test.

This is an indication that probably there isn’t a single reference from which a good measurement from all modes can be obtained and thus identified from a single reference set of FRFs. The best accomplishment in this type of testing is to define experimental mode shapes with as many DOFs as possible, what translates into using as few references as possible to measure FRFs. That is why a good pre-test planning is very important in order to accomplish this.

From equation 2.3, a mathematical manipulation can be done to do the inverse Fast Fourier Transform (FFT) \( FFT^{-1} \). The new relation stands for the forced response equation for the structure.

\[
\{x_f(t)\} = FFT^{-1}\{H(jw)\}\{F(jw)\}\]

(2.5)

For a fixed value of time \( t \), the forced response vector \( \{x_f(t)\} \) is the Operating Deflection Shape (ODS). By other words, an ODS is any forced motion of two or more points in a structure. When one two or more points have their motion defined, with a location and a direction (vector quantity) they define a shape, where one point has its motion defined in relation to all the other points.

From this new relation, several conclusions can be taken:

- The overall structural response is a sum of the transforms of all of the excitation forces times the columns of the FRF corresponding to the excitation DOFs, i.e., summation of responses due to each of its modes.
- The impulse response of the structure depends on where the impulsive force is applied.

The mode shapes are eigenvectors in this system, i.e, they can change in value, but not in shape. The denominators in Equation 2.4 are functions of frequency will cause the peaks in an FRF.
Each peak in the FRF is evidence of at least one mode or resonant condition. There can be cases where we have closely coupled modes. In this case two or more modes are included in one single resonance peak in a FRF. There can be even a different case where there are repeated roots, i.e., two or modes with the same modal frequency but different mode shapes. These details are going to be further discussed upon analysis of the measured FRFs.

### 2.3.3 FFT Windows

When performing FFT based measurements there can be a phenomenon called leakage. Leakage occurs when the FFT is computed from of a block of data which is not periodic, and this causes errors in the data measured, because, when doing FFT computation, it is assumed that the signal being processed is periodic in every data block.

The leakage phenomenon consists in having the signal energy being distributed over a wide frequency range in the FFT when it should be in a narrow frequency range.

In order to correct this problem, a windowing function, i.e. weighting functions, can be applied to minimize this effect, since it can never be truly eliminated, thus better representing the frequency spectrum of the data [11].

These window functions are defined so that the resulting data respects the periodic data rule for the calculation of a proper FFT. It is shaped starting and ending at zero, with a proper shape in between, to then be multiplied with the time data block, forcing the signal to be periodic. A weighting factor is also included so that the correct FFT signal amplitude level can be recovered after the windowing.

Usually, a type of signal that does not show any signs of leakage and thus, does not require windowing, is the transient signals, that are usually related to impact testing. Since it’s a type of signal that starts at zero at the beginning of the time window and then rises to some maximum and decays again to zero before the end of the time window, it complies with the periodic rule of proper data to do the FFT.

Since there are various windowing functions for every type of application, the engineer must choose the appropriate window for the task at hand. If not chosen correctly, it can introduce even more errors in the data. In order to facilitate this decision, the Table 2.2 was created.

### 2.3.4 Shaker Control

As said before, the shaker input is electronically controlled. This allows the user to decide from various input signals which one suits best for the testing being performed. The signals that can be used to drive the shaker and excite a system in order to determine frequency response functions can be organized in two groups: signals with continuous spectra and signals with discrete spectra.

Random signals, the only case that was considered for the testing from the first group, can only be defined by their statistical properties over some time period [12], because in any particular time period throughout the whole signal is one of a kind and it cannot be defined by any mathematical relationship. Random signals can be further classified as stationary or non-stationary. Stationary random signals are a special case where the statistical properties of the random signals do not vary with respect to
Table 2.2: Most Common Windows and their Features [10]

<table>
<thead>
<tr>
<th>Best for these Signal Types</th>
<th>Window</th>
<th>Frequency Resolution</th>
<th>Spectral Leakage</th>
<th>Amplitude Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Random</td>
<td>Barlett</td>
<td>Good</td>
<td>Fair</td>
<td>Fair</td>
</tr>
<tr>
<td></td>
<td>Blackman</td>
<td>Poor</td>
<td>Best</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Hanning</td>
<td>Good</td>
<td>Good</td>
<td>Fair</td>
</tr>
<tr>
<td></td>
<td>Hamming</td>
<td>Good</td>
<td>Fair</td>
<td>Fair</td>
</tr>
<tr>
<td></td>
<td>Kaiser-Bessel</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Tukey</td>
<td>Good</td>
<td>Poor</td>
<td>Poor</td>
</tr>
<tr>
<td></td>
<td>Welch</td>
<td>Good</td>
<td>Good</td>
<td>Fair</td>
</tr>
<tr>
<td>Transient &amp; Synchronous Sampling</td>
<td>None</td>
<td>Best</td>
<td>Poor</td>
<td>Poor</td>
</tr>
<tr>
<td>Sinusoids</td>
<td>Flat Top</td>
<td>Poor</td>
<td>Good</td>
<td>Best</td>
</tr>
</tbody>
</table>

translations with time. In even more detail, stationary random signals can be classified as ergodic or non-ergodic. A stationary random signal is ergodic when a time average on any particular subset of the signal is the same for any arbitrary subset of the random signal. All random signals which are commonly used as input signals fall into the category of ergodic, stationary random signals [4].

Deterministic signals, or signals with discrete spectra, have an explicit mathematical form which result in FRFs that are dependent upon the signal level and type and can be further divided into periodic and non-periodic signals. The most common inputs in the periodic deterministic signal designation are sinusoidal in nature while the most common inputs in the non-periodic deterministic designation are transient in form. These type of signals have the advantage of concentrating the energy in the signal on those frequencies which we estimate with the Fast Fourier Transformation.

When testing using either random signals or deterministic signals a set of FRFs for different signal levels can be obtained and used to try to analyse possible non-linear characteristics of the system. For this purpose random input signals can be used. When in the presence of non-linearities, the measurements result in a FRF that represents the best linear representation of the non-linear characteristics for a given level of random signal input. When facing small non-linearities, the choice between either a random input or a deterministic input will not differ in a way that shows that one is better than the other.

In the end, the possible signals that could be used while shaker testing are listed below:

- Pure Random Noise
- Burst Random Noise
- Periodic Chirp
- Stepped-sine Excitation

2.3.4.A Pure Random Noise

Pure random noise (or true random), is a continuous, normally distributed noise signal. Random noise is used commonly used for general vibration testing, even though is it not considered one of the best techniques for acquiring FRF measurements for modal testing.
The random nature of the signal excites the structure with varying amplitude and phase at the same time as averages are collected. This tends to average any slight non-linearities that may exist in the structure. Even with this phenomenon the signal never satisfies the periodicity requirement of the FFT process causing leakage. This signal has a continuous spectrum and since the signal is continuous in the time domain, it must be windowed by, for example, a Hanning or half sine window when carrying out the frequency analysis.

The process of applying a window to the data the resulting FRFs may help, but leakage will always be present. The peak amplitude will be affected and there will be an appearance of more damping in the structure due to the leakage and windowing effects. [13]

2.3.4.B Burst Random Noise

Burst random noise is a continuous noise which is turned off at a certain time during the measurement of each block, after which the force becomes zero and the responses (accelerations) die out during the remaining part of the block. This may prove itself sometimes a difficult task if the rigid body modes of a freely supported structure have low damping, but added time can be implemented between measurements in order to fix the problem. Possible leakage that can be found in the data is most likely linked to this kind of situation.

If a pre-trigger delay is also used, then the signal is totally observed within one sample interval, with all signals both begin and end at zero, thus satisfying the periodicity requirement of the FFT process. Of course, both the input and response signals need to satisfy this requirement. This means that no leakage will occur and no window is needed. In comparison with the pure random noise the peaks are much sharper and better defined with a very good coherence at the resonances frequencies. This signal is well suited for averaging out slight non-linearities that may be found in the measurements and it can be easily used when doing multi-input estimation.

2.3.4.C Sine Chirp

The sine chirp signal is a sinusoid signal, which rapid and continuously sweeps from low to high frequencies in a certain frequency range of interest in one sample interval of the analyser. The signal repeats and therefore satisfies the periodicity requirement of the FFT process. This means that there won’t be any leakage and no windowing is needed.

The advantage of using this type of signal compared with the random signals above is that the signal-to-noise ratio is the better. Even if the disturbance noise is negligible, multiple averages should be used to obtain a good result.

By changing the input force level applied to the system, linearity checks can be easily made using this excitation technique. The fact that the chirp signal consists of a sinusoid can also be an advantage if it is known that the structure is non-linear and it is desired to have an excitation signal that has the same amplitude throughout the test. The noise signals, however, are often better when one a non-linear structure is present and it is being measured a linear approximation of the system, because the noise signals have an amplitude distribution in which all amplitudes are randomly mixed.
2.3.4.D Stepped-sine Excitation

This is a different type of signal that, instead of doing broadband excitation as the previous cases, where the excitation signal contains frequencies within a wide frequency band [4], requires that a single frequency, coincident with an analyser spectral line, is used to excite the system at a time. For each individual frequency step, steady-state conditions are reached and the input and output signals are measured and the amplitude and phase relationships are determined between the output and the input force, i.e. the FRF is calculated, and only then the frequency steps up for the next level.

Since this excitation signal is usually processed with the FFT process and is done to be guaranteed by periodically setting the sampling frequency so that exactly an integer number of periods are measured, the leakage phenomenon does not occur.

Stepped-sine is thus a slow method, but it has the advantage of being able to cope with very low signal-to-noise ratios. The signal-to-noise ratio is the highest possible, which makes it possible to use relatively low levels (as all the signal power is concentrated at one frequency, while the extraneous noise is spread over all frequencies). Since it is not broadband in nature, this technique is a slow method and the slowest of all techniques presented here. On the other hand, since all the signal power is all concentrated at one frequency at the time, while the extraneous noise is spread over all frequencies, it is possible to use relatively low levels, thus makes it so it has the highest possible signal-to-noise ratio. This makes it likely to produce the best measurement of all the excitation techniques above.

Another advantage is that it is excellent for documenting non-linearities: with some systems it is possible to control either the excitation force or the response signal using feedback so that either the force or response signal is held constant. This feature also stands important when doing measurements measurements on lightly damped structures, avoiding with it excessive vibration levels, in this case, by controlling the response.

2.3.4.E Signal Choice

With all the possible signals that can be chosen, the choice relies on the engineer, the application and the structure being tested. In order to chose the best option and save time testing all the possibilities, it is best to rely on experience to make the decision. It can be seen in Table 2.3 the best choices when it comes to the type of structure being tested, and with the advantages and disadvantages of each signal being listed individually in Table 2.4.
Table 2.3: Best Signal Options for Each Structure Type [14]

<table>
<thead>
<tr>
<th>Structure Type of Excitation</th>
<th>Linear &amp; Burst Random</th>
<th>Moderately Linear Transient Excitation</th>
<th>Moderate Sine Excitation</th>
<th>Non-Linear (On Critical Modes Only*)</th>
<th>Heavy Stepped Sine</th>
<th>Non-Linear Swept Sine</th>
</tr>
</thead>
</table>

*Or known or found to exhibit non-linearity

Table 2.4: Excitation functions [3]

<table>
<thead>
<tr>
<th>Periodic*</th>
<th>Transient*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Sine Steady State</td>
</tr>
<tr>
<td>Minimize leakage</td>
<td>No</td>
</tr>
<tr>
<td>Signal to Noise Ratio</td>
<td>Very High</td>
</tr>
<tr>
<td>RMS to Peak Ratio</td>
<td>High</td>
</tr>
<tr>
<td>Test measurement time</td>
<td>Very Long</td>
</tr>
<tr>
<td>Controlled frequency content</td>
<td>Yes</td>
</tr>
<tr>
<td>Controlled amplitude content</td>
<td>Yes</td>
</tr>
<tr>
<td>Removes distortion</td>
<td>No</td>
</tr>
<tr>
<td>Characterize nonlinearity</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*In analyser window

After testing several signals the choice fell upon the periodic sine chirp and burst random excitation. All the testing done with the shaker followed the best results, which were the ones taken with the sine chirp excitation.

2.4 Recording the Data

Using a multichannel data acquisition system, a NI cDAQ-9188 with IEPE modules NI 9234 that connect to the accelerometers and impact hammer or shaker, the vibration response of the test object is measured and the signals from the sensors are then amplified, digitized, and stored in the system's memory as blocks of data, one data block for each measured DOF of the structure.

A very important requirement for the multichannel system is that it has to be able to simultaneously sample or digitalize the vibration signals at the same time it converts them from analogue signals.

Since we are dealing with an FFT-based system, it is necessary to satisfy the Nyquist criterion in order to prevent the aliasing, i.e. false frequencies in their frequency spectra. This is done by having the maximum frequency not exceed one half of the sampling frequency used to digitize them [8].

Along with the data recording software, it was used with the National Instruments software, LabView, that can be seen in Figures A.1 and A.2, both Graphical User Interfaces (GUIs), respectively, for the
recording of data information, with the coherence and FRF magnitude and phase information; and the control panel for the shaker.

The data is then pre-processed before being analysed in MEscope by a Matlab script written by myself and Stephen Warwick, which formats the data acquired in testing to a format that can be read by the software. This also enables to have the data either from the accelerometer from the starboard wing or the port wing or even the two at the same time. This allows to have a better understanding of the aircraft and the wings on their own.

2.5 Curve Fitting Methods

The first stage in modelling the dynamic behaviour of a structure is to determine the modal parameters as introduced below:

1. The modal (resonance) frequency;
2. The modal damping;
3. The mode shape.

Figure 2.4 shows the different ways in which modal parameters can be determined: analytically and experimentally. A ever growing use of finite element models is allowing to have modal parameters being determined from them allowing to solve numerically structural dynamics, before doing experimental modal analysis.

These parameters can be obtained by curve fitting the measured set of FRFs. Even tough it not going to be used in the present work, there is an alternative that didn’t use FRFs but Impulse Response Functions (IRFs), the inverse FFT of the FRFs.

Curve fitting is a process of matching a mathematical expression to a set of empirical data points. Only after determining the mathematical expression one can retrieve the requires modal parameters.
This process is done by applying a minimization of the squared error (or difference) between the analytical function and the measured data.

The three main requirements for a good curve fitting algorithm in a measurement system are [15]: execution speed; numerical stability and ease of use. By using MEscope one can use a curve fitting method from the following list:

- **Local Single Degree of Freedom or single mode method (SDOF) methods**: The easiest to use and of most common usage.
- **SDOF methods**: Can be used in the majority of FRF data sets with light modal density (presenting coupling).
- **Multiple Degree of Freedom or multiple mode method (MDOF) methods**: The best method to be used in cases of high modal density.
- **Global methods**: A better method than MDOF methods for FRFs data sets with local modes.
- **Multi-Reference (Poly Reference) methods**: This method can identify repeated roots (very closely coupled modes) where the others cannot.

From these major methods there are some similarities, but each one has its own characteristics. This list is enumerated in a crescent order of complexity used in the method. With the exception of the SDOF methods, which estimate modal parameters **one at the time**, all the other three (MDOF, Global and Multi-Reference methods) can simultaneously estimate parameters for **two or more modes at a time**. [1]

While Global and Multi-Reference methods can be used to fit an entire set of FRFs at once, Local methods are limited to one FRF at a time.

In relation to which method should be used, there isn’t a method that is best applied in every case, it relies on the engineer performing the analysis to choose the best option for the case at hand.

One particular Local MDOF method that was used and was deemed worthy of mentioning a small part of its procedure was the **Complex Exponential**. This particular algorithm does curve fitting of an analytical mathematical expression not of a FRF but, it actually calculates the IRF of a structure from the experimental frequency response data [1]. This makes it fast, numerically stable and able to handle multiple modes but it can have time domain leakage and one should always input a number over the actual number of modes and then delete computational modes.

### 2.5.1 Modal Parameter Estimation

After knowing which curve fitting methods can be used and when they are best to be used, the question is how do these methods actually work.

The common technique to all curve-fitting methods involves the curve-fitting of an analytical expression, unique for each method, to a set of measured transfer function data. This is performed in two steps [16] [17]:

---

[1] Reference 1

[2] Reference 2
1. The curve-fitting is performed in the available FRFs, minimizing the squared difference between the complex data and the complex valued analytical function form, i.e. a least squared error curve (fitting) estimate of the FRF data is determined. In this point it is already possible to determine the modal frequency and damping estimates.

2. Secondly, the modal residues are determined by having a second second least squared curve fit of the data. Then the mode shape is determined by a set of modal residues obtained from curve fitting a set of ODSs FRFs corresponding to a single reference response.

2.5.1.A Modal Frequency

The modal frequency, or resonance frequency, is easily determined. It is identified by being the peak frequency of the resonance mode on a magnitude FRF plot.

This peak frequency is related to the frequency resolution of the measurements, which means that if, unfortunately, the measurements have a poor resolution, the curve fitting will be severely impaired. A better accuracy can be obtained by reducing the frequency range of the baseband measurement, for resonances at low frequencies, or making a zoom measurement around the frequency of interest while in MEscope.

The resonance peaks should appear at the same frequency in every FRF measurement excluding only the cases where the measurement corresponds with a nodal point of the mode shape.

2.5.1.B Modal Damping

Damping represents the energy dissipation observed in structures. Without it, a structure would, once excited, vibrate forever. When it comes to modal testing, there is modal damping involved, which is applied to each mode separately.

Since usually damping is not uniform throughout the structure there can be cases where some modes can have very little damping while other may have relatively large damping

When modal damping is used, each mode has damping $c_i$, where

$$c_i = 2m_i\omega_i\zeta_i \quad (2.6)$$

and $m_i$ is the modal mass and $\zeta_i$ is the viscous damping ratio commonly used to specify the amount of damping as a percentage of the critical damping.

$$\zeta = \frac{c}{c_{cr}} \quad (2.7)$$

For a single degree of freedom (SDOF) system, critical damping is defined as

$$c_{cr} = 2\sqrt{km} = 2m\omega \quad (2.8)$$

$$\omega_d = \omega \sqrt{1 - \zeta^2} \quad (2.9)$$

where $\omega$ is the natural frequency of the undamped SDOF system and $\omega_d$ is the damped natural frequency.
When testing structures, their responses can be divided into two groups: responses where there is light modal density, usually the structures that show this are called simple structures, and the ones where there are heavy modal density.

![Figure 2.5: Light Versus Heavy Modal Density (Coupling)](image)

In simple structures, the modes are not closely spaced, and are not heavily damped, Figure 2.5, and at resonance, a simple structure behaves predominantly as a Single-Degree-of-Freedom system, and the modal parameters can be determined relatively easily.

![Figure 2.6: Example of an FRF of a Simple Structure Indicating the Modal Damping $\zeta_r$ for each Resonance Frequency $f_r$](image)

In these simpler cases, the modal damping can be determined fairly easily. The classical method of determining the damping at a resonance can be used and it consists on finding the so called half-power (with -3dB) points of the magnitude of the Frequency Response Function (FRF), like in Figure 2.6.

With the knowledge of these points, for a particular mode, the damping ratio $\zeta_r$ can be found from the following equation:

$$\zeta_r = \frac{\Delta f}{2f_r} \quad (2.10)$$

where $\Delta f$ is the frequency bandwidth between the two half power points and $f_r$ is the resonance frequency.

It should be noted that the resonance peak width, in order to have consistency, should also be the same for all FRF measurements, in order to calculate the same modal damping value in every FRF measurement. The accuracy of this method is strongly connected to the frequency resolution present in the measurements, this is because it determines how accurately the peak magnitude can be measured.
For lightly damped structures, in order to make proper measurements of the peak frequency and consequently the modal damping as well, high resolution analysis is needed.

2.5.1. C Modal Shape

A most common procedure to mode shape determination is called the Quadrature Picking method [18]. This method assumes that coupling between the modes is light, meaning that the structures are very lightly damped, as seen previously.

At any frequency, the magnitude of the frequency response function is the sum of the contributions, from the same frequency, from all modes. Being the case of a lightly damped structure the structure is going to have its response mainly determined by the mode present at that particular frequency.

Since the value of the imaginary part of the FRF at resonance, for simple structures, is proportional to the modal displacement, by examining the magnitude of the imaginary part of the FRF at every point on the structure, the relative modal displacement at each point can be found.

Consequently, from these displacements, the mode shapes can be determined. This method is then applied repeatedly until all the modes shapes are determined for the range being studied. If necessary, by having an excitation and response measurement from the same point and in the same direction, the mode shape can be scaled in absolute units.

With these last topics it is shown how simply all the three modal parameters, frequency, damping, and mode shape, are easily determined directly from a set of FRF measurements.

2.6 Boundary Conditions

In the experimental work done in this thesis, two different boundary conditions were simulated in laboratory: the fixed cantilever condition and the free free body condition.

The setting up of these different testing environments are very important because they allow the engineer to measure important modes that are sensitive to the critical FE-model parameters and they enable, if necessary, to measure the reaction forces at the component's or structure's attachments. These boundary conditions are specially important when testing new components in real life instead of components or structures that the engineer is already familiarized with, like in the case of follow-on aircraft developments.

2.6.1 Fixed Cantilever Condition

For the fixed cantilever condition, the wings were removed from the body of the aircraft and bolted down to an vibration absorbent table with a specially made metal adaptor that allowed simultaneous connection between the wings and table. The adaptor connects to the wings by their center, in the same way the wings would be connected to the fuselage of the aircraft, as can be seen in Figure 2.7.

This testing configuration is the closest that one can get to the fixed cantilever boundary condition that was defined in the FE model previously.
2.6.2 Free Free Condition

For this case, two different set-up configurations were used: a set-up using bungee chords and foam mattresses, as can be seen in Figures A.3 and A.4

The first free-free configuration was simulated by suspending the structure through the use of bungee chords. These are connected to a frame that attached to the structure of the aircraft. This frame was carefully installed in a way that wouldn’t change the aircraft’s center of gravity.

Another approach was to use what can be called as a soft support systems, that in this case was foam mattresses in order to simulate the free-free condition. These types of support systems are known to have the effect of reducing the frequency of the rigid-body modes and are commonly used in testing of larger structures [19].

2.7 Modal Correlation Analysis

Having determined either experimentally or analytically the modal parameters of a structure there is always a need to validate these results and see how well they match another set of results available. In theory, two different techniques, in the experimental realm, should result in the same sets of results. Unfortunately, this is rarely the case. For that, a modal correlation analysis is made and it uses parameters as the Modal Assurance Criterion (MAC) and the coherence between results to access how similar the results are.

The techniques can also be used, more specifically, to quantitatively and qualitatively evaluate the correspondences and differences between analytically and experimentally obtained modal parameters. This type of analysis can be performed to mode shapes, also static and operational shapes and FRFs.

There is a range of different levels of correlation analysis. They can do visual comparison of the mode shapes, global and local correlation, and perform the calculation of the correlation coefficients that are calculated from the weighted relative differences between different modal parameters.
2.7.1 MAC Criterion

The Modal Assurance Criterion (MAC) is a measure of the degree of orthogonality between two vectors. Given two vectors $\mu_1, \mu_2$, it is defined by [20]

$$MAC(\mu_1, \mu_2) = \left( \frac{|\mu_1^* \mu_2|}{||\mu_1|| \cdot ||\mu_2||} \right)^2 = \cos^2(\mu_1, \mu_2)$$

(2.11)

where $^*$ designates the conjugate transpose of a complex vector.

In more practical terms, the MAC values can go from 0 to 1. If the MAC value between two shapes is 1, it means that they are identical shapes, where 0 would mean that there is nothing similar at all between the two.

While performing analysis, if the MAC value is superior to 0.9 it is common practice to consider that in fact, the two shapes are similar. If the value is inferior to 0.9, they are usually discarded as different mode shapes [17].

2.7.2 Modal Phase Collinearity Criterion

The Modal Phase Collinearity (MPC) as defined in [21] is a measure for the degree of complexity of a mode shape by evaluating the functional linear relationship between the real and imaginary parts of the mode shape coefficients. For lightly damped structures, physical modes behave as real modes and the MPC value approaches unity. A mode with a low index is rather complex, a sign indicating a computational or noisy mode. If there are present complex physical modes, which are typical for highly damped systems, caution is advised when analysing the results.

2.7.3 Coherence

The coherence indicates the correlation between output spectrum and input spectrum, thus a power transfer between input and output of a linear system can be estimated.

In the testing environment it is usually used to verify how good the experimental results are in comparison while doing a series of measurements in the same conditions, e.g. while doing impact testing in a specific point, the coherence value can indicate how similar are the results in relation to the previous ones, thus showing if the measurements are being done properly or not. For example, if the FRF shows a peak, but the coherence is low for that frequency, there is a possibility that it could not be a real resonance. In which case, it is advised to redo the measurement at that point.

The coherence can have values in the range from 0 to 1. Low values are an indication of a weak relation between measurements and values close to 1 show a representative measurement.

2.8 Summary

In this chapter all the basis for Ground Vibration Testing (GVT) was covered in order to provide the knowledge needed for the best understanding of the following work presented. Summarizing the current modal testing method, it can be listed as:
• Accelerometers are used as response sensors;
• Excitation is either done by impact hammers or shakers;
• Shakers can be controlled by a various number of different signals according to the structure being tested;
• If shaker testing is done, the amplitude of the free signal has to be tested;
• Frequency Response Function (FRF) are determined from the excitation and response measurements;
• Modes are identified from the FRFs;
• The analysis of the data can then be performed.
State of the Art

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3.1 Introduction

For many years all kinds of aircrafts, either production aircrafts or just prototypes, go through very thorough processes of verification and validation. More recently, more complex aircraft designs have added additional testing requirements that are connected to the increased use of composite materials, active systems and the need to quantify non-linear behaviour.

Ground Vibration Testing allows engineers to obtain experimental vibration data of the whole aircraft structure and use that data to validate its most various models, including structural dynamic models and flutter and aerodynamic models afterwards. One of the key aspects that are obtained from the updating off these models is the prediction the flutter behaviour, allowing thus to carefully plan the safety-critical in-flight tests.

The main objective of a GVT is to experimentally determine the low-frequency modes of the whole aircraft structure for validating and improving its structural dynamic model as part of the flutter clearance process [22].

Usually GVT is done in a later part in the project development timeline and the fact that the aircraft subjected to study is only available for a short period of time usually implies that GVT is done under a lot of pressure. Along with the fact that multiple configurations are usually tested, pre-test planning is very important and strongly advised and consequently it makes every improvement on the GVT environment a very welcomed help.

The ever increasing complexity and accuracy of the tests performed that follow more complex aircraft designs, and also the time stringent deadlines that demand GVT time reduction has motivated and increased international research in this field.

The cost of having an airplane standing still to have modal testing done very high. Consequently the path to the state of the art in has to take into consideration new measurement techniques but they can only truly be proven useful if they manage to reduce installation time and cabling requirements, reduced testing time, or if they otherwise deliver increased value to the testing itself.

In the past years a lot of effort was spent on trying to improve GVT: Optimisation of test strategies, new hardware developments and some small improvements of tools for test time reduction. With these improvements also the standstill period of the was reduced, along with the cost for the entity performing the testing, usually the aircraft manufacturer.

The new developments that are now being applied and the future of Ground Vibration Testing is what is going to be presented in this following chapter.

3.2 State of the Art in Large Aircraft Testing

The standard way of testing for the past decades has been, for large aircrafts, the phase-resonance method, or the commonly called normal mode testing [23] [24]. This method is very appealing for this application because it is capable of doing a separation of closely spaced modes.

Normal mode testing consists on doing a single sine excitation at the natural frequencies of the modes with a shaker system. With a proper choice of shaker position and the phase relation between
the sine excitation signals, the aircraft will be forced to move as a single-degree-of-freedom system, and
the vibration response will only contain a contribution from the mode of interest.

This method presents various advantages:

- The real modes of the corresponding undamped structure are directly measured.
- Each eigenvectors are excited at a high energy level.
- Linearity tests can be easily performed.

But it has one main disadvantage that is the testing time. Unfortunately it is a very time consuming
method and usually, to cover for that, it is complemented and sometimes partially substituted by phase
separation techniques.

These techniques will determine the aircraft modes by evaluating its FRF and so what happens is
that the majority of modes will be determined by these “side-techniques” but the the most important
modes are still based on normal mode testing.

The modes determined by the normal mode testing are called critical modes and they are consid-
ered critical if they [25]:

- Significantly differ from the predictions
- Show non-linear behaviour
- Are important for flutter calculations

In the meanwhile, further technological breakthroughs have been happening and some new tech-
niques have grown and proven to be able to produce good results.

In the case of the partners French ONERA and the German DLR [22] it is being used a combination
of Phase Separation Methods (PSM) and Phase Resonance Method (PRM) making use of the
individual advantages of the two methods.

Along with the normal GVT set-up, described further in Chapter 2, special accelerometers (mea-
surement range 0.5 Hz to 1000 Hz), carbon push-pull rods and long stroke modal exciters are used, to
assure the obtaining of good results. Along with that special slip tables and tripods, developed specif-
ically for this purpose, served as mounts for the exciters in order to dynamically decouple the exciters
from the scaffolding.

During testing, different spectral analysis methods are applied to the measured time domain data,
such as FFT and Peak Reference Hold (PRH) technique.

In their previous experiences with GVT and performed studies it was determined that swept-sine
excitation signal with exponential sweep rates performs best for large aircraft applications.

Taking into consideration modes that show non-linear behaviour during testing, a new approach
was developped and implemented in order to characterize this non-linearities [26]. This new procedure,
based on the concept of equal complex power, consists in simulating a single input multiple outputs
(SIMO) measurement with a virtual driving point instead of a conventional multiple inputs multiple outputs
(MIMO, but correlated inputs) measurement which requires multiple measurements with independent
excitation force patterns for FRF estimation [22]. After creating the FRFs, in parallel to the ongoing measurements, this data is analysed using commercial Phase Separation Techniques.

The person in charge of the operations during testing can be supported in the correlation task by a so-called decision window, which graphically presents the quality criteria of the modes compared for a fast selection of the most reliable modal data.

The PRM, since it is a time consuming method, as stated before, but it is accurate and thus is only applied for critical modes. Linearity checks and optimal exciter force amplitudes can be performed simply by increasing the overall excitation force level and checking the resulting FRFs.

According to [22], and shown in Figure 3.2, shows that this test strategy has approached its limit regarding its potential for GVT time reduction. The next step to achieve test productivity enhancement can only be resizing the objectives of the GVT, but this would mean cutting short the test database, i.e. the experimental data retrieved and this could severely hinder the updating process of the FE models in the different configurations.
3.3 Spatial-Optical Approaches

There are several approaches to non-contact image or laser vibration response measurement systems that could lead to an higher spatial resolution without the time and labour cost of installing hundreds of accelerometers as it happens with the conventional GVT methods. Following, the new relevant methods are listed and explained.

3.3.1 Photogrammetry

Photogrammetry is defined as the process of making precise measurements of an object from photographs of such same object [27] and photogrammetric methods are now used for performing 3D digitizing of test objects for modal testing applications. They can measure displacement by locate identifiable points on the structure in successive or different images. Using two or more cameras, 3-dimensional motion may be obtained, similar to triangulation[28] [29].

There are two systems that can be used in order to locate measurement points on a structure and for determining the orientation of sensors mounted on the test object. The first is a classical photogrammetry system that uses a digital camera and image processing software. The second one, a reverse photogrammetry system, is a roving digitizer that uses a calibrated panel to determine the camera position at each digitized point.

3.3.1.A Camera Photogrammetry System

Even though this method has been used for the last decade, its previous digitize time was poor, but with new improvements it is now being considered a valid an proven way of reducing the testing time by being a very effective method to place sensor in the structure being tested.

With Camera Photogrammetry systems photographs are used to check on the locations and orientation of various components in the test system.

When doing modal testing there are plenty of points that can be used for sensor location, but usually due to sensor number limitations, only a small portion is actually used. Consequently these points are scattered over a wide area and a high number is required in order to use this method in order to be able to locate the said points. Another consequence is that a high number of points are created with the sole purpose of making the task of having references between pictures easier.

This creates a disadvantage of having a difficult time consuming task of locating and referencing these points between photographs. In recent breakthroughs in this method is the ability to automate the process, making this process more time efficient [29].

3.3.1.B Photogrammetric Digitizer

A photogrammetric digitizer system makes use of a roving probe that is placed at each digitized point the structure and acquires the data by having the probe flashing infrared light into the sensor location area for a infrared camera to registed, also located in the probe.
As the operator roves the probe across the structure, the camera inside the probe is pointed towards an array of retroreflective targets mounted on a flat panel [29].

Two types of targets are used on the panel: circular barcodes and simple circular dots. When an image of the panel is processed by an image processing algorithm in the computer, the barcodes are used for calculation of the gross orientation of the probe and the more numerous dots are used to provide redundant data to refine the solution.

Then the 2D digital image retrieved by the probe is processed by a computer and the 3D coordinates are calculated and can be printed to further use in structural dynamics analysis software.

This method can deliver accurately sensor locations and orientations but it can be complicated to use in very large structures, as was experienced by a Boeing modal testing team in [30] while static testing a Boeing 737.

### 3.3.2 Projected Dot Videogrammetry

Videogrammetry expands the methods and techniques of close-range photogrammetry and applies them to a sequence of images to generate time history data, enabling the object to be characterized dynamically [31].

Traditional videogrammetry relies on physically attached retro-reflective targets that may change the responses of the structures being measured by changing its mass and stiffness. The time to attach the target also is a serious disadvantage.

Videogrammetry, as photogrammetry, achieves the greatest accuracy with high contrast, solid-colored, circular targets, and thus a high-density grid of projected circular targets, called dot projection, is a viable alternative.

Knowing location of each target allows those points to be tracked in a sequence of images and since, over the time, the object changes shape or position independently of the dots, its dynamic characterization, such as deployment or vibration, can be characterized by tracking the overall 3D shape of the object instead of tracking specific object points. Motion is measured only in the direction towards the illuminating light source.

This method can have from two cameras up record the oscillating system, creating sequences of digital images used in the videogrammetric analysis [28]. At a sampling rate of five frames per second, the data acquisition system can have a set of 200 images over 40 seconds (in four 10 second cycles) for each camera position.

The processing software loads multiple sequences of images, associating each one with the correction parameters for the camera used to record, removing, thus, any distortions of the images due to imperfections in the cameras or lenses, enabling accurate measurements.

One thing that can happen during testing that must be accounted for is, as the object is moving and vibrating, the targets on the edges can slip in and out of visibility depending on the point in the cycle. Consequently, since these points are not visible in all the images they have to be discarded from every time step.
3.3.3 Laser measurements

Another field in modal testing that has shown substantial growth is the field of laser measurements. These laser-based response measurement techniques, such as: holography, ESPI and laser-Doppler velocimetry, show great advantages over the conventional measuring systems.

The fact that they don’t have to have any contact with the surface being measured is a very good fact and they give the possibility of doing full-field measurements, where most of the surface of the structure can be measured with a mesh density that is orders of magnitude greater than could be achieved with individual discrete transducers [28]. This feature can lead to newer technologies and methodologies that can show greater quality and an higher number of measurements that would be impossible to obtain using the conventional methods.

3.3.3.A Laser Doppler Vibrometer

There are various vibration mode shape measurement techniques that can be applied that use a Laser Doppler Vibrometer (LDV).

The Laser Doppler Vibrometer (LDV) operates by measuring the velocity of a point addressed by a focused laser beam, making use of the Doppler shift phenomenon between the incident light and scattered light returning to the measuring instrument.[32]

The fact that there is no contact with the structure is a great advantage, it avoids mass loading the structure and it doesn’t tamper with the measured results. It is a versatile method, where the points being measured can also be quickly changed by interposing adjustable beam-directing mirrors. Consequently, this method allows to obtain response measurements from hundreds to thousands of points without structural changes, that can happen when using a large number of accelerometers, and without multi-channel instrumentation.

One disadvantage that is present when using the LDV is that there is an always present optical phenomenon of speckle noise. What this means is that, in a typical point-by-point LDV survey, the LDV signal is spoiled in a minor number of points by “speckle drop-out” and consequently some form of data smoothing may be necessary to give a true picture of the spatial mode shape.

A new technique developed making use of the LDV is the Scanning Laser Doppler Vibrometer (SLDV). It continuously scans over the surface of a sinusoidal-excited structure, measuring the velocity response giving an amplitude-modulated sine wave and mode shapes, that after demodulation serve as output.

This method has some advantages over the previous stepped LDV: the amount of data required to describe the mode shape is much lower and the speckle noise which gives very short asynchronous random pulses with laser scanning which, in the frequency domain, corresponds a low-level broad-band background noise [32].

One disadvantage that is present with this method is that the SLDV methods can only obtain response measurements from a excitation a single input frequency at a time. Obviously it is possible to repeat this process for a series of different frequencies in a stepped-sine test to then establish FRFs with a reasonable frequency resolution, only at time expense, and only then modal analysis can be performed. Depending on the resolution that is acquired, the testing can have different testing times.
Other new methods have been demonstrated in the scientific community [28], among them there is:

- **A “fast scan” (point measurements):** It is used to measure a single or a low number of cycles of response to sine excitation and measurements can be repeated in a short time.

- **Continuous Scanning Laser Doppler Vibrometer (CSLDV):** It can be used to define mode shapes along a line, ellipse or other scan pattern using de-modulation. Coefficients of polynomials or Fourier series may be determined to describe the mode shape along the scan line.

- **Pseudo-point Scanning Laser:** It applies a Multiple Discrete Time Systems approach to extract equivalent point measurements from acquired CSLDV signals.

### 3.3.4 Test Planning

When discussing test planning, the name is very self-explanatory. It consists in simulating the test conditions and evaluate if the estimated results are good enough for the application in cause and if not, optimizing the parameters in order to have the best test setup possible.

This prevents wasting precious time that could be spent testing, which is a serious matter taking into consideration that in the vast majority of the time, testing is done in a very short window of time and it implies very high costs.

In this very money and time saving operation, there is usually a finite element model with which one can simulate the testing conditions. Having this model as base, the impact of each parameter to be measured in the final results and how the setup affects these measurements can be determined. This eliminates the risk of the past method that relied in the operator experience and judgement that could be proven wrong and was surely not 100% safe.

With the use of the FEM the following parameters can be optimized in order to have the best setup as possible [28]:

- Location of excitation position(s).
- Location of suspension and boundary conditions.
- Set of response DOFs to be measured.
- Location of measurements points and accelerometers

### 3.4 Further Developments

Even with all the state of the art procedures that have been stated here, there is still room for improvement in relation to Ground Vibration Testing. The next big leap will only possible to happen if there is a reduction of the information content of the test data, or more importantly, if there is a modification of the model verification process. Nevertheless, there can still be some medium advances in relation to test productivity and data quality enhancement with more modern tools [22].

Referring to other further developments in the field, there are some that can be mentioned, listed below:
• **Measurement hardware**: One example a hardware with ever growing capabilities is the computers used for data processing and data evaluation phases, that will keep reducing the testing time as their performance keeps getting higher. Special developed sensors, more specifically wireless sensors could save time during the installation phase. But, in the end, these systems are closely connected with the characteristics of the telemetry systems themselves, since they have limited channels and each sensor has to be transmitting in a individual separate frequency.

• **Tool and method developments**: With new methods and tools available, an increase in test productivity can be expected, especially if non-linear modes are present. Present day technologies are not very reliable in detecting non-linearities from swept-sine measurements. Consequently what happens is that these modes from non-linearities are identified as equivalent linear models instead of being properly identified and used to have better models with better predictions.

• **Test strategy enhancements**:
  
  – Development and integration of correlation-tools for in-test use;
  
  – Simple reorganisation of working stages can optimize the testing process greatly;

• **Use of the flight test accelerometers for modal identification while on the ground**. This new way of testing could have two major advantages:
  
  – The installation and dismantling time of external accelerometers can be greatly reduced;
  
  – The proper operation of the flight test sensors can be evaluated long before the Ground Vibration Testing (GVT) and consequently damaged sensors can be replaced early on.

• **Application of internal excitation**: The aircraft’s control surfaces can be used in order to excite the aircraft. This approach can simplify the testing phase very much, but there is one disadvantage that is that it may not be able to excite the higher modes with enough energy due to the constraints of the control surfaces themselves. This approach has already been successfully applied, has can be seen in [33].

• **Multi-objective testing**: One test can be used to accomplish several objectives by using the data in multiple ways. That implies that tests previously performed with a sole purpose that have to be done anyway included in the certification process can be used for modal identification purposes, e.g. take-off and landing tests.

### 3.5 Summary

With the ever increasing in already very high development costs in the aircraft design field, the common goal is to push technology and new methods and techniques to the next level in order to shorten even more the current testing time and at the same time increase the amount and quality of the test data.
In the past, the time spend doing GVT was shortened with the use of a combination of phase resonance method and phase separation techniques instead of using just the first. This change lead to a reduction in testing time, in large aircrafts, by \( \frac{1}{3} \) [22].

More small improvements are being accomplished due to test strategy improvements or enhanced softand hardware developments.

An reavaluation of the verification and validation process of aeroelastic models from a structural dynamics point of view is required and the role of the GVT has to be carefully taken into consideration if it is still the best procedure that can be taken or if new aproaches such as using information from other tests performed during aircraft development can be used and if they should be used.
4

Finite Element Models

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4.1 Introduction

This section discusses the use of Finite Element models during the GVT. The use of FEMs is common and is the standard tool for structural analysis.

The finite elements method is a very useful tool that allows engineers to have simulations of the behaviour of a structure. It can be used not only in structural analysis, with elastic and dynamic behaviours, but also in thermal and electromagnetism analysis for example. These numerical methods are important because they can be created and tested even before the construction of the structure itself, preventing redoing the physical model in case something was poorly planned. Optimum conditions can, thus, be found saving engineers precious time and money.

With these finite element models real conditions can be simulated and pre-planned, a very useful feature for engineers. However, caution is advised when using FE models because, in the end, these models are an approximation of reality. In order to have the best reliable models, verification and validation with experimental data should be used. This topic will be discussed later on.

In structural analysis and dynamics, an analysis performed with finite elements can be used in order to study and solve vibration problems. Vibration related issues are very important because they are one of the biggest factor in problems involving noise and fatigue damage.

The effectiveness of this right-first-time design approach however strongly depends on the capability to validate the numerically obtained structural behaviour. If results cannot be trusted, nor can the numerical model and simulations of design modifications be correctly evaluated.

In the model used, one common uncertainty can come from each finite element having to have defined its element type and formulation. Using different types can lead to different results, so the engineers should take this into consideration.

A problem when dealing with modelling is that the structure designed can be proven to be different from the actual built one, thus causing errors that should be accounted for in the beginning due to building procedures and possible manufacturing problems.

Another situation is that the geometrical representation of the structure and the mesh density. The accuracy of the model is intertwined to its mesh density. This requires that equivalent stiffness and masses have to be estimated and input.

Being available prior to the GVT, it can be used to make predictions on the aircraft dynamic behaviour, sets of analytical mode shapes and to optimize the test arrangement and duration. The numerical mode shapes obtained can be used to locate optimum references, by avoiding the nodal points present. With the analysis of the early FE model we can have:

- Estimated aircraft GVT masses
- Defined GVT boundary conditions

The normal modes obtained from the FE model before the actual GVT represent an early estimation of the aircraft modal information such as the frequencies and mode shapes. This information can be used to plan the test and decide before any actual testing takes place the best excitation condition, possible shaker locations and sensors (accelerometers) locations.
4.2 Finite Element Model

The finite element model serves as the basis for all the testing that was performed and also for further analysis, thus the need for verification validation and updating of this model.

This model will be used to perform structural analysis in Nastran and serve as basis for the aerodynamic analysis update as well. In Figure 4.1 it can be seen the steps of the overall updating process.

![FEMtools and ASWing Updating Diagram](image)

**Figure 4.1: Overall Updating Process**

From the updating procedure that is going to be done for the Nastran FE model, the results obtained from the experimental testing are going to be used to update the models in order to perform further analysis:

- Flutter analysis;
- **Structural mode shapes and frequencies**;
- Stability derivatives;
- Static deflections.

The model uses beam elements and this is because the purpose is to analyse the aircraft as a whole and study its dynamic behaviour instead of analysing specific parts of it. This makes the model fast to run and guarantees a better matching with the model with ASWing.

The model also has aerodynamic features where the elements used are not linked to the structural parts of the aircraft, in a way that they won’t influence the data for the structural and modal analysis. The aerodynamic elements were created in order to have Doublets model, with the panels theory.

For the purpose of this work, only the structural parts were important and dealt with, i.e., only the relevant parts for static and/or modal analysis.

This modal was made in the Nastran software and the way it combines the two meshes, the aerodynamic and the structural one can be explained briefly. The aerodynamic mesh, that uses specific elements for the panel method (CAERO1 and CAERO2), and the structural mesh, that uses beam elements CBEAM and rigid elements, are interpolated both using a splining method that transfers forces and displacements between the two. The major advantage of this method is that it works even if the two meshes are not coincident in the model.

The rigid elements present in the model, i.e. ribs, are only present to:

- Place point masses connected to those elements;
• To be able to interpolate the nodes in the extremity of the ribs with the aerodynamic mesh: it creates an interpolation surface, since the beams of the spar are just lines making interpolation a strenuous task;

• Create nodes to be able to perform modal analysis and be able to have better correlation and mode shapes

![Beam Representation](image1.png) ![Stick Representation](image2.png)

Figure 4.2: Finite Element Model In Different Visualizations

### 4.2.1 Modal Results

From the completed Finite Element Model (FEM) we can have the basis for the estimations of the results that can be found while performing Ground Vibration Testing (GVT). This values are very important to have, if possible, before the GVT because the instrumentation decisions can be done beforehand and prevent situations were modes cannot be measured because of unfundamented decisions.

This results allow to have a better choice of hardware as:

• The shaker system that can excite in the frequencies of interest;

• Data collection systems, and their resolution and accuracy;

• Support systems without resonance modes close to the modes of the aircraft.

But in the end, these results prove very useful to provide a better first feeling if the results obtained from the GVT are any close to them. If great disparities are found, it could mean that one of the two is probably wrong, but its always given priority to the GVT results.

Another advantage taken from this model is that optimal excitation positions can be simulated and determined in order to guarantee that a good excitation of the set of modes required for model updating of the complete aircraft structure.
Table 4.1: FE model Modal Frequencies for the Rigid and Flexible Wings Configuration

<table>
<thead>
<tr>
<th>Rigid</th>
<th>Fixed</th>
<th>Flexible</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Free</td>
<td>7.39</td>
</tr>
<tr>
<td></td>
<td>Free</td>
<td>21.70</td>
</tr>
<tr>
<td></td>
<td>Free</td>
<td>37.07</td>
</tr>
<tr>
<td></td>
<td>Free</td>
<td>53.06</td>
</tr>
<tr>
<td></td>
<td>Free</td>
<td>65.20</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>103,00</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>53,00</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>63,00</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>90,00</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>91,00</td>
</tr>
<tr>
<td></td>
<td>Cantilever</td>
<td>108,00</td>
</tr>
</tbody>
</table>

In Table 4.1 we have the frequency results prior to the model updating for all the simulated cases in order to compare later on to the experimental results.

4.3 Summary

In the process of developing a new aircraft, a FE model is created and developed. This model is usually used for structural, aeroelastic and modal analysis.

When developing this model, all the correct masses (test-measured) and boundary conditions are input. Then the analysis of the flexible modes is done and it is determined, in comparison with the GVT obtained data, if the model needs to be updated or not.

Usually a 5% margin is used for the difference in all the frequencies of the flexible modes present, if the results lie inside this margin, the model does not need to be updated, however, the updating should be done nevertheless to prevent wrong results in future phases of the project that is being developed. If this margin is not respected, then the model needs to be updated without a doubt.

An important note is that this updating of the model needs to account for local effects from the testing itself or absent components.
5.1 Introduction

Experimental testing is the most effective method to determine the true characteristics of a structure as it truly is. Experimental tests are still found, to this day, to be expensive and time-consuming, but, in order to be able to produce high quality, reliable structures, they are indispensable, being used to validate and update computation models.

Experimental testing can deliver limited, but various results. In structural dynamics resonance frequencies can be measured with high accuracy, but limited in range. Mode displacements can also be targeted when testing, but the results are generally less accurate.

The real structures being tested can have infinite number of DOFs and an infinite number of modes that could be measured. There is no limit to the number of unique DOFs between which FRF measurements can be made and it comes to the engineer in charge to define and select the range of the results and number of DOFs to fit the purpose of the results. Since experimental testing is expensive, test locations are limited and usually reduced to the minimum of locations possible without hindering the results obtained.

Experimental Modal Analysis (EMA) to be done properly requires expertise and care along with a big amount of instrumentation. During testing noise sources should be reduced to an absolute minimum since they can distort the results, where the most common sources can be: [34]

- Related to the test configuration: fixtures, excitation method, transducer locations and dynamic loading;
- The instrumentation: calibration, distortions and data acquisition;
- Actual noise from the working environment.

The important role of Ground Vibration Testing (GVT) has been and will remain to provide the reference data for finite element model updating. The most suitable reference data are in decreasing order of accuracy: resonance frequency, mode shape, modal displacement and modal damping. Some model updating methods also make use of the Frequency Response Function (FRF). The processing of the data resulting from the experimental test was processed using Vibrant Technologies’s MEscope program.

5.1.1 Beam Analysis

As an introduction to EMA and as an introduction to the type of results obtained by doing Ground Vibration Testing (GVT), a modal analysis of a simple cantilevered beam was made. This served as a learning tool before testing in the real aircraft and accelerate the learning curve of all the tools, procedures and analysis to be done in modal testing.
Figure 5.1: Beam Being Tested in Fixed Cantilever Boundary Condition

The analysis performed on the beam was done with the impact hammer and the accelerometers mentioned in Chapter 2.

Figure 5.2: Graphic Representation of the Overlaid FRF Response of the Cantilevered Beam

This beam, being a simple case, has a response that is also simple. It can be seen in Figure 5.2, as an overlaid image of the FRF response, that the modes are very well defined with very sharp peaks not only in the magnitude part of the data but also in the imaginary part.

In Table 5.1 the modes frequency is listed and, predictably, there are 5 very distinct modes that coincide with the number of peaks present in the magnitude and imaginary overlaid data. This, at first would be interpreted as the final frequencies, but, it is only after the curve-fitting that those values can truly be determined, including, not only the modal frequencies, but also the modal damping and mode shapes.

Table 5.1: Modes extracted from the ODSs and their respective modal frequencies

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>13.3</td>
</tr>
<tr>
<td>2</td>
<td>78.4</td>
</tr>
<tr>
<td>3</td>
<td>216</td>
</tr>
<tr>
<td>4</td>
<td>383</td>
</tr>
<tr>
<td>5</td>
<td>418</td>
</tr>
</tbody>
</table>
With the FRF data, the curve-fitting process can be done. From the data gathered, an analysis was made using three different curve-fitting methods:

- AF Polynomial
- Complex Exponential
- Z Polynomial

A good way of ensuring that the results are coherent with each other, is to make use of the MAC. The MAC number displays the relation between the modes being analysed, is a number, with values that can go from 0 to 1, with the value 0 meaning that the modes have nothing in common and 1 that we are comparing identical modes.

![MAC Plots comparing the different curve fitting methods](image)

Figure 5.3: MAC Plots comparing the different curve fitting methods

From the previous figures it can be seen the correlation between the results of the 3 curve-fitting methods and all three methods have a fairly good correlation between them. The method that approach the reality better was considered to be the Z Polynomial, since its results were the ones that best approximated the measured data, and so the final results can be summarized in Table 5.2.

For the results of the other methods, they can be consulted in Tables A.2, A.3 and A.4.

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
<th>Damping</th>
<th>Damping [%]</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>13.2</td>
<td>0.555</td>
<td>4.19</td>
<td>0.992</td>
</tr>
<tr>
<td>2</td>
<td>78.3</td>
<td>0.581</td>
<td>0.742</td>
<td>0.966</td>
</tr>
<tr>
<td>3</td>
<td>216</td>
<td>1.83</td>
<td>0.847</td>
<td>0.962</td>
</tr>
<tr>
<td>4</td>
<td>382</td>
<td>2.65</td>
<td>0.693</td>
<td>0.894</td>
</tr>
<tr>
<td>5</td>
<td>418</td>
<td>2.77</td>
<td>0.664</td>
<td>0.842</td>
</tr>
</tbody>
</table>

5.2 QT1 Aircraft

After this introduction to EMA with the simple case of the cantilevered beam, the testing proceeded to the next phase that is testing the actual aircraft, the QT1.
The QT1 aircraft has two wing configurations to be tested: one is a conventional rigid wing and a so-called “flexible wing” where the inboard part of the wing is made of an less stiffer aluminium spar.

The format of these two wings is exactly the same, so that the existing FE model has the same basis for both cases, only different structural properties that have to do with the inboard parts of the wings, that were built differently.

The two configurations were tested in order to update the existing finite element models so that they can be used for aerodynamic and aeroelastic analysis so that the QT1 can be flight cleared. For that reason, a tight schedule was imposed and a very long series of test were done in a very short time span.

**Over 200 hours of testing** were input into testing, with the various configurations and testing parameters. The results that are going to be presented only show a small part of the work put into the testing, since all those hours can be briefly resumed into a couple of result tables. Ground Vibration Testing may be considered a trivial type of work because its results can be so rapidly shown, but it is nothing of the sort. Experimental work comes with a lot of challenges that need to be fixed fast and efficiently in order to be able to keep up with the schedule that an engineer must comply with.

### 5.2.1 QT1 Wings

For each wing set, over 160 points were measured, including leading edge points, relevant for in-plane mode measurements. The points were mapped and were then marked on the wings for easier and faster testing.

For the case of the flexible wings, the inboard part of the wing, were positioned differently, because due to the different structure properties of the wing, the points were placed on the spar and ribs instead of the same points as in the rigid case, the outer skin.

This happens because in the inboard part of the flexible wings the existing skin panels are not structural elements, just aerodynamic. Which means, if they were to be impacted, no relevant data would be collected, because even though they are connected to the structure, they are not truly a part of it.

For the testing purposes, only the top panels were removed in order to not change the proprieties of the wings with the missing weight related to it. It can be see in Figure 5.5 that the measured points are only placed in the structural parts: the spar and the ribs, where proper modal testing can be performed.

![Figure 5.4: Impact Test Points on the Rigid Wing](image)

51
For both cases, no impact points were placed at the control surfaces because, in the same way as the wing panels for the flexible wings, they wouldn’t add relevant data to the testing.

So the logic behind the points placement position was to have center points as near as possible to the center of the structural parts of the wings and they have them spread out until the leading edge and as far to the leading edge without including the control surfaces, in order to have the best representation of the mode shapes, specially the torsion modes that need spread out points further from the center of the wing in order to be identified.

Before analysing the results, a comment is made here about the unpredictability of experimental testing. When testing the first QT1 configuration there was a point halfway through the wing that showed really poor results. When hitting it with an impact hammer a distinctive sound, different from the rest of the wing, that was probably due to an air bubble or to a structural element/connection.

This difference can be easily seen in Figure 5.6, as displayed in the FRF magnitude plot.

In it we have a representation of the FRFs corresponding to the impact points numbers 55, 56 and 57. The impact point 56 presents results that are very low in magnitude compared to the nearest adjacent impact points in the same transversal line in the wing, 56 and 57. Several tests were taken at this point.
but even with different hammer tips, no valid results could be obtained. Consequently, this point was removed from the data analysis.

### 5.2.2 Conventional Rigid Wings Testing Results

In Figure 5.7 it can be seen the QT1 with the conventional wings installed and ready for shaker testing with the foam simulating the free free boundary condition.

Since this was the first time testing an aircraft, everything was slower at first, as can be imagined, and setted up the testing procedures for the rest of the testing phase.

![Figure 5.7: QT1 Aircraft with Rigid Conventional Wings](image)

Since all the testing procedures were already explained in previous chapter, it is stated here only the results obtained for testing configurations: fixed cantilever and free free boundary conditions.

This set of wings didn’t present any major challenge since the results, as expected from the what was seen in the FE models, were all linear.

For the fixed cantilever condition impact testing was performed, as for the free free conditions, shaker testing was the chosen technique.

**Table 5.3: Results for the Rigid Wing in Fixed Cantilever Boundary Condition (Impact Testing)**

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency</th>
<th>Damping [Hz]</th>
<th>Damping (%)</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7.31</td>
<td>0.711</td>
<td>9.68</td>
<td>0.936</td>
</tr>
<tr>
<td>2</td>
<td>36.1</td>
<td>2.54</td>
<td>7.02</td>
<td>0.987</td>
</tr>
<tr>
<td>3</td>
<td>61.3</td>
<td>3.02</td>
<td>4.92</td>
<td>0.041</td>
</tr>
<tr>
<td>4</td>
<td>82.2</td>
<td>2.86</td>
<td>3.48</td>
<td>0.67</td>
</tr>
<tr>
<td>5</td>
<td>106</td>
<td>5.34</td>
<td>5.01</td>
<td>0.366</td>
</tr>
<tr>
<td>6</td>
<td>138</td>
<td>7.23</td>
<td>5.21</td>
<td>0.0565</td>
</tr>
<tr>
<td>7</td>
<td>179</td>
<td>5.75</td>
<td>3.2</td>
<td>0.65</td>
</tr>
</tbody>
</table>
Table 5.4: Results for the Rigid Wing in Free Free Boundary Condition (Shaker BC Foam)

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
<th>Damping</th>
<th>Damping (%)</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7.48</td>
<td>0.0728</td>
<td>0.973</td>
<td>0.836</td>
</tr>
<tr>
<td>2</td>
<td>19.2</td>
<td>0.0193</td>
<td>0.1</td>
<td>0.913</td>
</tr>
<tr>
<td>3</td>
<td>26.7</td>
<td>0.338</td>
<td>1.26</td>
<td>0.995</td>
</tr>
<tr>
<td>4</td>
<td>30.8</td>
<td>0.272</td>
<td>0.883</td>
<td>0.919</td>
</tr>
<tr>
<td>5</td>
<td>33.5</td>
<td>0.373</td>
<td>1.11</td>
<td>0.99</td>
</tr>
<tr>
<td>6</td>
<td>39.3</td>
<td>0.788</td>
<td>2.01</td>
<td>0.318</td>
</tr>
<tr>
<td>7</td>
<td>46.1</td>
<td>0.237</td>
<td>0.515</td>
<td>0.964</td>
</tr>
<tr>
<td>8</td>
<td>50.2</td>
<td>0.643</td>
<td>1.28</td>
<td>0.964</td>
</tr>
<tr>
<td>9</td>
<td>67.3</td>
<td>1.26</td>
<td>1.88</td>
<td>0.96</td>
</tr>
<tr>
<td>10</td>
<td>89.9</td>
<td>1.54</td>
<td>1.71</td>
<td>0.663</td>
</tr>
<tr>
<td>11</td>
<td>126</td>
<td>5.64</td>
<td>4.46</td>
<td>0.367</td>
</tr>
</tbody>
</table>

5.2.2.A Boundary Conditions Testing

For this set of wings, different boundary condition simulations were done, firstly the fixed cantilever was done as seen in Figure A.5, where the wing is mounted to a rig that simulates the placing of the wings in the aircraft and that is afterwards bolted down to a table ready for this type of tests.

For the free free condition, both bungees and foam conditions were used, as seen in Figures 5.7 and 5.8.

Figure 5.8: Bungee Configuration For Shaker testing With Rigid Wings
5.2.3 Flexible Wings Testing Results

For the flexible set of wings, two cases were analysed: uninstrumented wings and instrumented wings. The purpose of this was to see how much it would affect the presence of all the cabling, sensors and actuators in the vibration modes of the wings.

The structural parts remain the same and the testing points are the same in both cases so that a complete correlation can be done for these two cases. The instrumented wing added strain sensors for a different mid-flight testing and static load tests for purposes beyond this thesis.

With all the wiring and sensors, the points remained in the same position but in some cases the points were no longer accessible, either from cables or sensors placed at the same spot or very near it, but in the end less of 5% of the locations were lost. Fortunately, this difference was not noticeable in the results.

In Figure 5.9 it can be seen the uninstrumented wing set in its full span. It is relevant to say that the outboard part of the wing, from where the visible part of the spar ends to the wing tips (the major grey components), has the exact same dimensions and test points as the rigid wings.

The same testing procedure performed for the rigid wings involving shaker testing was done with this different wings set. Unfortunately, the results were far from optimal. The fact that this set of wings is less stiff because of its own structural configuration made it harder to acquire proper measurements.

A good shaker installation was very hard to find and countless hours were spent trying to improve it by having different support systems, as can be seen in Figures A.6, A.7 and A.8 different shaker mounts and even different shaker control signals.

In the end, the testing was finally performed resorting to impact testing, that was able to capture the lowest frequency modes with much better results than previously, thus guaranteeing the projects requirements.

![Figure 5.9: Flexible Wings](image)

5.2.3.A Non Instrumented

In relation to the non instrumented case, this served as a testing platform for the actual important test that was the instrumented wing, which is the one that is going to be used to update the FE model, since it is the one closer to reality.

Consequently, for this case, all the testing was done not only to perfect the testing techniques and methods in the laboratory but also to actually determine the difference that the installation of the instrumentation on the wing has on the modal parameters to be determined.
5.2.3.B Boundary Conditions Testing

As it was stated earlier, the non instrumented case served as a pre-test for the instrumented case. Being so, it served to compare the two techniques developed to test the free free boundary condition: the use foam or bungees.

The objective is to see if the two techniques yield the same results and if not, find out why, and choose only one to perform later on the instrumented wing. In theory, both cases should be the same since they simulate the same condition, but it is know that all that involves experimental testing does not always follow the theoretical assumptions.

With the results from Figure 5.10, the two conditions are now compared in Figure 5.11 in relation to their frequency shift having the bungees case as the reference.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Bungees [Hz]</th>
<th>Foam [Hz]</th>
<th>% Shift</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.25</td>
<td>2.69</td>
<td>9,846154</td>
</tr>
<tr>
<td>2</td>
<td>18.5</td>
<td>18.8</td>
<td>1,621622</td>
</tr>
<tr>
<td>3</td>
<td>55.9</td>
<td>58.1</td>
<td>3,935999</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mode</th>
<th>Bungees [Hz]</th>
<th>Foam [Hz]</th>
<th>% Shift</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>13.1</td>
<td>12.1</td>
<td>7,633588</td>
</tr>
<tr>
<td>2</td>
<td>16.4</td>
<td>16.6</td>
<td>1,219512</td>
</tr>
<tr>
<td>3</td>
<td>71</td>
<td>70.7</td>
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<tr>
<td>4</td>
<td>75.1</td>
<td>75</td>
<td>0,133156</td>
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</table>

<table>
<thead>
<tr>
<th>Mode</th>
<th>Bungees [Hz]</th>
<th>Foam [Hz]</th>
<th>% Shift</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>38.3</td>
<td>38.2</td>
<td>0,261097</td>
</tr>
</tbody>
</table>

Figure 5.10: GVT Results for the Non Instrumented Wings

Figure 5.11: Frequency Shift for the Non Instrumented Wings in the Free Free Boundary Conditions

Analysing first the Figure 5.11 it can be seen that there isn’t a big frequency shift percentage for the majority of the cases, excluding the two lower energy modes, the 1st Out of Plane mode and the 1st In of Plane mode. This is probably due to the fact that since they are the two lowest energy modes, a slight difference in the energy distribution for the two configurations can have a more profound impact in their results.
Taking a look at Figure 5.12 it can be seen that, except for the 1st Out of plane mode, that due to its low energy has a non satisfactory mode shape and causes it to have a low MAC value of 0.604, but with visual inspection it was determined that the modes are, in fact, the same.

With very high MAC values overall between the two results of the different boundary condition’s test it can be deemed that the two testing procedures result equivalent measurements.

Consequently, even though there is a slight difference in some modes, it can be said that: with this results we can assume that the two boundary conditions are, with a small margin of error, similar.

### 5.2.4 Instrumented Wings

After the non instrumented analysis, where only one free body condition in the laboratory was proved to be needed in order to simulate this condition.

Comparing the two conditions, it must be written a side effect that was seen when using foam mattresses was that, due to the weight of the QT1, it would sink into the foam rendering the test results while testing impaired.

Being based on experimental experience, with a major problem stated above with the use of foam mattresses and where the bungees technique is easier and safer to test, the Bungees boundary condition was the chosen one.

After this choice, the testing proceeded with the instrumented wings rendering the final results shown in Figure 5.13. These results are the ones that are going to be used for the model updating since they are the ones closer to reality.

### 5.2.5 Instrumented Vs Non-Instrumented Wings Results

Now that the results from both instrumented an instrumented wings are available, a comparison between the two can be made and the effect of placing all the sensors, actuators, cabling and so on, in the aircraft can be quantified.

In Figures 5.14(a) and 5.14(b) it can be seen the frequency shift percentage for both the fixed cantilever an bungees case.

In these two figures it can be seen that the difference is in general around 5% for the fixed cantilever case and fluctuates from 1.5% to 8% in the bungees case. The steady difference on the fixed cantilever
Figure 5.13: Final Results for the Instrumented Wings

<table>
<thead>
<tr>
<th>Fixed Cantilever</th>
<th>Bungees</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Out of Plane Modes</strong></td>
<td><strong>Out of Plane Modes</strong></td>
</tr>
<tr>
<td>Mode #</td>
<td>Frequency [Hz]</td>
</tr>
<tr>
<td>1</td>
<td>2.89</td>
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<tr>
<td>2</td>
<td>17.8</td>
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<td>3</td>
<td>55.9</td>
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<tr>
<td>4</td>
<td>84.1</td>
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</table>

<table>
<thead>
<tr>
<th><strong>In Plane Modes</strong></th>
<th><strong>In Plane Modes</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Mode #</td>
<td>Frequency [Hz]</td>
</tr>
<tr>
<td>1</td>
<td>12.9</td>
</tr>
<tr>
<td>2</td>
<td>35</td>
</tr>
<tr>
<td>3</td>
<td>22.7</td>
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</table>

<table>
<thead>
<tr>
<th><strong>Torsion Modes</strong></th>
<th><strong>Torsion Modes</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Mode #</td>
<td>Frequency [Hz]</td>
</tr>
<tr>
<td>1</td>
<td>37.4</td>
</tr>
</tbody>
</table>

(a) Fixed Cantilever Boundary Condition  
(b) Bungees Boundary Condition

Figure 5.14: Frequency Shift Between the instrumented and the Uninstrumented Case for the Fixed Cantilever and Bungees Boundary Conditions

case can be explained with the constant mass loading that doesn’t affect the modes themselves, just their frequency.

In the bungees case, there isn’t exactly a common ground to be able to offer a proper explanation, but the two biggest shifts are seen in the 1st Out of Plane mode, where the 6% difference reflects on a 0.21 Hz difference, which is not truly significant, and on the 2nd In Plane mode where the difference is at 8.54% difference where it is the biggest value on this comparison. This could be due to the fact that the actual placing of all the instrumentation affected more this case in particular, causing a more profound shift.

5.3 Pre-test Analysis and Planning

In order to save time while testing and planning, the FEMtools pre-test analysis tool proves itself very useful. Unfortunately, there wasn’t access to FEMtools in the early stages of work and so the planning was done non-numerically, using engineering judgement. Consequently now it can only be verified if this judgement was close to the optimal case and if was satisfactory.
The first step in this pre-test analysis, since this was performed after the actual testing, is to determine if the actual sensor placing was an actual good choice. For that, it was input, in the same FE model node correspondent at the installation point on the real wing, the 2 sensors in each wing tip. The mass of the accelerometer was another input, with 7 grams each tri-axial accelerometer. The virtual sensor placing can be seen in Figure 5.15(b).

To be able to determine if this configuration is a good one or not, an auto MAC, that stands for a virtual calculation of the MAC values of the modes that would be determined with this sensor configuration, was calculated. The graphic representation of this Auto MAC can be seen in Figure 5.15(a).

For the optimum case, it would have shown a perfect diagonal in the 3D plot, meaning that the results would have a perfect correlation between the testing and the FE model, which clearly is not the case. The complete diagonal is present, but it is apparent that there is what can be called "noise" results that scatter the values throughout the plot. This means that can happen that some test modes can be correlated to the actual FE model without being the actual mode, which is a problem.

Luckily, as seen in the previous chapter, there wasn’t a situation where there was this type of situation, mainly because all the modes presented linear behaviour. If that wasn’t the case, there would have been a high difficulty in correlating the test modes to the FE model ones.

Since, it is perfectly clear that this was not the optimum sensor location for measuring the modal data for this model, it is necessary to find it in order to know how it could have been done better and to improve it, if it is there any further testing in the future.

5.3.1 Optimum Sensor Placement

The optimal sensor placement position can be determined with FEMtools, and a variable number of sensor can be input and 3 different cases were tested.

The placement of the sensors is done throughout the model, in a way that the sensors can measure with the best quality the biggest number of modes. The placement can be seen in Figure 5.16 and the respective Mac plots in Figure 5.17, all of them simulating triaxial accelerometers.
The difference between the case with the lowest amount of sensors, 15, and the actual one used, with just 2, is tremendously notorious. This has to do with the fact that there is a higher number of sensors and they are placed all over the model, being able thus to have a much better measurement of the responses of the structure.

Nevertheless, the Auto MAC, Figure 5.17(a), is not yet perfect. It can be seen that using the double of the sensors can show very good results. There are still two modes that can have a still considerable high correlation that could be a problem in the post process analysis, but, if accounted for in the beginning, it can be safely averted.

For the case of 53 sensors, a exaggerated and impracticable number of sensor, it was tested with the purpose of trying to see where the limit of the best measuring capability was.

It can be seen that the difference between Figure 5.17(b) and 5.17(c), is practically none, which means, that at 30 sensors, the optimal sensor placement location diagram was already achieved.

After the determination of the optimum point, there has to be a critic analysis if it would be feasible to do so. With the placement of 30 sensors in the aircraft would mean that a weight of 210 grams would be added, without considering the cabling that would add non negligible weight. Mass loading would be an unavoidable problem. Another very important fact would be the cost of such operation.

Accelerometers of high precision are very expensive pieces of equipment and performing a test of such magnitude would be impracticable for most companies. Considering this factors, a mid-way configuration would be ideal: one where there are just enough sensors to be able to have good measurements,
but not too many that would cause non negligible mass loading and high costs.

The best option would be to use the 15 sensor scheme, but only use the sensors placed in the wings, the most important part of the measurements. This would reduce the number of sensors to 8 and would be a practical and feasible work configuration.

5.4 Shaker Placement

While using the shaker, in order to perform the tests with the best possible results, the shaker must be placed carefully. There is a need to avoid placing the shaker in a node of a mode in analysis. A node is a point where there is no displacement, as previously mentioned in Chapter 2. In order to avoid the said nodes, a node location analysis was made to all the modes identified, where the results can be seen in Figure 5.18.

![Figure 5.18: Nodes Location for the Aircraft’s Modes](image)

From this picture and taking into consideration the actual wing, the choice for the shaker installation point was made, and it has defined to be placed at point 17 as seen in the picture. The basis for this choice was that it should be as further away from the nodes present on the wings but since the only available space is the one closest to the root of the wing, it couldn’t be held to close to the root because of node 2.

Being so, point 17 is located just halfway between the wing root and the closest node, an acceptable engineering choice for this test.

5.4.1 Mode Participation Analysis

After the decision of the installation location of the shaker, once again, since unfortunately FEMtools and this tool wasn’t available at the time, it can be analysed how good this placement was in relation to other possible points.

Taking into consideration that there was a limitation of the installations points due to the nodes of the modes present, it was compared the chosen location to another one further away from the root of the wing, but very close to one of the nodes.
Since the proper excitation of the lower modes is very important, it can be seen that a point further away from the root could have given a better excitation of the lower modes, since Table 5.19 shows the distribution of energy of the excitation to each mode.

Unfortunately, the simulated point could have been a much better choice, but this analysis wasn’t possible at the time. Here is can be seen the importance of pre-test analysis: it allows to optimise the test procedures and consequently the results.

If the tests were to be repeated, a different position should be chosen taking into consideration these results.

### 5.5 Non-linearities Check

While performing modal testing with a shaker, as it is with an impact hammer, it is best to use low levels of excitation while trying to identify the structure’s characteristics as it is not supposed to induce operating level excitation. Over excitation of the structure could induce the structure to vibrate in a non-linear way that does not really exists normally. The excitation of non-linear characteristics can distort and render measured data useless for the purpose of modal testing [2].

When testing relatively simple structures, the use of a single shaker usually doesn’t represent a problem. Using a proper force level can result in good measurements. But, if more complicated or larger structures are being tested, a single shaker may not be enough to excite the structure properly.

Sometimes when providing excitation from a single shaker is not enough and the intensity levels are increased until a more satisfying response level is obtained.

If there comes a point where the intensity level itself reaches a point where non-linearities are being introduced into the system by the high-level alone, there can be a better configuration by using multiple shakers using each one a lower intensity level than the one that would be used with a shaker alone. A single shaker sometimes is not enough because with the increase of components in the structure it becomes more complicated to excite every location, and identifying the modal parameters can become a very hard task. Using multiple shaker can reduce the effect of exciting non-linearities thus preventing

<table>
<thead>
<tr>
<th>Mode</th>
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<th>MPA (%)</th>
<th>Mode</th>
<th>Freq (Hz)</th>
<th>MPA (%)</th>
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<td>64.14</td>
<td>0.00</td>
</tr>
</tbody>
</table>

**Figure 5.19:** Modal Participation for Different Shaker Placements
the data from being degraded.

![Response Comparison With Different Intensity Levels](image)

(a) Left Wing  
(b) Right Wing

**Figure 5.20:** Response Comparison With Different Intensity Levels

To prevent these kinds of situations, a non-linearity check, or an intensity test was performed. It can be seen in Figure 5.20 the results for both wings.

The scale used was level 0 for no excitation and level 10 the maximum excitation level that could be output by the shaker. Performing this test on site, it was verified that, starting in level 5, non-linear vibration appeared. The effects were:

- Rumbling sounds of equipment present inside the aircraft, meaning that components were vibrating as single components and not as part of a bigger structure
- The shaker attachment started vibrating unnaturally and started shifting position, causing even bigger non-linear responses.

With these results it was decided that a level below 5 was the best choice, and the optimum point was determined to be between level 3 and 4, where there was enough excitation to have good measurements but without exciting non-linear modes. For that, the level 3.5 was the final choice.

### 5.6 Shaker Reciprocity Tests

When performing shaker testing, the set-up is much more complicated than in impact testing. It is necessary to guarantee that the shaker is well positioned, the stinger perpendicular to the excitation surface and have a proper attachment of the shaker to the structure and the shaker must be placed on a proper surface for modal testing.

Even after all these procedures it is necessary to perform reciprocity tests. It means that the reciprocity of the of the shaker in relation to the response point has to be verified. In order to do that, impact testing is performed at the response point and measured at the shaker installation point and then compared to the excitation point with the shaker measured at the final response location.

This comparison allows to verify if the shaker set-up was properly done and assure that the testing is not compromised. Looking at Figure 5.21 it can be seen a reciprocity test performed when shaker testing. Even though the phase is plot is out of phase by 180 degrees, a sign of sign switch, it can be
seen that in terms of magnitude the two results are very similar, almost coincident throughout all the frequency range. This indicates that the shaker is set-up correctly and the testing can proceed.

**Figure 5.21: Reciprocity Results for the Rigid Wing Shaker Testing**
Update of the Finite Element Models

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6.1 Introduction

The final step in this work is related to the updating of the Finite Element models, thus ending the cycle of the verification and validation process.

Even when a good model is created using finite elements, it only becomes reliable to use when backed up by some real information of the model. That makes the model creation a not easy thing to do.

The model updating process is a very long process, it has to be done in a way that ensures that the FE model is as accurate as it can be in order to be used in later analysis, usually for flutter calculations of critical speeds of the aircraft, which is obviously a very important part of the design and safety process.

As, when modelling a new aircraft, engineers are trying to recreate something to resemble reality, there can be some issues that cannot be fixed with numerical methods alone. Validation and updating of models is very important because it helps to deal with some problems:

- Modelling problems (model geometry, material properties or boundary conditions);
- Unknown or incorrect physical properties (non-linear phenomenons, damping, and so on);
- Errors and uncertainties in the analysis performed (Use reduced set of degrees-of-freedom instead of the complete set, lumped masses matrices can be used instead of consistent mass matrices etc.).

The goal in model updating is to see and comprehend how modifying the design variables influences the system's responses and how errors can be avoided, accounted for and minimized. Usually, the errors and some inaccuracies are resultant from approximations made and they only are detected after the results have been validated with experimental data. This is best achieved by sequentially validating and updating of modelling of mass, stiffness and damping using data from static and dynamic testing.

The updating of done using the measured modal data, ensuring the best basis for the verification and validation process, and the updated model allows that small modifications can be performed without the need to redo the whole GVT process, at least in relation to the flutter calculations.

Before the realization of the GVT there is little or no information regarding experimental data, thus, the whole verification and validation process of the FE model starts as soon as the GVT has been performed. Consequently it is only considered in a very late phase of the certification process and it is not usually taken into consideration during the aircraft development phase, even though the FE model and nearly all parts of the aircraft are available several months before the scheduled time for the GVT.

6.1.1 The Updating Software

For the updating of the Finite Element models, the Dynamic Design Solutions’ FEMtools software was used. FEMtools is a versatile program that can be used, among other things, in experimental modal analysis. In this case it will allow to do the verification, validation and updating of FE models for structural analysis by doing the following:

- Pre-test analysis and planning
6.2 Sensitivity Analysis

The first step involves doing a sensitivity analysis of the model being used. Sensitivity analysis is done in order to obtain a better comprehension of how the structural responses of the FE model are influenced by modifying its properties. It can be changed parameters as boundary conditions, material stiffness, geometry and so on.

Sensitivity Analysis is a very useful tool and can be applied in various ways:

- Design Optimization
- Identifying sensitive and non-sensitive structure areas in relation to certain parameters
- Estimation of new parameter values
- Pretest analysis

6.2.1 Sensitivity Parameters and Responses

A parameter is a physical quantity that is used to model the real structure and that is estimated by the engineer in charge of the analysis [34]. Which parameters will be selected depends on the type of application.

Since the sensitivity analysis has the purpose to choose which parameters are used in the model updating, it should be used the maximum number of parameters possible in order to determine which ones are truly critical for the updating. In that process it will be clear which parameters are relevant, with high sensitivity, and which aren’t, with low sensitivity. Having high or low sensitivity implies how much the FE model changes with small changes in those parameters. Parameters with high sensitivity will cause great changes within the model with small parameter variation and parameters with low sensitivity are ones which that, even with a great parameter variation, the FE model change is not relevant enough to be considered.

When making the parameter selection for sensitivity analysis, there are some considerations that should be taken into account:

- It should be used as many parameter types as possible, including stiffness related ones in order to take into consideration possible modeling errors
- If possible group parameters with the same properties, such as geometry and tolerance, scatter, sensitivity and so on.
A comparison of the sensitivity analysis and the error localization results should be done.

There is no best selection for sensitivity analysis for all cases, so it should change by the application being targeted when model updating. The parameters present in the FEMtools software can be a part one of the following groups:

- Element material or geometrical properties;
- Nodal properties like spring stiffness;
- Boundary conditions;
- Lumped mass properties.

Adding to this choice of parameters, they can be split in two classes that can be used: local and global parameters. Local parameters are physical properties, as listed above, of a single element or node. Global parameters represent a simultaneous change of a physical property at a set of elements or nodes and can, consequently, be considered as a group of linked local parameters [34].

The selection of responses, as it is for parameters, is chosen taking into consideration the final application. When doing design optimization, the target behaviour will be described in terms of structural responses as the resonance frequencies. If, in the other hand, is model updating that is being done, all the experimental response results will be selected. There is a special care that should be noted: FEMtools automatic pairing is faulty an consequently the manual selection of the test responses has to be paired to an analytical ones, for unpaired single responses will not be considered by FEMtools and errors in the analysis can be made if one has that assumption that the responses were all used.

The sensitivity calculation method uses the FE model element matrices, structure eigenfrequencies and eigenvectors estimates and the sensitivity coefficients are evaluated at a particular state of the parameters and are defined as the rate of change of a particular response quantity $R$ with respect to a change in an analysis model property $P$, that has being referred to as a parameter.

### 6.2.2 Chosen Parameters and Responses

For the case being analysed, the QT1, the important parts of the model to be updated are the wings, more specifically the wing’s inboard and outboard parts. Consequently, these where the elements subjected to this sensitivity analysis to later perform the model updating. As a note regarding FEMtools and the FE model, it will appear in the results numbered elements, where the element set number one will be correspondent to the inboard section of the wings and the element set number two will be the outboard part of the wings.

In Figure 6.1(b) it can be seen the wing inboard area highlighted in a red tone colour and the outboard section, in the same line, in a lighter blue.

From all the parameters that could have been chosen from the FEMtools software, the list of the ones that were picked is displayed at Table 6.1 as the responses picked, for the modal updating of the modal, the frequencies of the FE model were used, as shown in Table 6.2.
Table 6.1: Parameter Selection (in color the selected ones for the 2nd Sensitivity Analysis)

<table>
<thead>
<tr>
<th>Parameters Used</th>
<th>Response Selection</th>
</tr>
</thead>
<tbody>
<tr>
<td>JX Mass Inertia about X</td>
<td>#</td>
</tr>
<tr>
<td>JY Mass Inertia about Y</td>
<td>1</td>
</tr>
<tr>
<td>JZ Mass Inertia about Z</td>
<td>2</td>
</tr>
<tr>
<td>GE Structural Element Damping</td>
<td>3</td>
</tr>
<tr>
<td>E Young's Modulus</td>
<td>4</td>
</tr>
<tr>
<td>RHO Mass Density</td>
<td>5</td>
</tr>
<tr>
<td>AX Cross Section Area</td>
<td>6</td>
</tr>
<tr>
<td>AY Shear Stiffness Area for Plane XY</td>
<td>7</td>
</tr>
<tr>
<td>IX Torsional Stiffness</td>
<td>8</td>
</tr>
<tr>
<td>IY Bending Moment of Inertia About Y</td>
<td>9</td>
</tr>
<tr>
<td>IZ Bending Moment of Inertia About Z</td>
<td>10</td>
</tr>
<tr>
<td>NSM Non Structural Mass</td>
<td>11</td>
</tr>
<tr>
<td>MG Lumped Mass</td>
<td>12</td>
</tr>
</tbody>
</table>

Table 6.2: Responses Selected: All the Frequencies from the FE Analysis

<table>
<thead>
<tr>
<th>#</th>
<th>Scatter (%)</th>
<th>Value [Hz]</th>
</tr>
</thead>
<tbody>
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<td></td>
</tr>
<tr>
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</tr>
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<td>3</td>
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<td>4</td>
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<td>196.49</td>
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</tbody>
</table>

Doing an analysis of Figure 6.1(a) it can be seen that only a handful of parameters are relevant for the responses selected and they were highlighted in Table 6.1. These selected parameters were picked for a second run of the sensitivity analysis so that the updating software can be aware of their relevance during the updating and so that it can be known which parameters are truly critical for this model updating.

In the set of Figures A.12, A.11(a) and A.11(b), it can be seen the results from the final sensitivity analysis. In Figure A.12 the 3D sensitivity plot is presented for this case, where sensitivity can be evaluated from the responses point of view or from the parameters point of view. These parameters are the ones that are going to be used from now on for all the model updating process.

6.3 Model Updating

When updating a model the values of the selected parameters are adjusted until a certain stopping mechanism is activated. The stopping mechanism are called the convergence criteria and when they are minimized the updating process comes to a halt and it comes to the engineer to verify if the results are plausible and good enough or if another update process should be done with different convergence criteria or other inputs.
6.3.1 Convergence Criterion

In model updating, the correlation coefficients will be interpreted as an objective function that needs to be minimized. With each iteration loop the values of the correlation coefficients will be verified to check if a convergence criterion is satisfied. The following criteria are used:

• The value of the reference correlation coefficient is less than an imposed margin:

\[ CC_t < \epsilon_1 \] (6.1)

where:

– \( CC_t \) Reference correlation coefficient at iteration \( t \);
– \( \epsilon \) Convergence margin.

• Two consecutive values of the reference correlation coefficient are within a given margin.

\[ | CC_{t+1} - CC_t | < \epsilon_2 \] (6.2)

• The number of iterations exceeds the maximum number that is allowed. This test puts a practical limit on the number of iterations in case that very small convergence margins are used.

The iteration loop in model updating will be stopped as soon as one of these tests is satisfied. This convergence criterion can then be used with the correlation coefficients as reference to perform the update.

6.3.2 Multi-Model Updating

For the QT1 case the normal updating method was not possible to be used, since the results were spread out through different sets of data, making it impossible to have the model updated with each data set at the time. So, to be able to perform a good model updating, the multi-model updating method was used.

It is the same process as normal updating with just one data set, but the FEMtools program updates the model in cycles using the different data sets. When performing model updating, a Multi Model Updating (MMU) technique can be used. Using it FE models with different properties, i.e., different boundary conditions, lumped masses or if the case applies, different models with the same properties can be simultaneous updated.

This technique is useful when the number of updating parameters is bigger than the the number of updating equations (responses). In theory, this case would have an infinite set of solutions, but using the different the MMU, allows to increase the number of updating equations and thus constrain the solution set for the case at hand.

A important factor in MMU is that the engineer should have every updating variable must be a physical property common in all the models being updated. All the variables not being updated can be different and shall remain the same.
To be able to use the MMU the engineer working on the model updating must follow these steps in order to guarantee that everything runs smoothly:

- Create the database of models to update (for every structure if there is more than one), including the FE models;
- Input the test model and data;
- Create the tables describing the common DOFs;
- List the modal parameters in each model;
- Select the updating targets and updating variables.

This step-by-step operation is the same as it would be if just one model was to be updated, but it is very important to emphasize again that the updating variables have to be of the same type and level in each model.

6.3.2.A Correlations Coefficients (CC)

Several correlation coefficients (CC) can be computed. They are based on the discrepancy between the numerical and experimental value of the responses by having the contribution of the various responses weighted in. The following standard correlation functions available in FEMtools:

- Weighted relative difference, **CCMEAN**, between resonance frequencies:

\[
CCMEAN = \frac{1}{C_R} \sum_{i=1}^{N} C_{R_i} \frac{\Delta f_i}{f_i}; \quad C_R = \sum_{i=1}^{N} C_{R_i}
\]  

(6.3)

where N is the number of active frequency responses.

- Weighted absolute relative difference, **CCABSOLUTE**, between resonance frequencies (Same equation 6.3 , but N is the number of active frequency responses)

- Average MAC values **MACMEAN**:

\[
MACMEAN = 1 - \frac{1}{C_R} \sum_{i=1}^{N} C_{R_i, MAC_i}; \quad C_R = \sum_{i=1}^{N} C_{R_i}
\]  

(6.4)

where N is the number of active frequency responses.

This is the average margin of the MAC-values for the paired mode shapes that correspond with N resonance frequencies of the active frequency responses. The confidence values are those of the resonance frequencies. The computation of MACMEAN does not require MAC responses.

- Average MAC values **CCMAC**: This is the average margin of the MAC-values of the active MAC responses. (Same equation as 6.4, but N is the number of active MAC responses)
• Weighted absolute difference between modal displacements \( CCMDISP \):

\[
CCMDISP = \frac{1}{C_R} \sum_{i=1}^{N} C_{R_i} \frac{\Delta \Psi_i}{\Psi_i} ; C_R = \sum_{i=1}^{N} C_{R_i}
\] (6.5)

where \( N \) is the number of active displacement responses.

• Weighted absolute difference between exact mass and calculated mass \( CCMASS \):

\[
CCMASH = \frac{1}{C_R} \sum_{i=1}^{N} C_{R_i} \frac{\Delta m_i}{m_i} ; C_R = \sum_{i=1}^{N} C_{R_i}
\] (6.6)

where \( N \) is the number of active mass, center-of-gravity and inertia properties.

The contributions of all the mass, center-of-gravity and inertia properties are added to \( CCMASH \).

• Total weighted relative differences \( CCTOTAL \):

\[
CCTOTAL = CCABS + CCMAC + CMDISP + CCM ASS
\] (6.7)

6.4 QT1 Model Updating Results

With the results from the GVT ready after being analysed and processed, the FE model updating can be done. The information regarding the FE model is input into FEMtools and the modal analysis performed right after. The test modal is another input, that in this case it is the same FE model, since there was no need to simplify it being it already very simple, and the GVT modal results as well.

With every modal information present in FEMtools, both from the FE model and the experimental testing, the updating can be performed. In order to do this updating a couple of steps are needed beforehand:

• Create Node-Point pairs: These pairs will do the matching between the FE model and the test model. Since in this case, both models are the same, the correlation will be total;

• Create DOF Pairs: These pairs are going to let the program know which of the test points are actually moving and are going to be used when comparing information, specially regarding mode shapes, from both models;

• Create Mode Shape Pairs: Having modal information from both models a pairing of the modes present in each model has to be done. This is can be done automatically by having only the mode shape pairs with high MAC values be paired or it can be done manually by matching each mode the its corresponding one on the other model. This can be specially useful if FEMtools doesn’t perform the pairing well or if it can be seen visually that the modes are the same, but by computational errors, the MAC value is not high. This can happens if the experimental data is non optimal.

• Define the Model Updating Settings: With everything paired and ready, the settings for the updating procedure must be defined and there are two major parameters:
Table 6.3: Updating Setting Steps

<table>
<thead>
<tr>
<th>Updating Step</th>
<th>1st</th>
<th>2nd</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\epsilon_1$</td>
<td>0,10</td>
<td>0,09</td>
</tr>
<tr>
<td>$\epsilon_2$</td>
<td>0,06</td>
<td>0,05</td>
</tr>
<tr>
<td>Max $\frac{dP}{P}$ (%)</td>
<td>10.00</td>
<td>5.00</td>
</tr>
<tr>
<td>Correlation Criterion</td>
<td>CCTOTAL</td>
<td>CCTOTAL</td>
</tr>
</tbody>
</table>

- The Convergence Control: Settings are chosen in order to define the what Correlation Criterion (CC) is going to be used and the stopping criteria, that include the maximum number of iterations, absolute CC target, defined previously as $\epsilon_1$, the minimum relative CC improvement (in %), defined previously as $\epsilon_2$;

- The Parameter Estimation: Regarding the update parameters it is defined the minimum and maximum parameter ($\frac{dP}{P}$) change while updating.

6.4.0.B Model Updating Settings

Regarding the previously mentioned Model Updating Settings the procedure, in order to have good results and a proper updating, it has to be well defined. To do that a two-step procedure was used: to follow the first parameter and then perform a continuation of the update with more stringent settings, as shown in 6.3.

This two steps method was found to be the most effective one throughout the updating phase. Since the first step is already very strict, the second step only serves as a confirmation that the model is indeed converged.

6.4.1 Rigid Wings

For the rigid wings, the updating was processed smoothly and occurred without problems and the results can be seen in Figure 6.2 and in Tables 6.4 and 6.5.

It should be noted a very high discrepancy in the CCTOTAL plot in Figure 6.2(a) that is probably due to a poorly done updating step by FEMtools but, as can be seen, it was quickly fixed and it can be seen that the updating converged and reached the lowest possible value of CCTOTAL, implying the lowest difference between the FE model and the test data reached by the program.

In Table 6.4 it can be seen that the modal results in the end are really good, with MAC values in almost every case higher than 90 % and low frequency differences fin two of the modes, where the 4th pair managed to be updated to perfection in relation to the frequency value, as the 1st as well. In terms of modal parameters it can be said that the updating was highly successful.
Figure 6.2: Results from the Rigid Wings Model Update

Table 6.4: Results from Mode Shape Pairs in the Rigid Wings FE Model Update

<table>
<thead>
<tr>
<th>Pair #</th>
<th>FEA #</th>
<th>Hz</th>
<th>EMA #</th>
<th>Hz</th>
<th>Diff. (%)</th>
<th>MAC (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>7.4625</td>
<td>1</td>
<td>7.4752</td>
<td>-0.17</td>
<td>92.6</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>30.151</td>
<td>3</td>
<td>26.704</td>
<td>12.91</td>
<td>93.2</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>30.151</td>
<td>5</td>
<td>33.539</td>
<td>-10.10</td>
<td>88.5</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>46.057</td>
<td>7</td>
<td>46.058</td>
<td>-0.00</td>
<td>93.0</td>
</tr>
</tbody>
</table>

Regarding the updating parameters in Table 6.5 it can be seen that the in the majority of the cases there is a very large difference from the starting point to the updated results, with differences up to 40%.

Table 6.5: Results from Parameter Changes in the Rigid Wings FE Model Update

<table>
<thead>
<tr>
<th>Parameter #</th>
<th>Type</th>
<th>Elem/Set</th>
<th>Old</th>
<th>Actual</th>
<th>Difference (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>E</td>
<td>1</td>
<td>7,00E+10</td>
<td>6,46E+10</td>
<td>-7.72</td>
</tr>
<tr>
<td>2</td>
<td>E</td>
<td>2</td>
<td>7,00E+10</td>
<td>5,66E+10</td>
<td>-19.11</td>
</tr>
<tr>
<td>3</td>
<td>RHO</td>
<td>1</td>
<td>1,28E+03</td>
<td>1,70E+03</td>
<td>32.86</td>
</tr>
<tr>
<td>4</td>
<td>RHO</td>
<td>2</td>
<td>6,40E+02</td>
<td>5,78E+02</td>
<td>-9.66</td>
</tr>
<tr>
<td>5</td>
<td>AX</td>
<td>1</td>
<td>1,43E-03</td>
<td>1,90E-03</td>
<td>32.93</td>
</tr>
<tr>
<td>6</td>
<td>AX</td>
<td>2</td>
<td>1,43E-03</td>
<td>1,29E-03</td>
<td>-9.56</td>
</tr>
<tr>
<td>7</td>
<td>IX</td>
<td>1</td>
<td>7,19E-08</td>
<td>8,01E-08</td>
<td>11.49</td>
</tr>
<tr>
<td>8</td>
<td>IY</td>
<td>1</td>
<td>1,93E-08</td>
<td>2,47E-08</td>
<td>28.06</td>
</tr>
<tr>
<td>9</td>
<td>IY</td>
<td>2</td>
<td>1,93E-08</td>
<td>1,60E-08</td>
<td>-17.03</td>
</tr>
<tr>
<td>10</td>
<td>IZ</td>
<td>1</td>
<td>1,50E-06</td>
<td>9,18E-07</td>
<td>-38.90</td>
</tr>
<tr>
<td>11</td>
<td>IZ</td>
<td>2</td>
<td>1,50E-06</td>
<td>1,48E-06</td>
<td>-1.79</td>
</tr>
</tbody>
</table>

6.4.2 Flexible Wings

For the case of the flexible wings model updating, the situation was not so easy. Considering that the GVT results were not optimal, the updating process was more difficult.

This problem was created because, since the wings are more flexible than normal, the results, that previously could be taken of just one of the accelerometers, now only showed good results for one of the wings. For that reason, where previously only one data set had to be used, now it had to be used two. This implied that Multi-Model Updating had to be done, where the model is updated taking into consideration multiple data sets.
The way FEMtools works obliges to create a unique project containing just one set of data for each wing. With that separation, computation problems were found, since it had its results divided into left wing, right wing and leading edge results. In the rigid case all the information could be set in just one project.

After being able to figure out how to manage the split data set, the updating process could be done. On a first try the results were by far non satisfactory. Using the same process used in the rigid wing was proved unsuccessful. The fact that the update was being done with separate projects wings caused the MAC values to be much lower that in reality. This was the main cause for the updating problems.

The update of the model had as a correlation coefficient the CCTOTAL, that according to 6.7 takes into consideration the MAC values from the mode shape pairs and since there is no mass updating or modal displacements take into consideration, the only relevant factors are CCMAC and CCABS. Knowing that the value of CCTOTAL should be as low as possible, values in the order of 70 % are not acceptable, as can be seen in 6.3(a).

The problem referred before related to the calculated MAC values by FEMtools in this situation can be seen perfectly in Figure 6.3(b) where there is a very non linear behaviour of the CCMAC values, nothing like the smooth convergence presented in the case of the rigid wings. The problem with the MAC goes beyond CCMAC and it is apparent that is affecting the CCABS factor, where the frequencies are almost converged at a fixed value, but the fluctuation accompanies the CCMAC value shifts.

![Figure 6.3](image)

**Figure 6.3:** 1st Results from the Flexible Wings Model Update

Since it is apparent that an update using the CCTOTAL coefficient won’t lead to good results another approach was taken. Instead of CCTOTAL, it was chosen the CCABS coefficient, where, used with manual mode shape pairs, guarantees that the model is updated taking into consideration the proper model-experimental results combinations and still manages to update properly the model taking into consideration the modal frequencies.

With the approach change, the results are clear and the difference obvious, taking a look at Figure 6.4(a) it can be seen that the updating process is now much better. Reduced from 183 iterations to just 21 and with a very good convergence curve.

The values are now better as well: previously CCABS converged at the value of almost 28% and now it is at 15%. The improvement is very good.
Figure 6.4: Final Results from the Flexible Wings Model Update

Table 6.6: Mode Shape Pairs for the Flexible Wings FE model Update

<table>
<thead>
<tr>
<th>Pair #</th>
<th>FEA [Hz]</th>
<th>EMA [Hz]</th>
<th>Diff. (%)</th>
<th>MAC (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.7182</td>
<td>2.8903</td>
<td>-5.96</td>
<td>96.0</td>
</tr>
<tr>
<td>2</td>
<td>17.067</td>
<td>17.844</td>
<td>-4.35</td>
<td>82.3</td>
</tr>
<tr>
<td>3</td>
<td>47.114</td>
<td>54.503</td>
<td>-13.56</td>
<td>29.4</td>
</tr>
<tr>
<td>4</td>
<td>53.516</td>
<td>37.371</td>
<td>43.20</td>
<td>1.7</td>
</tr>
<tr>
<td>5</td>
<td>92.121</td>
<td>84.135</td>
<td>9.49</td>
<td>30.3</td>
</tr>
</tbody>
</table>

Table 6.7: Results from Parameter Changes in the Flexible Wings FE Model Update

<table>
<thead>
<tr>
<th>Parameter #</th>
<th>Type</th>
<th>Elem/Set</th>
<th>Old</th>
<th>Actual</th>
<th>Difference (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>E</td>
<td>1</td>
<td>6,89E+10</td>
<td>6,90E+10</td>
<td>0,14</td>
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<tr>
<td>2</td>
<td>E</td>
<td>2</td>
<td>7,00E+10</td>
<td>6,14E+10</td>
<td>-12,32</td>
</tr>
<tr>
<td>3</td>
<td>RHO</td>
<td>1</td>
<td>2,70E+03</td>
<td>2,74E+03</td>
<td>1,38</td>
</tr>
<tr>
<td>4</td>
<td>RHO</td>
<td>2</td>
<td>6,40E+02</td>
<td>6,97E+02</td>
<td>8,91</td>
</tr>
<tr>
<td>5</td>
<td>AX</td>
<td>1</td>
<td>1,71E-04</td>
<td>1,86E-04</td>
<td>8,73</td>
</tr>
<tr>
<td>6</td>
<td>IX</td>
<td>1</td>
<td>9,27E-09</td>
<td>6,63E-09</td>
<td>-28,49</td>
</tr>
<tr>
<td>7</td>
<td>IY</td>
<td>1</td>
<td>2,82E-09</td>
<td>3,20E-09</td>
<td>13,27</td>
</tr>
<tr>
<td>8</td>
<td>IZ</td>
<td>1</td>
<td>4,36E-08</td>
<td>5,27E-08</td>
<td>20,89</td>
</tr>
</tbody>
</table>

In comparison to the results from the rigid wing, it can be seen in Table 6.7 that for this case the model was much closer to the reality than the rigid case. The most important parameters are the Young’s modulus for the proper simulation of the new installed spars in the flexible wings and it can be seen that had a a very accurate value to begin with.

6.5 Summary

The update of the finite element models was successful and was able to be improved with the support of the results of the ground vibration tests. The case of the rigid and the flexible wings results were discussed in terms of modal parameters and structural parameters.

When a converged FE model solution has been obtained it can be used to make an interpretation of the structural modes and can now be used to update the aeroelastic analysis. As a side note it can be said that with the FE models validated and updated, they can be also be used for structured health monitoring.
7

Conclusions and Future Work

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7.1 Overall Remarks

The major objectives of the work were completed successfully and excellent results were obtained for both the rigid and the flexible wing configurations and the finite element models were updated based on the experimental data.

The experimental modal testing yielded all the required modal parameters for the two set of wings. The updating process of the FE models did present some challenges that were properly overcome regarding the best correlation factors to be used for updating the aeroelastic models for flight clearance of the QT1 UAV. A minimum set of ti-axial accelerometers were used.

It was found that higher frequency modes were more difficult to obtain and showed small nonlinearities, and these were not included in the finite element model updating. Taking a closer look at these modes, it can be inferred that the methods and techniques used were not compliant with the proper detection and analysis of nonlinear modes for the flexible wing.

Comparing the two excitations methods used, it was found that impact testing was more versatile than the shaker testing method. It took less time, it had a far easier set-up, showed roughly the same quality in the data and performed better in the flexible wing case. Overall, impact testing was better. However, the shaker testing presents advantages when the excitation range is limited or when it is necessary to excite a particular frequency of interest.

The differences between the two excitations methods became more distinct when testing the flexible wings. The flexible wing had a lower damping than the rigid wing and the placement of the shaker becomes critical. It was concluded that the use of a shaker in lightly damped structures is not a very good option, unless there is a very good energy distribution throughout the structure.

There were some difficulties regarding the model updating using FEMtools, more specifically in the case of the flexible wing, since the data used was not optimal. This is a consequence of the lack of a computational pre-test planning in the beginning of the experimental phase, that was not possible due to not having access to FEMtools at that time. It is inferred that pre-test planning is a necessary step in modal updating. It allows to save precious time when facing tight testing schedules and to acquire better results.

Pre-test planning is a very useful tool specially when it comes to the main shaker testing disadvantage that is the time it takes to complete the tests. Shaker testing can have roughly 40% of the time used in the set-up phase. Unlike impact testing, if an error is found during the testing, it is very time consuming to set it up again.

With over 200 hours of testing, it can be said that Ground Vibration Testing is a very resource intensive aspect in aircraft design and certification. Experimental testing is still a critical part of the whole process of designing a new aircraft and it enables validation and evaluation of the structural dynamics response of the aircraft and subsequently for aeroelastic certification.
7.2 Future work

A more detailed analysis of the nonlinear modes of the flexible wings would be a very interesting topic to pursue, since it was found that the methods used were not ideal for nonlinear mode analysis. Nonlinear modal analysis is an issue of contemporary interest in aeronautics due to the new interest in high aspect ratio wings and should be pursued in order to propose new methods and processes that enable the engineers to have better results, improving the whole process of Experimental Modal Analysis (EMA) and FE model updating and structural analysis in general.

[2] Pete Avitabile, “Why can’t I run a modal test with one big shaker and just “crank up the signal?””, SEM Experimental Techniques (Page 1), Apr 2011


[11] Pete Avitabile, “Which window is most appropriate for the various types of modal tests performed?”, SEM Experimental Techniques, February 2002


[18] Svend Gade, Henrik Herlufsen and Hans Konstantin-Hansen, How to Determine the Modal Parameters of Simple Structures, Bruel & Kjaer, Denmark


[36] Brian Schwarz & Mark Richardson, Pete Avitabile, Locating Optimal References for Modal Testing, Vibrant Technology & Inc. U. of Massachusetts, IMAC XX Conference.


Table A.1: Impact Hammer PCB 086C03 Specifications

<table>
<thead>
<tr>
<th>Performance</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensitivity (±15 %)</td>
<td>2.25 mV/N</td>
</tr>
<tr>
<td>Measurement Range</td>
<td>± 2224 N pk</td>
</tr>
<tr>
<td>Resonant Frequency</td>
<td>≥ 22 kHz</td>
</tr>
<tr>
<td>Non-Linearity</td>
<td>≤ 1 %</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Electrical</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Excitation Voltage</td>
<td>20 to 30 VDC</td>
</tr>
<tr>
<td>Constant Current Excitation</td>
<td>2 to 20 mA</td>
</tr>
<tr>
<td>Output Impedance</td>
<td>&lt;100 Ohm</td>
</tr>
<tr>
<td>Output Bias Voltage</td>
<td>8 to 14 VDC</td>
</tr>
<tr>
<td>Discharge Time Constant</td>
<td>≥ 2000 sec</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Physical</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensing Element</td>
<td>Quartz</td>
</tr>
<tr>
<td>Sealing</td>
<td>Epoxy</td>
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<tr>
<td>Hammer Mass</td>
<td>0.16 kg</td>
</tr>
<tr>
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</tr>
<tr>
<td>Tip Diameter</td>
<td>0.63 cm</td>
</tr>
<tr>
<td>Hammer Length</td>
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</tr>
<tr>
<td>Electrical Connection Position</td>
<td>Bottom of Handle</td>
</tr>
<tr>
<td>Extender Mass Weight</td>
<td>75 gm</td>
</tr>
<tr>
<td>Electrical Connector</td>
<td>BNC Jack</td>
</tr>
</tbody>
</table>

Figure A.1: LabView GUI for the Measured Data
Figure A.2: LabView GUI for the Shaker Control

Figure A.3: Testing Configuration with Bungees for the Free Free Boundary Condition
Figure A.4: Testing Configuration with Foam for the Free Free Boundary Condition

Table A.2: Modal Frequencies From the AF Polynomial Results for the Beam Case

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
<th>Damping [Hz]</th>
<th>Damping [%]</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
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<td>382</td>
<td>2,48</td>
<td>0,649</td>
<td>0,884</td>
</tr>
<tr>
<td>5</td>
<td>417</td>
<td>3,76</td>
<td>0,902</td>
<td>0,853</td>
</tr>
<tr>
<td>6</td>
<td>433</td>
<td>2,48</td>
<td>0,573</td>
<td>0,268</td>
</tr>
</tbody>
</table>

Table A.3: Modal Frequencies From the Complex Exponential Results for the Beam Case

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
<th>Damping [Hz]</th>
<th>Damping [%]</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12,9</td>
<td>0,134</td>
<td>1,04</td>
<td>0,976</td>
</tr>
<tr>
<td>2</td>
<td>78,4</td>
<td>0,602</td>
<td>0,768</td>
<td>0,966</td>
</tr>
<tr>
<td>3</td>
<td>216</td>
<td>1,81</td>
<td>0,839</td>
<td>0,888</td>
</tr>
<tr>
<td>4</td>
<td>383</td>
<td>2,51</td>
<td>0,656</td>
<td>0,884</td>
</tr>
<tr>
<td>5</td>
<td>419</td>
<td>2,45</td>
<td>0,584</td>
<td>0,791</td>
</tr>
</tbody>
</table>

Table A.4: Modal Frequencies for the Side Measurements From the Z Polynomial Results

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency [Hz]</th>
<th>Damping [Hz]</th>
<th>Damping [%]</th>
<th>MPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>73,2</td>
<td>0,37</td>
<td>0,506</td>
<td>0,999</td>
</tr>
<tr>
<td>2</td>
<td>382</td>
<td>1,65</td>
<td>0,432</td>
<td>0,5</td>
</tr>
<tr>
<td>3</td>
<td>430</td>
<td>1,21</td>
<td>0,281</td>
<td>0,98</td>
</tr>
</tbody>
</table>
Figure A.5: Rigid Wings in the Fixed Cantilever Boundary Condition

(a) Sensitivity Plot in Function of Parameters  (b) Sensitivity Plot in Function of Responses

Figure A.11: Sensitivity Plot in Function of Parameters and Responses
Figure A.6: Shaker Configuration Flexible Wings

Figure A.7: Shaker Configuration Flexible Wings
Figure A.8: Shaker Configuration Flexible Wings
Figure A.9: Instrumented Flexible Wings
Figure A.10: Instrumented Flexible Wings Impact Testing
Figure A.12: 3D Sensitivity Plot