UDC 622.823

V.S. Loveikin, Dr. Sci. (Tech.), Professor,
Yu.V. Chovniuk, Cand. Sci. (Tech.), Assoc. Professor,
A.P. Liashko

National University of Life and Environmental Sciences of Ukraine, Kiev, Ukraine, e-mail: sachyk.anastasiy@mail.ru

THE CRANE'S VIBRATING SYSTEMS CONTROLLED BY MECHATRONIC DEVICES WITH MAGNETORHEOLOGICAL FLUID: THE NONLINEAR MATHEMATICAL MODEL OF BEHAVIOR AND OPTIMIZATION OF WORK REGIMES

В.С. Ловейкін, д-р техн. наук, проф.,
Ю.В. Човнюк, канд. техн. наук, доц.,
А.П. Ляшко

Національний університет біоресурсів і природокористування України, м.Київ, Україна, e-mail: sachyk.anastasiy@mail.ru

ВІБРАЦІЙНІ СИСТЕМИ КРАНІВ, КЕРОВАНІ МЕХАТРОННИМИ ПРИСРОЯМИ З МАГНІТОРЕОЛОГІЧНОЮ РІДИНОЮ: НЕЛІНІЙНА МАТЕМАТИЧНА МОДЕЛЬ ПОВЕДІНКИ ТА ОПТИМІЗАЦІЯ РОБОЧИХ РЕЖИМІВ

Purpose. To validate the nonlinear mathematical model of behavior and optimization of work regimes for the crane's vibrating systems controllable by mechatronics' devices with magnetorheological fluid.

Methodology. We used the methods of mechanics and mathematical physics.

Findings. The nonlinear mathematical model of behavior and optimization of work regimes for the study of the cranes' vibrating systems controllable by mechatronics' devices with a magnetorheological fluid are presented. The behavior of controllable viscosity fluid (CVF) under applied external (magnetic) field is presented as well. We have calculated the equivalent damping factor based on the principle of energy dissipated during one cycle of damper work under the external field of constant strength. When mass or stiffness is variable the equivalent damping factor can be set by adjusting the strength of external field to have crane's vibrating damping system purposely/continuously working in the critical or other chosen mode.

Originality. Use of fluid with magnetorheological effect in crane's system (in modes start/stop) can reduce significantly unwanted overload, vibrations and resonances arising from them. Also it allows us to optimize the trajectory of their motion depending not only upon standard characteristics (displacement, velocity, acceleration), but also on time derivative of motion law of third-order and fourth-order systems (jerk).

Practical value. This paper also presents cases of applying periodically changing strengths of an external synchronized with cycles of periodical crane's motion of the vibrating system to continuously optimal control the damping force within each cycle.

Keywords: *noise control, crane's optimal vibration control, smart materials, nonlinear mathematical model, behavior, optimization, work regime*

Introduction. Statement of the problem. It is known that magnetorheological (MR) and electrorheological (ER) fluids may be used for the crane's vibrating systems controllable by mechatronic devices.

Magnetorheological fluids are suspensions consisting of ferromagnetic particles in a low permeability base liquid, usually oil (in some cases water) with surfactants to prevent sedimentation. Electrorheological fluids are suspensions of electrostatically polarizable particles. Very fast reversible changes (usually in milliseconds) of rheological properties, especially apparent viscosity and elasticity are caused by the polarization induced in the suspended particles under applied external magnetic flux or electrical field. The particle chain formation and later changes from chains to columns are observed. This is known as the rheological effect. Thus MR or ER fluids behave as a Newtonian liquid (if base fluid has this property) without the presence of polarizing magnetic flux or electrical field and as a semi-solid when exposed to the field. This phenomenon is associated with changes of yield stress of the suspension. In effect, external field fluid strength changes according to applied external field. This fluid (or suspension) under an external field behaves as a Bingham semi-plastic until the shear stress becomes equal to the yield stress, which begins the onset of flow. The ER fluid behaves in the same manner as the MR fluid when as external electrical field is applied.

Content analysis of the recent investigations and articles. The known applications of MR fluids are in brake/clutch design [1], engine mounts [2,3] and in vibration dampers [4–6].

Early investigations of sound transmission loss (STL) in the stiffness controlled space between two barriers with ER fluid between them under DC and AC voltage [7, 8] shows, that due to increased mechanical coupling strength, the STL decreases. The STL was investigated for various kinds of ER suspensions in the frequency range from 100Hz up to 2kHz. Laboratory results showed that the normal stress developed in ER fluid has a significant influence on the magnitude of STL. The tangential (shear) stress had a negligible effect on the STL.

As an example, the (crane's) vibration of a two degree of freedom system with a MR damper is used to illustrate the separation of the vibrating excitation source from the system to reduce the negative effect from the unwanted crane's vibrations. This is very common example of an airplane taxiing over a wavy surface of a runway or a vehicle driving over a wavy road surface. The MR fluid (MRF) damper in this suspension design is used to separate, to same extent, motion of mass m_1 which represents wheel with attached masses, from m_2, which is airplane or vehicle body mass. This RF damper, with a controlled value for its damping factor by associated control system (such as mechatronics control system), allows optimizing for minimization of the amplitude of motion or force transmitted to the airplane or vehicle body. The passive, the most common design, vibration suspension works in optimal conditions only when the mass of the system varies in a narrow range and in a certain frequency space. To improve/expand suspension performance over a wide range of payloads and frequencies, the active vibration control technique can be used, however, associated with this design, complexity, cost and power requirements limits its application. With some compromises in crane's control effectiveness, the active vibration control system can be replaced by a semi-active vibration control system of the crane. In many practical applications semi-active vibration control systems can be nearly as effective as active vibration control systems used in some passenger vehicles. The positive characteristics of this system are:

1. The semi-active system/suspension still works in a passive regime even when the control system and/or power supply fails.
2. The power requirements to control the damping force of the damper with rheological fluid (RF) are relatively low.
3. By using MR fluid in a damper, a common passenger vehicle 12 V DC electrical system is sufficient to create effective the damping force. The force controlling the electrical current usually doesn't exceed a few amperes.

Response of the crane's vibration system damper with a rheological fluid to the external field. The principle of application of a magnetorheological fluid in damper design to control the magnitude of a damping force F_d in the crane's vibration system by applying electromagnetic field resulted from electrical current (i) flowing in coils around piston's orifices is shown in fig. 1. The response of the damper under an applied external field in this example results from the changes in apparent viscosity of the MRF suspension.

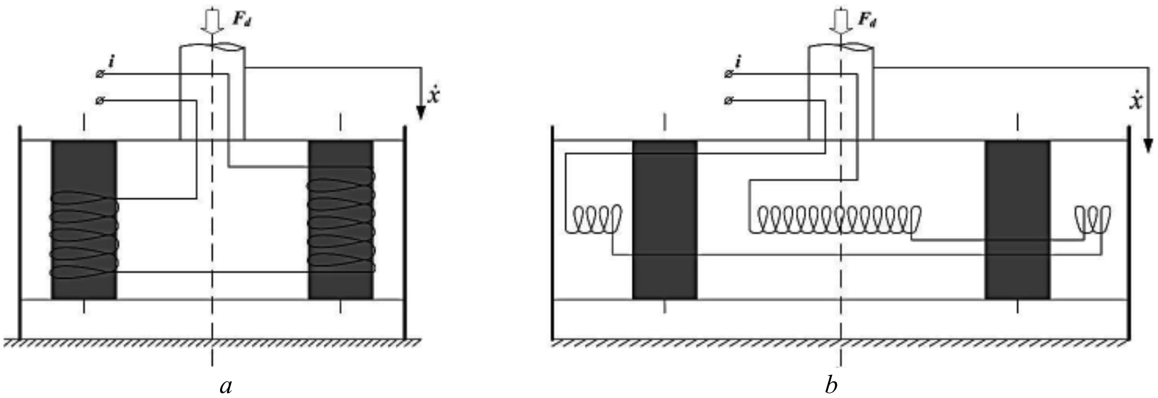

Fig. 1. The principle to control damping force F_d by applying variable electrical current i to change the apparent viscosity of the MRF in the orifices: a – tangential stress control; b – normal stress control

The damping force F_d is proportional to the apparent viscosity of the RF in the orifices and its velocity ($\dot{x}$). The viscosity (after I. Newton) is described as a relationship between shear stress in a fluid (τ) and observed velocity gradient ($\partial\dot{x}/_{\partial h}$) in a fluid subjected to motion. Characteristics $\tau_{RF} = f(\dot{x})$ of a typical MRF are shown in fig. 2. In the absence of an applied external field the RF often exhibits Newtonian-like behavior associated mostly with the base fluid physical properties. An applied external field changes this behavior and the rheological fluid in the piston's orifices shows a variable yield stress which depends on the strength of that field. The apparent shear stress of the RF depends of two components. One of them is Newtonian, proportional to the viscosity of the base fluid and velocity gradient. The second is controllable by the applied external field. The controllable external field stress shown in fig. 1, *b* is significantly higher in amplitude than in the design shown in fig. 1, *a*.

Equation (1) describes the property of apparent shear stress observed in the piston's orifices when an external field is applied.

$$\tau(RF_i) = \tau_0(RF_i) + \eta\cdot(\partial\dot{x}/_{\partial h}), \qquad (1)$$

where yield stress $\tau_0(RF_i)$ as a function of the external field caused by the magnetic flux density for $\dot{x} = 0$ and Newtonian shear stress $\eta\cdot\left(\partial\dot{x}/_{\partial h}\right)$ proportional to dynamic viscosity of the base fluid η and velocity gradient $\partial\dot{x}/_{\partial h}$.

In the absence of the external field, the shear stress $\tau(RF_i)$ of the rheological fluid behaves viscoelastically. fig. 2 shows the behavior of the apparent shear stress of a rheological fluid in a damper under an applied external electrical or electromagnetic field. The electromagnetic field can also be represented by the electrical current i flowing in coils placed around piston's orifices.

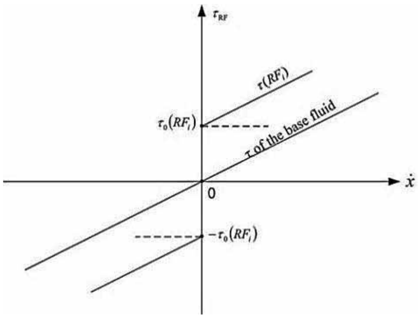

Fig. 2. The shear stress versus velocity of a MRF under applied electromagnetic field represented by current I, where $\tau_0(RF_i)$ – yield stress, $\tau(RF_i)$ *– the shear stress*

According to fig. 2 the shear stress of a RF can be expressed as

$$\tau(RF_i) = \tau_0(RF_i) + \frac{\partial \tau_{RF_i}}{\partial \dot{x}} \cdot \dot{x}\,.$$

The equivalent damping factor C_{RF}, is:

$$C_{RF_i} = \left[\left\{\tau_0(RF_i) + \frac{\partial \tau_{RF_i}}{\partial \dot{x}} \cdot \dot{x}\right\} \cdot A\right] \cdot \frac{1}{\dot{x}}\,,$$

where A is chosen oblique area.

The damping force F_{di} at point of work is

$$F_{di} = \tau(RF_i) \cdot A\,.$$

The ratio of

$$\frac{\partial \tau_{RF_i}}{\partial \dot{x}} = f\left[\tau_0(RF_i)\right],$$

needs to be established experimentally.

RF damper model for the crane's vibration system. The balance of internal damper forces in equilibrium with an external force (free body diagram) of a RF damper of the crane's vibration system is shown in fig. 3. The complex damping force F_{di} (which is also a response force from the damper in motion) has two components, $F_{d\eta}$, which depends on a damping constant C_η (related to the piston's orifice design and physical properties of the base fluid) and velocity $\dot{x}$, and F_{doi}, which depends only on the external, in this case electromagnetic field, represented by electrical current i. In the absence of an external electromagnetic field and/or current i, the internal force F_{doi} becomes zero and the damping force becomes $F_{di}=F_{d\eta}$.

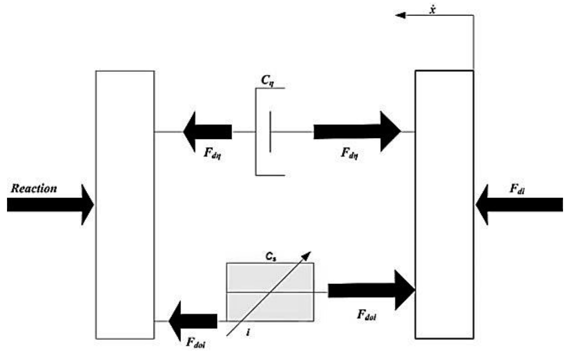

Fig. 3. Model of the rheological fluid damper for a crane's vibration system, where $F_{d\eta}$ *– viscous damping force;* F_{doi} *– damping force controlled by external field;* C_s *– the apparent damping coefficient of the RF;* C_η *– a damping constant;* F_{di} *– the complex damping force.*

The relationship between force, shear stress and velocity is called the Rheological Fluid Model and can be expressed in the general form as

$$F_{di} = \begin{cases} \tau_0(RF_i) \cdot A + \dfrac{\partial_i RF}{\partial \dot{x}} A\dot{x} & \dot{x} > 0 \\ 0 & \dot{x} = 0. \\ -\tau_0(RF_i) \cdot A + \dfrac{\partial_i RF}{\partial \dot{x}} A\dot{x} & \dot{x} < 0 \end{cases}$$

Considering that

$$\tau_0(RF) \cdot A = F_{do}(RF),$$

represents damping force controlled by external field and:

$$\frac{\partial \tau_{RF}}{\partial \dot{x}} A\dot{x} = F_{d\eta}\,,$$

which represented damping force proportional to the velocity $\dot{x}$ (see fig. 3) and is

$$F_{di} = F_{doi} + F_{d\eta}\,, \qquad (2)$$

represents the complex damping force.

Response of the crane's vibrating system with RF damper. ***One degree of freedom crane's vibrating system with RF damper.*** The free body diagram of a one degree of freedom (1DOF) crane's vibrating system with a RF damper is shown in fig. 4.

In this model the instantaneous equilibrium of forces is

$$m\ddot{x} + C_\eta \cdot \dot{x} + F_{do}(RF) \cdot \operatorname{sgn}(\dot{x}) + kx = 0\,, \qquad (3)$$

where

$$C_\eta \cdot \dot{x} + F_{do}(RF) \cdot \operatorname{sgn}(\dot{x}) = F_d\,,$$

is the complex damping force.

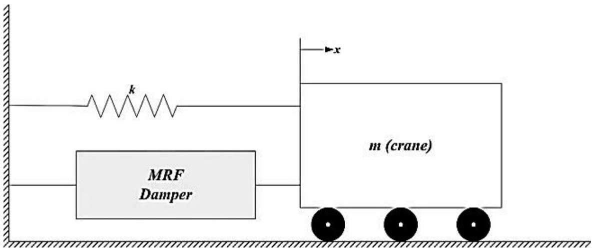

Fig. 4. Model of a 1DOF of crane's vibrating system with a RF damper, where m is mass of crane and k is stiffness of crane

This can be expressed as a product of equivalent damping C_{eq} and velocity $\dot{x}$

$$F_d = C_{eq} \cdot \dot{x} .$$

Two degree of freedom crane's vibrating system with base excitation and RF damper. This model represents two degree of freedom (2DOF) crane's vibrating systems with a base excitation system, having stiffness k_1 and mass m_1 in the first stage and connected by a spring with stiffness k_2 and a parallel attached controllable (by mechatronic system) MR damper to the second mass m_2.

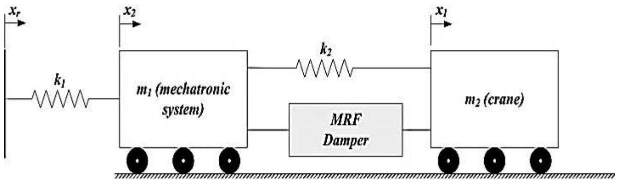

Fig. 5. The model of two degree of freedom crane's vibrating system with base excitation, where де m_1 – mechatronics system, k_1 – stiffness of mechatronics system; m_2 –crane; k_2 – stiffness of spring; x_1, x_2 – axial coordinates of mechatronics system and crane.

Behavior of this two degree of freedom crane's vibrating system can be described by a set of two equations which are the instantaneous equilibrium of the acting forces

$$\begin{cases} m_2 \cdot \ddot{x}_2 = (x_1 - x_2) \cdot k_2 + (\dot{x}_1 - \dot{x}_2) C_{RF} \\ m_1 \cdot \ddot{x}_1 = (x_2 - x_1) \cdot k_2 + (\dot{x}_2 - \dot{x}_1) C_{RF} + (x_2 - x_1) \cdot k_1 \end{cases},$$

where

$$(\dot{x}_2 - \dot{x}_1) \cdot C_{RF} = F_d$$

represents the magneto-rheological damping force.

The rheological complex damping coefficient complex damping coefficient (C_{RF}) depends on the mechanical and electrical design of the damper and rheological fluid used.

Equivalent damping in the crane's vibrating system. The response of the 1DOF crane's vibrating system with RF damper presented in fig. 4 under harmonic excitation force $F_0 \cdot \sin \omega t$ applied to mass m is

$$m\ddot{x} + F_{do}(RF) \cdot \mathrm{sgn}(\dot{x}) + C_\eta \cdot \dot{x} + kx = F_0 \cdot \sin(\omega t) \cdot \qquad (4)$$

In this equation the damping force has two components.

One of them is a Newtonian type and it is proportional to the velocity $\dot{x}$, and a second, a semi Bingham one, which depends on the strength of the external field and direction of motion expressed by $sgn(\dot{x})$.

The energy dissipated, ΔE_η in the viscously damped system per one cycle with viscous damping coefficient C_η is

$$\Delta E_\eta = \oint F_{d\eta} dx = \int_0^{2\pi/\omega} C_\eta \cdot \dot{x} \cdot \frac{dx}{dt} dt = \int_0^{2\pi/\omega} C_\eta \cdot \dot{x}^2 dt .$$

Substituting $x = X \cdot \sin(\omega t)$ and $\dot{x} = \omega X \cdot \cos(\omega t)$ into above equation,

$$\Delta E_\eta = C_\eta \cdot \int_0^{2\pi/\omega} \left[\omega^2 \cdot X^2 \cdot \cos^2(\omega t)\right] dt ,$$

then integrating, results in

$$\Delta E_\eta = C_\eta \cdot \pi \cdot \omega \cdot X^2 .$$

The second damping component represented by force, $F_{do}(RF)$ in (2) yields the following expression for dissipated energy

$$\Delta E(RF) = F_{do}(RF) \cdot \int_0^{2\pi/\omega} [\mathrm{sgn}(\dot{x}) \cdot \dot{x}] dt .$$

Then dissipated energy in one cycle of the damper becomes

$$\Delta E(RF) = F_{do}(RF) \cdot X \cdot \left[\int_0^{\pi/2} cos(\omega t) d(\omega t) - \int_{\pi/2}^{3\pi/2} cos(\omega t) d(\omega t) + \int_{3\pi/2}^{2\pi} cos(\omega t) d(\omega t) \right].$$

Solving the integration yields that the energy dissipated by a controllable by mechatronic system damping force $F_{do}(RF)$ is

$$\Delta E(RF) = 4 \cdot F_{do}(RF) \cdot X .$$

To create a viscously damped system of equivalent energy loss, we obtain

$$\pi \cdot C_{eq} \cdot \omega \cdot X^2 = 4 \cdot F_{do}(RF) \cdot X + C_\eta \cdot \pi \cdot \omega \cdot X^2 .$$

Thus the equivalent damping coefficient C_{eq} yields

$$C_{eq} = \frac{4F_{do}(RF) \cdot X + C_\eta \cdot \pi \cdot \omega \cdot X^2}{\pi \omega X^2} .$$

In terms of equivalent damping ratio ξ_{eq}

$$C_{eq} = 2 \cdot \xi_{eq} \cdot \omega_n \cdot m$$

and

$$\xi_{eq} = \frac{4 \cdot F_{do}(RF) \cdot X + C_\eta \cdot \pi \cdot \omega \cdot X^2}{2\pi \cdot \omega \cdot \omega_n \cdot X^2}.$$

The 1DOF crane's vibrating system with equivalent damping C_{eq} which will dissipate as much energy as the system described by (3) is

$$\ddot{x} + 2\xi_{eq} \cdot \omega \cdot \dot{x} + \omega_n^2 \cdot x = f_0 \cdot sin(\omega t), \qquad (5)$$

where $f_0 = {F_0}/{m}$ and $\omega_n = \sqrt{{k}/{m}}$.

This is also an approximation of the (4).

Conclusion. In this paper the analytical model of rheological fluid was formulated and the equivalent coefficient of damping of the damper with a magneto-rheological (MR) fluid based on the dissipated energy principle was calculated. The major parameter in these calculations is apparent viscosity associated with shear stress of the MR fluid under an applied external field. This equivalent coefficient of damping allows the performance of crane's vibration calculations and the design of mechanical systems to control with a help of mechatronic devices unwanted crane's vibrations in wider payload and frequency range that the crane's system with uncontrollable damping. In addition, when a variable strength external field synchronized with the period of crane's system oscillations is applied an almost unlimited characteristic of a damping force can be obtained. The rheological phenomenon can also be used to control sound transmission loss of a multibarrier system with rheological fluid placed between them. The increasing mechanical strength of the fluid between barriers increases apparent/equivalent stiffness of the system, thus the control of sound transmission loss in a stiffness control space is achievable.

References / Список літератури

1. Choi, S.B., Hong, S.R, Cheong, C.C. and Park, Y.K. (1999), "Comparison of field-controlled characteristics between ER and MR clutches", *Journal of Intelligent Material Systems and Structures*, vol. 10, pp. 615–619.

2. Choi, S.B., Hong, S.R., Sung, K.G., Sohn, J.W. (2008), "Optimal control of structural vibrations using a mixed-mode magnetotheological fluid mount", *International Journal of Mechanical Sciences*, vol. 50, pp. 559–56.

3. Kamath, G.M., Werely, N.M. and Jolly, M.R. (1999), "Characterization of magnetorheological helicopter lag dampers", *Journal of the American Helicopter Society*, vol. 44, pp. 234–248.

4. Pang, L., Kamath, G.M. and Wereley, N.M. (1997), "Analysis and testing of a linear stroke magnetorheological damper", *AIAA/ASME Adaptive Structures Forum*, Paper No. AIAA 98–2040, Long Beach, CA.

5. Spencer, Jr.B.F., Dyke, S.J., Sain, M.K. and Carlson, J.D. (1997), "Phenomenological model for a magnetorheological damper", *Journal of Engineering Mechanics ASCE*, vol. 123, pp. 230–238.

6. Weiss, K.D., Duclos, T., Carlson, J.D., Chrzan, J.M. and Margida, A.J. (1993), "High strength magneto- and electro-rheological fluids", *Society of Automotive Engineers*, SAE Paper 932451.

7. Weiss, K.D., Carlson, J.D. and Nixon, D.A. (1994), "Viscoelastic properties of magneto- and electrorheological fluids", *Journal of Intelligent Material Systems and Structures*, vol. 5, pp. 772–775.

8. Carlson, J.D., Cantanzarite, D.M. and Clair, K.A.S. (1996), "Commercial magnetorheological fluid devices", *International Journal of Modern Physics B*, vol. 10, pp. 23–24, 2857–2865.

Мета. Обґрунтування нелінійної математичної моделі поведінки та оптимізації робочих режимів для вібраційних систем кранів, що керовані мехатронними пристроями з магнітореологічною рідиною.

Методика. Використані методи механіки та математичної фізики.

Результати. Наведена нелінійна математична модель поведінки та оптимізації режимів руху (робочих режимів) вібраційних систем вантажопідйомних кранів, керованих мехатронними пристроями з магнітореологічною рідиною. Також наведена характеристика в'язкості рідини, якою можна керувати докладанням зовнішнього (магнітного) поля. Був розрахований еквівалентний коефіцієнт загасання на основі принципу енергії, що розсіюється протягом одного циклу роботи демпфера, при постійній силі зовнішнього поля. Коли маса або жорсткість – змінні величини, еквівалентний коефіцієнт демпфірування може бути встановлений відповідним регулюванням сили зовнішнього поля, щоб мати навмисну/постійну роботу коливальної, демпферної системи крана у критичному чи іншому обраному стані.

Наукова новизна. Застосування рідини з магнітореологічним ефектом у кранових системах у режимах пуску/зупинки дозволяє істотно зменшити небажані перевантаження, коливання й резонанси, що виникають у них. А також дозволяє оптимізувати траєкторію їх руху, що визначається не тільки стандартними характеристиками (переміщення, швидкість, прискорення), але й похідними у часі від закону руху системи третього та четвертого порядків (ривок).

Практична значимість. Робота представляє випадки застосування періодичної зміни сил зовнішньої, синхронізованої з періодами періодичного переміщення крана, коливальної системи для постійного оптимального управління силою демпфірування всередині кожного циклу.

Ключові слова: *управління шумом, вібрація кранів, нелінійна математична модель, магнітореологічна рідина*

Цель. Обоснование нелинейной математической модели поведения и оптимизации рабочих режимов для вибрационных систем кранов, управляемых мехатронными устройствами с магнитореологической жидкостью.

Методика. Использованы методы механики и математической физики.

Результаты. Приведена нелинейная математическая модель поведения и оптимизации режимов движения (рабочих режимов) вибрационных систем грузоподъемных кранов, управляемых мехатронными устройствами с магнитореологической жидкостью. Также представлена характеристика вязкости жидкости, которой можно управлять приложением внешнего (магнитного) поля. Был рассчитан эквивалентный коэффициент затухания на основе принципа энергии, рассеиваемой в течение одного цикла работы демпфера, при постоянной силе внешнего поля. Когда масса или жесткость – переменные величины, эквивалентный коэффициент демпфирования может быть установлен соответствующей регулировкой силы внешнего поля, чтобы иметь намеренную/постоянную работу колебательной, демпфированной системы крана в критическом или другом выбранном состоянии.

Научная новизна. Применение жидкости с магнитореологическим эффектом в крановых системах в режимах пуска/остановки позволяет существенно уменьшить нежелательные перегрузки, колебания и резонансы, возникающие в них. Также позволяет оптимизировать траекторию их движения, которая определяется не только стандартными характеристиками (перемещение, скорость, ускорение), но и производными по времени от закона движения системы третьего и четвертого порядков (рывок).

Практическая значимость. Работа представляет случаи применения периодического изменения сил внешней, синхронизированной с периодами периодического перемещения крана, колебательной системы для постоянного оптимального управления силой демпфирования внутри каждого цикла.

Ключевые слова: *управление шумом, вибрация кранов, нелинейная математическая модель, магнитореологическая жидкость*

Рекомендовано до публікації докт. техн. наук О.М. Шашенком. Дата надходження рукопису 04.12.13.