

Modal analysis and acoustic noise characterization of a grain crusher

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Abstract

Introduction. Noise is one of the most important physical factors occurring in private farmers' working environment. Hazardous noise exposures and hearing loss have been documented among farmers and farm workers for many years. In most cases, reducing the amplitude of vibratory motion of elements in a machine will reduce the noise generated by the machine element. The authors present the results on studies performed on a grain crusher with the aim of optimizing their acoustic behaviour.

Materials and method. The dynamic behaviour of a grain crusher was investigated by identification of its natural frequency and damping parameters. The study was conducted using the experimental modal analysis technique. The excitation was generated at one single point by a vibration exciter, and the response signals were acquired by an accelerometers fixed at different points of the grain crusher. The modal parameters were determined from a set of frequency response measurements between a reference point and a number of measurement points of the structure.

Results. The introduced structural modifications reduced the noise level by 3dB (A)5 dB[lin] for the hopper component frequencies related to the vibration of the hopper (180 Hz) and the crusher support structure (240 to 480 Hz). The level of these components determines the level of noise at the operator's work station for the average conditions of filling the hopper with grain. The total noise level at the crusher operator was reduced by 2.6 dB (A).

Conclusions. Reducing the amplitude of vibratory motion of the elements in a machine will reduce the noise generated by the machine element. The obtained results confirm that structural dynamic modification is an effective tool for changing the dynamical properties of vibrating systems.

Key words

noise minimization, agricultural machinery, experimental modal analysis

INTRODUCTION

An experimental study of the sound emitted from a grain crusher. The noise at the work station of the grain crusher operator was measured as a function of filling the hopper with wheat grain. Sound level was measured with a microphone located near the ear of the grain crusher operator.

Figure 1 shows comparison of the sound pressure level (SPL) dB(linear) with the A-weighted sound pressure level dB(A) during work at the work station of a grain crusher operator. The greatest noise at the operator's work station occurs at the final stage of crushing wheat when there a small amount of grain remains in the hopper. The highest noise level was 97 dB(A) (105 dB(lin)).

Graphs of noise spectra for individual filling conditions of the hopper of a grain crusher are shown in Figures 2–3.

Reduction of the noise emitted by the grain crusher is required. The experimental modal analysis method (EMA) will be used to reduce the noise. This method allows the introduction of structural modifications to the mechanical system to improve its vibroacoustic properties.

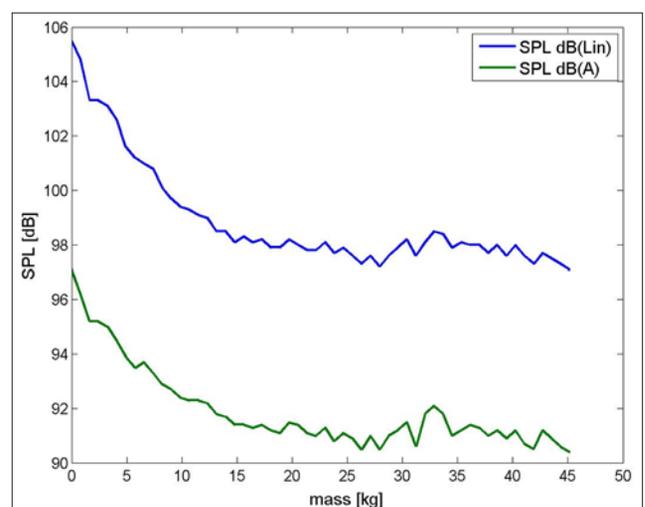


Figure 1. Comparison of sound pressure level (SPL), dB(linear) and A-weighted dB(A) at the work station of a grain crusher operator

Experimental modal analysis method. Two different ways exist for calculating the modal parameters [8]. The first method, called the theoretical modal analysis, assumes knowledge of the structural matrices: stiffness matrix K , mass matrix M and damping matrix C . The modal model is

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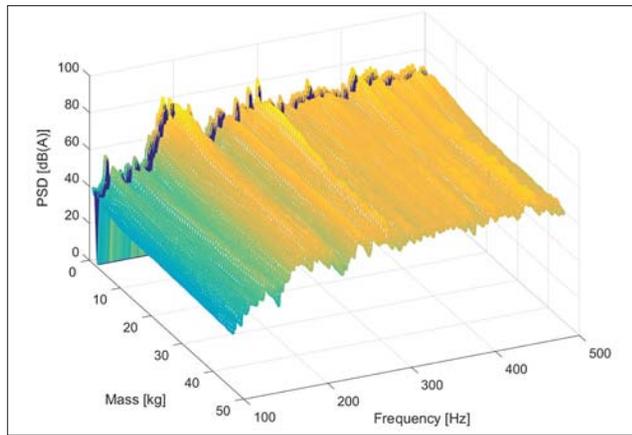


Figure 2. Sound pressure level(SPL) dB(A) spectra at work station of a grain crusher operator as a function of filling the hopper with wheat grain

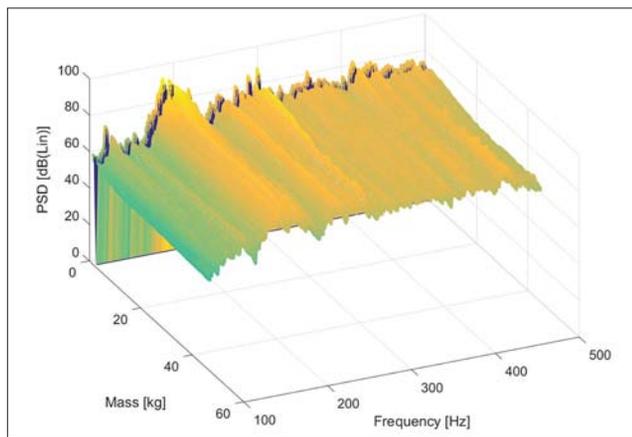


Figure 3. Sound pressure level(SPL) dB(Lin) spectra at work station of a grain crusher operator as a function of filling the hopper with wheat grain

represented by a spectral matrix (diagonal matrix containing eigenvalues) and modal matrix (eigenvectors in the columns). The second approach, called the experimental modal analysis, exploits the system response functions and requires modal analysis identification techniques for the computation of the modal parameters (eigenfrequencies, damping ratios, mode shapes, and modal scaling factors) [4, 5].

The modal parameters can be determined from a set of frequency response measurements between a reference point and a number of measurement points of the structure. Frequency response function $H(f)$ in the frequency domain and impulse response function $h(t)$ in the time domain are used to describe input-output (force-response) relationships of any system, where the signal $a(t)$ and $b(t)$ represent input and output of the physical system. The frequency response function is defined as the complex ratio of the Fourier transformation of an output response $X(\omega)$ (e.g. acceleration) divided by the Fourier transformation of the input force $F(\omega)$ that caused the output.

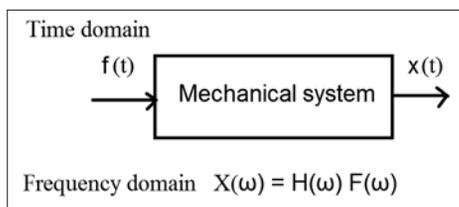


Figure 4. Block Diagram of an FRF

The system equations of motion for the model can be expressed in matrix form as follows:

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

where: $[M]$, $[C]$ and $[K]$ are mass, damping, and stiffness matrices, respectively; $\{f\}$ is the force vector due to external excitation. By taking the Fourier transformation of equation (1), the following matrix form of equation can be obtained:

$$\{X(j\omega)\} = [[K] - \omega^2[M] - j\omega[C]]^{-1} \{F(j\omega)\} \quad (2)$$

where, $\{X(j\omega)\}$ and $\{F(j\omega)\}$ are the complex Fourier transformation vectors of $\{x\}$ and $\{f\}$, respectively, ω is the excitation frequency.

The FRF matrix H can be expressed as follows:

$$X_i(\omega) = H_{ij}(\omega) F_j(\omega) \quad (3)$$

$$H = [-\omega^2 M + j\omega C + K]^{-1} \quad (4)$$

Knowing the resonance frequencies of the system, the damping for the resonant frequencies and the modal matrix (modal shapes) and the modal mass of the FRF matrix can be derived from the relation:

$$H_{i,j}(\omega) = \sum_{r=1}^n \frac{\phi_{ir} \phi_{jr} / m_r}{\sqrt{(\omega_r^2 - \omega^2)^2 + (2\xi_r \omega \omega_r)^2}} \quad (5)$$

where: ϕ_{ir} – modal shape at point i for the r th natural (modal) frequency;
 ξ_r – damping for r th natural (modal) frequency;
 m_r – modal mass for r th modal frequency;
 ω_r – r th modal frequency.

Experimental modal analysis deals with the estimation of modal parameters from vibration data obtained in laboratory conditions.

The component H_{ij} of the FRF matrix, H , corresponds to a particular output response at point i due to an input force at point j . Combinations of excitation and response at different locations lead to a complete set of frequency response functions (FRFs), which can be collectively represented by an FRF matrix of the system.

Basic assumptions in the modal analysis are linearity, time invariance, reciprocity and observability. Reciprocity means that the frequency response function measured between two degrees-of-freedom must be the same regardless of which degree-of-freedom is input or output.

RESULTS

Experimental modal analysis of grain crusher. The dynamic behaviour of a grain crusher under operational conditions was investigated by identification of its natural frequency and damping parameters. The study was conducted using the experimental modal analysis technique. Excitation was generated at one single point by a vibration exciter (i.e. single-input, multiple-output, SIMO, analysis was performed), and the response signals were acquired by accelerometers fixed at different points of the grain crusher.

Dynamic characteristics of the system were determined in the form of frequency response function. The measured

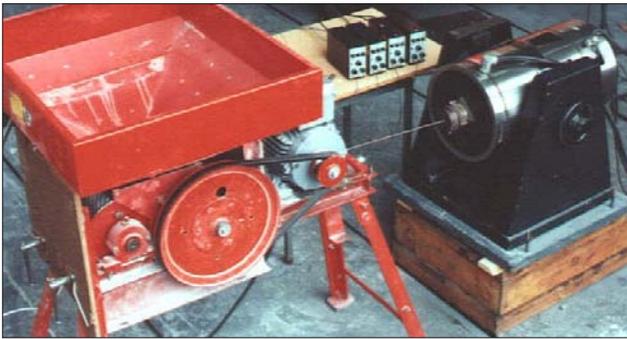


Figure 5. Forced vibration tests of grain crusher

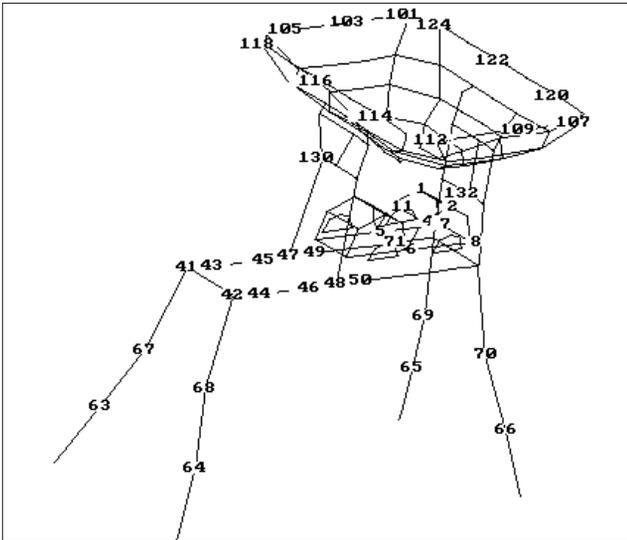


Figure 6. Geometric model of grain crusher introduced to the experimental modal analysis system

characteristics were used to determine the modal model of the system.

Structural dynamic modification of a grain crusher.

Structural dynamic modification (SDM) as an application of modal analysis is a technique for studying the effect of physical parameter changes of a structural system on its dynamic properties, which are in the forms of natural frequencies and mode shapes. The modification may be expressed in terms of the mass, damping, and stiffness matrices [7].

A structural modification was introduced to increase the stiffness coefficients of the machine frame (points: 42–44, 44–46, 46–48, 41–43, 43–45, 45–47) by 6.9 MNm^{-1} . The introduced modification led to a significant reduction in the magnitude of the frequency response function in the 240–280 Hz frequency range. These are the frequencies that determine the vibration level and noise of the crusher in the case of an average filling of the hopper.

Figure 7 shows an example of a comparison of the frequency response function before and after introduction of the structural modification (excitation point No. 5 located on the crushing roller; response point No. 117 located on the hopper of a grain crusher)

Figure 8–9 compare the noise spectra (in dB [A]) at the grain crusher operator’s workstation during grain crushing (0.5 kg of grain in hopper) for the following cases: initial condition and after increase the stiffness coefficients of the machine frame (stiffening of the engine support structure).

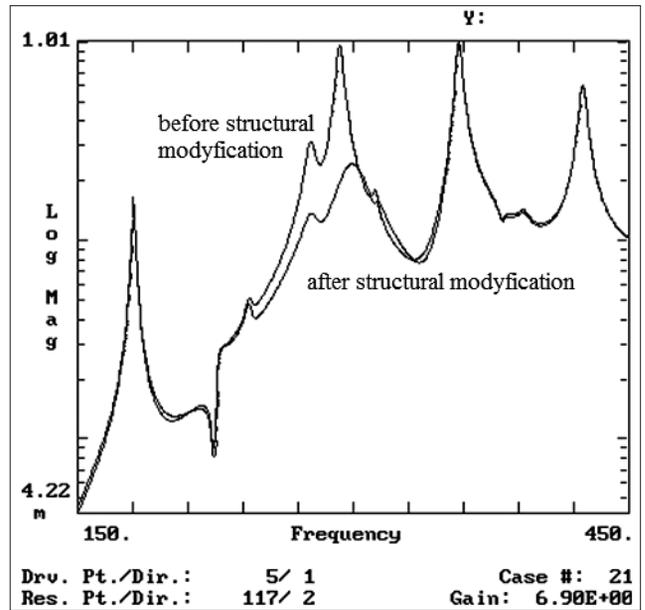


Figure 7. Comparison of the frequency response function before and after introduction of the structural modification

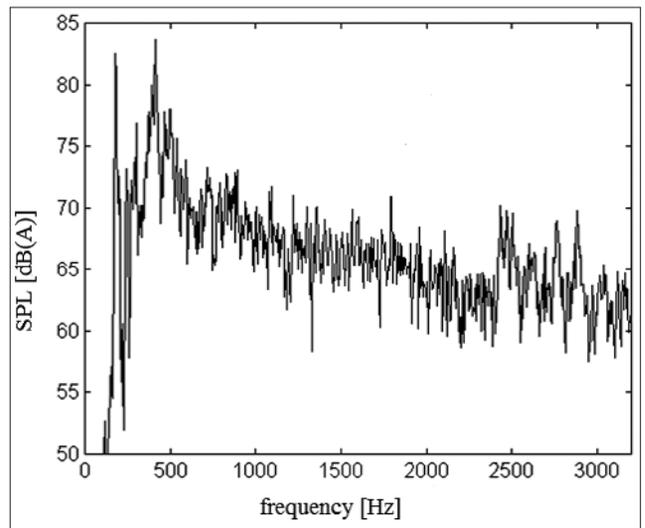


Figure 8. Noise spectra (in dB [A]) at the grain crusher operator’s workstation during grain crushing for initial condition

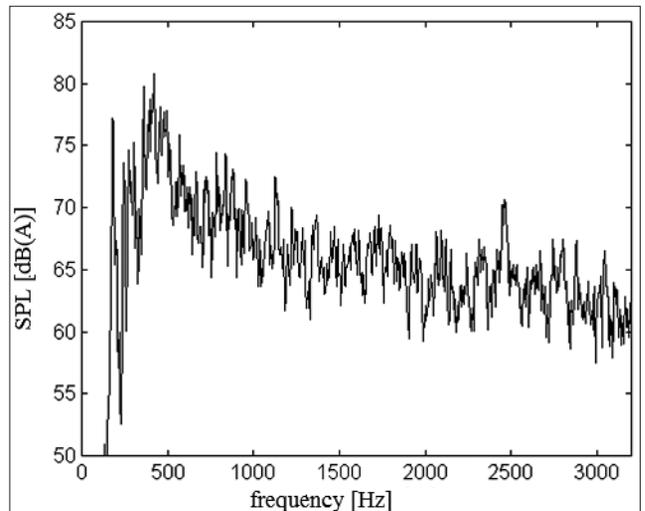


Figure 9. Noise spectra (in dB [A]) at the grain crusher operator’s workstation during grain crushing after increase in stiffness coefficients

The introduced modifications reduced the noise level by 3dB(A) (5 dB[Lin]) for the hopper component frequencies related to the vibration of the hopper (180 Hz) and the crusher support structure (240 – 480 Hz). The level of these components determines the level of noise at the operator's workstation for the average filling conditions of the hopper with grain. The total noise level at the crusher operator was reduced by 2.6 dB (A).

CONCLUSIONS

The article presents an example of the use of experimental modal analysis to reduce the noise of a grain crusher. The sound pressure level at the at work station of the grain crusher operator was high and reached the level 98 dB (A) at the end of the crushing process (small amount of grain in the hopper).

Structural dynamic modification was used to reduce the noise of the crusher. This method allowed the introduction of structural modifications to the mechanical system to improve its vibroacoustic properties. After increasing the stiffness coefficients of the machine frame, reduction of vibration and noise emitted by the machine was achieved. The total

noise level at the crusher operator was reduced by 2.6 dB (A). The obtained results confirm that structural dynamic modification is an effective tool to change the dynamic properties of vibrating systems.

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