APPLICATION NOTE

Information on this document

Contents

This document is the second in a series of four application notes on structural dynamics. The document includes an introduction to experimental modal analysis, providing information on the required equipment and the general procedure.

Target group

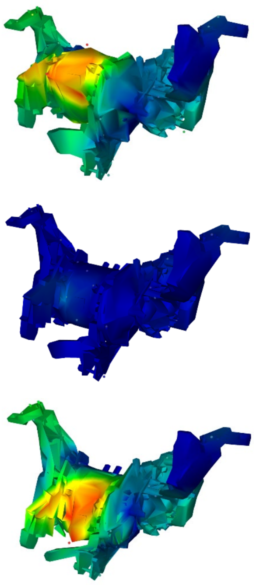
This document serves as an introduction to structural dynamics and is intended for acousticians who would like to get informed about the basics of experimental modal analysis.

Questions?

Do you have questions? Your feedback is appreciated!
For questions on the content of this document: Imke.Hauswirth@head-acoustics.com
For technical questions on our products: SVP-Support@head-acoustics.com

Structural dynamics – Part 2 (EMA)

1. Introduction



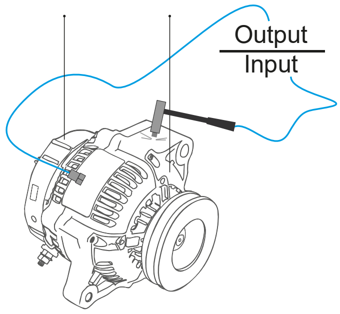
What is modal analysis?

Experimental modal analysis (EMA) is a method for analyzing the dynamic properties of linear, time-invariant structures. The dynamic properties are determined by the mass distribution, rigidity and damping of the structure and influence its vibration behavior. Modal analysis is categorized as a system analysis. In this type of analysis, measurements are linked to a model (in this case, the modal model) to determine inherent properties of the structure.

Modal parameters

Based on measurements on the real structure, modal quantities are determined and visualized by means of appropriate software. Modal quantities are natural frequencies, eigenmodes and modal damping.

Basic principle of the EMA



The aim of experimental modal analysis is to determine the vibration behavior of a real structure based on measured transfer functions. The transfer functions describe the correlations between an excitation force applied to the structure and the vibrations thus excited at different points of the structure in the frequency range. In order to determine these transfer functions, the time signals of the excitation force and the system response are measured simultaneously. In this process, there must be no significant secondary excitation of the system. Thus, the system is not in operation, and is decoupled from its environment in order to prevent unwanted structure-borne noise. The only excitation is provided by the impact hammer or shaker. The excitation force is not static in this context. It rather has an oscillating history, the frequency and amplitude of which are measured by means of a force gauge. The system response can be measured using displacement, velocity or acceleration sensors. In practice, accelerometers are most commonly used for these measurements. With regard to the selection and number of the measurement points, it must be taken into account that only with a sufficient number of measurement points and their suitable arrangement as well as a suitable excitation can all natural frequencies be detected and the corresponding eigenmode geometrically resolved.

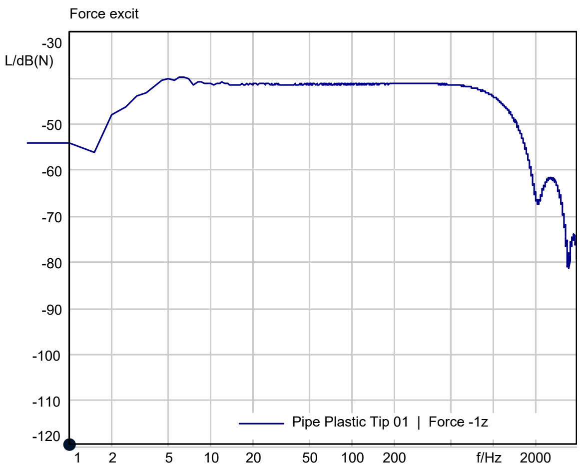
Modal analysis is a linear analysis. Any non-linearity, e.g., the flexibility of a component, would falsify the measured transfer functions and the conclusions drawn from them. The linearity of a system can partly be investigated by observing the coherence at different excitations, e.g., by an impact hammer. With excitations of different intensities, the coherence of non-linear structures collapses over a broad band. Other influences, e.g., temperature, humidity or sensor mass influence, need to be tested by other methods.

2. Generation of excitation signals

Two approaches have been established for exciting the structure:

Excitation by using an impact hammer

Excitation by using an impact hammer: Excitation by using an impact hammer involves force pulses applied to the structure by hammer strokes in order to achieve broadband excitation. Impact hammers have a force sensor that measures the applied force during striking. The mass and rigidity of both the impact hammer and the structure affect the form of the excitation spectrum. The steepness of the impulse is decisive for the excited frequency range. An ideal hammer stroke would be a Dirac impulse, thus a constant excitation over all frequencies. With a real hammer stroke, however, the excitation spectrum falls off towards high frequencies. The aim is to achieve a power spectral density remaining as constant as possible over the relevant frequency range. If a rigid structure is excited by a relatively light hammer, the rigidity of the hammer tip is the main factor influencing the excitation spectrum. The standard accessories of an impact hammer usually include an additional mass and various impact tips (steel, aluminum, plastic, rubber). By selecting a suitable hammer tip, the excitation spectrum can be adjusted to the relevant frequency range. It may be advisable to use hammer tips of different materials, and to combine the resulting transfer functions. Impact hammers are offered with very different masses (from one gram to several kilograms). Combining a small mass and a stiff hammer tip allows for a broad spectrum to be excited, while combining a large mass and a soft hammer tip only allows for a narrow spectrum to be excited. The usable frequency range that can be excited with a very heavy impact hammer extends to approximately 10 Hz. With small impact hammers, frequencies of up to more than 5,000 Hz can be excited. The curve of the excitation force should be relatively flat in the relevant frequency range (see graph on the left). As a rule of thumb, sufficient energy input is achieved when the amplitude of the force excitation drops by a maximum of 10 to 20 dB. In the case of non-linear structures with a pronounced dependence on the force amplitude, automatic hammers are used that reproducibly introduce a very similar impulse into the structure.



Excitation by using a shaker

Excitation by using a shaker system: Shakers usually consist of a vibration source and a force sensor to record the applied force.

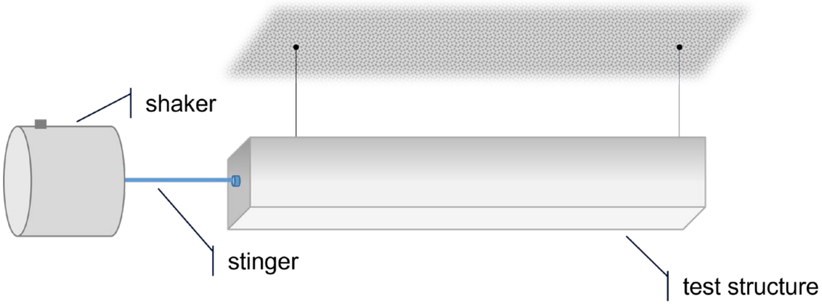
Different types of shakers are available (e.g., electrohydraulic, electromagnetic or piezoelectric shakers). The selection should be made taking into account the requirements, such as desired frequency range and static preload. For excitation, the shaker system is fed with an excitation signal. When using multiple shakers (MIMO measurement), uncorrelated excitation signals must be used. The excitation signal should be selected according to the user's application (e.g., sweep, random noise, burst). In addition, when designing the measurement setup, it must be taken into account that coupling the shaker to the structure may influence the dynamic response of the latter.

The following are some examples of what should be considered in the measurement setup:

- The force application by means of a shaker only occurs in the desired direction (no transverse forces).
- The test structure can move freely despite the shaker.
- The shaker has little or no influence on the mass distribution of the test structure.
- The shaker excites the structure at a suitable point (i.e., not at a vibration node, for example).

Using a stinger

A stinger may be used, for example, to transmit force from the shaker to the structure in order to avoid transverse forces.



Advantages and disadvantages

Theoretically, there should be no difference between the results of measurements with shaker excitation and those with impact hammer excitation. However, differences do occur, as measurements are made on real structures in practice. In the case of shaker excitation, coupling to the structure may cause changes in the eigenmodes. Typically, many accelerometers are used when measuring with shaker excitation and the Roving Accelerometer method. By attaching these sensors, however, the mass distribution of the structure is changed for each measurement. This change, in turn, may affect the vibration behavior of the structure.

Excitation by using a shaker	Excitation by using an impact hammer
Advantages: • Defined excitation signal, thus selectable frequency range • Long-term measurement • Suitable for measurements of very highly damped structures	**Advantages:** • Cost-effective and mobile use • Ready for use quickly • Little time required for a measurement
Disadvantages: • Rather unsuitable for mobile use • More complex and expensive to implement	**Disadvantages:** • Reproducibility problematic with manual excitation • Not suitable for very highly damped structures • Limited frequency range

3. Detection of accelerations

Notes on acceleration sensors

Acceleration sensors can be used to measure the accelerations of the structure. In most cases, piezoelectric acceleration sensors are used. When attaching the acceleration sensors, it must be taken into account that applying additional masses to the structure causes natural frequencies to shift downward. The weight impact of acceleration sensors must not be underestimated. To reduce this effect, the sensor mass should be very small compared to the mass of the structure. Furthermore, the cable of the sensor needs to be laid in a way that disturbing influences are minimized.

Apart from the characteristics of the sensor, the attachment of the sensor to the structure also determines the detectable frequency range. In addition to permanent attachment by screws or connectors, the sensors can also be attached to the structure magnetically or with wax or adhesive (e.g., superglue or X60). The higher the desired frequency range, the stiffer the connections between the sensor and the structure must be. Accelerometers usually have a measuring range between 1 and 10,000 Hz. Care must be taken to ensure that the natural frequency of the sensor is not within the desired measuring range.

Selecting the coordinate system

The alignment of the accelerometers must be adjusted to the local coordinate system of the corresponding measurement point (see also step 1 in the following chapter), as otherwise errors will occur during modal analysis. The local coordinates are automatically transformed into a global coordinate system by ArtemiS SUITE in the background.

By selecting a suitable local coordinate system, the application of accelerometers by means of mounting plates[1] can be avoided in many cases.

The perfect accelerometer

A perfect accelerometer should feature high sensitivity, a wide frequency range, and low mass. In reality, however, compromises usually have to be made, as, for example, high sensitivity often requires a higher mass of the sensor.

Laser vibrometer

In some applications, using contacting sensors is not reasonable or technically not possible, e.g., in case of an unfavorable mass ratio between sensor and measuring structure. In such cases, the use of a laser Doppler vibrometer as a non-contact measurement method should be considered, as this measurement method can determine the movement of the structure without applying additional mass.

4. Process of a modal analysis

Step 1

The basic procedure for experimental modal analysis can be divided into four sections. In a first step, the substitute model of the structure under investigation is created. For this purpose, the structure is approximated by a finite number of structure points. It is important to use uniform coordinate systems and uniform measurement point designations for the model creation and all measurements. This is the only way to ensure error-free allocation of measurement results during evaluation. In addition, the number and position of the measurement and structure points must be selected during model generation in such a way that the vibration modes can actually be identified in the desired frequency range. As with the temporal discretization of analog signals, structural oscillation can only be validly identified if an adequate number of measurement points at suitable positions are considered. Otherwise, the corresponding eigenmodes will not be identified correctly in the case of higher-frequency oscillations, but will instead show the same spatial oscillation patterns as eigenmodes with lower frequencies (spatial aliasing effect). Furthermore, it must be taken into account that a sensor applied to a node of a mode shape cannot record this mode shape.

For a more realistic visualization, model points in the form of a 3D model of the measured object can be created in ArtemiS SUITE in addition to the measurement points.

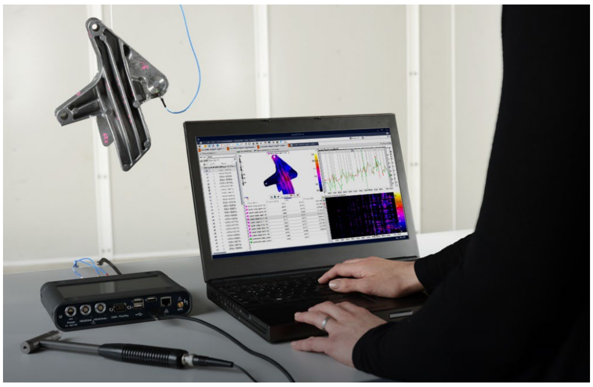
Step 2

In a second step, suitable excitation points must be defined on the structure. Not all points on a structure are equally suitable for force application. It is necessary to find the points on the structure whose excitation actually excites all the mode shapes of interest. This is not guaranteed for all points. If experience with a test object is already available, the suitable excitation points are usually known. If this is not the case, they can be determined by numerical methods or by test measurements. In some applica-

[1] Mounting plates are usually used to adapt the angle of the sensor to the global coordinate system.

tions, especially when multiple mode shapes of a structure are relevant, a single excitation point may not be sufficient.

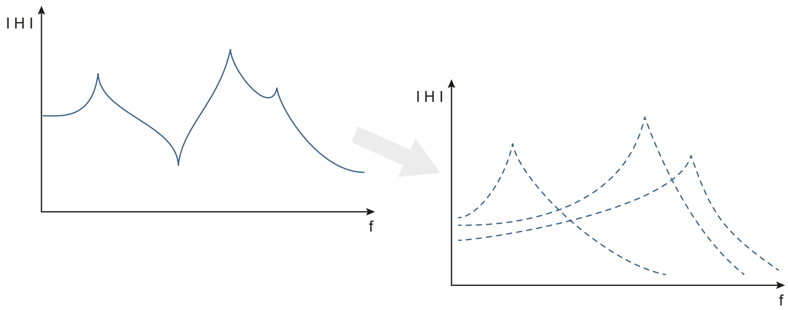
Step 3



In the next step, the measurements are performed and the transfer functions are determined. For this purpose, the structure under investigation is excited with a force (impact hammer or shaker) at the previously defined point(s) and the system response is determined at the same time. The amplitude of the system response changes depending on the frequency of the excitation force. If the system is excited with an excitation frequency close to one of its natural frequencies, the system response is at a maximum. The detected time signals are transferred to the frequency range. The transfer functions and coherence are determined from the averaged frequency spectra. To eliminate irregularities, e.g., interference or noise in the individual measurements, the analysis results of several measurements are averaged.
The coherence can be used to evaluate the quality of the determined transfer function.

Step 4

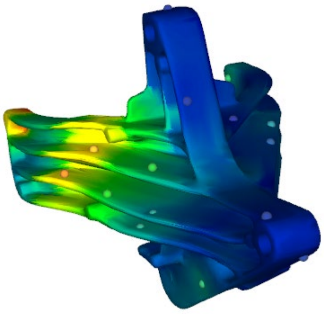
In the fourth step, the modal parameters are determined from the measured transfer functions using curve fitting. For this purpose, different methods are used, e.g., SDOF and MDOF methods. SDOF methods can only be used if weakly coupled modes are present, i.e., if there are natural frequencies in a weakly damped system that are distributed sufficiently far apart. If the eigenmodes of the test object are coupled, MDOF methods are to be used. Moreover, a distinction is made between local and global methods. Local methods use one single transfer function for curve fitting, whereas global methods evaluate a whole set of transfer functions.



The procedure is called curve fitting, because the modal parameters are determined by approximating the measured transfer functions as closely as possible. If the procedure is carried out for multiple transfer functions at the same time, the eigenmodes of the structure associated with the natural frequencies can also be determined.

Visualization

In the visualization process, the information obtained on the modal parameters is applied to the structural model with the various measurement points. In this way, the eigenmodes can be graphically animated for each frequency on the structural model and thus clearly displayed.



Further evaluations

The results of the modal analysis can directly be used to analyze the dynamic response of structures and their vibration problems, and are the starting point for optimizing the dynamic vibration behavior. Furthermore, the results can be used to validate mathematical models and to adapt their parameters to the real behavior. For this purpose, the results of the modal analysis are compared with the calculated vibration behavior from numerical simulations, such as the finite element method, and, if necessary, the FE model is optimized. It may be necessary to adapt the mathematical models, if, for example, material parameters and the geometry are not known in sufficient detail for the simulation, or if the geometry is interrupted, for example by joints that can only be represented as approximations in the modeling. Conversely, comparing measurements and the simulation can also provide an indication of inaccuracies in the measurement. Using the determined modal damping as input parameter for finite element models is another application for the results of a modal analysis.

Proceed to the [third application note on structural analysis] providing an introduction into operational deflection shape analysis.