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AN EXPERIMENTAL STUDY ON THE DYNAMIC BEHAVIOR OF A TRACTOR AGGREGATED WITH BALER-WRAPPER

Summary

The article presents methods of testing the dynamic behavior of a machine. The analysis of structural vibration is necessary in order to calculate the natural frequencies of a machine, and the response to the expected excitation. The dynamic properties of the machine can be determined by experimental methods. Presents the results of the tractor–baler-wrapper unit tests using operating deflection shapes (ODS) analysis for determining the vibration patterns of machine under various operating conditions.

***Key words**: machine dynamics, modal analysis, operational deflection shapes, operational loads*

BADANIE EKSPERYMENTALNE ZACHOWAŃ DYNAMICZNYCH AGREGATU CIĄGNIK–PRASOWIJARKA

Streszczenie

W artykule przedstawiono metody badania zachowań dynamicznych maszyny. Analiza drgań jest niezbędna do obliczenia częstotliwości drgań własnych maszyny oraz odpowiedzi na oczekiwane wymuszenie. Właściwości dynamiczne maszyny można określić metodami eksperymentalnymi. Przedstawiono wyniki badań agregatu ciągnik–prasoowijarka z wykorzystaniem eksploatacyjnej postaci drgań (ODS) pozwalające na wyznaczenie odkształceń dynamicznych konstrukcji na rzeczywiste wymuszenia działające w różnych warunkach polowych.

***Słowa kluczowe**: dynamika maszyn, analiza modalna, eksploatacyjna postać drgań, obciążenia eksploatacyjne*

1. Introduction

Agricultural machines are characterized by high dynamic loads during their operation. An example of such a machine is a baler-wrapper. At the design stage, it is important to determine the nature of changes and the value of dynamic loads in the designed machine. The correctness of design calculations and, as a result, operational reliability and the cost of machine production depend on the determination of the state of these loads. The first step in analyzing dynamics is usually to determine natural frequencies of machine elements and systems.

The analysis of structural vibration is necessary in order to calculate the natural frequencies of a machine, and the response to the expected excitation. The dynamic characteristics of the machine can be determined by experimental methods. Modal testing is an experimental technique used to obtain the modal model of a system. The theoretical basis of the technique is the relationship between the vibration response at one location and excitation at the same or another location as a function of excitation frequency. The natural frequencies are inherent properties of a structure. They don't depend on the forces or loads acting on the structure. Modes change when the material properties (mass, stiffness, damping properties), or boundary conditions of the structure change.

Operating Deflection Shape (ODS) analysis is a method used for visualization of the vibration pattern of a machine as influenced by its own operating forces. ODS results differ significantly from modal shapes. They depend on external forces and working loads and will change if the load changes [7].

2. Material and methods

Experimental identification of structural dynamic models is usually based on the modal analysis approach. However, in many applications, the real operating conditions may differ significantly from those applied during the modal test. In operational modal analysis modal parameters are estimated in operational condition from response data without knowing the input loading force.

2.1. Experimental modal analysis

If you have a structure for testing on which measurements can be made, it can be assumed that you can define a parametric model that correctly characterizes the structure.
The modal analysis can be performed as theoretical, experimental or operational modal analysis. Theoretical modal analysis usually applies Finite Element Method (FEM) to solve dynamic problems of the analyzed structure [6]. Modal analysis is the process of determining the dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes, and using them to formulate a mathematical model of its dynamic behavior.

Experimental modal analysis (EMA) is the process of determining the modal parameters by way of an experimental approach. 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 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 transform of an output response X(ω) (eg. acceleration) divided by the Fourier transform of the input force F(ω) that caused the output.

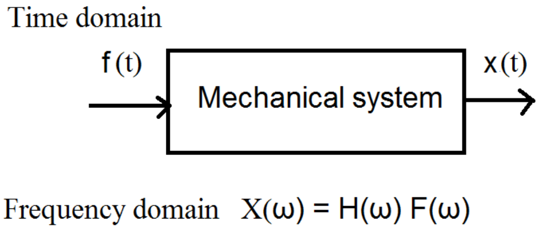

Fig. 1. Block Diagram of an FRF
Rys. 1. Schemat blokowy FRF

For an *n* degree-of-freedom (DOF) linear vibratory system, the equation of motion can be expressed as:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \quad (1)$$

where M, C, K are the system mass, damping and stiffness matrices respectively.

Transforming equation (1) in the frequency domain by Fourier, it gives:

$$(-\omega^2 M + j\omega C + K)X(\omega) = F(\omega)$$

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

Transforming equation (1) in the frequency domain, it gives:

$$(-\omega^2 M + j\omega C + K)X(\omega) = F(\omega)$$

where, {X(jω)} and {F(jω))} 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:

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

$$X(\omega) = H(\omega)F(\omega)$$

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:

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

where: ϕ_{ir} -modal shape at point i for the rth natural (modal) frequency,

ξ_r - damping for rth natural (modal) frequency,

m_r - modal mass for rth modal frequency,

ω_r - rth 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.

2.2. Operational deflection shapes

All experimental modal parameters can be obtained from measured . That is, experimental modal parameters can be obtained by artificially exciting a machine or structure, measuring its operating deflection shapes (motion at two or more DOFs), and post-processing the vibration data [1, 4, 7].

ODS is defined as the displacement of the structure at the chosen vibration frequency with external excitation acting on the given structure. In this way, displacement of the structure during excited movement can be determined – understood as relative movement of the chosen point with reference to the remaining points. Since the movement is a vector (acceleration, velocity or displacement) it has a point of application, direction and magnitude, which are defining by the way of displacement of the structure during its movements. ODS differs from modal model parameters (modal vector). ODS depends on excitation; and when the applied loading changes - ODS also changes. ODS has the dimension of the displacement, velocity or acceleration, depending on the values collected during the measurement [2, 3]. In an experimental way, the measurement is carried out using mechanical vibration sensors. The first one is located in the reference point, the second sensor is moved between the selected (nodal) points of the structure. As the result, one obtains individual and relative (cross-) spectra of the accelerations, as well as the functions of spectral transmissibility T(ω). T(ω) is the complex transmissibility function of acceleration/acceleration type [1].

$$T_{ij}(\omega) = \frac{X_i(\omega)}{X_j(\omega)} = \frac{\sum_k H_{ik}(\omega)F_k(\omega)}{\sum_k H_{jk}(\omega)F_k(\omega)}$$

where "j" is the index of the reference point, whereas "i" refers to the test-point monitoring of the structure.

3. Results and discussion

3.1. Identification of modal parameters of baler-wrapper

The study of the operational mode of vibrations was carried out for during the travel of the baler-wrapper in the field at a speed of 10 km/h (Fig. 2).

Measurement points are located at:

- C1 - rear wrapping beam, left side
- C2 - rear wrapping beam, right side
- C3 - frame, above the left wheel
- C4 - baler drawbar at the hitch
- C5 - upper beam above the chamber
- C6 - frame, above the right wheel.

Source: own study / Źródło: opracowanie własne

Fig. 2. Tractor baler-wrapper set during field tests
Rys. 2. Agregat prasoowijarka–ciągnik podczas badań polowych

Measuring signals from above-mentioned vibration sensors were recorded simultaneously. Fig. 3 shows the location of the vibration acceleration sensors used to determine the ODS in the baler-wrapper.

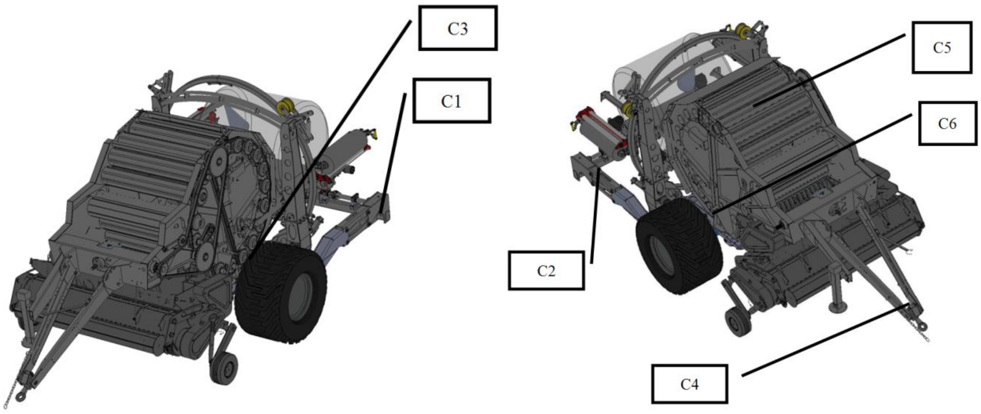

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Fig. 3. Test point location
Rys. 3. Rozmieszczenie punktów pomiarowych

Processing of measuring signals was carried out with the LMS Test.Lab and Matlab software.

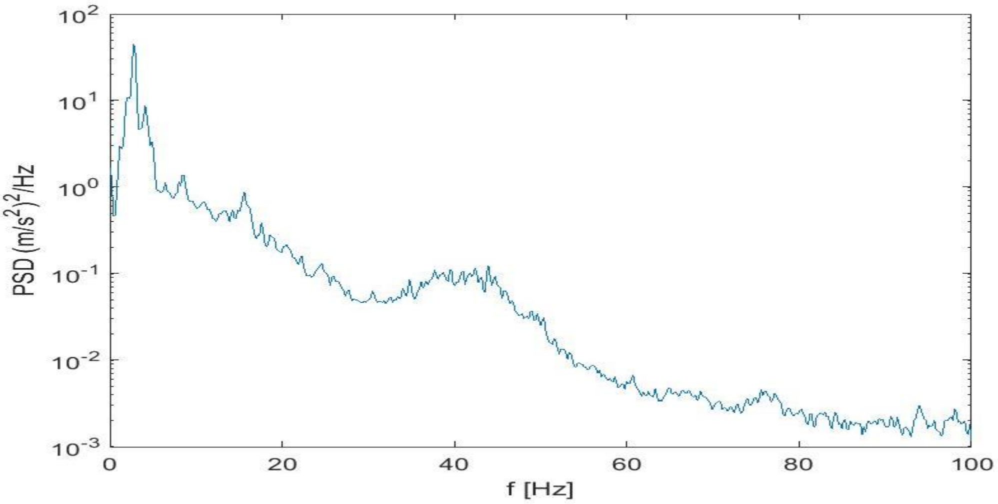

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Fig. 4. Resultant spectrum of accelerations of mechanical vibration appearing during the travel of the baler-wrapper in the field at a speed of 10 km/h
Rys. 4. Wypadkowe widmo przyspieszeń drgań mechanicznych występujących podczas przejazdu prasoowijarki po polu z prędkością 10 km/h

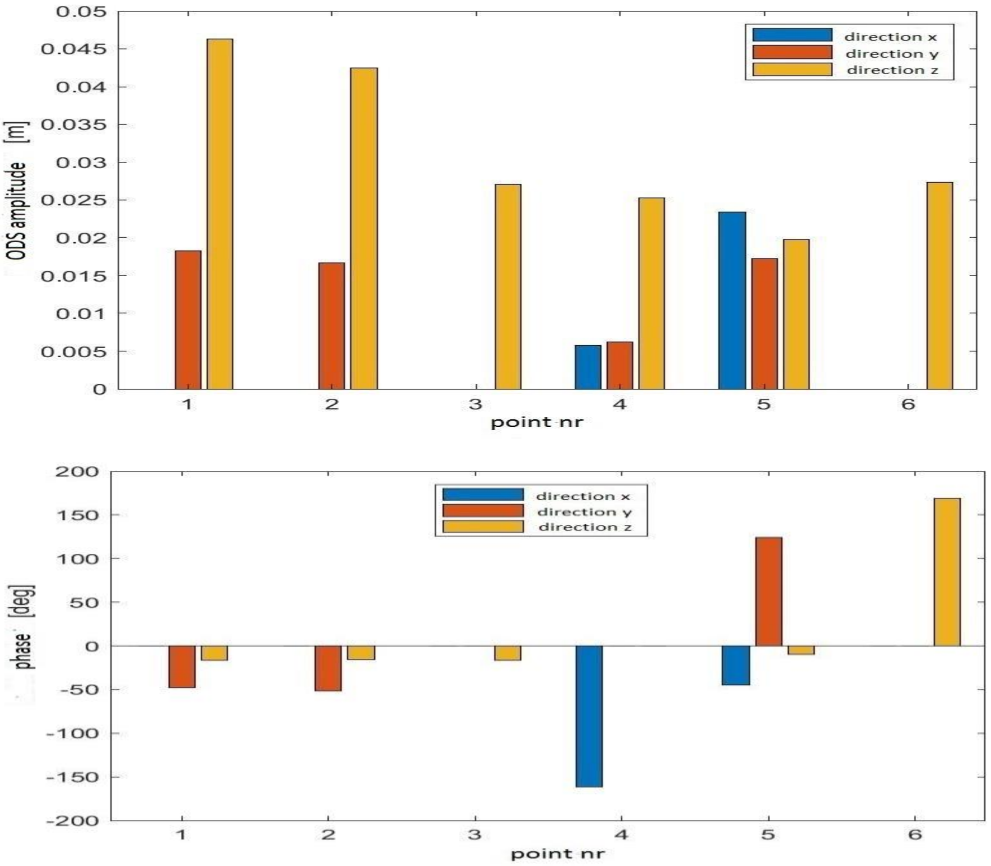

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Fig. 5. The amplitude and phase (ODS) of the 2,74 Hz operating condition occurring when a baler-wrapper is driven on a field
Rys. 5. Amplituda i faza postaci eksploatacyjnej (ODS) 2,74 Hz podczas przejazdu prasoowijarki po polu

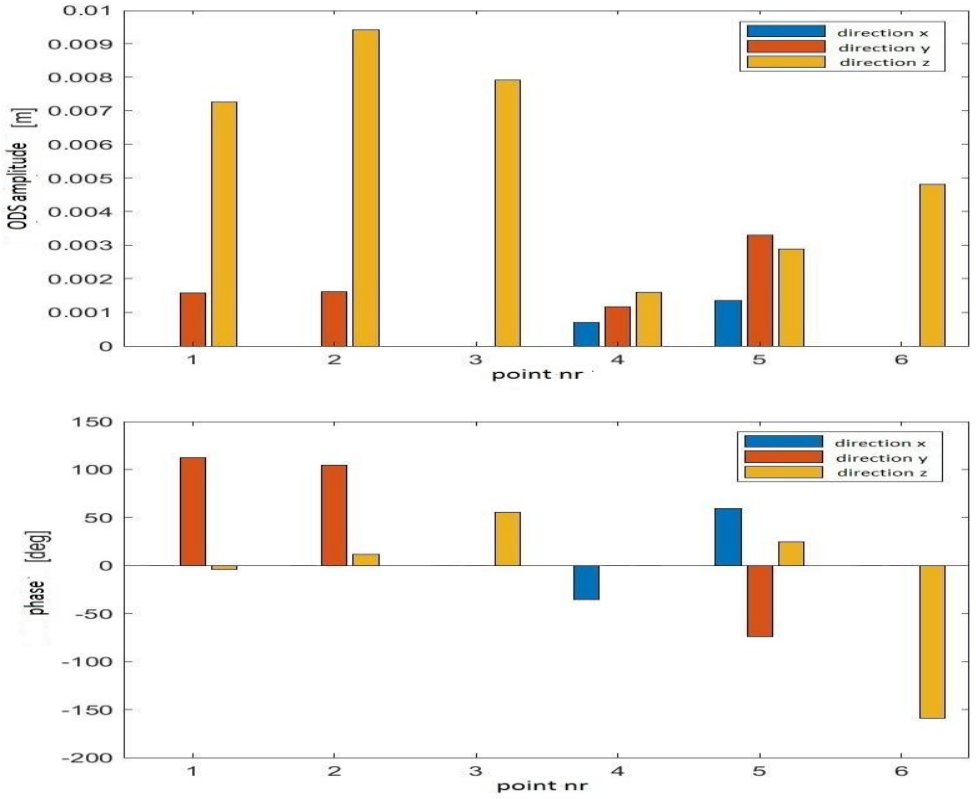

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Fig. 6. The amplitude and phase (ODS) of the 4,10 Hz operating condition occurring when a baler-wrapper is driven on a field
Rys. 6. Amplituda i faza postaci eksploatacyjnej (ODS) 4,10 Hz podczas przejazdu prasoowijarki po polu

Low-frequency components (approx. 2.7 Hz) determine the state of vibrations of the baler-wrapper. The greatest amplitudes of displacements during the travel of the baler-wrapper in the field occur at points located on the left and right side of rear wrapping beam in vertical directions. These displacements are in phase.

4. Conclusions

The representation of the dynamical behavior of structures by the modal parameters and operational deflection shapes allows for a better physical interpretation. It is a way to get condensed experimental data suitable for mechanical design purposes.

The method of testing operational deflection shapes of a structure, as opposed to the method of modal model identification, gives the results depending not only on the object properties but also on the method of loading the structure during the measurement.

Low-frequency components (approx. 2.7 Hz) in vertical direction determine the state of vibrations of the baler-wrapper. The results of experimental research can be used to verify and improve analytical models (FEM) of agricultural machines.

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