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Procedia Engineering 111 (2015) 858 – 863
Procedia Engineering
www.elsevier.com/locate/procedia

XXIV R-S-P seminar, Theoretical Foundation of Civil Engineering (24RSP) (TFoCE 2015)

Experimental (OMA) and numerical (FEM) modal analysis of ball mill foundations

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Abstract

The paper presents the results of the modal analysis of a ball mill foundation, an element of the processing technological line in an ore enrichment plant in Poland. The modal analysis was performed in two ways: numerically, using FEM, and experimentally, using OMA. The analyses results were validated and the FEM model was tuned appropriately. The results show a surprisingly good supporting structure design in an era when modern design methods, such as the FEM method, were not yet available, i.e. in the 1960s. The study results ruled out the state of the ground on which the foundation rests as the cause of large structure vibration. Instead, the poor technical condition of the machinery (ball mill bearings) was indicated as the main cause.

Peer-review under responsibility of organizing committee of the XXIV R-S-P seminar, Theoretical Foundation of Civil Engineering (24RSP)

Keywords: OMA; FEM; Ball mill; Foundations

1. Short description of the problem

The immediate reason for the dynamic analysis was the occurrence of clearly perceivable vibration in the flotation and mill halls. The vibration was large enough to concern the production supervisors and the maintenance personnel working in these halls. Some concerns even arose as to the stability of the hall constructions and the conservation of the foundations of the mills and other structures.

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Peer-review under responsibility of organizing committee of the XXIV R-S-P seminar, Theoretical Foundation of Civil Engineering (24RSP)
doi:10.1016/j.proeng.2015.07.159

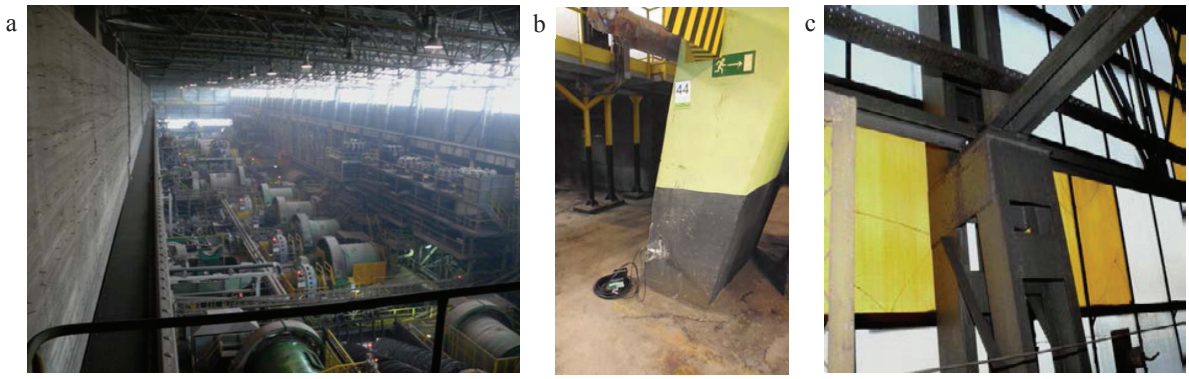

Fig. 1. (a) view of mill hall; (b) reinforced concrete column; (c) steel column.

The structure as a whole is very large and heavy, with dimensions of 351,0 × 88,1 × 30,5 m, and would therefore seem to be immune to local vibration caused by machines functioning within it. However, with the large number of devices causing intense vibration (there are about 30 of the mills alone, Fig. 1a), it was decided that a study should be performed to establish whether the vibration could lead to a destabilization of the structures of the halls and of the engineering objects within them. This was especially justified in that the hall is a multi-nave, relatively complex mixed structure partially made of reinforced concrete (Fig. 1b), partially of steel (Fig. 1c).

As the cause of the larger vibration of some of the mill foundations was not known, the following were to be established:

- whether the dynamic characteristics of some of the machine foundations were different than in other machine foundations (which would indicate they are damaged)
- whether some of the machines generate greater vibration due to their worse technical state.

2. Brüel & Kjær's platform for vibration analysis.

2.1. System PULSE™

Dynamic measurements were performed with the use of a Brüel & Kjær 34 channel PULSE™ system, see Fig. 2. The PULSE™ system we used can be divided into two independent parts, each of 17 input channels, all featuring the frequency range of DC to 25.6 kHz. Because the 3560 PULSE measurement system contains Dyn-X acquisition modules, all inputs reach the dynamic range of 160 dB with ideal linearity and phase matching. The system has been equipped with 17 input channels multi-analyzer PULSE 3560-C (see Fig. 2a) and transducers intended for the purpose of experimental Modal Analysis – a set of 15 seismic high sensitivity accelerometers DeltaTron 8340 (Fig. 2b). Each accelerometer DeltaTron 8340 has a mass – 775g and sensitivity – 1mV/ms^{-2}.

Fig. 2. System PULSE™ (a) multi-analyzer PULSE 3560-C; (b) seismic accelerometer DeltaTron 8340.

2.2. Software – Experimental Operational Modal Analysis (OMA)

The OMA application has been equipped with all of the newest achievements in the field, including automatic search and detection of mode shapes, as well as automatic detection and removal of harmonic contents in the measured signal. The OMA package (Operational Modal Analysis Pro developed by: Structural Vibration Solutions A/S) applied by the authors contains 6 algorithms for obtaining eigenfrequencies and eigenforms: FDD (Frequency Domain Decomposition); EFDD (Enhanced Frequency Domain Decomposition); CFDD (Curve-Fit Frequency Domain Decomposition); SSI-UPC (Stochastic Subspace Identification-Unweighted Principle Components); SSI-PC (Stochastic Subspace Identification-Principle Components); SSI-CVA (Stochastic Subspace Identification-Canonical Variate Analysis). The first three algorithms belong to frequency domain, the rest of the them – to time domain analyses [1]. The method is based on the idea of replacing the response of a discrete system with a sum of responses of many one-degree-of-freedom systems. In algorithms 4-6, the stochastic subspace identification method is used. This method is based on the equation of state of a dynamic system and its decomposition using Kalman filtration [2].

3. Initial dynamic measurements

The initial dynamic measurements were performer in order to identify the type, level and energy of the vibration generated by the machines functioning in the mill and flotation halls. To that end, measurements were taken on the columns supporting the hall structures. The measurements were performed using the multi-analyzer PULSE 3560-C, see Fig. 2a, and the Brüel & Kjær 8340 seismic accelerometers, see Fig. 2b. The transductors (accelerometers) were mounted on the columns using especially designed cubes, which make it possible to measure vibration in one point and in three directions perpendicular to one another, see Fig. 3a.

The analysis results were presented in the form of the so-called „vibration map." Fig. 3b presents examples of RMS values of the measured vibration levels in the horizontal direction x in the 1/3-octave band with a middle value of 2 Hz. The RMS values were analyzed, since they are a certain averaging of the signal (while the amplitude value may distort the signal in non-stationary vibration). There are many different vibration sources within the hall, all giving rise to excitation of differing frequencies. Therefore, the narrow-band analysis (octaves, 1/n-octaves or Fourier's transforms) constitute a better approach than the wide-band analysis. The narrow-band approach makes it possible to extract the signal components corresponding to each excitation frequency – which allows to establish whether the registered vibration stems from one or more sources. The analyses were performed in accordance with the norm [3] in the range 1-100 Hz. The legend on the right side of Fig. 3b ascribes the values of the vibration accelerations to chosen colors.

The areas of heightened vibration were determined using such maps and the mills within those areas (red color) were analyzed in detail, see Fig. 3b.

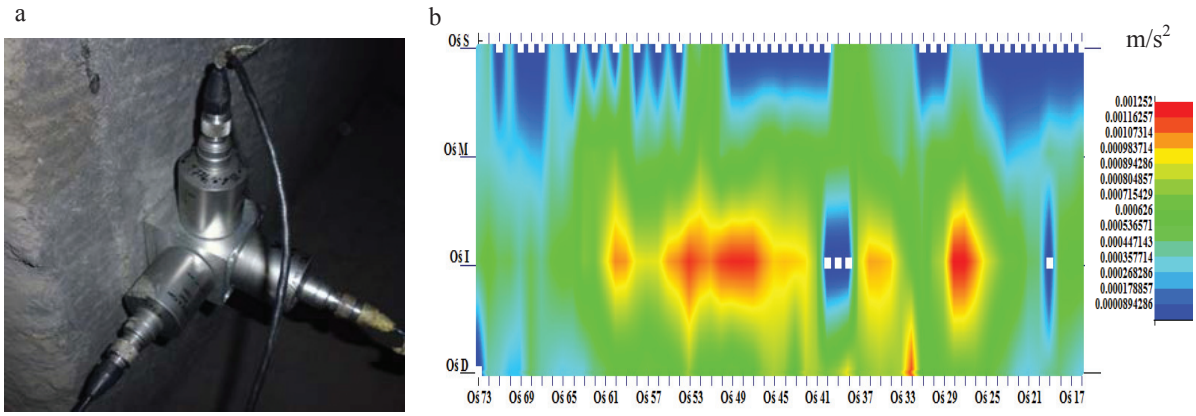

Fig. 3. (a) mounting of the seismic accelerometers DeltaTron 8340; (b) map of hall vibration in 1/3-octave band with a middle value of 2 Hz.

4. Dynamic analysis of mills and their foundations

4.1. Numerical (FEM) modal analysis

Tab.1 shows the first 4 eigenfrequencies of the mill MK 221 determined using FEM algorithms. Fig. 4a presents a FEM scheme of a mill with its foundation, Fig. 4b – an example of the first vertical eigenform of the foundation.

Table 1.FEM and OMA eigenfrequencies of mill MK 221 (mass 159,65 t)

N°	Description of eigenform	Eigenfrequency FEM [Hz]	Eigenfrequency OMA [Hz]
1	rotation around *x* axis	2,25305	2,519
2	rotation around *y* axis	2,97202	–
3	rotation around *z* axis	4,72485	–
4	vertical vibrations (along *z* axis)	5,08093	4,886

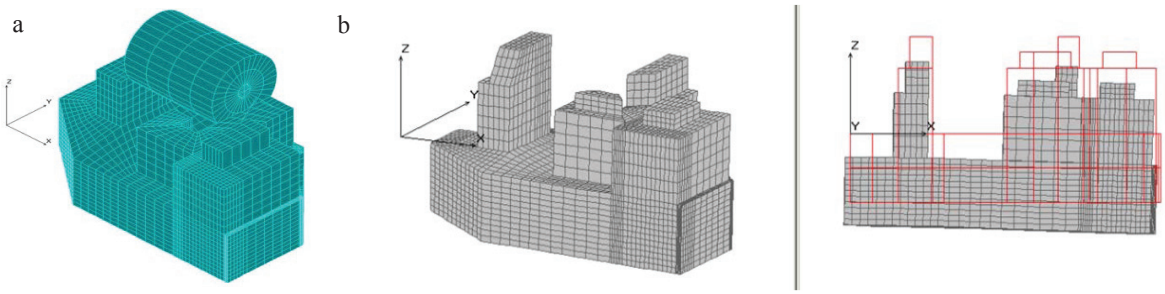

Fig. 4. (a) FEM scheme of a mill with its foundation; (b) first vertical eigenform of the foundation – 5,08093Hz .

4.2. Operational modal analysis (OMA)

The main measurements were performed for three mill foundations, one hour for each. The results presented here are for mill MK 221, which was not functioning during the measurements. The excitation was due to the influence of other working mills and machines. However, the work of those mills and machines is of a deterministic nature, not a stochastic one. Therefore, the excitation of the analyzed mill did not fully fulfill the basic assumption of OMA excitation – it is not white noise. Due to this difficulty, the analysis was limited to identifying two eigenforms: the first rotational and the first vertical ones. The vertical one was used in FEM model validation. The OMA measurement scheme is presented in Fig. 5a. The vibration was measured simultaneously in four points and in three directions in each point. The closest cuboid represents the part of the foundation block that supports the engine, the gear train and the toothed-wheel rim; the farthest one – the part supporting the bearings of the mill drum. The OMA results are presented in Tab. 1, and the identified vertical eigenform is presented in Fig. 5b.

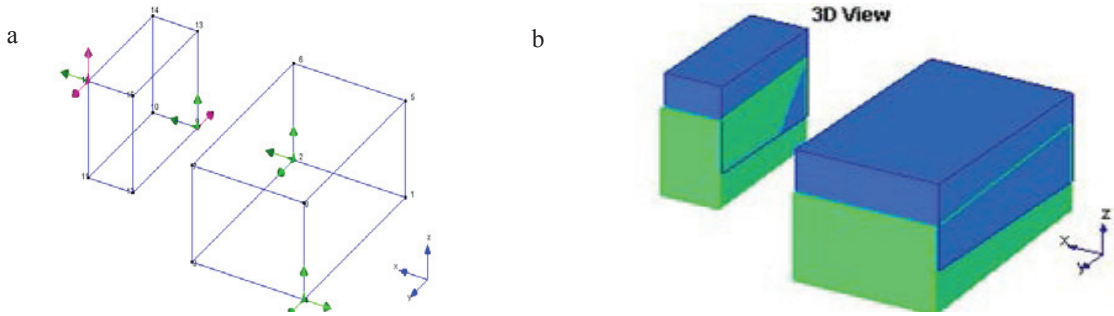

Fig. 5. (a) OMA scheme of the foundation; (b) identified vertical eigenform – 4,886.

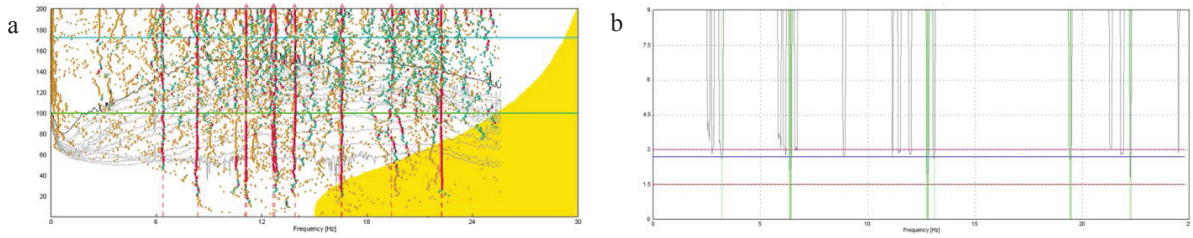

Fig. 6. (a) Stabilization Diagram of Stochastic Subspace Identification (SSI-CVA); (b) detection of harmonic excitation using kurtosis

All six OMA methods were used to determine the eigenfrequencies. The better results were obtained for stochastic algorithms in time domain (SSI). An exemplary stabilization diagram of the SSI-CVA method is shown in Fig. 6a. Using kurtosis, harmonic excitation was detected (Fig. 6b). The harmonic excitations differed from the frequencies of the vibration registered in the bearings and foundations. It was therefore concluded that the dominant vibration was not caused by rotating motion of the mill drum; instead, it was mainly due to the gear teeth engaging.

4.3. ODS analysis of exploitation vibration

The measurements for the ODS analysis were taken using 17 accelerometers, see Fig. 7a. Green color marks the concrete foundations of the drum bearings. Red color marks the bearing. During ODS analysis, a set of ODS Frequency Response Functions was obtained, which are presented in Fig. 8. For some frequencies, their values noticeably peak. These frequencies and the vibration forms corresponding to them were interpreted to be dominant exploitation vibration forms. There are visible excitations with the 4th, 8th, 12th etc. harmonic of the basic frequency of the pinion shaft, i.e., 11,1; 22,2 Hz etc., where 66,6 Hz is the frequency at which the gears engage. The figures below present vibration forms corresponding to frequencies of 11,1Hz (Fig. 7b) and 66,6 Hz (Fig. 7c).

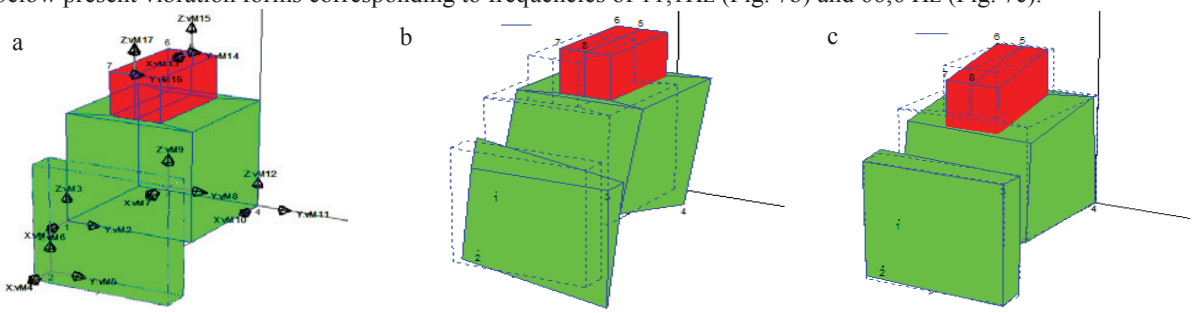

Fig. 7. (a) ODS scheme; (b) ODS form – 11.1 Hz; (c) ODS form – 66,6 Hz

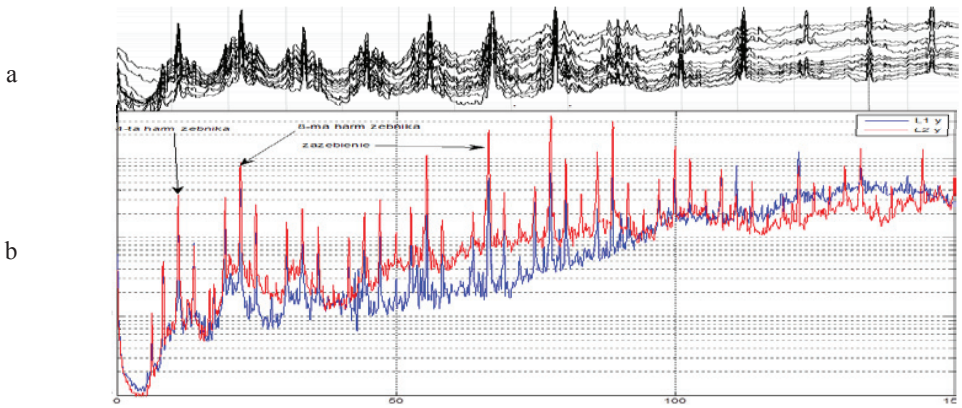

Fig. 8. (a) set of ODS Frequency Response Functions; (b) time history of bearings L1 and L2 vibrations and characteristic frequencies

4.4. Transient resonance analysis and order tracking analysis

Fig. 8b presents the results of transient resonance analysis in the range 0-150 Hz ([3]). Blue (L1) and red (L2) mark the vibration registered in the drum bearings. A comparison between the results shown in Fig. 8a and Fig. 8b leads to the conclusion that the fourth harmonic of the pinion (11,1 Hz, Fig. 8b) corresponds to the amplification of the vibration occurring for this frequency in the ODS function graphs (Fig. 8a). Such an amplification is characteristic of damage to the tooth clutch. Similar results were obtained using order tracking analysis, which is a diagnostic method that allows to evaluate the state of a machine on the basis of the order spectrum. Taking measurements is more difficult, as the rotational speed must also be registered using a tacho probe. Examples of vibration spectrograms are presented in Fig. 9a and Fig. 9b respectively.

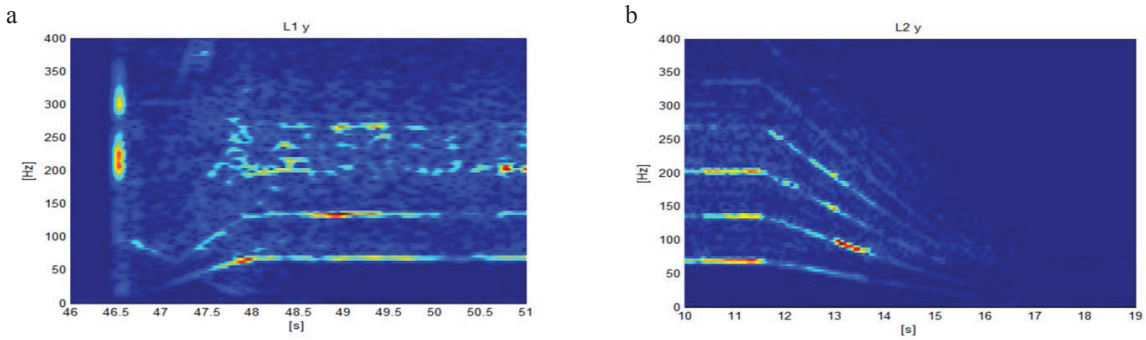

Fig. 9. (a) spectrogram (frequency-time analyses) of bearings L1*y* accelerations; (b) spectrogram of bearings L2*y* accelerations

5. Conclusions

It was found that:

- there is a correlation between large foundation vibration and the poor condition of the machines
- the modal parameters of the analyzed foundations were very similar, which means there were no significant damages to any of the foundations
- the frequencies of the dominant fundament vibration correspond to harmonics of the shaft and engagement
- exceeding the normative vibration levels on the bearings of practically all the mills indicates their overload, which accelerates the rate of wear of the propulsion elements; this in turn causes the vibration to increase, leading to a progressive wear of the pinion teeth; the result is a vicious circle of ongoing deterioration

Finally, the large foundation vibration of the mill MK 221 were diagnosed to be caused by reducing the number of pinion teeth from 27 to 24. This alteration reduced the mill's power consumption, but it was detrimental to the dynamics of the mill's propulsion system. At the engine's rotational speed of 2,778 Hz, the sixth harmonic frequency of the damaged clutch, i.e. 6 × 11,11 Hz = 66,67 Hz becomes equal to the basic frequency of engagement which results from multiplying the rotational speed of the pinion shaft by the number of pinion teeth, i.e. 24 × 2,778 Hz = 66,67 Hz. The frequency of the damaged clutch is four times greater than the rotational speed of the shaft (4th harmonic, i.e. 4 × 2,778 Hz = 11,11 Hz).

If the original 27 teeth remained in the pinion, the basic frequency of engagement would be 27 × 2,778 Hz = 75,00 Hz and would not be equal to the sixth harmonic frequency of the damaged clutch, i.e. 66,67 Hz.

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