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APPLICATION OF OPERATIONAL MODAL ANALYSIS TO INDUSTRIAL MACHINERY AND PLANTS

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ABSTRACT

This paper presents experiences and detailed results of Operational Modal Analysis (OMA) applications to industrial machinery and plants. In these projects, OMA techniques supplemented vibration/pulsation assessment and analysis.

First case was part of a refrigeration system with asynchronous drive motor (520 kW) and screw compressor on a vibration isolated frame. Here, experimental modal analysis of the frame was significantly disturbed by nearby machinery. Second case was a gearbox (1 MW) in a test center drive train. OMA algorithms allowed data combination of a permanently installed condition monitoring sensor set with a mobile triaxial vibration sensor. Third case was a study for the development of a new passive and adaptive pulsation damping device in industrial piping.

Keywords: *case study, vibration assessment, screw compressor, gearbox, pulsation damping*

1. INTRODUCTION

Following chapters present backgrounds, methods, and results of three case studies utilizing OMA. In the first two cases, OMA gave additional insight into exceeding machinery vibrations. Results led to specific vibration-reduction measures. Third case study is part of the research and development of a new pulsation damping device for industrial piping. These examples illustrate present-day applicability of OMA to industrial machinery and plants.

2. METHODS

Data was acquired with a mobile system (CS 8008, IMC). Data analysis utilized commercial software (Famos, IMC). Operational modal analysis were performed with Enhanced Frequency Domain Decomposition (EFDD) or Curve-fit Frequency Domain Decomposition (CFDD) methods implemented in the software ARTeMIS Modal, SVS.

Measurements and data assessments of the case study *Refrigeration System* followed standard DIN ISO 10816-3 [1] and guideline VDI 3836 [2] with mobile velocity sensors (PCB IMI) and sample rate 5 kHz. For case study *Gearbox* signals were sampled with 20 kHz from mobile velocity sensors (PCB IMI) and from screwed acceleration sensors (B&K Vibro). All impact tests were conducted with a 1.5 kg modal hammer (KISTLER). Synchronized measurements and data assessments followed standard DIN ISO 10816-3 [1]. Case study *Pulsation Damping* had a 2 kHz sample rate for dynamic pressure sensors (KISTLER) with charge amplifier (B&K).

3. CASE STUDIES

3.1. Refrigeration System

Motivation of this study were noticeable vibrations in a refrigeration system, namely at a compressor package (see Fig. 1 right). The package consisted of a fixed speed asynchronous electric motor (6 kV, 2,995 rpm ≙ 49.9 Hz, 520 kW), clutch, and screw compressor. All elements were mounted on a single, vibration-isolated frame. Root cause for the noticeable vibrations was unknown and therefore a detailed vibration analysis was executed. Beside operational vibration measurements, it was possible to conduct impulse tests for Experimental Modal Analysis (EMA) during standstill of the compressor package. However, an identically constructed secondary unit operated nearby with similar operating conditions during the impact tests for EMA.

Fig. 1 left shows details of data in the vicinity of 50 Hz for two measurement points (M1 and M2) in vertical direction. Frequency Response Function (FRF; mean of five impacts) indicates an Eigenfrequency at 49.9 Hz though collapse of Coherence (Coh) indicates insufficient signal-to-noise ratio for interpretation. Therefore, extended analysis was necessary and hence OMA methods were applied.

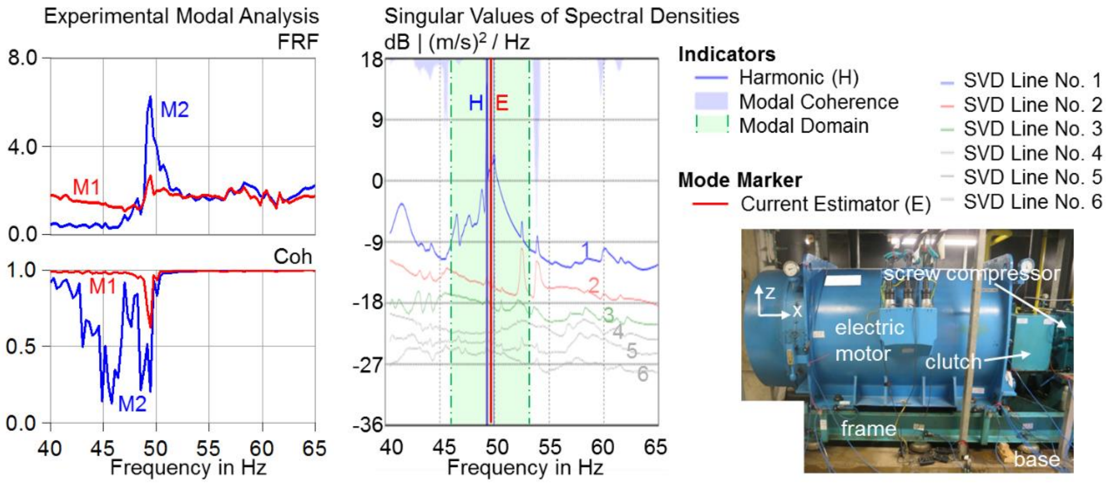
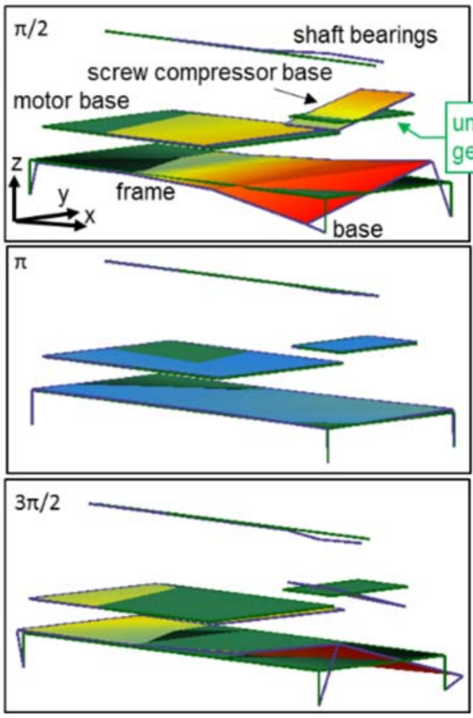


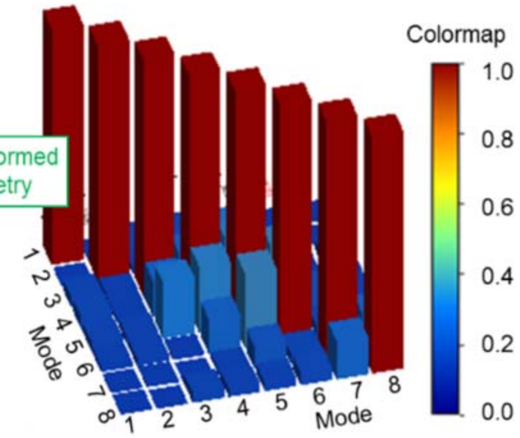
Figure 1. Left: Details of Frequency Response Function (FRF) with peak and Coherence (Coh) with collapse at 49.9 Hz. Center: Corresponding details of SVD data for OMA. Right: Photo of the compressor package.

Fig. 1 center shows a detail of OMA results from (automated) CFDD data analysis for the refrigeration system under study. For this analysis, multiple data sets each including multiple impacts with varied impact points and impact directions have been put together.

In Fig. 1 center appears the rotational frequency of the secondary unit, which is identical with the rotational frequency of the system under study, as harmonic at 49.3 Hz within the modal domain next to the mode estimator at 49.8 Hz. This CFDD result proves the initial hypothesis of an Eigenfrequency at 49.8 Hz. Additional validation for this conclusion provides the Modal Assurance Criterion (MAC) matrix in Fig. 2: all non-diagonal elements are $\ll 1$. Fig. 2 additionally illustrates the mode shape at 49.8 Hz for three phase angles.

Results of this case study show that the noticeable vibrations result from a resonance: rotational frequency and one Eigenfrequency of the compressor package coincide. The corresponding mode shape is a combined bending/torsion of the frame mainly below the compressor. This finding leads to specific vibration-reduction measures: renovation of the vibration isolation elements and stiffening of the frame below the compressor.





Mode	Eigenfrequency in Hz	mode shape
1	2.1	bending of frame
2	3.5	bending of frame
3	7.8	rigid body mode, pendulum
4	14.6	vertical rigid body mode
5	25.4	rigid body mode, pendulum
6	38.0	torsion mode
7	49.8	bending/torsion of frame
8	79.1	bending of frame

Figure 2. Left: Wire frame model and CFDD mode shape freeze images of the refrigeration system at 49.9 Hz. Right: CFDD MAC-matrix (top) and mode list (bottom).

3.2. Gearbox

Vibrations of a gearbox (1,000 kg, 1 MW, max. 13,000 rpm, transmission ratio 13/1) led to exceedance of alarm-threshold values in a test center drive train. It was possible to record data from a permanently installed vibration monitoring system during operation and synchronously from a triaxial mobile sensor. Fig. 3 shows mode shape results from this study. This mode is located in the vicinity of the first tooth mesh frequency of the gear input stage at critical 5,200 rpm.

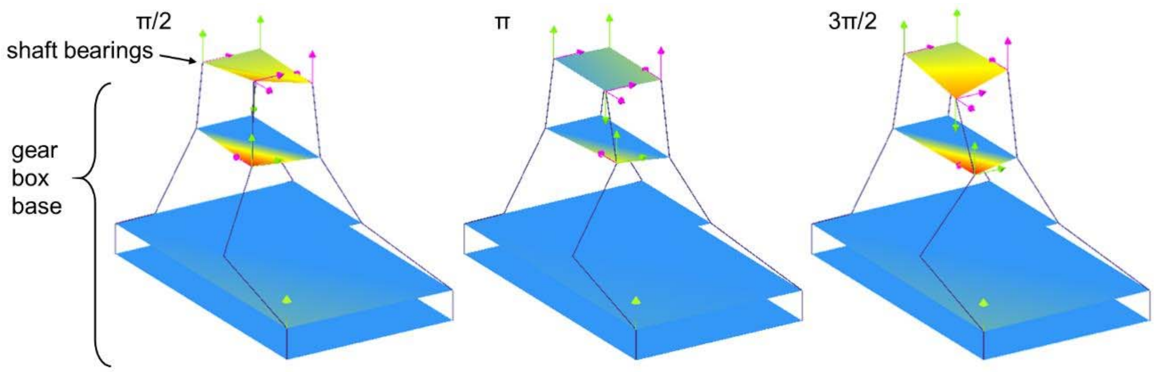


Figure 3. Wire frame model and EFDD mode shape freeze images of the refrigeration system at 2,158 Hz.

The mode shape in Fig. 3 shows that a “soft spot” exists in the gearbox base below the shaft bearing level. Since change of stiffness within the gearbox base was not an option, additional mass and increase of structural damping were vibration reduction measures of choice.

3.3. Pulsation Damping

Pipeline vibrations play a major role in industrial applications. These structural vibrations are often induced by pulsations. A common pulsation reduction measure is the installation of orifices in the pipe system. However, standard orifices induce significant undesirable pressure loss. An alternative pulsation reduction principle is the concept of an adaptive pulsation damper with dynamically varying flow cross section. Theory and design background of this device are presented in detail elsewhere [3].

Part of the research and development (R&D) process was a pilot study at a closed loop air test rig for pulsating flow before the installation of an adaptive pulsation damper prototype. Motivation for this study was the experimental evaluation of acoustic modes in the test section. Fig. 4 right illustrates the test section.

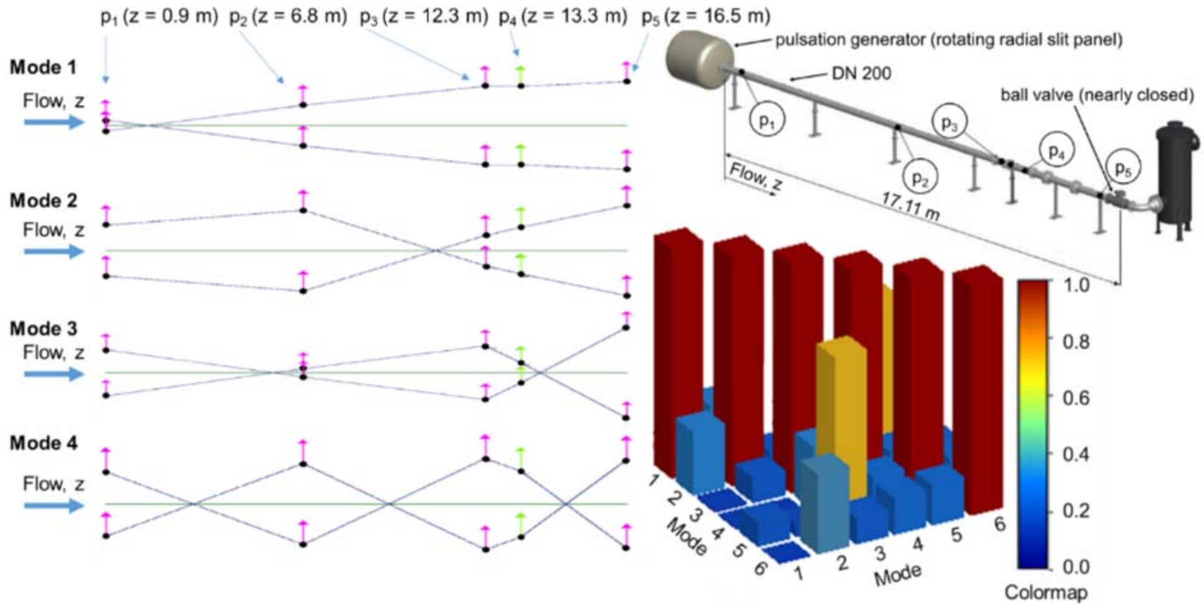


Figure 4. Left: First four pulsation mode shapes of the test section without spatial aliasing. Right: Drawing of test section (top) and MAC-matrix (bottom).

EFDD analysis revealed the acoustic modes of the test section. Fig. 4 shows the results in terms of mode shapes and MAC-matrix. Mode shapes increase node point number with increasing mode order as anticipated. Note the MAC-value of 0.73 between mode 3 and 5. This MAC-value does not indicate linear dependency between these two modes but results from spatial aliasing.

Results of this case study show that it is possible to apply OMA methods to pulsation data in pipelines in order to identify acoustic modes. In general, turbulent flow superimposes harmonic pulsations induced from industrial machinery operation. This stochastic excitation of acoustic modes is a text book excitation type for OMA. Possible applications are validation of Computational Fluid Dynamics (CFD) results and optimization of pulsation reduction measures.

4. CONCLUSIONS

OMA methods beneficially provide additional information about root cause and for possible vibration-reduction measures of exceeding vibrations. This paper presented OMA methods as tools for EMA validation, monitoring data upvaluation, and pressure pulsation analysis in three case studies from the field of industrial machinery and plants.

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