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# SUPPRESSION OF CRANE VIBRATIONS WITH CONTROLLED BOOM SUPPORT SYSTEM

Application of a hydropneumatic boom support system with controlled stiffness and dissipation parameters in a crane in its ready-for-transport position allows for modification of dynamic properties of the crane and for vibration mode control during the ride. Adjusting the support system flexibility to disturbances being the result of uneven terrain may help to reduce the loading of the bearing structure and to increase speed still maintaining the required safety and comfort standards. That improves the functional quality of mobile machines. The results of this study may be used as the basis for evaluation of vibration control methods in mobile cranes in which the boom acts as a dynamic absorber.

## 1. Introduction

As wheeled cranes have large mass and dimensions which determine the inertia moment, the natural frequencies of their vibrations are rather low. Furthermore, the specific mass distribution and small speeds whilst in operation generate low-frequency disturbances. As the result, all dynamic reactions during the ride are enhanced.

Low-frequency vibrations worsen work safety and comfort, and have a negative effect on human organs. That refers mainly to vibrations 3-30 Hz. Vibrations of small intensity can also directly or indirectly impact on work effectiveness and quality.

As far as machine structure is concerned, dynamic forces resulting from vibrations cause overloading of machine elements and equipment, and in consequence lead to premature failures.

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Effective suppression of vibrations 0-10 Hz in mass of sprung vehicles and heavy machines caused by excitations of kinematic mode due to rough terrain cannot be achieved using passive vibro-control systems only, no matter whether their physical characteristics are linear or nonlinear [4].

A more effective technique involves flexibility control in elements connecting machine units. Because of the frequency range and loads carried by connecting elements, systems providing dynamic flexibility control can be implemented by the use of hydraulic or pneumatic systems.

## 2. Model of a hydropneumatic, flexible system

A hydropneumatic flexible system can be implemented to support the crane boom by the use of steel and gas-loaded springs connected in parallel. The steel spring carries the main static loads and provides the required natural frequency of the whole subsystem comprising the mass on the flexible support. The hydropneumatic spring facilitates control of natural frequency over a certain range. It allows for flow throttling and hence the control of damping factor. In the case of crane, the control of the damping factor in the boom support system is equivalent to control of resonance amplitudes.

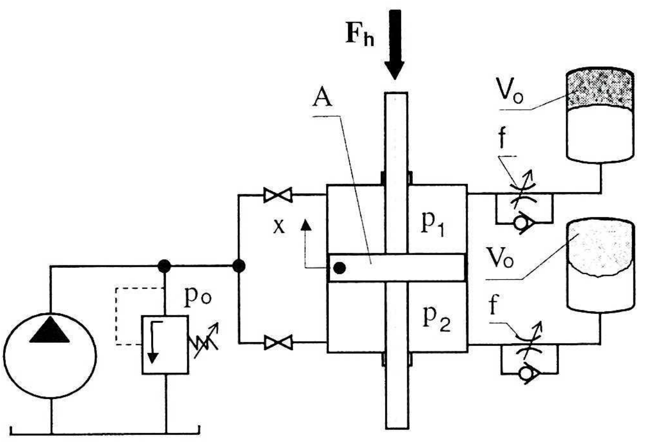


Fig. 1. Schematic diagram of a hydropneumatic system for damping of boom vibrations

The hydropneumatic spring shown schematically in Fig. 1 is implemented by the use of a double-action cylinder and gas-loaded accumulators connected to the cylinder via throttling valves controlling the flow of the fluid from the cylinder to the accumulator. The flow in the opposite direction is not throttled, owing to the presence of non-return valves. The supply system consisting of a hydraulic pump with an overflow valve is used for filling the cylinder and accumulators and for refilling system with the fluid.

Piston displacement x with respect to the cylinder is accompanied by reaction of the force $F_h$, and for small displacements this force is assumed to be given by the formula [7]

$$F_h = \frac{\xi\rho A^3}{2f^2}\left(\frac{dx}{dt}\right)^2 \cdot \mathrm{sign}\left(\frac{dx}{dt}\right) + \frac{2\kappa p_o A^2}{V_o}x, \tag{1}$$

where: $\xi$ — dimensionless coefficient of throttling losses, $\rho$ — density of the fluid, f — surface area of the throttling interstice, $p_o$ — accumulator loading pressure, A — surface area of the piston, $\kappa$ — power exponent depending on the type of thermodynamic process, $V_o$ — gas volume in the accumulator related to the loading pressure.

Taking into account the criterion of equal work of dissipating forces for linear and nonlinear systems, the effects of damping can be linearised in the conditions of steady state vibrations. Accordingly, the approximate value of the reaction force given by (1) can be written as:

$$\tilde{F}_h = \alpha_h \frac{dx}{dt} + c_h x, \tag{2}$$

while the linearised damping factor (which depends on the amplitude H and vibrations frequency $\omega$) and the coefficient of flexibility can be derived from the formulas:

$$\alpha_h = \frac{4}{3\pi}H\omega\frac{\xi\rho A^3}{f^2}, \qquad c_h = \frac{2\kappa p_o A^2}{V_o}. \tag{3}$$

As far as the system shown in Fig. 1 is concerned, the values of damping factor $\zeta_o$ can be controlled by precise setting of the surface area of the throttling interstice, in accordance with the formula

$$f = B\cdot\frac{1}{\sqrt{\omega\zeta_o}}, \quad \text{where:} \quad B = \sqrt{\frac{2\xi\rho A^3}{3\pi m_r \omega}}, \quad \zeta_o = \frac{\alpha_h}{2m_r\omega}, \tag{4}$$

while $m_r$ represents the mass of movable elements reduced to the mass of the cylinder piston.

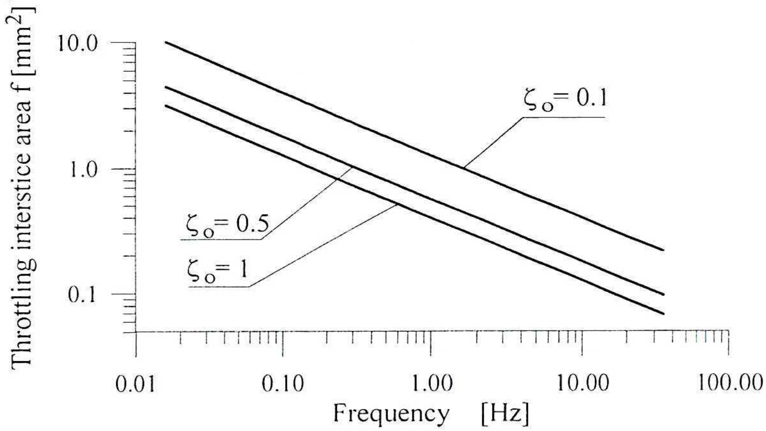


Fig. 2. Relationship between the throttling interstice area and vibrations frequency

Fig. 2 presents the relationship between the surface area of the throttling interstice f and vibrations frequency, providing the constant value of the damping factor $\zeta_0$ in the system having such parameters that $B = 10^{-4} m^2 s^{1/2}$.

## 3. Model of a wheeled crane with controlled, hydropneumatic boom support

### 3.1 Physical model

The solution providing the boom as a dynamic absorber was employed in cranes Omega 100 (made in USA, with the lifting capacity 14-28 t). The vibration control system belongs to the group of passive systems operating over a small range of crane speed (45-55 km/h). Flexible elements (pneumatic springs) were applied in booms lifting cylinders and so the boom vibrations were out of phase with induced vibrations.

Frequencies of natural vibrations of the systems are adjusted to the frequency of induced vibrations thanks to the vibration damper built in the booms lifting cylinder, so the boom together with a part of the damper becomes a dynamic absorber with respect to the crane body. In this way, the vibrations of the crane body can be damped. A serious drawback is that boom movements cannot be transmitted smoothly to the piston rod in ready-for-transport position, since the longitudinal axis of the cylinder is at a small angle to the lateral axis of the boom [5].

The support of the crane boom in ready-for-transport position (in most typical designs the support is in the front part), allows for modification of dynamic properties of the crane and for active control of vibration intensity during the ride provided the rigid support is replaced with a hydropneumatic support with controllable stiffness and dissipation parameters. This solution involves adjusting the flexibility of the support system to disturbances of the kinematic mode produced during the ride over rough terrain.

The model used for studies of dynamic loading of the crane during the ride is the plane model with 3 DOFs, as shown in Fig. 3. Such design is characteristic of heavy, multi-axle cranes, for example Hydros T 351, where two independent front axles are suspended on leaf springs.

The ends of rear axles are connected with lengthwise rocker arms normal to the given pair of vehicle axles, thus forming the so called Clarc axles. As a result, the signal acting upon each rear axle is transmitted onto the frame at one point only.

Dynamic flexibility of tired wheels and suspension elements were considered in the light of Voigt-Kelvin model. The relationship between damping factor and frequency of vibrations was considered. The mass of wheels, which is small in relation to the mass of the crane body, is neglected.

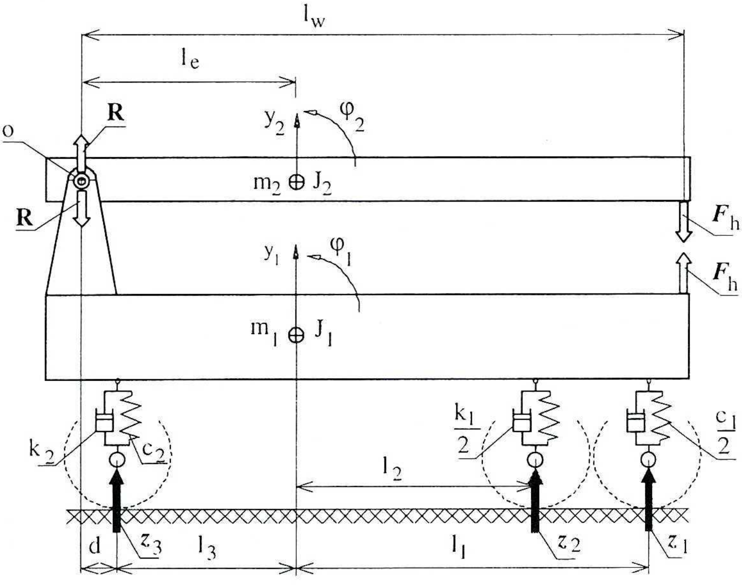


Fig. 3. Model of a wheeled crane during the ride as a three-input dynamic system with 3 DOFs

### 3.2. Mathematical model

The equations of motion of the crane components with respect to the static balance point for vertical and angular displacements of the crane body and for relative displacements of the boom can be written as:

$$M \cdot \ddot{q} + K \cdot \dot{q} + C \cdot q = K_z \cdot \dot{z} + C_z \cdot z + F, \tag{5}$$

where: $q = \begin{bmatrix} y_1(t) \\ \varphi_1(t) \\ u(t) \end{bmatrix}$, $z = \begin{bmatrix} z_1(t) \\ z_2(t) \\ z_3(t) \end{bmatrix}$, $F = \begin{bmatrix} 0 \\ 0 \\ F_h(t) \cdot l_w \end{bmatrix}$, $u(t) = [\varphi_2(t) - \varphi_1(t)] \cdot l_w$,

and: M — inertia matrix, K — matrix of damping within the system, C — flexibility matrix, $K_z$ — matrix of damping in wheel suspensions, $C_z$ — matrix of flexibility of chassis springing, F — control vector.

## 4. Evaluation of the system

Considering equation (5), vibration control during the ride involves adjusting the hydraulic reaction $F_h$ to instantaneous disturbances of the kinematic mode so as to meet the specified objectives. In order to improve work comfort, the accelerations of vertical vibrations at the point where the operator's seat is mounted should be minimised. One can also attempt to minimise dynamic forces in axle suspensions thus improving safety features. In most cases, however, those criteria are in conflict so it is necessary to seek some sort of a compromise.

When chassis flexibility is neglected, it is possible to improve the safety features and work comfort at the same time by minimising vertical vibrations and angular vibrations of the chassis. The elements of the inertia matrix in (5) beneath the main diagonal are zero, the values of those related to the second derivative of displacement u(t) are smaller than the elements related to the second derivative of displacements $y_1(t)$ and $\varphi_1(t)$ by two orders of magnitude while the control vector F directly impacts only on boom acceleration with respect to the chassis, so its impact on chassis vibration control will be restricted.

Spectral characteristics in relation to vibration acceleration at the point where the driver's seat is mounted were determined taking into account the boundary values of the damping factor $\zeta_o$ variability range. That was done to determine the effectiveness of vibration control methods involving the variations of dynamic flexibility of the boom support. The crane Hydros T 351 was considered in that simulation.

Disturbances of the kinematic mode resulting from rough terrain and acting at support points are taken to be stochastic processes. An assumption was made that roughness amplitudes decrease with the squared vibration frequency. It was also assumed that $K_z = \delta/\omega C_z$. The results corresponding to the crane ride at 40 km/h over rough terrain and taking into account damping in wheel tires characterised by the damping factor $\delta = 0{,}2$ are presented in Fig. 4.

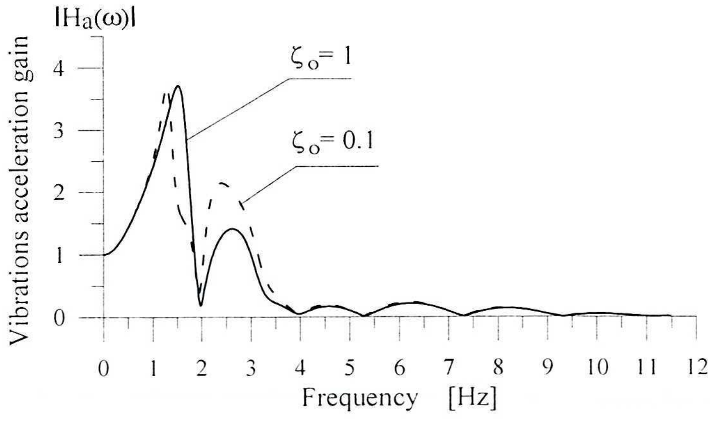


Fig. 4. The effects of damping on vertical vibrations acceleration gain at the point where the driver's seat is mounted

The plots are not shifted (apart from a very narrow frequency range 1,5–2 Hz), which means that the properties of the system can be improved using passive techniques only (choosing the optimal damping characterised by the damping coefficient $\zeta_o = 1$). Similar conclusions as to possibility of vibration control in multi-axle cranes taking into account frame flexibility can be found in [1].

Damping of transient state vibrations is another problem. These vibrations are generated when the crane has to overcome abrupt obstacles. The accelerations cause overloading of the order of several g, they are dangerous both for the operator and the machine itself.

Fig. 5 presents the results of numerical simulation of the process when the crane rides onto the step 0,1 m high at 20 km/h. The effects of energy dissipation in wheels are neglected. Comparison was made between the effectiveness of the passive system of boom support with the maximal damping factor ($\zeta_o = 1$) and the active system operating in accordance with the damping control strategy presented below:

$$\begin{cases} v_k \cdot \dfrac{du}{dt} \geq 0 \rightarrow \zeta_o = \zeta_{o\max} \\ v_k \cdot \dfrac{du}{dt} \langle 0 \rightarrow \zeta_o = \zeta_{o\min} \end{cases}, \tag{6}$$

where: $v_k$ — horizontal component of chassis vibration velocity at the point where the driver's seat is mounted, $\zeta_{o\max}$, $\zeta_{o\min}$ — the boundary values of damping factor.

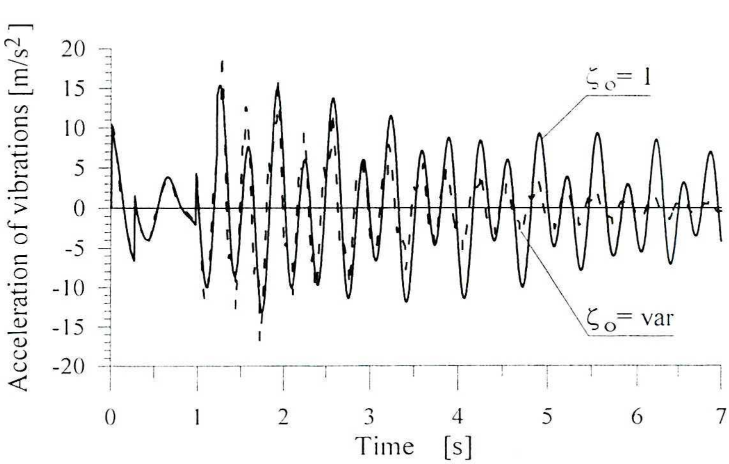


Fig. 5. The effects of damping control on acceleration of chassis vibrations at the point where the driver's seat is mounted

In heavy cranes of older design, such as Hydros T351, there are no absorbers in wheel suspensions; the boom is supported on a rigid steel support while high static and dynamic stiffness of wheel tires (high pressures) limits the dispersion of the energy of vibrations. When flexible boom support with controllable parameters is added, the traction features of the machine will be improved.

## 5. Conclusions

This study confirms that traction properties of heavy, multi-axle cranes can be vastly improved by the use of passive techniques and choosing the optimal stiffness and dissipation parameters in the system comprising the boom and chassis connection, rather than active systems providing for controllable dynamic flexibility.

Introduction of the flexible boom support with controlled parameters may effectively improve the dynamic properties in comparison to older designs since the wheel suspension systems would require no intervention. The hydropneumatic system shown schematically in Fig. 1 may be used in active vibro-control of operator's seats in mobile machines. The plots in Fig. 5 indicate that an appropriate control strategy allows for effective damping of operator's seat vibrations, particularly during transient states.

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**Redukcja drgań żurawia ze sterowanym układem podparcia wysięgnika**

Streszczenie

Zastosowanie hydropneumatycznego układu podpierającego wysięgnik żurawia w położeniu transportowym, o sterowanych parametrach sprężysto-dyssypacyjnych, umożliwia modyfikowanie własności dynamicznych żurawia i wpływanie na charakter drgań w czasie jazdy. Dostosowanie podatności układu podpierającego wysięgnik do wymuszeń kinematycznych związanych z pokonywaniem nierówności podłoża może wpływać na zmniejszenie obciążeń ustroju nośnego oraz umożliwiać zwiększenie prędkości jazdy w warunkach zapewnienia bezpieczeństwa i komfortu. W ten sposób można poprawić parametry trakcyjne pojazdu. Otrzymane wyniki stanowią podstawę oceny efektywności metody tłumienia drgań trakcyjnych żurawi, polegającej na wykorzystaniu wysięgnika, jako aktywnego eliminatora drgań nadwozia.