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OPTIMAL STABILIZATION OF EQUIPMENT IN UNSPRUNG MOBILE MACHINES

Vibration intensity in mobile machines depends on the road roughness profile, ride velocity and dissipative properties of machine components. To reduce vibrations of a mobile machine with a boom equipment one of the available passive methods, utilizing a hydropnematic system for boom support to improve flexibility, the system incorporating throttling valves. Energy dissipation in a hydropneumatic system controls the decay of vibrations of the machine body and equipment. In the range of large velocities, passive methods prove inadequate. When ride velocity is to be increased, at the same time the required safety features and stabilization of the position of machine equipment are to be provided, further dynamic analyses are fully merited to identify processes taking place in the driving system. The final result should be the synthesis of the LQR control system to modulate the loading characteristics of the motor and to control the flow in a hydraulic boom-support system.

1. Introduction

Extensive research is undertaken to explore new solutions to be incorporated in heavy machine designs to improve their operational parameters. New, effective methods are sought controlling vibrations of traction, wheeled machines, particularly machines with a boom equipment that are widely employed in materials handling and cargo lifting. Those machines have a characteristic operating cycle and their rides on the site have to be frequent [1, 4].

In unsprung mobile machines, dissipation of vibration energy can be achieved only due to elastic-damping properties of the wheel tires. On account of their small deformations and slow deformation rates, reduction of

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excited vibrations of the machine body and equipment is minor, which is particularly dangerous in the case of a resonance that occurs when frequency of base-excited vibrations due to ride velocity and road roughness coincides with the natural frequency of the machine's vibrations. When the driving system incorporates a controller modulating the motor's loading characteristic, dynamic loading of the machine during the ride should not be affected at velocities other than those at which resonance occurs [5].

In transport machines, such as loaders, the effects of road unevenness on the behaviour of the equipment and the payload are of major importance. The position of the transported load has to be stabilized, and that is typically achieved by stabilizing the absolute position of the boom. Incorporation of gas-loaded accumulators in hydraulic boom support systems and throttling control of viscous damping vastly improve controllability of the system.

The dynamic analysis utilizes a 2D model of a two-axial wheeled machine, together with a loading equipment, as shown in Fig. 1.

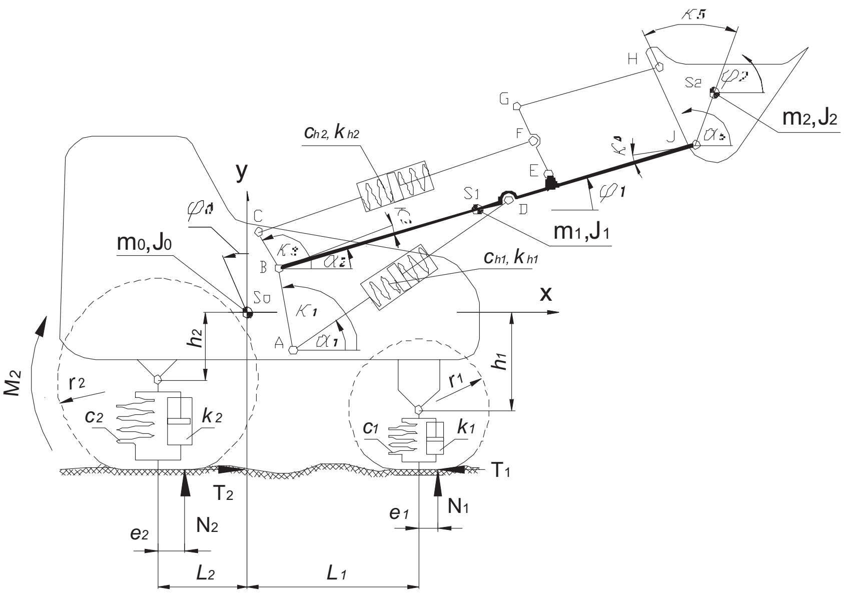

Fig. 1. Physical model of a two-axial wheeled machine

In the proposed controlled system, the values of two control variables shall be adjusted in the feedback system so as to ensure the optimal implementation of the quality criterion. The value of the driving torque transmitted onto the riding wheels is adjusted at the stabilized average ride velocity (active vibration control) and so is the cross-section area of a slit in a proportional value in the hydraulic line of a cylinder supporting the boom (semiactive control). The purpose is to minimize vibrations of the equipment with the payload and oscillations due to longitudinal rolling of the machine body.

2. Heavy machines during the ride – as dynamic systems

In the loader model shown in Fig. 1, the machine frame is rigidly connected to the wheel axles. Real axle wheels are driven with the torque M_2. Assuming that wheels roll without slipping, the number of the system's dofs depends on flexibility of fluid cells in the equipment, i.e. the cylinders in the boom and bucket driving systems. In a conventional hydraulic control system, i.e. with no additional flexibility provided by the gad-loaded accumulators, the rotational angles of the machine frame $\varphi_0(t)$, of the boom $\varphi_1(t)$ and the bucket $\varphi_2(t)$ should be equal. The machine itself, tog with the working equipment, is thus treated as a rigid solid performing flat motion. When gas-loaded accumulators are incorporated in the boom support system, the number of the system's dofs goes up to four, whilst the relationship $\varphi_1(t) = \varphi_2(t)$ is still valid.

A model of a tired wheel and velocity characteristic of the compression-ignition engine, reduced to the driving axle, as well as differential equations governing the machine motions are discussed elsewhere [4, 5]. Underlying the differential equations of motion is the assumption that while the machines moves in the rough terrain, the system performs not only the progressive motion, but also slightly oscillates around the position of static equilibrium. Applying the matrix notation, we get the mathematical model in the form:

$$\mathbf{M} \cdot \frac{\mathrm{d}}{\mathrm{dt}}\dot{\mathbf{q}} = \mathbf{F_s} + \mathbf{c} \cdot \mathbf{q} + \mathbf{k} \cdot \dot{\mathbf{q}} + \mathbf{r} \cdot \mathbf{w}, \tag{1}$$

where: $\mathbf{M}$ – inertia matrix, $\mathbf{F}_s$ – vector of generalized static forces, $\mathbf{c}$ – stiffness matrix, $\mathbf{k}$ – damping matrix, $\mathbf{r}$ – matrix of dynamic flexibility of the driving system, $\mathbf{q}$ – vector of generalized coordinates, $\mathbf{w}$ – vector of kinematic excitations acting upon the wheels. Recall that $\dot{\mathbf{q}} = [\Omega, \dot{y}_0, \dot{\varphi}_0, \dot{\varphi}_1]^{\mathrm{T}}$ and $\mathbf{w} = [w_1, w_2, \dot{w}_1, \dot{w}_2]^{\mathrm{T}}$.

Taking into account the characteristic of the drive $M_N(\Omega)$ and variability of the mass moment of inertia of the driving system, reduced to the wheel axles, we can incorporate the following relationship into the system of equation (1):

$$M_2 = M_N(\Omega) - (m_0 + m_1 + m_2) r_{2D} \left(2\Omega \frac{dr_{2D}}{dt} + r_{2D} \frac{d\Omega}{dt} \right), \tag{2}$$

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where: Ω – angular velocity of the driving wheel's rotational speed with respect to the predetermined value, r_{2D} – dynamic radius of the driving wheel.

3. Model of a hydro-pneumatic boom support

Conventional boom support systems in loaders display minor flexibility. Elastic-dissipative properties of such systems can be modified by parallel connection of a gas-loaded accumulator in the section of a hydraulic line between the distributor and the oil-supplying pipe on the piston end of the cylinder, or on the side of two accumulators, in the manner shown in Fig. 2.

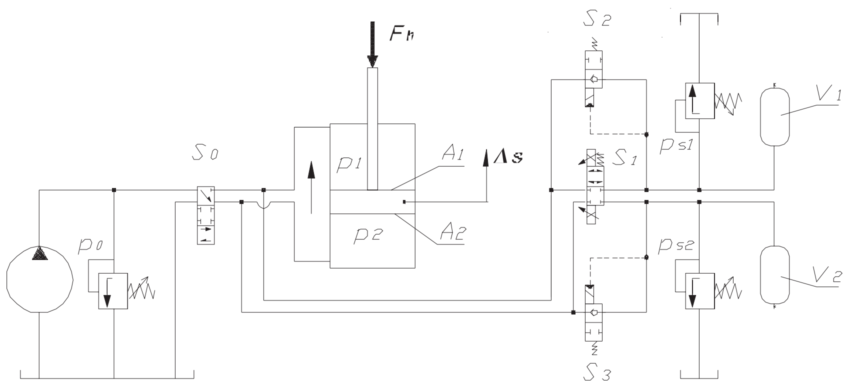

Fig. 2. Hydro-pneumatic model of the boom support system

Before they can be connected to the hydraulic system, the accumulators are initially charged with nitrogen gas up to the pressures p_{10}, p_{20} to fill in the nominal working volumes V_{10}, V_{20}, respectively. To ensure the smooth operation of the boom support system, the initial charging pressures have to be adjusted to the cylinder loading on the piston- and rod-end, associated with the bucket filling level and the configuration of the equipment during the ride.

Valves S_2, S_3 enable the charging of accumulators whilst in service. The valve S1 might be an on-off distributor, used by the operator to connect the accumulators with the volume in the piston and rod ends of the cylinder during the ride. Alternatively, it might be a proportional distributor enabling throttling control. Response of the hydraulic cylinder F_h depends on parameters of the hydro-pneumatic system and the variables of state. Accordingly, we get:

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$$F_h = -\kappa\left(p_{01}\frac{A_1^2}{V_{01}} + p_{02}\frac{A_2^2}{V_{02}}\right)\Delta s - \left[A_1\left(\frac{A_1}{\alpha_1 f_1}\right)^2 + A_2\left(\frac{A_2}{\alpha_2 f_2}\right)^2\right]\left(\frac{d}{dt}\Delta s\right)^2 sign\left(\frac{d}{dt}\Delta s\right), \tag{3}$$

where: Δs – piston displacement with respect to the cylinder, κ – adiabatic exponent for nitrogen, A_1, A_2 – surface areas on the piston and rod end, respectively, f_1, f_2 – surface areas of the throttling slits, α_1, α_2 – slit contraction factors.

To implement throttling control, the response F_h is represented in the form of a linear function (4). Values of equivalent stiffness and damping coefficients are derived by comparing (3) and (4) for the assumed damping control variable ζ :

$$F_h = -c_h \Delta s - k_h \zeta, \tag{4}$$

The change of the total slit cross-section area is implemented by the proportional valve S_1, in accordance with formula (5), basing on the value of ζ determined by the controller determined by the controller.

$$\zeta = \left(\frac{\frac{d}{dt}\Delta s}{f_1 + f_2}\right)^2 sign\left(\frac{d}{dt}\Delta s\right). \tag{5}$$

In the case of control strategies that cannot be physically implemented, for example when the signs of relative velocity $d\Delta s/dt$ and of the control variable ζ prove to be opposite, the control algorithm shall switch off the throttling control, thus generating the value $\zeta = 0$.

4. Heavy machines during the ride – as control plants

4.1. Control plant in the space of state variables

When we consider a machine as a control object, kinematic excitations become disturbances to be minimized by applying the feedback action, associated with the state variables.

Fig. 3 shows a block diagram of a control system with feedback, where: **A** – matrix of state, **B** – control matrix, **R** – matrix of disturbances, **C** – measurement matrix, **K** – desired matrix of gain coefficients for the designed controller, **z** – vector of state, **u** – vector of control inputs, **w** – vector of input excitations; **y** – output vector, **z** – vector of reference variables (in the case analyzed here we assume that $\mathbf{z}_s = 0$).

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On account of the clearly defined goal of control action- i.e. load stabilization, the measurement matrix **C** represents the vertical displacements of a pin bolt connecting the boom to the bucket.

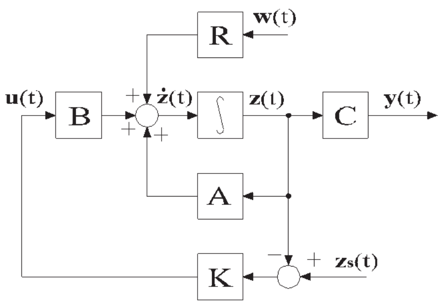

Fig. 3. Block diagram of a feedback system

The proposed vibration control system can be described in the space of state with the matrix equation [3]:

$$\frac{\mathrm{d}}{\mathrm{d}t}\mathbf{z} = \mathbf{Az} + \mathbf{Bu} + \mathbf{Rw}. \tag{6}$$

Zeroing of elements in the first column of the stiffness matrix in Eq. (6) eliminates the direct impacts of angular displacements of the driving wheel on system dynamics, which is accounted for in the state vector formula:

$$\mathbf{z} = [y_0, \varphi_0, \varphi_1, \Omega, \dot{y}_0, \dot{\varphi}_0, \dot{\varphi}_1]^{\mathbf{T}}, \tag{7}$$

Accordingly, the matrix of state becomes:

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ \mathbf{-M^{-1}c} & \mathbf{-M^{-1}k} \end{bmatrix}. \tag{8}$$

where: **c** – stiffness matrix 4x3, leaving aside the column of zeroes, **0** – zero matrix 3x4, **I** – identity matrix 3x3.

4.2. Controllability of a dynamic system

Controllability of a dynamic system governed by the equation of state (6) was investigated by analyzing the matrix:

$$\mathbf{S} = [\mathbf{B}, \mathbf{AB}, \mathbf{AAB}, ..., \mathbf{A^{n-1}B}],\ n = 8. \tag{9}$$

It was found out that control of a backhoe loader vibration is achieved either by modulation of the driving characteristics or by throttling the flow

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in the hydraulic line of the boom supporting cylinder. LQR controller is able to determine these two variables simultaneously.

The optimal feedback matrix $\mathbf{K}_{opt}$ is obtained by solving the problem of minimization of the quality functional [2, 3]:

$$J = \lim_{T\to\infty} \int_0^T [\mathbf{z}^{\mathrm{T}}\mathbf{Q}\mathbf{z} + \mathbf{u}^{\mathrm{T}}\mathbf{P}\mathbf{u}]dt, \tag{10}$$

where: $\mathbf{Q}$ – matrix of weights for state variables, $\mathbf{P}$ – matrix of weights for control actions.
Eq. (11) yields the vector of optimal controls which minimize the quality indicator (10):

$$\mathbf{u}_{\mathbf{opt}}(\mathrm{t}) = -\mathbf{P}^{-1}\mathbf{B}^{\mathrm{T}}\mathbf{K}_{\mathbf{opt}}\mathbf{z}(\mathrm{t}). \tag{11}$$

5. Conclusions

The influence of the ride velocity on traction properties of a mobile loader on wheels is shown in Figs 4, 5, 6, 7 in relation to the boom support type and operation of the LQR controller.

Continuous lines indicated as 1 represent machines with a hydraulic boom support system; broken lines 2 represent machines with a hydropneumatic system, continuous lines 3 represent machines complete with a hydropneumatic boom support system, with the LQR switched on.

Those characteristics were collected from computer simulations in Matlab-Simulink, in the conditions of predetermined kinematic excitations. The amplitude of roughness profile is taken to be 2 cm. It appears that vibration reduction and load stabilization during the loader ride is possible when the stiff boom – support system is replaced by a flexible one, at the same time the slit cross-section area in the valve S1 is precisely controlled.

No evidence was found to show that the method of boom support should affect the safety features, understood as the ratio of vertical amplitudes of dynamic forces to static force acting upon the axle. In order to finally verify that hypothesis, further research is merited to simulate various throttling conditions. At higher velocities of the ride, passive vibration control methods prove inadequate. In a limited range, certain improvement of traction properties of heavy machines is achieved through the application of a LQR controller modulating the loading characteristics of the engine and controlling the flow in a hydraulic boom – support system.

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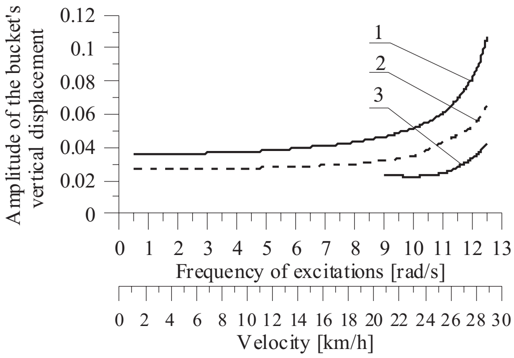

Fig. 4. Amplitude of the bucket's vertical displacement vs ride velocity

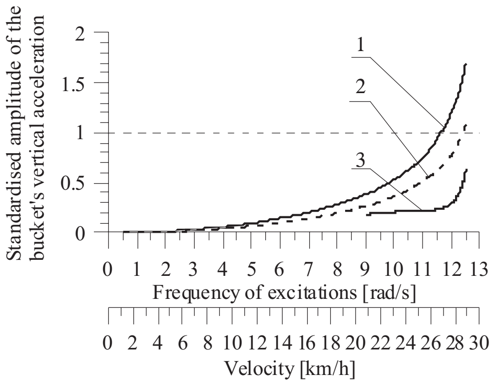

Fig. 5. Normalized amplitude of the bucket's vertical displacement vs ride velocity

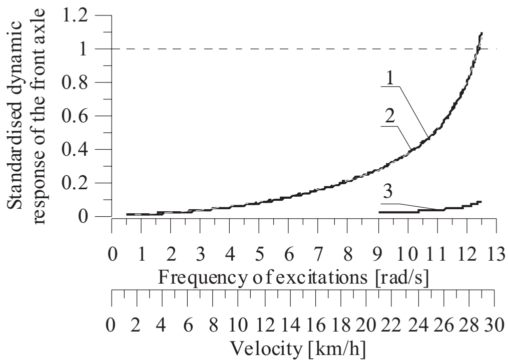

Fig. 6. Safety feature of the front axle vs ride velocity

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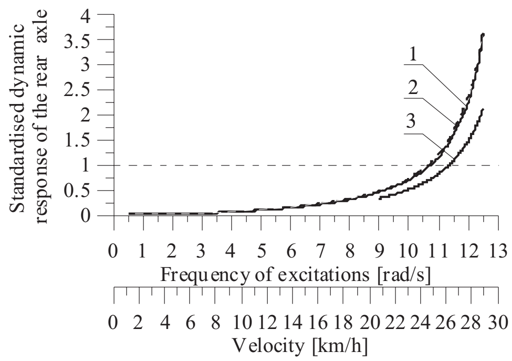

Fig. 7. Safety feature of the rear axle vs ride velocity

Dynamic loading of the front axle can be reduced nearly to zero, which vastly improves maneuverability. In the case of the rear axle, the safety improvement is rather minor.

Fig. 8 compares the oscillations of the driving torque in the engine loading characteristics (limited to the rear axle wheels) for on- and off-state of the LQR controller. Drive velocity is 23 km/h, which corresponds to kinematic excitations of 12 rad/s. It is worthwhile to mention that when the LQR controller is on during the ride, pulsation of rotational speed is reduced.

The limited range of vibration reduction when the LQR controller is employed is attributable chiefly to control potentials of applied actuators. One has to bear in mind that in the control strategy applied here the controller influences the loading characteristics of the engine and oscillations of the driving torque are transmitted onto the dynamic system in a minor degree only, as shown in [5]. Hence, strong modulation of the driving torque is required to ensure significant reduction of the machine frame vibrations, which might cause the engine to choke. Besides, the variability range of the slit cross-section is limited too and must contain only positive numbers.

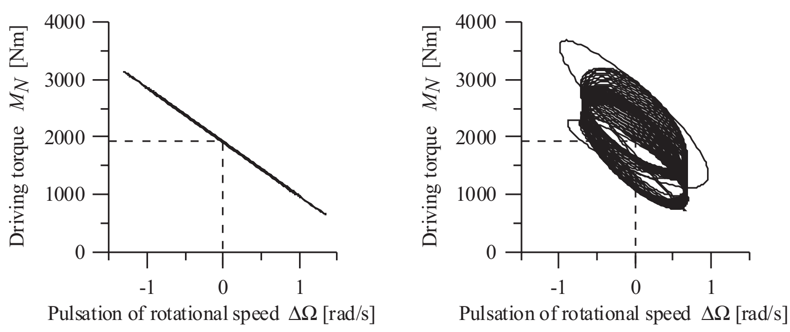

Fig. 8. Driving system performance (on- and off- state of LQR controller)

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The proposed simulation model can be easily modified to study the machine dynamics in the conditions of stationary stochastic excitations. Transition from the time to frequency domain enables us to represent the throttling control in the form of a linear function (4). Hence, we get $\mathbf{A}$ = const, $\mathbf{B}$ = const, $\mathbf{Q}$ = const, and finally, the optimal feedback matrix $\mathbf{K_{opt}}$ = const.

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Stabilizacja optymalna osprzętu nieresorowanych maszyn na podwoziach kołowych

S t r e s z c z e n i e

Intensywność drgań przejezdnej maszyny roboczej na podwoziu kołowym zależy od profilu nierówności drogi, prędkości jazdy oraz własności dyssypacyjnych zespołów maszyny. Jedną z metod pasywnych obniżenia wibroaktywności podczas jazdy maszyn z osprzętem wysięgnikowym może być zwiększenie podatności poprzez zastosowanie hydropneumatycznego układu podparcia wysięgnika zawierającego w swojej strukturze zawory dławiące. Rozpraszanie energii w układzie hydropneumatycznym ma zasadniczy wpływ na przebieg procesu zanikania drgań nadwozia i osprzętu maszyny. W szerokim zakresie prędkości jazdy metody pasywne redukcji drgań mogą okazać się niewystarczające. Dążenie do podwyższenia prędkości jazdy maszyn roboczych ciężkich przy zapewnieniu wymaganego bezpieczeństwa oraz stabilizacji położenia osprzętu wymaga dodatkowo analizy procesów dynamicznych zachodzących w układzie napędowym.

W wyniku możliwa jest synteza układu sterowania typu LQR – modulującego charakterystykę obciążeniową silnika i sterującego przepływ w układzie hydraulicznym podparcia wysięgnika.