KÖTTER CONSULTING ENGINEERS

MACHINE DYNAMICS

OMA expands the KCE service range.

Resonances may cause inadmissibly high vibrations. In the planning phase or if there are any vibration problems, the natural frequencies therefore must be known precisely. For more precise and reliable determination of the natural frequencies – also under difficult conditions – KÖTTER Consulting Engineers (KCE) now also uses methods of Operational Modal Analysis (OMA).

Operational Modal Analysis (OMA) is a further development of the modal analysis. Natural frequencies, natural shapes and associated damping values of structures can be determined with both procedures. At experimental modal analysis, the structures are specifically excited with known (measured) excitations and the vibration response is measured. The ratio of response to excitation (transmission function) can be used to derive the required information. If the excitation is too small or not known due to external disturbances, modal analysis cannot be used. OMA is used in such cases. It was originally developed for very large structures such as buildings and bridges. Such structures usually do not technically permit targeted vibration excitation without any additional disturbances by, e.g., wind. These problems can be solved easily: only the vibration response is analysed and excitation takes place via the environment – or by the structure itself in operation. If the (unknown) excitation has certain properties, the response signal alone can be used to derive natural frequencies, natural shapes and dampenings. Current further developments and new algorithms now permit use of OMA in the areas of machines and plants as well, as the following project during measurement example of KCE shows quite impressively. In this project, OMA was used to examine increased vibrations in the coolant compression plant from figure 1. The examined plant comprised a fixed-speed asynchronous motor (6 kV, 50 Hz, 520 kW), coupling and screw-type compressor. These were set up with vibration insulation on a shared basic frame. The staff complained about increased vibrations in operation. The cause-effect mechanism for the noticeable vibrations was not known. Therefore, the customer charged KCE with an objective vibration-technical examination of the units and the substructure. As part of the examination, a natural frequency determination by impact tests with a modal hammer took place. These measurements were carried out with the plant standing still. However, the adjacent system of the same build was in operation in parallel.

Analysis of the impact tests showed that several natural frequencies of the plant were in the range up to 100 Hz. In particular the frequency range at approx. 50 Hz stood out. The data quality (coherence) was low, since the adjacent plant impaired the measurement in this frequency range, but the transmission function showed a clear peak in direct proximity of the 49.9 Hz rotation frequency of the plant. To determine whether this was a natural frequency, the recorded data were subjected to expanded analysis by OMA.

For the OMA, the data of all impact tests with different excitation positions and directions were combined. Use of reference sensors made it possible to assign the individual sensor data to the correct phases. In the next step, signal influences of interferences with harmonic excitation ranges (e.g. rotary frequencies of e-motors, grid disturbances) were removed from the data. Two different calculation algorithms were used for vibration: Enhanced Frequency Domain Decomposition (EFDD) and Curvefit Frequency Domain Decomposition (CFDD). The additional provisions of the natural modes make it possible to prove by the Modal Assurance Criterion (MAC) that the modes are linearly independent. Complete linear independence makes MAC = 0, which means a natural frequency.

Table 1 shows that the different calculation procedures only had small deviations in the determined natural frequencies. Additionally, the MAC values of the CFDD calculation are presented as examples. It becomes clear that all secondary diagonal values are near zero and that the modes are therefore all linearly independent. This makes the values of table 1 valid natural frequencies. The OMA results showed that the noticeable frequency at 49.8 Hz actual was a natural frequency of the structure that was in direct proximity to the rotation or main excitation frequency.

Figure 2 schematically shows the associated natural shape of the plant: an overlaid bend/torsion at the compressor-side basic frame.

In this example project, OMA permitted successful determination of natural frequencies in spite of detrimental measuring conditions, which made it possible to develop fitting reduction measures.

You cannot shut your system down? Adjacent units impair your vibrations? We will gladly develop solution approaches for your vibration problems as well. Call us or meet us at the 7th International Operational Modal Analysis Conference IOMAC 2017 in Ingolstadt.

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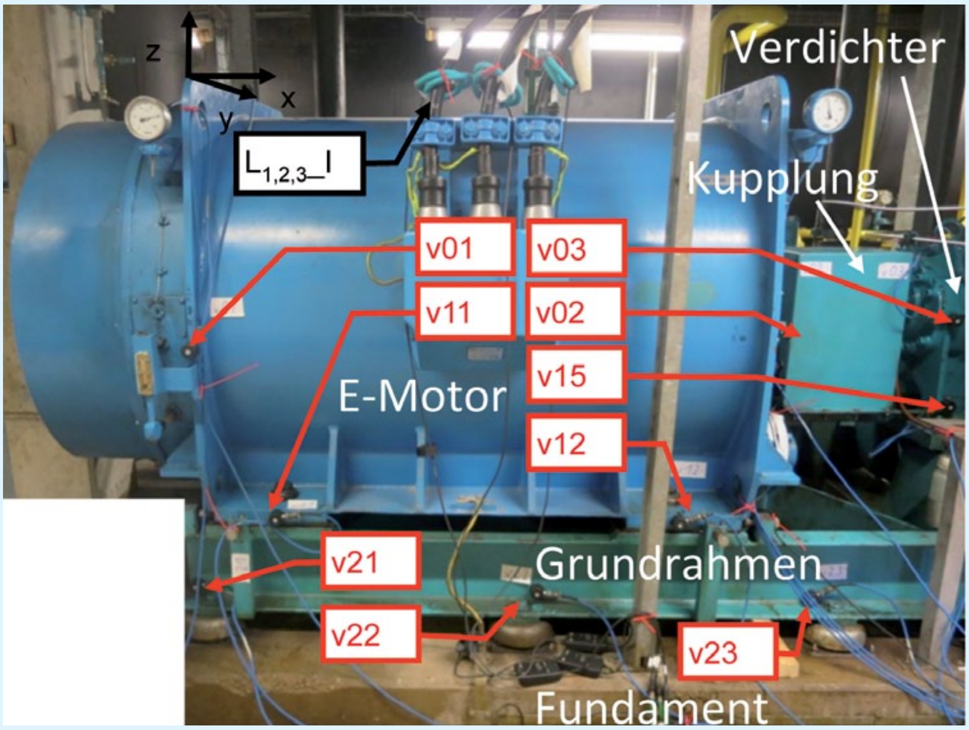
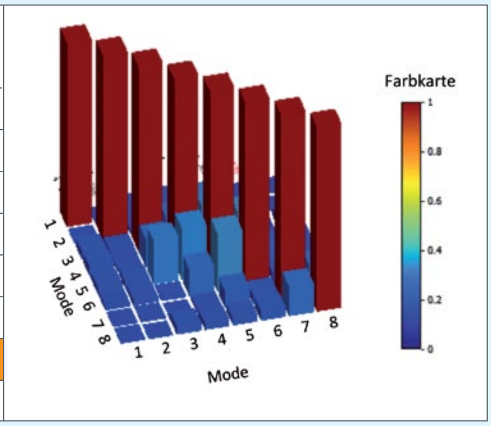


Drive of the coolant compressor plant with vibration and current measuring points

Mode	EFDD	CFDD	Natural shape
	Natural frequency in Hz		
1	2.1	2.1	Bend of the basic frame
2	3.5	3.5	Bend of the basic frame
3	7.8	7.8	Rigid mode, swinging
4	14.6	14.6	Vertical rigid body mode
5	25.4	25.4	Rigid mode, swinging
6	38.0	38.0	Torsion mode
7	49.9	49.8	Bend/torsion of the basic frame
8	79.1	79.1	Bend of the basic frame



Natural frequencies of the plant and 3D illustration of the MAC matrix at the example of the CFDD results

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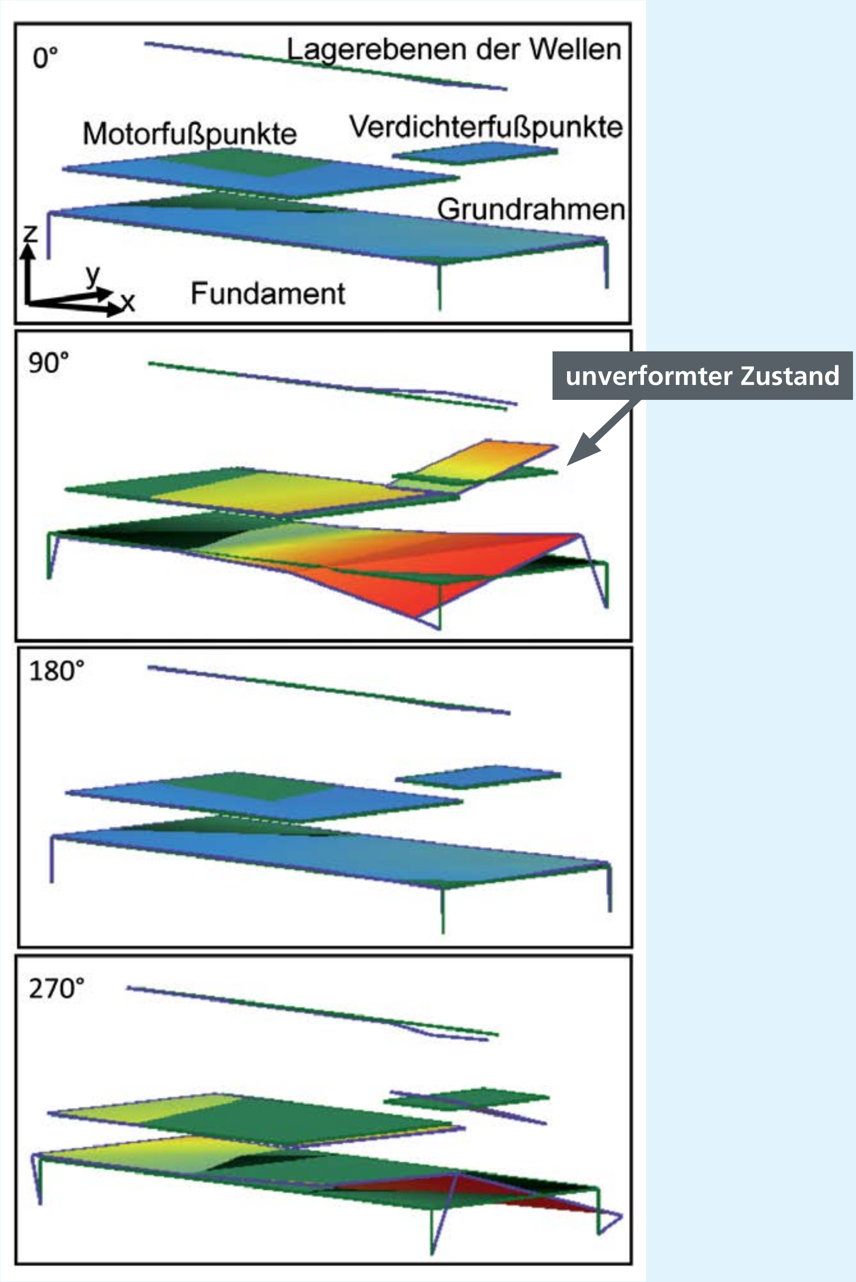

Own shape at 49.8 Hz as still images
(deformation not to scale)

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