IEEE Instrumentation and Measurement
Technology Conference
Budapest, Hungary, May 21-23, 2001

Structural Dynamics Modeling using Modal Analysis: Applications, Trends and Challenges

Herman Van der Auweraer
LMS International
Interleuvenlaan 68, B-3001 Leuven, Belgium
Phone: +32 16 384 200, Fax: +32 16 384 350
E-mail: herman.vanderauweraer@lms.be

***Abstract* - System identification plays a crucial role in structural dynamics and vibro-acoustic system optimization. The followed approach is based on the "Modal Analysis" concept. The Eigenmodes of the system model can be visualized and allow a direct physical interpretation. Using the modal models, design improvements can be predicted and the structure optimized. The present paper not only reviews the test procedure and system identification principles of modal analysis, but also discusses the main practical problems with which engineers, performing modal analysis on industrial structures, are confronted on a daily basis. New trends in modal analysis that specifically address these problems are reviewed and illustrated with case studies. This includes the issues of instrumentation, test definition, measurement principles, and parameter estimation.**

***Keywords* – Vibration, System Identification, Modal Analysis**

1. INTRODUCTION: MODAL ANALYSIS

The dynamic behavior of mechanical structures is typically done using a linear system modeling approach. The inputs to the system in general are forces ("loads"), the outputs the displacement or acceleration responses. Using these variables, classical system analysis can be applied.

Specific to the mechanical problem is the straightforward physical interpretation that can be given to the system's Eigenvalues and Eigenvectors. System poles in structural dynamics usually occur in complex conjugate pairs, each pair corresponding to a structural "mode". The pole's imaginary part relates to the resonance frequency and the real part to the damping. Structural damping is typically very low (a few % of the critical damping), hence this damping is usually expressed as a ratio with respect to the critical damping. As a consequence, resonance effects are very outspoken, easily observed, and directly linked to many structural dynamics problems. The system's Eigenvectors, expressed in the basis of the physical coordinates on the structure, then correspond to characteristic structural vibration patterns, referred to as the system's "mode-shapes".

Each mode can be considered as an independent single-degree-of-freedom system. The dynamic system response {x(t)} of a mechanical system to an arbitrary load {f(t)} can hence be written as a linear superposition of the independent single-degree-of-freedom "modal" responses $\{q_i(t)\}$. The derivation and use of this system model based on resonance frequencies, damping ratios and mode-shapes is the essence of the "modal analysis" approach. An example of modes of an aircraft and of a car are shown in a wire-frame geometrical representation in Fig. 1.

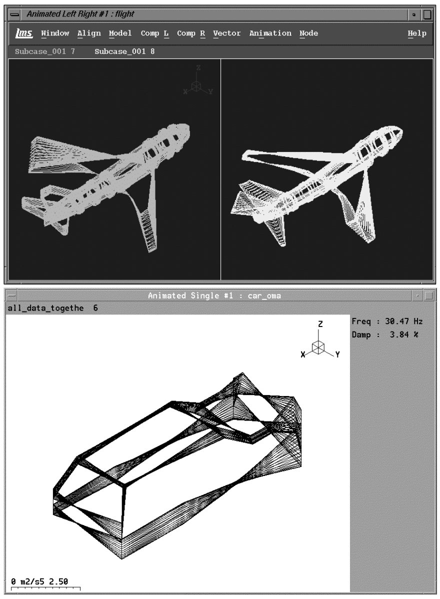


Fig. 1: Aircraft (upper) and car (lower) mode shapes

The modal representation of a mechanical structure can be determined analytically if a lumped mass-spring system is concerned. In the general case of a continuous structure, a numerical approximation by means of a Finite Element Model (FEM) is made, discretizing the structure in a finite number of physical coordinates.

The equations of motion describing this approximated system in the time and Laplace domain are given by:

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\} \tag{1}$$

$$\left[s^2[M] + s[C] + [K]\right]\{X(s)\} = \{F(s)\} \tag{2}$$

with [M], [C], [K] resp. mass, damping and stiffness matrices

The solution of these equations leads to an Eigenvalue problem that is solved in terms of the modal parameters [1].

The limitations of the FEM approach lie in the increasing model size required to properly describe complex structures with appropriate detail (models with over 1 million degrees of freedom are used today in the car and aircraft industry). This leads to higher model construction and calculation times, but even more important, there remain inherent modeling accuracy limitations, related to the modeling of structural junctions, non-homogeneous elements, complex materials etc.

To address these limitations, an experimental approach to modal analysis was developed yielding results which can be used either as a model by itself, or to validate and improve the FE models [2]. The resulting Experimental Modal Analysis (EMA) approach has become a standard element of the mechanical product design and engineering process.

Using the EMA models or the updated FE models, structural design problems related to vibrations, fatigue and noise can be addressed, a diagnosis on the causes made and appropriate design modifications designed and their effect predicted.

2. EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis can be regarded as a “Black Box” or input-output approach, extracting the modal model from input-output measurements. Hereto, the structure under consideration is discretized by means of a grid of “test” locations (which is much coarser than a FE discretization).

Typically, artificial excitation forces are applied at a subset of locations and the corresponding excitation force signals (the “inputs”) as well as the vibration responses at all locations (the “outputs”) are measured. From these data, the modal parameters are extracted using system identification methods. Mechanical reciprocity makes that it is not necessary to excite all inputs as long as all outputs have been measured.

Several particular constraints however make the process of system identification for structural dynamics largely different from this in electrical engineering or process control.

First of all, a continuous structure will have an infinite number of modes. In practice, the analyst will be interested only in a limited number of these modes, up to a certain frequency or only in a certain frequency band. Still, model orders of 50 to 100 are no exception. While some of the modes are rather separated in resonance frequency, others may be very close leading to highly overlapping responses. To illustrate this, a typical Frequency Response Function (FRF) on the engine block of a car is shown in Fig. 2.

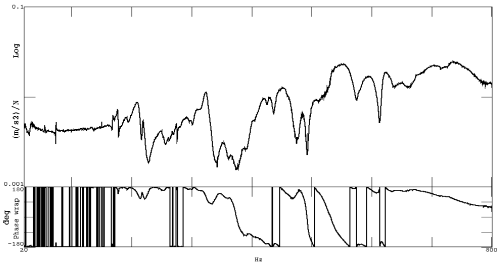


Fig. 2: Typical FRF on a car engine block

Another issue is the large number of measurements to be made and processed. A car body may require over 1000 measurement locations, at each of which accelerations in 3 directions are measured. In many cases, such large number of measurements cannot be made for all responses simultaneously due to equipment (number of sensors, number of data-acquisition channels) or setup (mass-loading) limitations. Hence a full analysis will often be performed in several “patches”, over periods ranging from one day to several weeks. The combined processing of all “patches” (often with inconsistencies in between them) is hence a must [3].

Finally, limitations to the level of applicable excitation forces combined with the large size and distributed damping of many structures make that the response signal levels are often very low and contaminated by measurement noise.

The consequence of these constraints is that classical system identification approaches extracting the parameters of a discrete-time state-space model or of an ARMA model directly from the sampled input-output data are often not practical or even not feasible. In general system theory terms we are speaking of systems with 4 to 8 inputs, over 1000 outputs, on which long time histories (often several thousands of samples) are measured non-simultaneously.

As a result, a large variety of dedicated modal parameter identification approaches have been proposed over the years. In most methods, the raw input-output data are first processed into a non-parametric system description matrix, consisting of Frequency Response Functions (FRFs) or Impulse Responses (IRs). This corresponds to a significant data reduction (allowing long data observations to reduce variance errors) and permits the use and combination of non-simultaneously measured data. This also fits the test reality where the modal testing process is often separated in time (and location) from the actual analysis process.

The matrix of FRFs or IRs can then be used as input data to establish the parameters of a system model such as a frequency or time domain state-space model (a process which is referred to as "system realization"), or to directly identify the structural parameters of a model described by the constitutive equations. The former methods are referred to as Eigensystem Realization - ERA [4] or Subspace Identification (due to the inherent SVD model reduction [5]), the latter as Direct Parameter Estimation methods [6,7].

Alternatively, many model formulations are used which directly exploit specific properties of the FRFs (or IRs). For example, the FRF and IR matrices can be expressed directly in terms of the modal parameters:

$$[H(j\omega)] = \sum_{r=1}^{N} \left(\frac{Q_r \{\psi\}_r \{\psi\}_r^t}{(j\omega - \lambda_r)} + \frac{Q_r^* \{\psi\}_r^* \{\psi\}_r^{*t}}{(j\omega - \lambda_r^*)} \right) \qquad (3)$$

$$[h(t)] = \sum_{r=1}^{N} \left(Q_r \{\psi\}_r \{\psi\}_r^t e^{\lambda_r t} + Q_r^* \{\psi\}_r^* \{\psi\}_r^{*t} e^{\lambda_r^* t} \right) \qquad (4)$$

where
Q_r : modal scaling factor
$\{\psi\}_r$: modal vector r
λ_r : system pole: $\sigma_r + j\omega_r$
σ_r : damping factor
ω_r : damped natural frequency
[V]: = $\left[\{\psi\}_1 \cdots \{\psi\}_N \; \{\psi\}_1^* \cdots \{\psi\}_N^* \right]$: modal vector matrix

This leads to methods such as the non-linear frequency domain method [8], the Least Squares Complex Exponential method [9,10] or the Ibrahim Time Domain Method [11]. Other formulations express the FRF in a rational matrix polynomial formulation using least squares [12,13] or maximum likelihood [14] estimators.

A review of the modal analysis process, including a detailed discussion of the most popular parameter estimation methods, can be found in [15,16].

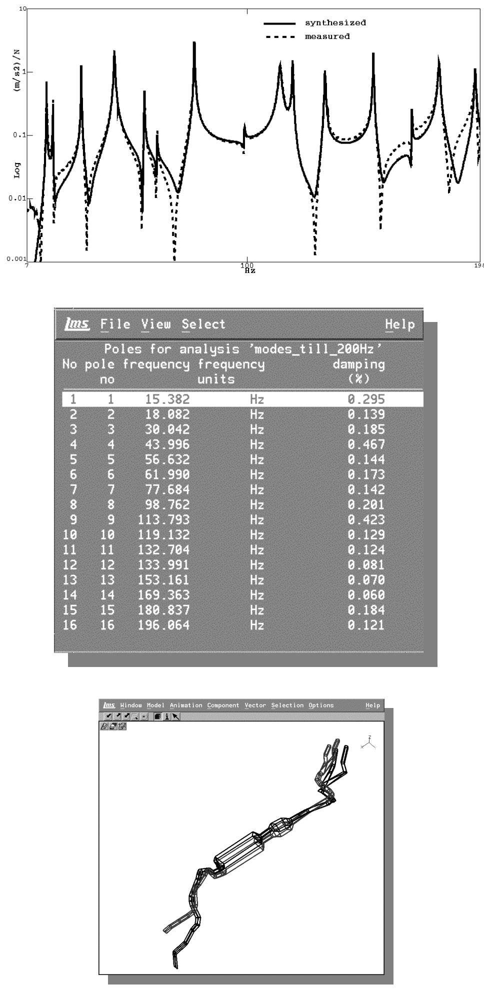
3. INDUSTRIAL APPLICATIONS OF MODAL ANALYSIS

Modal analysis has become a standard approach in today's structural dynamics studies. Typical examples include:

- analysis of a car body, car components (engine, suspension, exhaust, brake,..), a fully equipped car
- analysis of aircraft by Ground Vibration Tests, of aircraft components (landing gear, control surfaces, engines..), in-flight testing in view of aero-elasticity stability (flutter)
- analysis of space launchers, of payloads, payload fixtures, of equipment racks, on-board systems…
- in the process industry: analysis of pumps, compressors, piping systems, shafts and bearings, turbine blades, machinery foundations, high precision equipment…
- civil engineering studies related to bridges, dams, high-rise buildings, off-shore platforms…
- audio and household systems such as washing machine, loudspeaker, CD-drive, computer rack…

In all these applications, it is the purpose to obtain adequate system models to perform troubleshooting as well as system optimization in terms of mission-critical functional performance parameters such as safety, stability, fatigue life, vibration comfort, interior and exterior noise etc.

An example of the process for modal analysis applied to a car exhaust system is given in Fig. 3 where a typical FRF, a table with modal parameters and a mode shape are shown.



Poles for analysis 'modes_till_200Hz'

No	pole no	frequency	frequency units	damping (%)
1	1	15.382	Hz	0.295
2	2	18.082	Hz	0.139
3	3	30.042	Hz	0.185
4	4	43.996	Hz	0.467
5	5	56.632	Hz	0.144
6	6	61.990	Hz	0.173
7	7	77.684	Hz	0.142
8	8	98.762	Hz	0.201
9	9	113.793	Hz	0.423
10	10	119.132	Hz	0.129
11	11	132.704	Hz	0.124
12	12	133.991	Hz	0.081
13	13	153.161	Hz	0.070
14	14	169.363	Hz	0.060
15	15	180.837	Hz	0.184
16	16	196.064	Hz	0.121

Fig. 3: Car exhaust; FRF (upper), pole table (middle) and mode shape (lower)

4. TRENDS AND CHALLENGES

The critical steps and main trends in the application and technology of modal analysis in industry will be reviewed below.

4.1. Test Definition

A modern modal analysis test can easily comprise more than 1000 degrees of freedom. This is due to the complexity of many built-up products, where many components can exhibit a local behavior. In particular, when modal analysis is applied to validate and update a FE model, a sufficiently dense grid of test points is needed. The FE model grid is anyway much finer than the test one, requiring a substantial reduction. So optimally selecting the minimal set of needed test points is very valuable in optimizing the test efficiency.

In case a FE model is available, this model can be used precisely to this purpose: to select the optimal location of (a minimal set of) measurement sensors and excitations for the optimal control and observation of a number of selected target modes [15,16]. This “pre-test analysis” is very important to maximize the test efficiency, reduce test costs and limit the unavailability of the prototype under test. An example is shown in Fig. 4 where the optimal sensor locations for a test on a prototype for the space-station return vehicle are derived from a FE model.

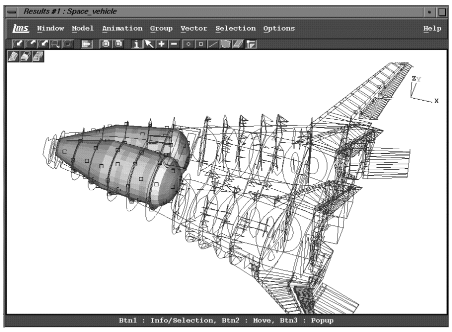


Fig. 4: FE-based sensor grid definition for a space structure

4.2. Test Instrumentation

The increasing complexity –and size- of the tests that are conducted make that instrumentation set-up takes up a significant part of the total test duration. State-of-the-art instrumentation consists of multi-channel data-acquisition front-ends with local conditioning, ADC and signal processing, connected to powerful computers. Channel counts range between 16 and 64, up to as much as 1000. Signal bandwidths range from 100 Hz to 20 kHz, allowing the use of audio-type DSP components such Sigma-Delta ADCs [17].

A schematic overview of a modal test set-up is given in Fig. 5.

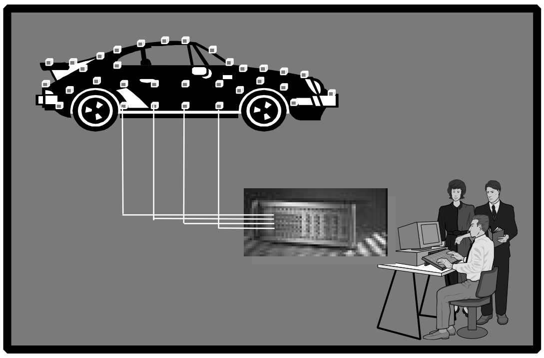


Fig. 5: Modal Test Set-up

As errors in cabling and sensor identification become more difficult to be noticed and corrected, automated procedures including the use of “Smart Transducers” with embedded information in terms of calibration, position etc. become more widespread [18]. The concept of a “Transducer Electronic Data Sheet” or TEDS, embedded in the transducer is in the process of being taken up in the IEEE 1451 standard [19].

An important efficiency gain is furthermore that the structure under test can be pre-instrumented (including sensor calibration) outside the test room, only requiring “blind” cable connection to start the actual test. This optimizes the occupancy and throughput of costly test-rooms (for example semi-anechoic rooms for studying vibro-acoustic effects).

4.3. Optical Measurements

One of the instrumentation constraints with modal analysis is the effect of the transducer mass on the structure [16,20]. Especially with increasing frequency, the influence of the transducer mass increases (larger inertia effect), while the number of transducers has to increase (higher spatial complexity of the modes). Non-contact optical vibration measurement techniques have been developed. hereto

Essentially two approaches are used. One is based on Laser Doppler Vibrometers (LDV) which scan the vibrating surface in a sequential way [21]. This approach supports broadband as well as sinusoidal testing and is frequently used. Fig. 6 shows a typical set-up for measuring the modes of a car door.

Fig. 6: LDV measurement of a car door

The second approach is based on full field Electronic Speckle Pattern Interferometry (ESPI) [22]. This approach uses strobed continuous or pulsed lasers and requires the use of sinusoidal excitation. The imaging takes place using a special CCD camera and specific image processing transforms the qualitative interferograms to quantitative FRF values at the excitation frequency. The complete FRF is then built up frequency by frequency. Special data reduction methods allow to reduce the pixel-density vibration response fields to the spatial resolution needed for a proper modal analysis [23]. The first industrial applications of this approach on cars are documented [24]. An example of ESPI-obtained vibration shapes from the floor plate of a car are shown in Fig. 7.

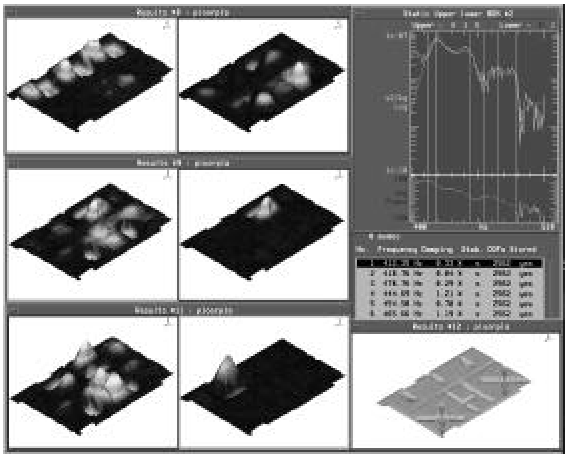


Fig. 7: FRF and ESPI mode shapes from a car floorplate

4.4. *Parameter Estimation*

While many modal parameter estimation techniques have been developed, they all remain highly interactive. The key problems are the selection of the correct model order and the fact that not all resulting Eigenvalues have a physical meaning. Many are due to mathematical effects or to noise and result from the forced fulfillment of the specified model order rather than from dynamic system properties. To overcome this, the concept of the "stabilization diagram" is used. Hereto, a repeated analysis of the same data set is performed, each time for a different model order. The pole values from each analysis are combined in one single diagram, with as horizontal axis the pole frequency and as vertical axis the solution order. The pole is indicated by a symbol in this diagram. Poles corresponding to the physical system appear at nearly identical locations for every analysis, which is readily visible in the diagram. To point out that the frequency (resp. damping and mode shape vector) of a pole falls within certain bounds of the result obtained at a lower system order, this is additionally indicated (for example by a "f", "d" or "v"). This is illustrated in Fig. 8.

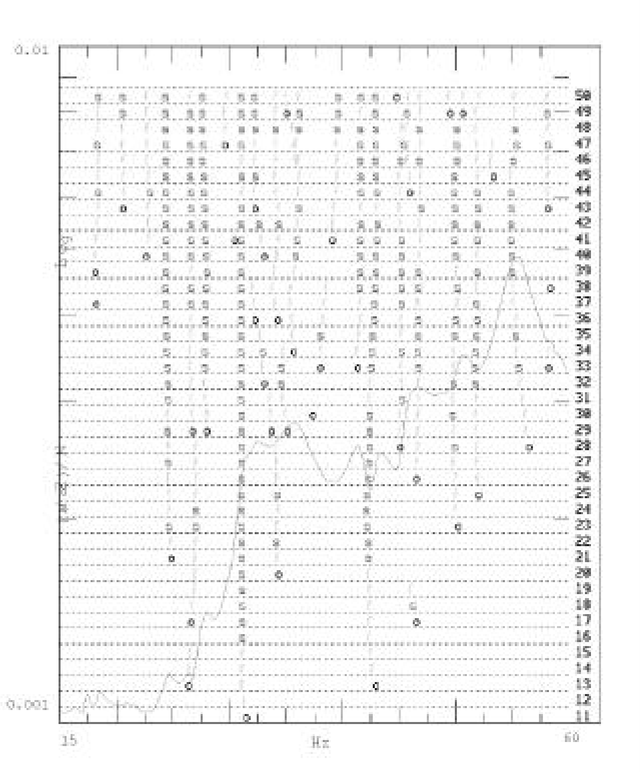


Fig. 8: Stabilization diagram

While this procedure proves to be adequate for interactive selection of valid system solutions, the challenge is to automate the process. Researched solutions include estimation methods which are much more robust with respect to the appearance of spurious poles [13], fully automated, self tuning, algorithms such as the maximum likelihood estimator [26] and also the implementation of the heuristic pole selection rules in an automated procedure. The latter solution is for example pursued to automate the modal identification which has to be done on each Space Shuttle after landing [27,28]. This topic will remain a major research area in the coming years.

4.5. *Vibro-acoustic Modal Analysis*

In many interior noise problems, not only the structure, but also the cavity acoustics show resonance behavior. An example is the "booming" noise in a car at certain speeds, where a cavity resonance is excited by engine or road induced vibrations. It has been shown that the concept of modal analysis can also be applied to acoustical or mixed structural-acoustical systems using as acoustical input variable a source volume velocity and as acoustical output the pressure.

The corresponding system equations, expressed again in a finite element formulation, are [29]:

$$\begin{bmatrix} K^S & -K^C \\ 0 & K^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} - jw \begin{bmatrix} C^S & 0 \\ 0 & C^f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} - w^2 \begin{bmatrix} M^S & 0 \\ M^c & M_f \end{bmatrix} \begin{Bmatrix} x \\ p \end{Bmatrix} = \begin{Bmatrix} f \\ p\dot{q} \end{Bmatrix} \quad (5)$$

Characteristic to this approach is that the Eigenvalues are due to the coupled behavior of structure and acoustics. In general, they can not be separated in a purely acoustical or purely structural cause. The Eigenvectors of course each have a specific acoustic and a structural part. Some special scaling considerations related to the vibro-acoustic reciprocity principle have to be taken into account [29]. An example of the acoustic mode of a car interior is shown in Fig. 9.

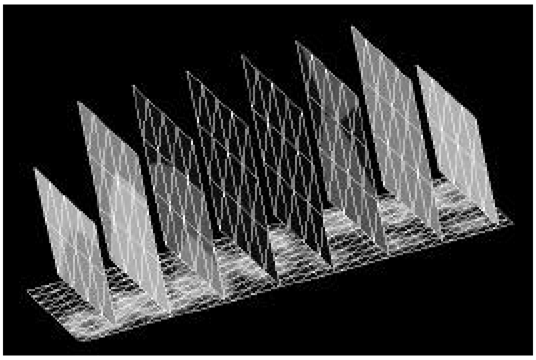


Fig. 9: Acoustic mode shape of a car interior

The acoustic pressure is represented by gray levels in sections of the car. The modal behavior is clearly seen. In this particular case, the acoustic cavity mode coincided with a structural suspension mode, resulting in a strong transmission of wheel vibrations as noise into the car interior [30].

4.6. In-operation Modal Analysis

In the classical modal parameter estimation approach, the baseline data that are processed are FRFs or IRs measured in laboratory conditions. However, in many applications, the real operating conditions may differ significantly from those applied during the modal test. Hence the request to derive models in real operational conditions.

As all real-world systems are to a certain extent non-linear, the models obtained under real loading will be linearized for much more representative working points. Additionally, they will properly take into account the environmental influences on the system behavior (pre-stress of suspensions, load-induced stiffening, aero-elastic interaction, …). Another interest in in-operation modal analysis stems from the fact that in many cases (swell excitation of off-shore platforms, traffic/wind excitation of civil constructions) forced excitation tests are very difficult to conduct, and operating data are often the only ones available. Hence, a considerable interest exists in extracting valid models directly from operating data. Finally, one may observe that in many cases (car road tests, aircraft/spacecraft flight tests, …), large in-operation data sets are measured anyway, for level verification, for operating field shape analysis and other purposes. Hence, extending classical operating data analysis procedures with modal parameter identification capabilities allows a better exploitation of these data.

In most cases, only response data are measurable while the actual loading conditions are unknown. Consequently, over the last years, several modal parameter estimation techniques have been proposed and studied for modal parameter extraction from output-only data [31-33]. They include Auto-Regressive Moving Averaging models (ARMA), Natural Excitation Technique (NexT [34]) and stochastic realization methods [35, 36]. The underlying principle of the NExT technique is that correlation functions between the responses can be expressed as a sum of decaying sinusoids. Each decaying sinusoid has a damped natural frequency and damping ratio that is identical to the one of the corresponding structural mode.

Below, three typical results are presented. In Fig. 10, the mode-shape from tests on a highway bridge, excited by the wind and the traffic underneath, are shown. Despite the low vibration levels and the fact that only response data were available, the resulting mode-shape is of high quality. More details can be found in [37,38].

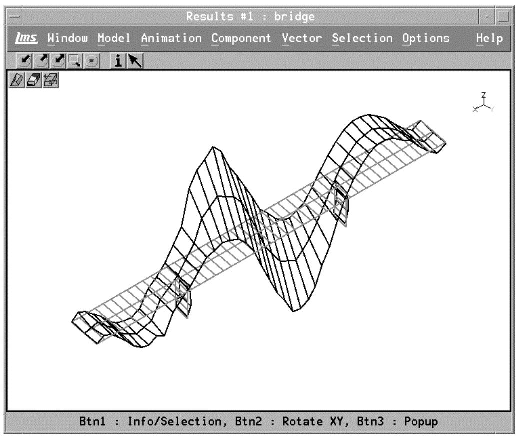


Fig. 10: output-only modeshapes from a bridge

Fig. 11 shows vibration data measured on Ariane flight 501, during and shortly after launch. The launcher as well as internal components such as fuel tanks were instrumented wit over 100 accelerometers. All data were sent by telemetry to a ground station for analysis [39]. From the estimation of the modal parameters in different segments of the flight, the evolution of some of the resonance frequencies as a function of the diminishing fuel amount was clearly observed.

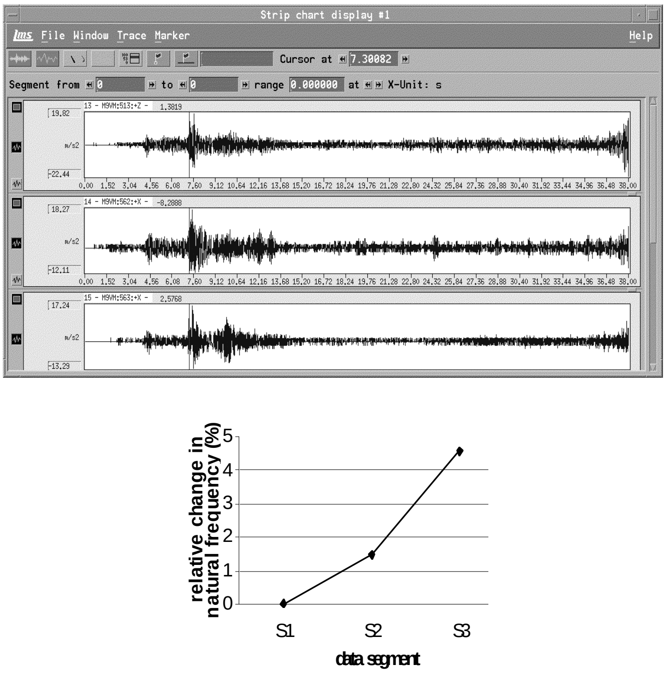


Fig. 11: Launcher flight vibrations (upper) and resonance frequency evolution over 3 time segments (lower)

Finally, in Fig. 12, helicopter mode-shapes, resulting from ground vibration tests as well as from in-flight tests under different conditions, are compared. Though the flight data are of somewhat lower quality, identical mode-shapes are identified.

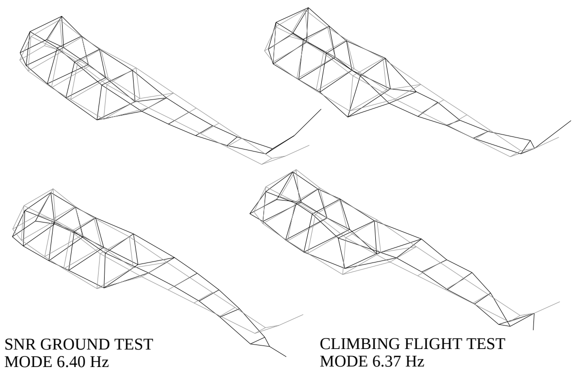


Fig. 12: helicopter mode shapes (ground and flight tests)

4.7. Change Detection

Finally, it should be noted that, in many applications, the problem of structural dynamics identification is closely related to the one of detecting changes in this dynamic behavior. An example is the flight envelope qualification for aircraft, requiring the repeated in-flight modal analysis at different airspeeds. At each airspeed, resonance frequencies and damping ratio's of critical modes are checked to verify the absence of any aero-elastic instability (flutter). Since during the progress from one flight condition to the next one, the dynamics may change due to imminent flutter, the dynamics must be monitored continuously. Other examples are in the use of changes in the modal system model to detect structural damage (e.g. in the Space Shuttle survey's, or for civil structures damage detection) or in the assessment of the integrity of a structure after the forced loading during a qualification test (e.g. a satellite submitted to shaker tests simulating launch vibrations).

Several approaches can be distinguished. The most popular one is where modal analysis is repeatedly performed and changes in the modal parameters are interpreted in terms of changes in the structure. A complete review of related methods can be found in [42]. In a second approach, the modal model is only determined for the "nominal" structure, and new test data are confronted with this model through the calculation of a residual [43,44]. Also purely statistical approaches analyzing low-order statistical models are used [45]. The key problem in all approaches is the capability to recognize and separate the effect of environmental influences, test and analysis errors from the physical changes occurring on the structure. In this context, it should be noted that promising first results have been published on the concept of "model distances" [46]. In [37], a review of methods in view of civil engineering applications is made; in [47], the application to the failure analysis in a sports car is documented.

CONCLUSIONS

Modal analysis has evolved into a standard tool for structural dynamics problem analysis and design optimization. The essence of the experimental approach is the application of a "black-box" system identification approach, but several specific elements (large number of outputs, high system orders…) distinguish this problem form the classical one encountered in electrical engineering. The research area is very dynamic, with a focus on performance improvement, test cost reduction and the development of new application fields such damage detection. Many new system identification techniques are currently proposed, but the main interest of the modal analysis community is to critically evaluate the added value of each method for addressing their specific problems.

ACKNOWLEDGMENT

This research is conducted in the context of the EUREKA project 2419, FLITE. The financial support of the Flemish Institute for the Improvement of the Scientific and Technological Research in Industry (IWT) is gratefully acknowledged.

REFERENCES

[1] R. Craig, *Structural Dynamic*", John Wiley & Sons, 1981

[2] M. Friswell, J. Mottershead, *Finite Element Model Updating in Structural Dynamics*, Kluwer Academic Publishers, Dordrecht, 1995.

[3] H. Van der Auweraer, W. Leurs, P. Mas, L. Hermans, "On the problem of obtaining consistent estimates from multi-patch modal tests", Proc. ISMA 25, Leuven (B), pp. 119-1126, 2000.

[4] J-N. Juang, R. Pappa, "An eigensystem realization algorithm for modal parameter identification and reduction", *J. of Guidance, Control and Dynamics*, vol.8, no.5, pp.620-627, 1985.

[5] P van Overschee, B. De Moor, *Subspace identification for linear systems: theory, implementation, applications*, Kluwer Academic Publishers, Dordrecht, 1996.

[6] J. Leuridan, *Some Direct Parameter Model Identification Methods Applicable for Multiple Input Modal Analysis*, PhD dissertation, Univ. of Cincinnati (US), 1984.

[7] F. Lembregts, R. Snoeys, J. Leuridan, "Application and evaluation of multiple input modal parameter estimation", *Int. J. of Modal Analysis*, vol.2, no.1, pp.19-31, 1987.

[8] J. Busturia, J. Gimenez, "Multiexcitation multiresponse non-linear least squares algorithm", *Proc. 10-th ISMA*, Leuven (B), 1985.

[9] D. Brown, R. Allemang, R. Zimmerman, M. Mergeay, "Parameter estimation techniques for modal analysis", SAE Paper 790221, 1979

[10] H. Vold, J. Kundrat, T. Rocklin, R. Russel, "A multi-input modal parameter estimation algorithm for mini-computers", SAE paper 820194, *Trans. SAE*, vol.91, no.1, pp.815-821, 1982.

[11] S. Ibrahim, E. Mickulcik, "A method for the direct identification of vibration parameters from the free responses", *The Shock and Vibration Bul*., vol.47, no.4, pp. 183-198, 1977.

[12] H. Van der Auweraer, J. Leuridan, "Multiple input orthogonal polynominal parameter estimation", *Mechanical Systems and Signal Processing*, vol. 1, no. 3, pp. 259-272, 1987.

[13] H. Van der Auweraer, P. Guillaume, P. Verboven and S. Vanlanduit, "Application of a fast-stabilizing frequency domain parameter estimation method", Submitted to *ASME J. of Dyn. Systems, Meas. And Control*, unpublished, 2001.

[14] H. Van der Auweraer, J. Leuridan, R. Pintelon, J. Schoukens, "A frequency domain maximum likelihood identification scheme with application to flight flutter data analysis", *Proc. 8-th IMAC*, pp. 1252-1261, 1990.

[15] W. Heylen, S. Lammens, P. Sas, *Modal Analysis Theory and Testing*, Dept. Mech. Eng., K.U.Leuven, Leuven, Belgium, 1995, ed. Society of Experimental Mechanics.

[16] N. Maia, J. Silva, *Theoretical and Experimental Modal Analysis*, Research Studies Press, Taunton, (UK), 1997.

[17] R. Bono, M. Dillon, "New developments in multi-channel test systems", *Proc. ISMA 25*, pp.483-490, 2000.

[18] M. Schiefer, M. Lally, "A framework for smart transducer interface systems", SAE paper 1999-01-1858, *Proc. SAE Noise and Vibration Conference*, vol .P-342, pp.1559-1564, 1999.

[19] IEEE Std. 1451.2-1997, *IEEE Standard for a Smart Transducer Interface for Sensors and Actuators*, The IEEE Inc., New York, 1998.

[20] O. Dossing, "Prediction of transducer mass-loading effects and identification of dynamic mass", *Proc. 9-th IMAC*, pp.306-311, 1991.

[21] G. Cloud, *Optical Methods of Engineering Analysis*, Cambridge University Press, Cambridge (UK), 1998.

[22] R. Freymann, W. Honsberg, F. Winter, H. Steinbichler, "Holographic modal analysis", *Laser in Research and Engineering*, Springer Verlag Berlin, pp. 530-542, 1996.

[23] S. Vanlanduit, P. Guillaume, J. Schoukens, "Development of a data reduction procedure with noise extraction for high resolution optical measurements", *Proc. of the SPIE*, vol. 3411, pp. 357-365, 1998.

[24] H. Van der Auweraer, H. Steinbichler, C. Haberstok, R. Freymann, D. Storer, V. Linet, "Industrial applications of pulsed-laser ESPI vibration analysis", *Proc. IMAC 19*, 2001, in press.

[25] J. Cooper, "On the use of stability plots for modal parameter identification", *Proc. 2-nd Int. Conf. on Identification in Engineering Systems*, Swansea (UK), pp. 375-381, 1999.

[26] P. Guillaume, P.Verboven, S. Vanlanduit, "Frequency domain maximum likelihood identification of modal parameters with confidence intervals", *Proc. 23-rd ISMA*, Leuven (B), pp. 359-366, 1998.

[27] R. Pappa, G. James III, D. Zimmerman, "Autonomous model identification of the space shuttle tail rudder, *Proc. ASME Design Engineering Technical Conf*., DETC97/VIB-4250, 1997.

[28] K. Chhipwadia, D. Zimmerman, G. James III, "Evolving autonomous model parameter estimation", *Proc. 17th IMAC*, pp. 819-825, 1999

[29] K. Wyckaert F. Augusztinowicz, P. Sas, "Vibro-acoustical modal analysis: Reciprocity, model symmetry and model validity", *J. Ac. Soc. Am*., pp. 3172-3181, 1996.

[30] K. Wyckaert, H. Van der Auweraer, W. Hendricx, "Correlation of acoustical modal analysis with operating data for road noise problems", *Proc. 3rd Int.l Cong. on Air- and Structure-Borne Sound and Vibration*, Montreal (CND), pp. 931-940, 1994.

[31] G. James III, T. Carne, and J. Laufer, "The natural excitation technique (NExT) for modal parameter extraction from operating structures", *Int. J. of Analytical and Experimental Modal Analysis*, vol. 10, no 4, pp. 260-277, 1995.

[32] P. Van Overschee, B De Moor, "Subspace algorithm for the stochastic identification problem", *Automatica*, vol.29, no.3, pp. 649-660, 1993.

[33] U. Desai, P. Debajyoti, R. Kirkpatrick, "A realization approach to stochastic model reduction", *Int. J. Control*, vol. 42, no. 4, pp. 821-838, 1985.

[34] M. Desforges, J. Cooper, J. Wright, "Spectral and modal parameter estimation from output-only measurement", *Mechanical Systems and Signal Processing*, vol.9, no.2, pp.169-186, 1995.

[35] J. Cooper, M. Deforges, "Modal parameter identification from ambient response data using advanced identification methods", *Proc. DAMAS '97*, Sheffield (UK), pp. 309-318, 1997.

[36] L. Hermans, H. Van der Auweraer, "Modal testing and analysis of structures under operational conditions: industrial applications", *Mechanical Systems and Signal Processing*, vol.13, no.2, pp.193-216, 1999.

[37] B. Peeters, *System Identification and Damage Detection in Civil Engineering*, PhD dissertation K.U.Leuven, 2000, http://www.bwk.kuleuven.ac.be/bwm/

[38] B. Peeters, G. De Roeck, L. Hermans, T. Wauters, C. Krämer, C. De Smet, "Comparison of system identification methods using operational data of a bridge test, Proc. ISMA 23, Leuven (B), pp.923-930, 1998.

[39] L. Hermans, H. Van der Auweraer, A. Benveniste, M. Goursat, D. Haerens, P. Mourey, "Estimation of in-flight structural dynamics models of a spacecraft launcher", Proc. ISMA23, Leuven (B), pp. 427-433, 1998.

[40] T. Uhl, W. Bochniak, W. Lisowski, L. Hermans, H. Van der Auweraer, J. Malecki, "FE model correlation of a helicopter using ground test results", Proc. 18, pp. 940-946, 1999

[41] L. Hermans, H. Van der Auweraer, A. Hatami, J. Cooper, T. Uhl, W. Lisowski, A. Wasilak, "In-flight modal testing and analysis of a helicopter", Proc. IMAC 17, pp. 80-86, 1999.

[42] S. Doebling, C. Farrar, M. Prime, D. Shevitz, "Damage identification and health monitoring of structural and mechanical systems from changes in their vibration characteristics: a literature review", Los Alamos National Laboratory Report, LA-13070-MS, 1996.

[43] M. Abdelghani, M. Basseville, A. Benveniste, "In-operation damage monitoring and diagnostics of vibrating structures, with applications to offshore structures and rotating machinery". *Proc. IMAC 15*, pp. ,1997.

[44] L. Mevel, L. Hermans, H. Van der Auweraer, "On the application of a subspace based fault detection method to industrial structures", *Mechanical Systems and Signal Processing*, vol.13, no.6, 823-838, 1999.

[45] H. Soohn, M. Fugate, C. Farrar, "Continuous structural monitoring using statistical process control", *Proc. IMAC-18*, pp. 660-667, 2000.

[46] K. De Cock, B. De Moor, "Subspace angles and distances between ARMA models", *Proc. of the 14-th Int. Symp. of Mathematical Theory of Networks and Systems* (MTNS2000), Perpignan (F), Jun. 2000, 7 p

[47] L. Hermans, H. Van der Auweraer, L. Mevel, "Health monitoring and detection of a fatigue problem of a sports car", Proc. IMAC 17, pp. 42-48, 1999.