MEASURING VS MODELLING MODES

Understanding dynamic behaviour is essential in the development of precision systems to achieve the required system performance. This article discusses the value of experimental structural dynamics, highlighting its role in mitigating risks by verifying system dynamics and bridging gaps between model predictions and real-world performance. Two key techniques, experimental modal analysis (EMA) and transmissibility measurements, are explored, showing how they identify dynamics in complex systems. A practical precision-engineering use case illustrates how experimental structural dynamics uncovered unexpected dynamics.

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Introduction

At the start of a new development, dynamic requirements are defined and compiled in a dynamic error budget. Next, the system's architectures are developed and compared with these requirements. Finally, a detailed design is created and the system is built and verified.

In this precision engineering process, the term 'first-time-right' is frequently used. However, for highly accurate and complex systems, achieving first-time-right, based on the idea that all phenomena can be modelled and simulated, is challenging or even impossible due to the model complexities and model (input) uncertainties. As a result, a mismatch between model-based predicted and real system performance is not uncommon. This is where experimental dynamics adds its value in precision engineering by providing methods to make dynamic verifications possible.

Experimental dynamics is the identification of dynamic parameters by means of physical measurements. In every phase of the V-model, experimental dynamics can add value by verifying a requirement, a feasibility set-up and/or a realised (sub)system, which results in a feedback loop back to the design. This can reduce projects risks and time to market.

Experimental dynamics covers or has interfaces with many activities in the development process, such as acoustics, disturbance identification, motion control tuning, sensitivity analysis and system identification. This article focuses specifically on experimental structural dynamics, defined here as the estimation of dynamic properties of structures (via estimation methods often involving the determination of modal properties of the structures) from experimentally measured data in the time domain. The aim here is to explain fellow engineers the value of experimental structural dynamics during the development of high-precision systems.

Two key measurement techniques are further elaborated upon: experimental modal analysis (EMA) and transmissibility measurements. For both techniques it is important to know how to transform a time-domain signal to the frequency domain and how to go from signal analysis to system analysis. In both techniques, the frequency response function (FRF) is an important measurement, which establishes a linear time-invariant relation between input and output signal in the frequency domain. These topics have been further explained in three Mikroniek articles published in 2014 [1] [2] [3].

Experimental modal analysis

EMA is a method to determine the modal parameters of a structural system. The basis of EMA is the measurement of multiple FRFs. For measuring FRFs, a particular position and direction is excited and the response is measured simultaneously. If the point of excitation and measurement coincide, it is called a driving-point measurement, which is often the first measurement. The position of the driving point needs to be selected carefully to avoid that it is placed in a node of a certain mode shape; otherwise, the modal estimation of this mode will be of poor quality.

After the driving-point measurement, either the excitation point or the response position is moved or orientated in another direction, and another FRF is measured. This sequence continues until all measurements are done. The number of measurement points depends on which behaviour of the system needs to be identified; e.g., a higher-order mode shape has a higher spatial frequency and needs more measurement points to prevent spatial aliasing. Finite-element method (FEM) simulations, when available,

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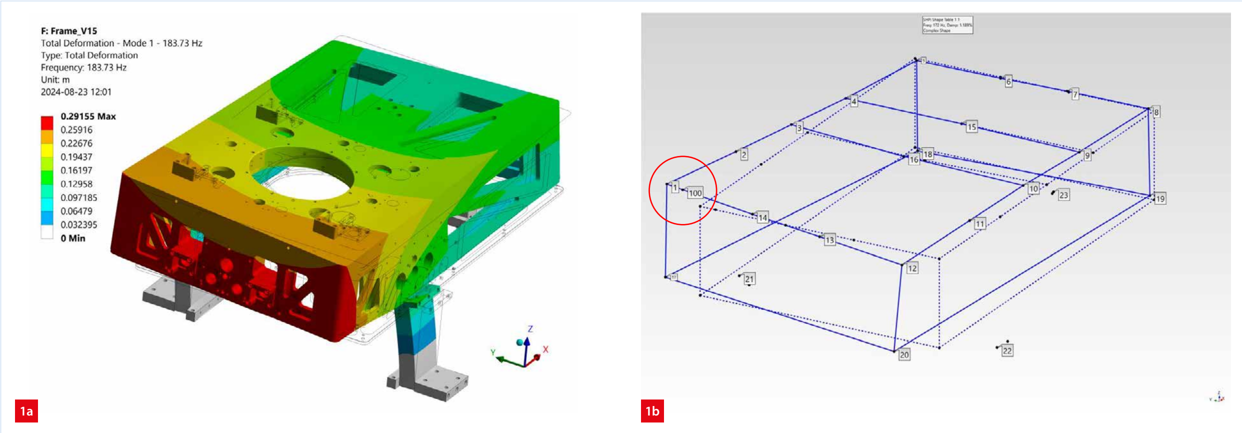


Visualisation of a flexible mode of a metro frame.
(a) Modal FEM analysis result predicting 184 Hz.
(b) Modal experimental analysis result identifying 172 Hz.

are useful to determine the position of the driving-point measurement and the number of required measurement points.

The most commonly used excitation equipment is a shaker or an excitation hammer. Alternatively, an existing actuator in the measured system can also be used as excitation source. However, it often turns out to be difficult to measure forces reliably with this kind of self-excitation. An accelerometer is often selected to measure the response. If the added mass of an accelerometer is relatively high with respect to the modal mass of the measured part or when accessibility is limited, the response can better be measured using laser-doppler vibrometry or photogrammetry.

After performing the measurement, the data can be used to quantify eigenfrequencies, damping and modal vectors using proper tooling based on modal parameter estimation techniques. The background of experimental modal analysis is explained in detail in [4].

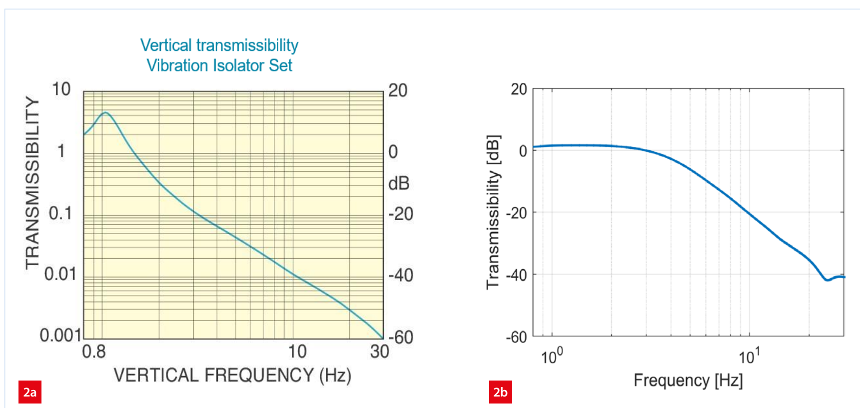
Figure 1 shows a practical example of a comparison between a FEM modal analysis and an experimental modal analysis. Figure 1b shows the wire-frame representation of the device under test indicating the measurement points. For the driving-point measurement, point 100 (encircled in red) was chosen. With roving hammer excitation, the FRFs between all indicated points and the reference point (100) were identified. The mode shape is comparable, but the eigenfrequency is 12 Hz lower in the measurement. This difference did not have a significant impact on the system performance.

Transmissibility

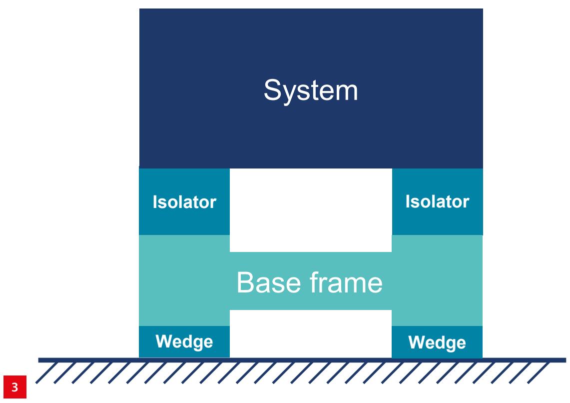
In addition to modal analysis, identifying the transmissibility of vibrations through the system is often required, i.e. the ratio of motion from one point to another point as a function of frequency. This method is a practical way to investigate how disturbances propagate through a particular transmission path. This is particularly useful for components with high damping, such as vibration isolation systems, bearings or dynamic links (cable slabs), as their dynamic behaviour is challenging to predict accurately.

The transmissibility function is a special case of the FRF by being dimensionless. Typically, the measured input and output signals are acceleration, velocity or displacement. While measuring these with two sensors, the system should be excited with, e.g., a shaker or excitation hammer to get a good signal-to-noise ratio (SNR). Additionally, care must be taken to ensure that the transmissibility is measured for the intended path, avoiding interference from unintended parallel paths.

Figure 2 provides an example of a transmissibility function of a vibration isolation table. This table was selected to attenuate floor vibrations for a precision system. In Figure 2a, the supplier's specified transmissibility of a general isolator is shown, while Figure 2b presents the transmissibility of the specific system version as measured using two seismic accelerometers. Comparing the two shows the resonance



Example of the transmissibility of a vibration isolation table.
(a) General definition by the supplier's datasheet.
(b) Experimental measurement of the specific system version.



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2D schematic representation of the use-case precision system.

FEM model 1: ball-joint connection [Hz]	FEM model 2: free-free [Hz]
258	162
275	227
285	264
312	278

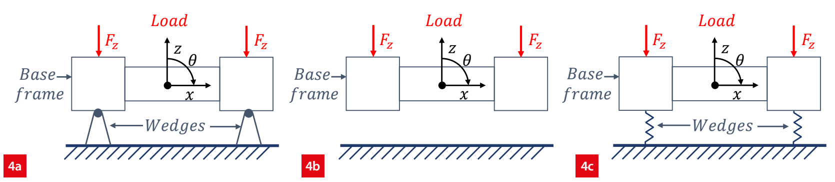
Table 1
FEM modal analysis results: first four non-zero eigenfrequencies.

frequency and damping differ significantly and the transmissibility at 10 Hz is –20 dB instead of –40 dB. Now that the actual transmissibility is known, it can be used to model the floor disturbance attenuation accurately. This example illustrates how experimental structural dynamics can help identify the actual system dynamics and reduce the mismatch between the model-based and the real dynamics.

Use case

This section zooms in on a practical project example of how experimental dynamics has been used at NTS to improve the performance of an existing nanometer-level precision system. Figure 3 presents a 2D schematic of the system. The system is mounted on vibration isolators, which are placed on top of a base frame (weld assembly) to position the isolators correctly. Wedges are positioned under the base frame, at the same horizontal position as the vibration isolators to level the system. The connection between the wedges and the base frame is preloaded by gravity, which will only give a limited interface stiffness between the modules.

In the redesign phase, FEM analyses showed that the system performance was limited by the base-frame compliance. Therefore, a redesign was initiated to increase the eigenfrequencies and thereby improve the system performance. During the redesign, the other components of the system had to remain the same.



Base-frame modelling.
(a) FEM model 1: ball-joint connection.
(b) FEM model 2: free-free condition.
(c) Expected behaviour.

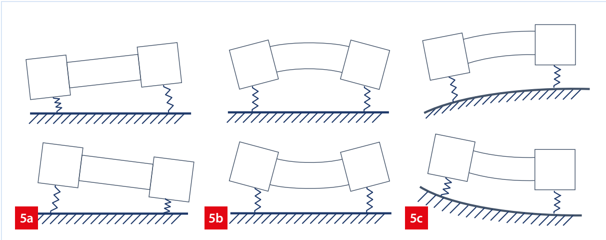
During the redesign phase, FEM analyses were conducted to predict the eigenfrequencies of the new base frame. Two FEM models with different wedge boundary conditions were created, in view of the uncertainty in estimating the exact interface stiffness between the wedges and the base. The first model assumed rigid ball-joint connections at the wedge-frame interface, constraining the base frame in all translational directions (Figure 4a). The second model assumed zero interface stiffness, leading to a free-free simulation with six additional rigid-body modes of the base frame (Figure 4b).

Table 1 lists the first four non-zero eigenfrequencies of both simulations. The expectation in the real system is that there will be a finite interface stiffness (Figure 4c), which results in 'rigid-body modes' with a non-zero frequency (Figure 5a). The true eigenfrequencies of the flexible modes will be in between the simulation results of models 1 and 2 (Figure 5b). In the worst case, when the base frame is mounted in the machine, six 'rigid-body modes' and one flexible mode can be expected below 200 Hz. Experimental modal analysis will be used to identify the real boundary conditions and verify the flexible modes.

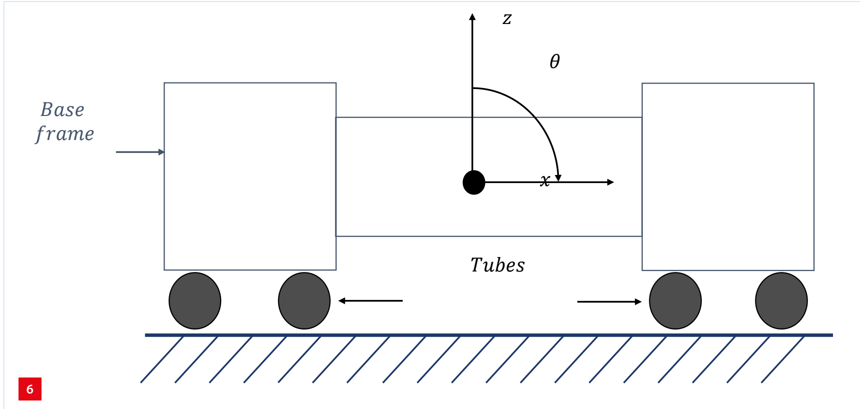
Model verification with EMA

For the verification with EMA, a two-step measurement approach was used to quantify where potential mismatches between the FEM prediction and the real system lie:
1. Conduct a stand-alone free-free EMA of the base frame.
2. Perform EMA with the base frame mounted in the system.

Step 1 verifies the modelling of the base frame itself, while step 2 determines the influence of the interface stiffnesses. All measurements are done using an excitation hammer and a 3D accelerometer. An additional free-free boundary condition FEM simulation without any external load is conducted to compare with the stand-alone EMA. The measurement points are chosen based on FEM simulations to capture the relevant eigenfrequencies and mode shapes. The base frame is placed on rubber tubes to generate the low support stiffness mimicking the free-floating condition (see Figure 6). As a rule-of-thumb, the frequency of the rigid-body modes must be 10x lower than the frequency of the first flexible mode of the device under test.



Base-frame mode shapes.
(a) Rigid-body modes.
(b) Flexible-body modes.
(c) Floor modes and influence on base frame.



Schematic representation of the stand-alone measurement.

Table 2 lists the results of the stand-alone simulation and EMA results. The two results are comparable, which proves that the base frame has been modelled correctly. The measurements show that the first four eigenfrequencies of the base frame are 3-7% higher, which can be attributed to small mismatches between the simulated welds and real welds in the base frame.

Modal verification after installation

After the base frame was installed in the system, the second measurement was performed. It was expected to find six rigid-body modes and one flexible-body mode in the frequency range up to 200 Hz. However, the first couple of measurements identified nine eigenfrequencies already below 200 Hz, indicating more compliancy in the system than anticipated. The stand-alone EMA measurement excluded the base frame as a suspect, which narrowed the root cause down to a difference in boundary conditions.

The mode-shape analysis identified five of the nine modes as rigid-body modes, a combination of *x*, *y*, *Rx*, *Ry* and *Rz* modes, which were expected beforehand. The other four modes showed some flexible behaviour in the base frame and were later identified as floor modes. The wedge stiffness connects the base frame to the floor and therefore the base frame becomes a part of the floor dynamics; see Figure 5c. As an extra verification, a modal analysis was performed of the floor to validate this observation. The first flexible base-frame mode of the mounted system was identified at 248 Hz, which is in the range of the simulation prediction in Table 1.

The results of the EMA analysis were used to update interface stiffnesses and include the floor behaviour in the FEM simulation. In the end, the FEM-simulation results corresponded very well to the measurement, with only 2% difference.

Conclusion

Experimental structural dynamics is indispensable in determining the true dynamics of the real system. This ability adds value by enabling dynamic verifications during the development process of precision systems, which can be used to:
1. Troubleshoot the dynamic behaviour of underperforming systems.
2. Improve the accuracy of dynamic models and performance predictions.
3. Verify the dynamic requirements of realised (sub)systems.

A use case demonstrated how experimental structural dynamics has been used at NTS to dynamically verify realised systems and troubleshoot a mismatch between model-based (FEM) predicted and real system dynamics.

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Table 2
FEM modal analysis results: first four non-zero eigenfrequencies.

FEM simulation [Hz]	Measurement [Hz]	Difference [%]
258	267	3
403	432	7
444	468	5
462	475	3