

Quality Control of Carbon Fiber Structures by Acoustic Method

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Abstract

A method for quality control, using impact excitation and sound measurements, is investigated. The Eigenmodes of a structure are unique if all properties, i.e. the shape, the resonance frequency, and the damping, are considered. It is, however, demanding to experimentally determine the vibration shape of the Eigenmodes, compared to getting estimates of the Eigenfrequency and the Damping. For a specific structure it may, however, be possible to use identified peaks in a measured sound spectrum, and from them make estimates of the Eigenfrequencies and Damping, establishing enough information for quality control, even without knowing the mode shapes. The risk associated with this approach is if a match of peaks in the response spectra is found in terms of frequency and damping, but this match is in fact for different Eigenmodes (i.e. the correlation in frequency and damping is for Eigenmodes with different shapes). This may happen for one pair of Eigenmodes, but is unlikely if more Eigenfrequencies are included in the comparison. Another risk with the used approach is if the acoustic radiation is poor from one or more of the Structural Eigenmodes, or if the excitation is sensitive to the impact (excitation) location. Further, with a single microphone used, there is a risk the sound radiation in this particular direction is low. The strategy to reduce such risks is to use multiple impact locations, and to use a non-fixed test rig for the test object. Demonstration of the method's applicability is made using aircraft components manufactured in Carbon Fiber Reinforced Polymer, and show promising results in terms of ability to detect variations in manufacturing quality, with a very simple test setup and short time required for the test.

Keywords: Carbon Fiber (1), Quality Control (2), Structural Dynamics (3), Acoustics (4).

1 Introduction

Components made of Carbon Fiber Reinforced Polymers [CFRP] may have deficiencies that are difficult, or impossible, to observe without testing. Such faults could be a result of mistakes in the production from e.g. misalignment of the “plies” in the forming of the component, or improper curing. Visual inspection and measuring the weight of the component may not reveal such imperfections that have an influence on the structural characteristics of the part. Dynamic testing of a structural component, such as Experimental Modal Analysis (EMA), is a very precise method to determine the structural properties. However, a full EMA is a rather complicated task, involving applying and measuring an excitation, as well as measuring the structural response without influencing the test object, or at least having an influence that can be compensated for. Ultrasonic testing (e.g. phased-array

or C-scan) is another used Non Destructive Testing (NDT) method applied to CFRP. It offers high reliability and excellent sensitivity to subsurface defects such as delaminations or porosity. However, layer orientation and/or layer order are more challenging to detect by ultrasonic testing. The same is valid for thermographic NDT methods, such as pulsed or lock-in active thermography, which offer fast, non-contact and large area inspection, and can reveal defects via thermal patterns, that are normally detected by infrared cameras. Ultrasonic testing also requires special equipment e.g. [1] and skilled operators as well as coupling media or immersion setups, which can make it complex and cost-intensive to implement. Thermographic systems though tend to require less operator training and lower operational cost, but still require more costly equipment than the investigated acoustic method [2]. The method evaluated, i.e. measuring the sound radiated

by a structural component when excited with a simple “tap” by a small metal rod, is easy to perform, requires less than 20 minutes of training for operating staff, and is efficient. There is, however, no guarantee that all structural Eigenmodes are possible to identify from the acoustic radiation. The present research studies the reliability and precision for the suggested quality control method.

2 Background

Modern acoustic methods for NDT began to be developed during the 1960s, and have subsequently been even further developed as both sensors and analysis equipment have become both more efficient and cheaper. A contemporary example is that ABB [3] uses acoustic emission (AE) to detect bearing wear, and from this predicts the best time for bearing replacement. More modern research has been done by Karlsson et al., 2023 [4] and Hosoya et al., 2014 [5]. In the first case it is described how float glass is tested, and in the latter case it is described how a nanosecond long acoustic pulse can be created by laser ablation. A very brief, and easy to understand, description of failure modes and NDT of CFRP is likewise given by Dwivedi et al., 2017 [6]. Different AE methods have also been described by, for example, Michalkova and Kadlec, 2016 [7] and Barile, 2019 [8]. The examples above show that some AE methods are already implemented in industrial use and hence at TRLs (Technology Readiness Level) 9 or above. However, AE methods, suitable for quality control of CFRP, in an industrial production environment, are still at TRLs 5 or below. Furthermore, for AE-tests, the typical application is to identify wear, or faults, requiring service or repair. The tests are also often made in operating conditions, having no need for specific excitation. The articles above also describe that high-frequency acoustic signals, ultrasound, or thermographic NDT can be used to detect cavities and non-homogeneous regions. Such testing does, however, not provide information on the component-level properties. A failure in layup order, thickness, or orientation, is hard to detect with such methods. And those characteristics are important since components and structures made of CFRP offer unique properties in terms of strength and stiffness in relation to the weight. The variability in carbon fiber type, density, number and thickness of each layer, and orientation can be used to optimize to a much greater extent than if the components are made in metals or other isotropic materials. But the large flexibility in design of the components can also result in high complexity in the production, typically with a combination and orientation of several CFRP layers in a mold before the component is pressurized and cured in an autoclave. The underlying structural dynamic properties are also well known as a measure for quality, as variations in the structural stiffness and mass are unmistakably revealed by these properties. However structural dynamics testing is quite complicated and requires the use of accelerometers or other vibration sensing devices. Ideally the vibration sensing should be performed with non-contact sensors to avoid any influence on the structure from the sensors. Such sensors are expensive and require special signal conditioning. A search for commercial products similar to the methods used in the

project was performed by the aircraft manufacturer Blackwing and revealed no such products. Also, a patent search for methods similar to those tested in the project did not reveal any technology comparable to the one evaluated.

3 Strategy

The setup should be simple, and a test is to be made in no more than 2-3 minutes. Some specific training of the regular production personnel will most likely be required, but this is expected to only have to be once, and no more than 15-20 minutes.

Measurement data are to be evaluated and compared to database results for resonances and damping values. Any anomalies for the pre-defined ranges shall be identified and may lead to a "disqualify" message for the evaluated part. Some parts are likely to be best evaluated with a low number of Eigenmodes (2-3) considered in the evaluation, while other parts may have many Eigenmodes (5-8) in the evaluation set. Eigenmode frequencies, identified from the central frequency of the peaks in the measured acoustic response spectrum, is the foundation in the evaluation. But also the damping estimates may support the evaluation procedure, by e.g. allowing for identification of "false peaks" not showing the properties of the sound radiated from real structural Eigenmodes.

4 Theoretical Basis

Structural Eigenmodes are very good to classify a structure. An Eigenmode has three properties; the Mode Shape, the Resonance Frequency, and the Damping. The Mode Shape is a distinct property, and any deviation from a reference Mode Shape is a result of deviation in the structural properties.

4.1 Structural Dynamics

A system in balance fulfill the Equilibrium Condition, Eq. (1).

$$\begin{bmatrix} \frac{\partial}{\partial x} & 0 & 0 & \frac{\partial}{\partial y} & \frac{\partial}{\partial z} & 0 \\ 0 & \frac{\partial}{\partial y} & 0 & \frac{\partial}{\partial x} & 0 & \frac{\partial}{\partial z} \\ 0 & 0 & \frac{\partial}{\partial z} & 0 & \frac{\partial}{\partial x} & \frac{\partial}{\partial y} \end{bmatrix} \begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \sigma_{xy} \\ \sigma_{xz} \\ \sigma_{yz} \end{bmatrix} + \begin{bmatrix} b_x \\ b_y \\ b_z \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \quad (1)$$

with b_x , b_y , b_z , representing the body forces. Using $\tilde{\nabla}^T$ for the matrix differential operator, the compact form is given by Eq. (2).

$$\tilde{\nabla}^T \sigma + b = 0 \quad (2)$$

4.2 Finite Element Formulation for Structural Dynamics

A weak form of Eq. (2) may be found by integration over the pertinent region and performing partial integration. Assuming linear elasticity, and following Ottosen and Petersson [9], the equation for the Stiffness Matrix in a Finite Element Formulation of a structure, Eq. (3), is established.

$$K_s = \int B_s^T D B_s dV \quad (3)$$

where the matrix B_s is given by applying the differential operator on the shape functions N_s .

$$B_s = \tilde{\nabla}^T N_s \quad (4)$$

The body forces, b , are in the case of dynamics proportional to the accelerations, and with the same Finite Element formulation as for the Stiffness Matrix, the basis for these forces are giving by the accelerations and the Mass Matrix, Eq. (5).

$$M_s = \int N_s^T \rho N_s dV \quad (5)$$

With the matrices of Eq. (3) and Eq. (5) the dynamic system is expressed in Eq. (6).

$$K_s X(t) + M_s \ddot{X}(t) = F(t) \quad (6)$$

and with

$$X(t) = \hat{X} e^{-i\omega t} \quad (7)$$

$$\ddot{X}(t) = -\omega^2 \hat{X} e^{-i\omega t} \quad (8)$$

$$F(t) = \hat{F} e^{-i\omega t} \quad (9)$$

the well known equation system Eq. (10) arises.

$$K_s \hat{X}(\omega) - \omega^2 M_s \hat{X}(\omega) = \hat{F}(\omega) \quad (10)$$

This system can be solved for its Eigenmodes to find the resonance frequencies.

4.3 Acoustics

In a general context fluid dynamics is best described by the Navier-Stokes equations. However, for Acoustics the viscosity of the fluid can be neglected, and the relation between the pressure and the density can be assumed to be linear. These simplifications results in the well-known Wave Equation Eq. (11).

$$\frac{\partial^2 P}{\partial t^2} - c^2 \nabla^2 P = Q \quad (11)$$

where c is the speed of sound and ∇^2 is the Laplace operator.

4.4 Finite Element Formulation for Acoustics

From the wave equation Eq. (11), the Stiffness Matrix, Eq. (12), and the Mass Matrix, Eq. (14), for acoustics analysis are established.

$$K_f = \frac{1}{\rho} \int B_f^T B_f dV \quad (12)$$

where the matrix B_f is given by applying

$$\begin{bmatrix} \frac{\partial}{\partial x} \\ \frac{\partial}{\partial y} \\ \frac{\partial}{\partial z} \end{bmatrix} = \nabla \quad (13)$$

on the shape functions N_f .

$$B_f = \nabla N_f \quad (14)$$

In the case of Acoustics, the mass matrix is give by:

$$M_f = \frac{1}{\rho c^2} \int N_f^T N_f dV \quad (15)$$

With the matrices of Eq. (12) and Eq. (14), and the same transformation to the frequency domain, Eq. (7) - Eq. (9), the dynamic system is expressed in (16)

$$K_f \hat{P}(\omega) + M_f \ddot{\hat{P}}(\omega) = \hat{Q}(\omega) \quad (16)$$

4.5 Vibro-Acoustics

Although often neglected, most structural dynamic systems are influenced by a surrounding fluid. If the fluid is air, and the structure is having a relatively large mass per surface unit, the effect of the fluid is indeed found to generally be small. However, for Eigen-modes with very low damping there may be a non negligible influence.

In a similar manner, any acoustic system will interact with non-rigid objects in contact with the fluid.

In both cases the interaction can be expressed by a load/source term. For the structure the acoustic pressure acting on the structure is directly giving a force vector $F(t)$, Eq. (17),

$$F_{fs}(t) = \int_A N_f N_s \vec{n} P(t) dA \quad (17)$$

with consideration of the surface normal. The influence of structural response on an acoustic system is proportional to the acceleration Eq. (18);

$$Q_{sf}(t) = c^2 \rho \int_A N_s N_f \vec{n} \ddot{X}(t) dA \quad (18)$$

In both cases the integral is to be evaluated over the area of interface. A coupled, Vibro-Acoustic, system can be formed by combining (10) and (16) with (17), and (18).

$$\left(\begin{bmatrix} K_s & K_{sf} \\ 0 & K_f \end{bmatrix} - \omega^2 \begin{bmatrix} M_s & 0 \\ M_{fs} & M_s \end{bmatrix} \right) \begin{bmatrix} \hat{X}(\omega) \\ \hat{P}(\omega) \end{bmatrix} = \begin{bmatrix} \hat{F}(\omega) \\ \hat{Q}(\omega) \end{bmatrix} \quad (19)$$

For details it is referred to [10]. The effect of the coupling Matrices K_{sf} and M_{fs} is easily evaluated by response analysis, with a selection of appropriate excitation $F(\omega)$. Eigenvalue analysis is more difficult, especially if the fluid system

is having frequency dependent factors as e.g. resulting from free radiation boundary conditions.

Everything described in above is implemented in most commercial software for analysis with the Finite Element Method. One example is ALTAIR/HyperWorks, and this software is used for the evaluations in the next section.

5 Evaluation of typical Aircraft parts

A first component used for investigation of Vibro-Acoustics is a Frame in the Horizontal tail structure, Figure 1.



Figure 1: Frame for Horizontal Tail, HTD009S.

This component is typical in size and weight for a lot of components used to make the BLACKWING BW 650.

5.1 Effect of the Vibro-Acoustic coupling

The bulkhead structure in the horizontal tail is used to evaluate the effect of the fluid on the structural dynamics. With no ("free") boundary conditions the frequency of the first four Eigenmodes of the structure are given in the table below (Table 1).

Table 1: Eigenfrequencies - Structure Only.

Eigenmode	1	2	3	4
Freq. [Hz]	370.4	410.5	957.0	1027.5

The surface weight is around 1.5 kg per square meter, coming mainly from about 1.0 mm thickness of CFRP. Adding a model for the Air surrounding the structure has a noticeable, but small, effect on the Eigenmodes (Table 2).

Table 2: Eigenfrequencies - Structure in Air.

Eigenmode	1	2	3	4
Freq. [Hz]	364.3	408.3	949.0	1009.5
df [Hz]	- 6.1	-2.2	-8.0	-18.0
df [%]	-1.6	-0.5	-0.8	-1.8

The percentage change of the Eigenfrequency is around -2%, which means a mass effect of 30 gram per square meter. This can be seen as a layer of 1.25 cm air moving in phase with the structure on both sides as a pure mass effect.

An evaluation is also made to investigate the change in the damping from the air. This evaluation is made by frequency response analysis, assuming a structural damping of 0.333 % (critical damping). With air, the damping increase to 0.38% for the first Eigenmode, and to 0.34%, 0.37% and 0.42%, for second, third, and fourth Eigenmodes. For the Eigenmode shape the effect of air is very small. The shape correlation factor, SAC, defined for two column vectors X and Y in Eq. (20),

$$SAC = \frac{X^H Y}{|X||Y|} \quad (20)$$

gives $SAC = 0.999$ for all of the Eigenmodes 1-4 (X : Eigenmode responses vector without air, Y : Eigenmode responses vector with air).

5.2 Testing of the HTD009S Component

The spectral levels from a test on the HTD009S are given in Figure 2.

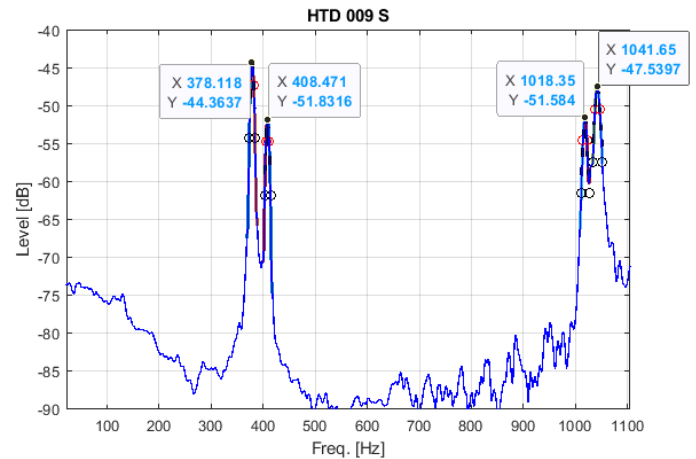


Figure 2: HTD009S Spectrum.

Just like in the model there are four distinct peaks in the spectrum below 1200 Hz. There is a bit of variation for individual specimens, but typically the peak with lowest frequency is at 380 Hz, and the second peak at 410 Hz. This is in good correlation with the results of the model (364 Hz and 408 Hz for the structure in air, Table 2). The peaks in the acoustic spectrum around 1000 Hz are at slightly higher frequency than the 3rd and 4th Eigenfrequencies of the model, at 1018 Hz and 1042 Hz, to be compared with 949 Hz and 1009 Hz for the model (Table 2).

5.3 The Main Landing Gear

One of the most vital parts of an aircraft is the landing gear. The BLACKWING aircraft has a nose landing gear as well as left and right main landing gear, Figure 3.



Figure 3: BLACKWING BW 635/650 Landing Gear.

The main structural part of the landing gear is made in one piece, figure 4.



Figure 4: Main Landing gear, "leg".

Landing Gears from the normal production are showing relatively consistent frequencies of the peaks in the spectrum tested, Figure 5.

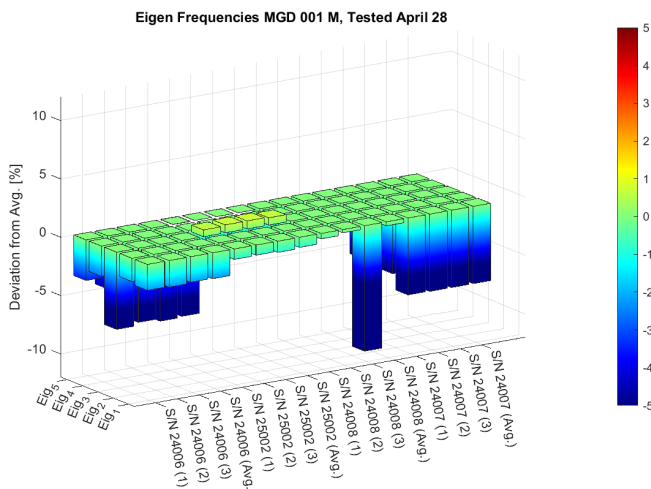


Figure 5: Test Results Main Landing gear Leg. Right hand Side (M).

The graph may at first appear a bit strange, with all results being below the Average. These results, however, are for the Right Hand Side "M" Landing Gear, generally showing lower identified resonance frequencies compared to the Left Hand side parts. In particular two of the Right Hand Side Landing Gear, S/N 24006 and in particular S/N 24007, are found to have lower frequency of the peaks in the spectrum than the average.

5.4 Comparison of change due to interchange of layers in Landing Gear

In order to evaluate the methods ability to detect a Landing Gear made with a fault, a part with modified layup was made. This component has interchanged layers of Carbon Fiber and Fiberglass plies, which should make it stiffer. Finite Element Analysis suggested this part to have resonance frequencies about 5% higher on average than the standard part, with the lowest change for Eigenmode 3 and 5, as given by Table 3. Measurement show an even larger increase in Eigenfrequencies, and follow the same pattern with highest change for Eigenmode 1, and less change of Eigenmode 3 and 5 (Table 3).

Table 3: Comparison of change due to interchange of layers in Landing Gear.

Eigen Mode	1	2	3	4	5
FE Model	+ 7 %	+6 %	+2 %	+ 6 %	+1 %
Test	+11 %	+ 7 %	+ 5 %	+ 7 %	+ 3 %

A graphic illustration of the increase in the frequency of the identified peaks in the sound radiation spectrum, is presented in Figure 6, with the landing gear having the structural modification to the far right identified with MG (1), MG (2), MG (3), and MG (Avg).

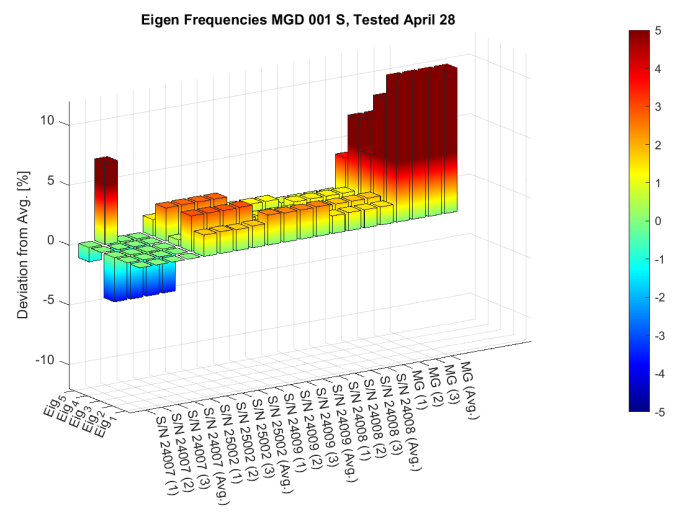


Figure 6: Test Results Main Landing gear Leg. Left Hand Side (S).

Figure 6 also show the generally higher frequencies for the Left Hand Side Landing Gears, compared to the Right Hand

Side Landing Gears (Figure 5). However, this variation is much lower than what is found for the landing gear with modified layers.

6 Summary and Conclusions

A method for Quality Control of structures using acoustic measurements is presented and illustrated for the application to Aircraft Components made of Fiber Reinforced Polymers. The equations for Vibro-Acoustic are established, and the coupling between Structural Vibrations and a surrounding acoustic fluid is demonstrated for a typical component of the BLACKWING aircraft.

A component with intentional altering of the Carbon Fiber plies and Fiberglass plies is used to investigate, and verify, the capability of the method.

The use of numerical models, by means of Vibro-Acoustic Finite Element Analysis, is helping in understanding the measurement results and giving guidance on specific structural properties to search for.

Criteria for acceptance, or rejection, of parts are starting to being established. More testing in a production environment is required before completing the quality control methodology. However, it can already be concluded that a simple mechanical impact, and analysis of the associated radiated sound, measured with a single microphone, can be used to detect deviation in structural properties arising from even small manufacturing variations.

Nomenclature

Designation	Denotation	Unit
$x(t)$	Displacement, function of time	m
$X(t)$	Displacement vector, function of time	m
$\hat{X}(\omega)$	Displacement vector, function of Frequency, Complex valued	m
$p(t)$	Acoustic Pressure, function of time	Pa
$P(t)$	Acoustic Pressure vector, function of time	Pa
$\hat{P}(\omega)$	Acoustic Pressure vector, function of Frequency, Complex valued	Pa

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