CONTROL EFFECTIVENESS COMPARISON OF WHEEL LOADER'S LONGITUDINAL VIBRATIONS BY MEANS OF PASSIVE AND ACTIVE VIBRATION STABILISERS

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Abstract

When wheel loaders move with velocity higher than 10 km/h, they are subject to intense longitudinal vibrations. These vibrations are experienced particularly unpleasantly by an operator. Passive vibration stabilisers are used for controlling their intensity. They make outrigger's support more flexible so that outrigger's vibrations counteract the wheel loader's longitudinal oscillations when moving. Unfortunately, the effectiveness of passive stabilisers is insufficient and at present more effective alternative means are sought. One of the solutions for angular longitudinal vibrations includes the replacement of passive stabilisers with active stabilisers. Nevertheless, the possibility of active stabilisers' use in minimising longitudinal vibrations requires a scientific assessment. Hence, this paper, meeting the needs, includes calculations, which allow for comparing the effectiveness of active and passive stabilisers in minimising the vibrations of a typical wheel loader. Active stabilisers, which are the subject of the study, worked based on state controller, whose synthesis was carried out on the basis of optimal control linear theory with square quality indicator.

The calculations were conducted on linear models under stochastic road excitations. Stochastic excitations were defined by means of power spectral density of road's unevenness. Active stabiliser's effectiveness was analysed, both for a loader moving with an empty bucket, and for a loader transporting a 3-tonne load. In order to facilitate the analysis of the results obtained, vibrations minimising quality indicator was introduced, defined as a root of a vehicle longitudinal deflections power.

Keywords: *wheel loader, longitudinal oscillations, active stabiliser, passive stabiliser, state controller synthesis*

1. Introduction

Contemporary wheel loaders, which are required to move relatively fast reaching the velocity of 60 km/h, are often subject to intense longitudinal vibrations. The background of this phenomenon and the influence of loaders' selected construction parameters on this phenomenon were analysed, inter alia, in the following literature: [1, 2].

Passive vibration stabilisers are currently used for controlling longitudinal vibrations in wheel loaders. The descriptions of their operation and discussions concerning their significant constructional solutions may be found in numerous scientific publications [3, 4].

The correction of wheel loaders' movement dynamics by means of passive vibration stabilisers is not satisfactory. As far as we know, significant improvement of passive stabilisers effectiveness, as a result of their optimisation, is quite unlike [5].

Therefore, structurally new solutions are currently sought, which could replace longitudinal vibration control means used, at the same time improving operators' vibration comfort. Contributing to these actions, the author endeavours to present his own assessment of effectiveness improvement possibility as regards loaders' longitudinal vibrations control by means of replacing their passive stabilisers with active stabilisers equipped with linear state controllers.

2. The structure of mathematical model of a wheel loader with passive stabiliser and of mathematical model of a wheel loader with active stabiliser

The performance of the task undertaken began with accepting a wheel loader's physical model with passive stabiliser and a wheel loader's physical model with active stabiliser. A graphic presentation of both models is provided in Figure 1. When elaborating the models, the following significant simplifications were introduced:

- wheels' point contact with the ground,
- lack of deformations within loaders' bearing structure,
- continuous wheels-ground contact,
- lack of influence of alternate movement resistances on linear velocity of loaders' movement,
- lack of loaders' snaking movement,
- lack of loaders' lateral oscillation etc.

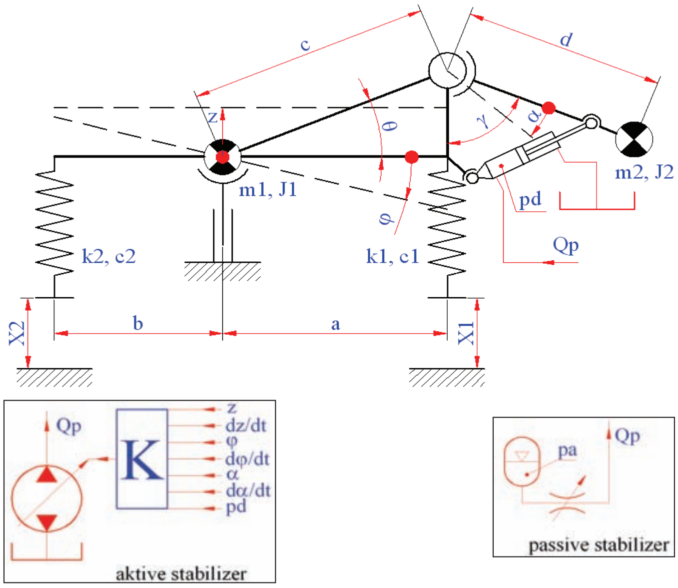

Fig. 1. Accepted wheel loader's physical model along with analysed stabilisers' diagrams

Additionally, for the active vibration stabiliser model, it has been assumed that its controlled hydraulic pump is ideal, which means that it sets to demanded capacity without time delay. It is obvious that real hydraulic pumps have dynamics similar to inertial object. The omission of pump inertion within the model could lead to overestimation of active stabiliser's performance.

Based on the physical models accepted, mathematical models equivalent to them were constructed in the form of differential equations. The wheel loader's mathematical model with the active stabiliser does not include, at this stage, the operation of a state controller. The following was accepted as state variables for both models: vertical displacement and centre of gravity velocity of a loader's frame (z, dz/dt), longitudinal angular deflection and longitudinal angular velocity of a loader's frame (φ, dφ/dt), angular displacement and angular velocity of a loader's outrigger in relation to a loader's frame (α, dα/dt) and pressure in lower chambers of hydraulic cylinders supporting the outrigger (p_d). With a loader's model with passive vibration stabilizer, the vector of state variables has been supplemented with liquid pressure in hydraulic battery (p_a). Constructed nonlinear mathematical models were subject to linearisation surrounded by balance points by means of development into Taylor series and rejection of nonlinear development expressions. As a subsequent step, mathematical model parameters were replaced with numerical values, appropriate for a wheel loader made in Poland, type Ł220. Significant model parameters accepted are provided in Table 1.

Tab. 1. The values of significant parameters of Ł220 type loader used in mathematical models

k_1	1 400 000 N/m (3.7 bar)	J_1	21 231 kg m^2
k_2	1 000 000 N/m (1.5 bar)	J_2	1 289 kg m^2
c_1	15 932 Ns/m	a	2.08 m
c_2	11 380 Ns/m	b	0.892 m
m_1	8 725 kg	c	1.479 m
m_2	2 328 kg	d	1.896 m
θ	32.4°	γ	52.9°
Sd (summarised area of hydraulic cylinder pistons supporting the outrigger)	0.02454 m^2	E (volume resilient modulus of working liquid)	1200 MPa
f_t (substitute equal friction coefficient in hydraulic cylinders)	12160 N*s/m	RQ_{kr} (R – hydraulic resistance. Q_{kr} – critical flow)	0.2482*10^9 [N*s/m^5]

For the purposes of ordering the notation and facilitating the calculations, linearised models have been expressed in a matrix form as below:

$$\begin{aligned}\dot{x} &= A\cdot x + B\cdot u \\ y &= C\cdot x\end{aligned}, \tag{1}$$

hence: A- state matrix, B – input matrix, C – output matrix which is a unit matrix, x – state variables vector, u – control and enforcement vector.

Being in possession of a loader's mathematical description, the synthesis of active vibration stabiliser's controller commenced. The calculations were carried out on the basis of optimal control linear theory with square quality indicator. According to this theory, control quality indicator was accepted in the following form:

$$J = \int_0^\infty \left(x^T(t)\cdot Q\cdot x(t) + u^T(t)\cdot R\cdot u(t)\right)dt\,, \tag{2}$$

hence: Q and R weight matrices. Matrix Q is symmetrical and semi-positively expressed, and matrix R is symmetrical and positively expressed. Matrices Q, R were used for defining the compromise between control effects (control of angular amplitudes of longitudinal vibrations), and regulation costs (necessity of supplying the system with power).

The selection of high values of Q matrix elements in relation to R matrix elements implies that we are concentrated on controlling the vibrations rather than on controlling the power, which will have to be supplied to the system. Usually, literature recommends accepting Q matrix value equalling the reciprocal (or reciprocal of squares) maximum permissible values of the state variables. Analogously, guidelines concern R matrix values as well; whereby, maximum values of state variables are obviously replaced here with control maximum values. Defining the value of Q matrix lack of limitations or little restrictive limitations were accepted in relation to all state variables, excluding angle amplitude of loader's longitudinal inclination. These steps were undertaken since the purpose of the study was not to build a real active stabiliser of longitudinal vibrations but to probe prospective possibilities of the said stabiliser. Such a solution was also supported by a common opinion that minimising a larger number of state variables than only one provides limited effects in most cases. Ultimately, the maximum value of control (maximum pump capacity of active stabiliser), which was accepted, equals Q_p=100 l/min, maximum permissible value of loader's longitudinal inclination angle φ_{max}=2° and α_{max}=20° and p_{dmax}=20MPa. The above establishments, upon using the units from SI system, allowed to express Q and R matrices in the below form

$$Q=\begin{bmatrix}0&0&0&0&0&0&0\\0&0&0&0&0&0&0\\0&0&28.65&0&0&0&0\\0&0&0&0&0&0&0\\0&0&0&0&2.86&0&0\\0&0&0&0&0&0&0\\0&0&0&0&0&0&5\cdot10^{-8}\end{bmatrix}\qquad R=[600]. \tag{3}$$

Upon defining weight matrix, applying calculation algorithms implemented in Matlab package, control matrix K was indicated as below.

$$K=\left[2.1\cdot10^{-4}\quad -1.8\cdot10^{-7}\quad -1.6\cdot10^{-5}\quad -6.7\cdot10^{-4}\quad -2.2\cdot10^{-4}\quad 0.014\quad 9.1\cdot10^{-6}\right]^{T}. \tag{4}$$

When analysing the obtained elements of control matrix K and possible values of state variables (tab. 2), it has been noted that pressure changes p_d have a determining impact on the operation of active vibration stabiliser. The influence of other state variables proved to be negligible. It means that in the event of proving the effectiveness of the device under analysis, there exists a prospective possibility of its construction simplification, at the same time controllingits manufacturing costs.

Upon concluding the synthesis of state controller, it was added to a mathematical model of a loader with active stabiliser.

In order to facilitate the assessment of properties of the devices under analysis based on their mathematical linear models in the form of state equations, models in the form of operative transmittance were designated as well. Two transmittances specifying dynamics of longitudinal deflections depending on excitations x_1 and x_2 were particularly interesting for each vehicle.

Tab. 2. Estimate assessment of influence on control signal of particular state variables

State variable x_i	Estimate typical value of state variable amplitude $\hat{x}_i$	Estimate maximum value of a component connected with a given state variable in the control signal $x_i \cdot K[i]$
z_1	0.1 [m]	$2.16\cdot10^{-5}$
dz_1/dt	2 [m/s]	$3.62\cdot10^{-7}$
φ	0.1 [rad]	$1.65\cdot10^{-6}$
$d\varphi/dt$	2 [rad/s]	$13.42\cdot10^{-4}$
α	0.1 [rad]	$2.20\cdot10^{-5}$
$d\alpha/dt$	2 [rad/s]	0.0288
p_d	500000 [Pa]	4.56

Assuming that excitations on the front axis of a vehicle are connected with excitations on vehicle's rear axis according to the relationships (5) and (6) upon eliminating enforcement x_2, the said transmittances were replaced with single transmittances:

$$x_1(t) = x_2(t + \Delta t)\,, \tag{5}$$

$$\Delta t = \frac{a+b}{V}\,. \tag{6}$$

Then, they were replaced with relevant spectral transmittances obtaining loaders' longitudinal dynamics descriptions in the following form:

$$T(j\omega) = \frac{\varphi(j\omega)}{x_1(j\omega)}\,. \tag{7}$$

Modelling process was completed with defining stochastic excitations by means of power spectral density of road's unevenness and defining power spectral density of vehicles' longitudinal deflections in accordance with the below relationship:

$$G_w(\omega) = \left|T\left(j\omega\right)\right|^2 \cdot G_d(\omega) = \left|T\left(j\omega\right)\right|^2 \cdot G_d\left(\omega_0\right) \cdot \left(\frac{\omega_0}{\omega}\right)^w\,, \tag{8}$$

hence: $G_d(\omega)$ – spectral density of road's unevenness with spatial frequency ω, w – sinuosity coefficient (accepted w=2), ω_0=spatial frequency of reference (accepted ω_0=1 m^{-1}), $G_d(\omega_0)$ – road unevenness indicator – power spectral density for spatial frequency of reference (accepted $G_d(\omega_0)$=0,000155 m^3).

3. Effectiveness assessment of active vibration stabiliser based on its linear model

In order to assess the effectiveness of active vibration stabiliser, applying linear models, power spectral density of loader's longitudinal deflections was designated: without a stabiliser, with passive stabiliser and with active stabiliser. Figure 2 presents differences between values of power spectral density of longitudinal deflections obtained for a loader with active stabiliser and loader with passive stabiliser.

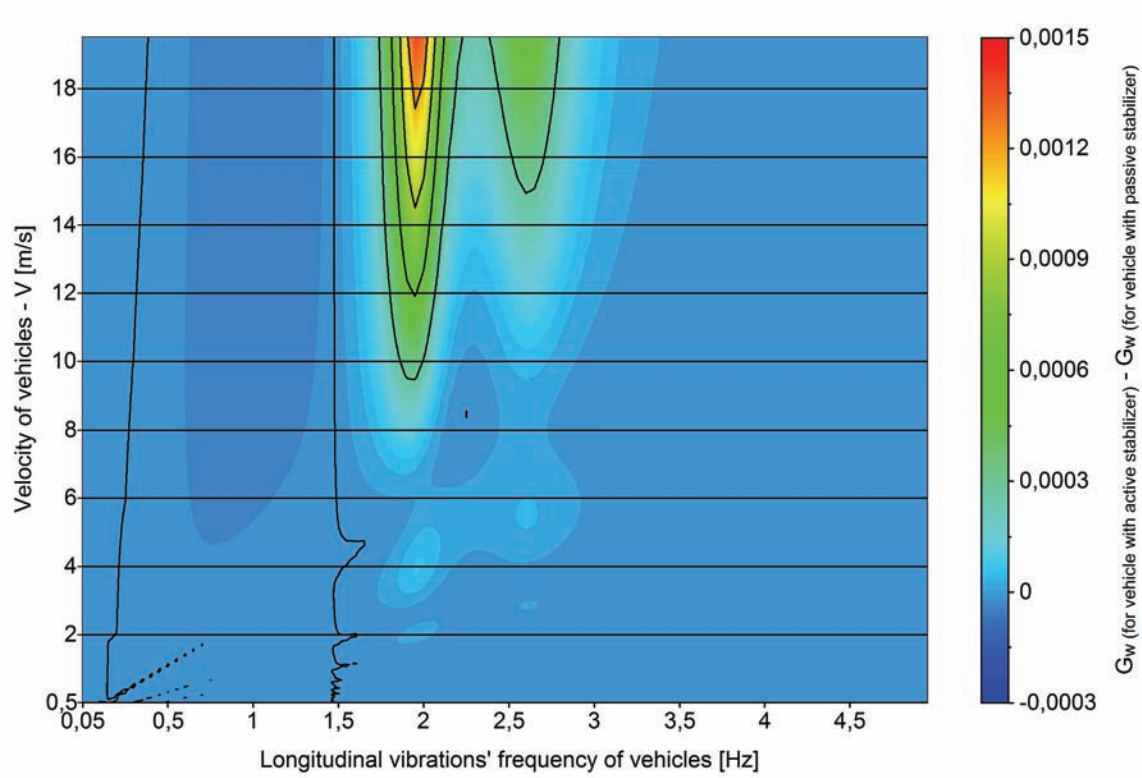

Fig. 2. Differences between the values of power spectral density of loader's longitudinal deflections with active and passive stabiliser

The results provided in Figure 2 demonstrate that harmonics with resonance frequencies minimise a passive stabiliser better, whereas harmonics with low frequency in the range from 0,5 to 1,5 Hz minimise an active stabilizer better. Nevertheless, the differences between stabilisers' efficiency at resonance frequencies are higher.

In order to compare the effectiveness of stabilisers more measurably, „P" indicator was introduced – root of vehicle's longitudinal deflections power. It may be interpreted in a similar way to a signal effective value. Its definition relationship is presented below:

$$P = \sqrt{\int_{0,8}^{4} G(f)df} \quad . \tag{9}$$

Signal power is specified here for vehicle's longitudinal deflections frequency in the range from 0,8 to 4 Hz. This limitation of integration range was supported by two facts. Firstly, this is frequency range of horizontal vibrations, to which a human is most sensitive. Secondly, amplitudes of all significant harmonic longitudinal vibrations of a typical wheel loader are within this range.

„P" indicator's value for a loader, both with passive and active stabilizer, is presented in Figure 3. They prove unequivocally better performance of passive stabiliser in relation to active stabiliser. Yet, the differences demonstrated are to be regarded as minor.

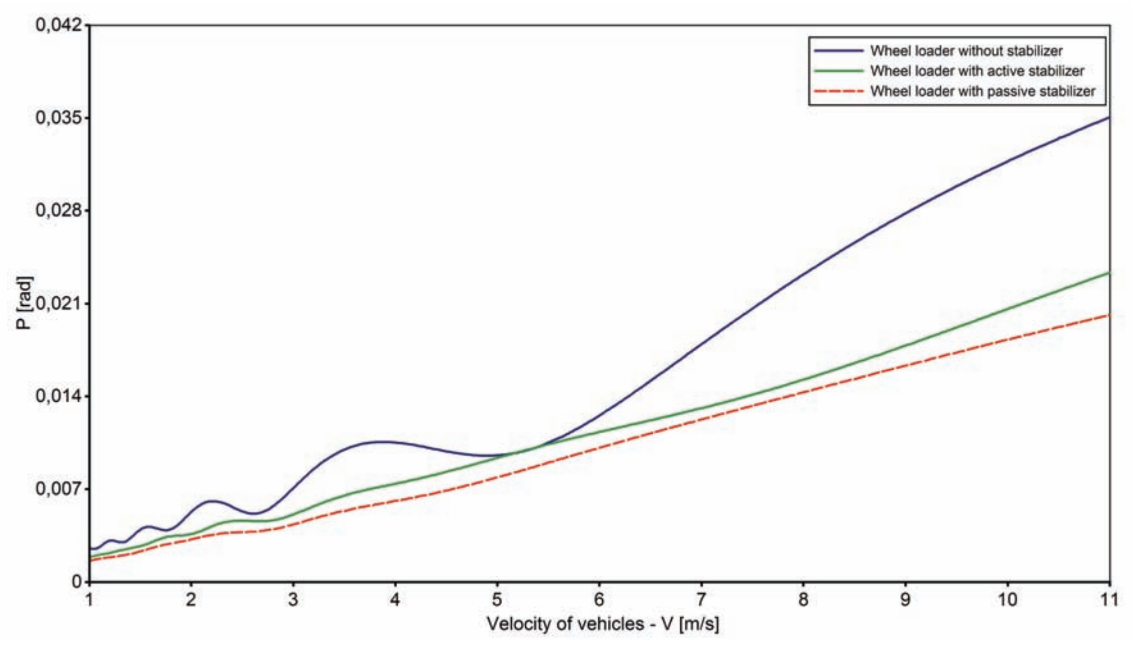

Fig. 3. „P" indicator values for loaders moving without load with various vibration stabilisers

The results discussed so far concern loaders moving without load with an empty bucket. In order to verify the influence of load on stabilisers' performance, additional calculations were conducted. The effectiveness of both stabilisers was compared in the conditions when loaders moved with a 3-tonne load in the bucket.

New control K matrix was designated for the purpose of these calculations, with the same weight matrices.

$$K = \begin{bmatrix} -433 & -11{,}9 & 28{,}9 & 52{,}04 & -3{,}66 & 0{,}0347 & 5{,}77 \cdot 10^{-6} \end{bmatrix}^T . \tag{4}$$

The increase of outrigger mass caused by load led to considerable changes within control matrices. Control signal in new conditions of active stabiliser operation, in practice to a comparable extent, depended on value changes of most state variables. In Figure 4, the difference between the power spectral density of longitudinal deflections is presented, obtained for a loaded loader with active stabiliser and analogous values obtained for a loaded loader with passive stabiliser. The results obtained unequivocally indicate higher effectiveness of active stabiliser compared to passive stabiliser. Active stabiliser minimises harmonic components with frequency above 2,2 Hz better and in the range from 0,5 to 1 Hz. Minimally worse performance is obtained in the range from 1 to 2,2 [Hz]. Considerably higher effectiveness of active stabiliser in relation to passive stabiliser is not confirmed by simulations on nonlinear models. Direct diagnosed cause of faulty determining of passive stabiliser was modelling of liquid flow resistances through its valve. These resistances were defined as linearly dependent on flow volume. However; it is only true for cases of laminar flows.

As simulations show, this flow was dominating in the case when a loader's bucket was empty. When a loader's bucket was loaded with 3000 kg, turbulent flow through stabiliser's valve became a dominating liquid flow. With turbulent flow, flow resistances are proportional to efficiency square of flowing liquid. Unfortunately, despite the trials, the influence of linearly changeable choking on minimising the vibrations by a stabiliser was not included in a linearised model with satisfactory precision. In real conditions, a passive stabiliser will be more effective than demonstrated in the results of simulation. In addition, if we assemble a system in a passive stabilizer, which will increase choking of flowing liquid along with mass increase of load transported, it will turn out that passive stabiliser's performance is not worse than active stabiliser's performance. It is noticeable in Figure 5, where next to diagrams analogous to diagrams in Figure 3, a diagram for a loader with passive stabiliser is also presented, in which choking is adapted to the mass of load transported.

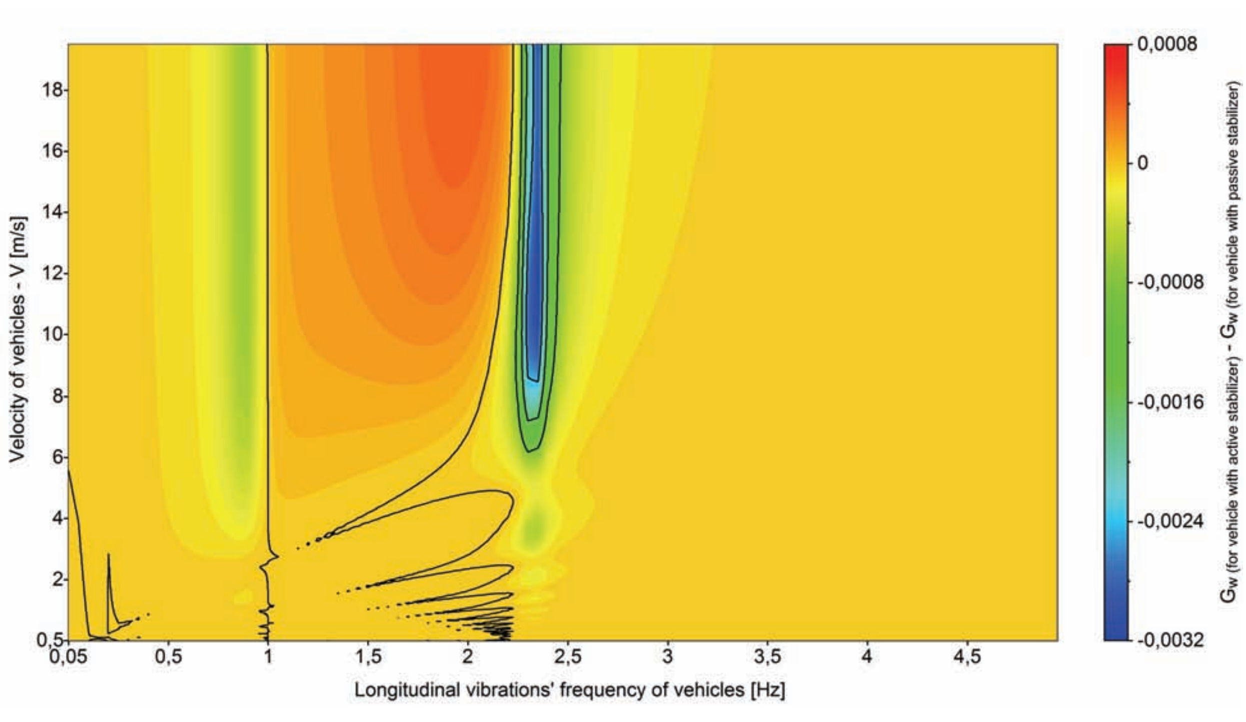

Fig. 4. Differences between the values of power spectral density of longitudinal deflections designated for a loader with active stabiliser and analogous values designated for a loader with passive stabiliser. The results concern vehicles moving with a 3-tonne load

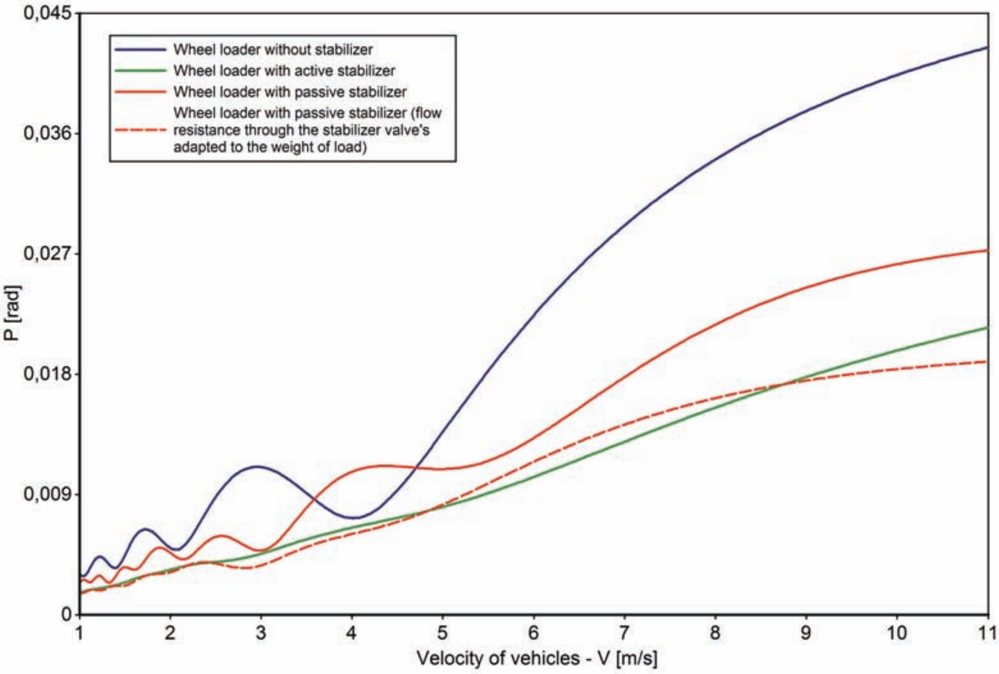

Fig. 5. „P" indicator values for wheel loaders with load of 3 tonnes and equipped with various vibration stabilisers

4. Summary

Based on the calculations and simulations carried out, it may be stated that active stabilisers with analysed control system do not demonstrate considerably better properties in relation to passive stabilisers, which are currently used. Therefore, there are no rational prerequisites for their use in order to reduce longitudinal vibrations of wheel loaders with a bucket. This statement is even more justified since the costs of constructing and operating active stabilisers are considerably higher compared to the construction and operation costs of passive stabilisers.

The results obtained also suggest that the problem of excessive vibrations in wheel loaders with a bucket may be unresolved by means of a minor interference in the structure of current wheel loaders.

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