ADAPTIVE CONTROL OF DRILL STRING VIBRATIONS

É. A. Petrovskii, K. A. Bashmur, and I. S. Nashivanov

The basic principles and methods of drill string vibration control are studied. The prospects of development of a semiactive vibration damping method are shown. The principle of semiactive vibration damping developed by the authors using the Matlab Simulink graphical programming environment for modeling is studied. Analytical correlations of the equations of motion of this model are presented. Resulting motion graphs are derived and conclusions are drawn about the behavior pattern of similar vibration damping systems.

Keywords: vibrations, drill string, damper, model, vibration damping.

Well drilling is attended by severe vibrations of various equipment. Vibrations often have a localized source, but they affect the whole equipment system [1, 2].

Because of the trend toward increasing well drilling depths and wide application of inclined and directional drilling, drill string (DS) vibrations cause substantial economic losses, so development of an efficient DS vibration control method is becoming an increasingly urgent task [3]. In terms of the theory of control, the current DS vibration control methods can be subdivided into three groups: passive, active, and semiactive (combined) [4].

A distinctive feature of passive DS vibration control methods is that the operating control system does not require external energy sources.

Active DS vibration control, in keeping with the dynamic characteristics of the control system, is accomplished by active application of a force that is equal in magnitude and opposite to the direction of the forces generated by external vibrations.

The semiactive vibration control technologies are distinguished by application of elements of active control in passive types of devices. To reduce string vibrations, the dynamics of the bottom-hole drill string assembly (BDSA) is controlled continuously and updated information is transmitted to the operator periodically (Fig. 1). This control method is realized in the design of the damper [5] whose operating principle is based on variation of the viscosity of the magnetic fluid (by controlling the inductance in the damper module) and, consequently, of the kinetic stiffness of the BDSA.

We developed a new principle of semiactive vibration damping for damping longitudinal vibrations, particularly of DS, based on the action of permanent or variable magnets. The proposed solution will allow enhancement of the reliability and efficiency and diminution of overall dimensions of the system without the aid of electronic vibration recording devices.

Let us review the calculation scheme (Fig. 2) where one M50 permanent neodymium magnet is attached each to the moving body (at the bottom) and to the lower limiter (at the top). To simplify the model, frictional forces are ignored. To monitor the change in disposition of each of the objects, the extents of their displacement relative to the origin of the coordinates are introduced in the scheme. The displacement x_i is the distance from the level of the origin of the coordinates to the level of location of the mid-point of the referred object.

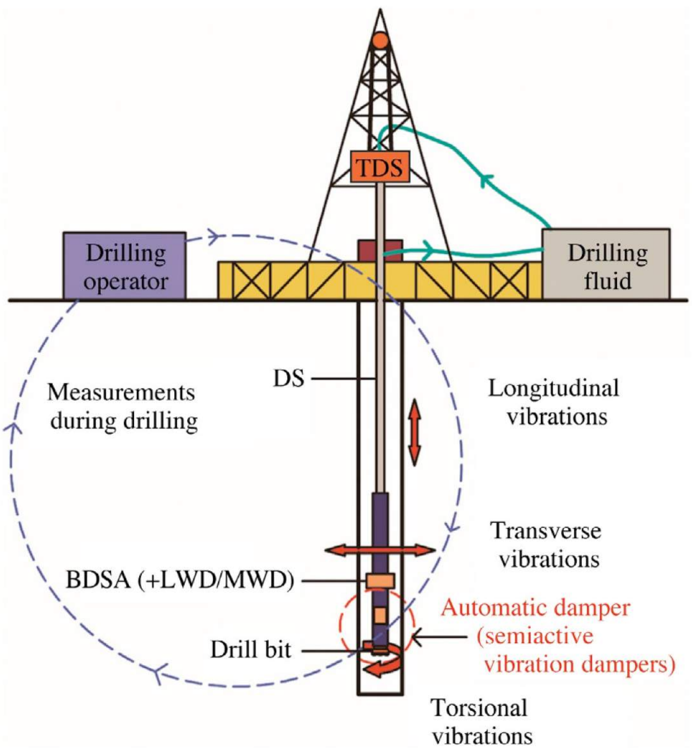

Fig. 1. DS vibration and shock control system.

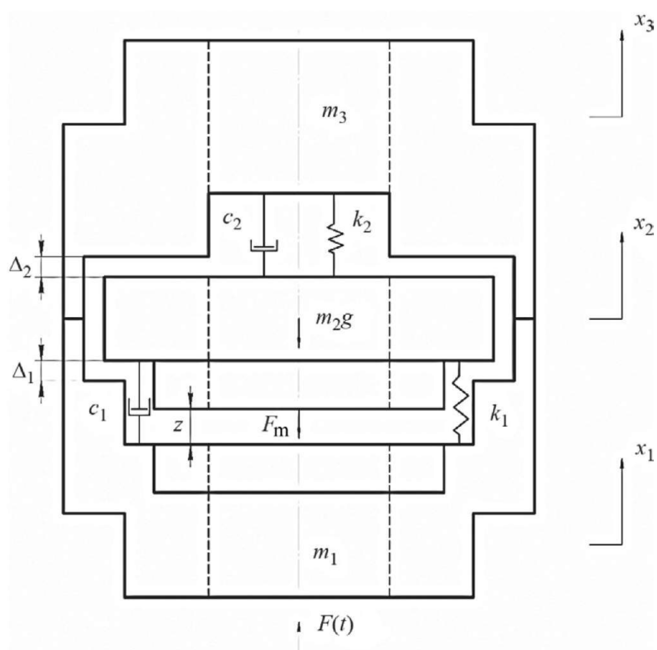

Fig. 2. Calculation scheme of magnetic vibration damper: m_1 — mass of bottom limiter, m_2 — mass of moving body; m_3 — mass of top limiter; m_2g — weight of moving body; c_1, k_1 — damping and rigidity in the first level; c_2, k_2 — damping and rigidity in the second level; Δ_1 — gap between moving body and bottom limiter; Δ_2 — gap between moving body and top limiter; z — distance between magnets; x_1, x_2, x_3 — generalized displacement coordinates of each object relative to origin of coordinates; $F(t)$ — perturbing force; F_m — magnetic force.

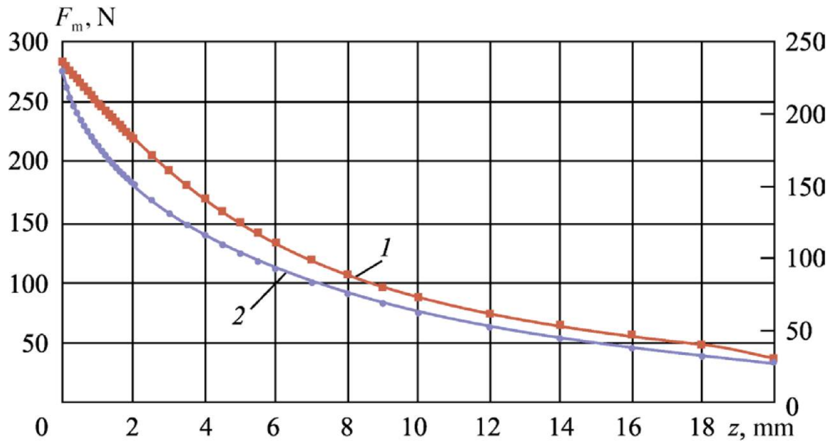

Fig. 3. Standard (*1*) and approximating (*2*) functions of magnetic force.

Translocation of components of the vibration damping system is possible only relative to the vertical axis. The top and bottom limiters (placed with the initial gaps Δ_1 and Δ_2) are considered fully elastic, so impact loads on them can be considered momentary. It is also accepted that the magnets lie at a distance z from each other, in which case $\Delta_1 < z$ and $\Delta_2 < z$, i.e., the possibility of collision of the magnets is ruled out.

The poles of the magnets are so disposed that when they come close to each other the magnets are repulsed with a force F_m, which counteracts the perturbing force $F(t)$ which increases with increase of the amplitude of $F(t)$. Based on this, we shall tentatively accept that the action of the force is directed downward [counter to the perturbing force $F(t)$].

The value of the force F_m can be varied and the efficiency of the vibration process can be controlled by varying the size, shape, type, and spatial configuration of the magnets.

One of the most important parameters affecting the change in the force F_m is the distance z between the magnets. This parameter can be evaluated by analyzing the standard and approximating functions, the comparative graphs (Fig. 3) of which are constructed in the ELCUT program developed for calculating and simulating physical fields.

The magnitudes of the force F_m are approximated depending the z value in conformity with the function that is a third-degree polynomial. At $z < 20$ mm approximation of the third-degree polynomial yields a good result and a low error with the standard function, including $z \geq 0$ (Fig. 3). At $z > 20$ mm the graph differs from the standard (the branch of the hyperbola intersects the OX axis and then descends rapidly), so it is inexpedient to consider the magnetic force F_m in the model for the values $z > 20$ mm.

For constructing a simulation model, it is essential to write the equations that describe the movement of the bodies. For the referred calculation scheme (Fig. 2), where there is a perturbing force, the differential equations system has the form:

$$\begin{cases} m_3\ddot{x}_3 + c_2(\dot{x}_3 - \dot{x}_2) + k_2(x_3 - x_2) = 0, \\ m_2\ddot{x}_2 + c_1(\dot{x}_2 - \dot{x}_1) + c_2(\dot{x}_2 - \dot{x}_3) + k_1(x_2 - x_1) + k_2(x_2 - x_3) = m_2 g + F_m, \\ m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = F(t) - F_m, \end{cases} \tag{1}$$

where x_i, $\dot{x}_i$, $\ddot{x}_i$ are the generalized coordinates and their time-linked derivatives.

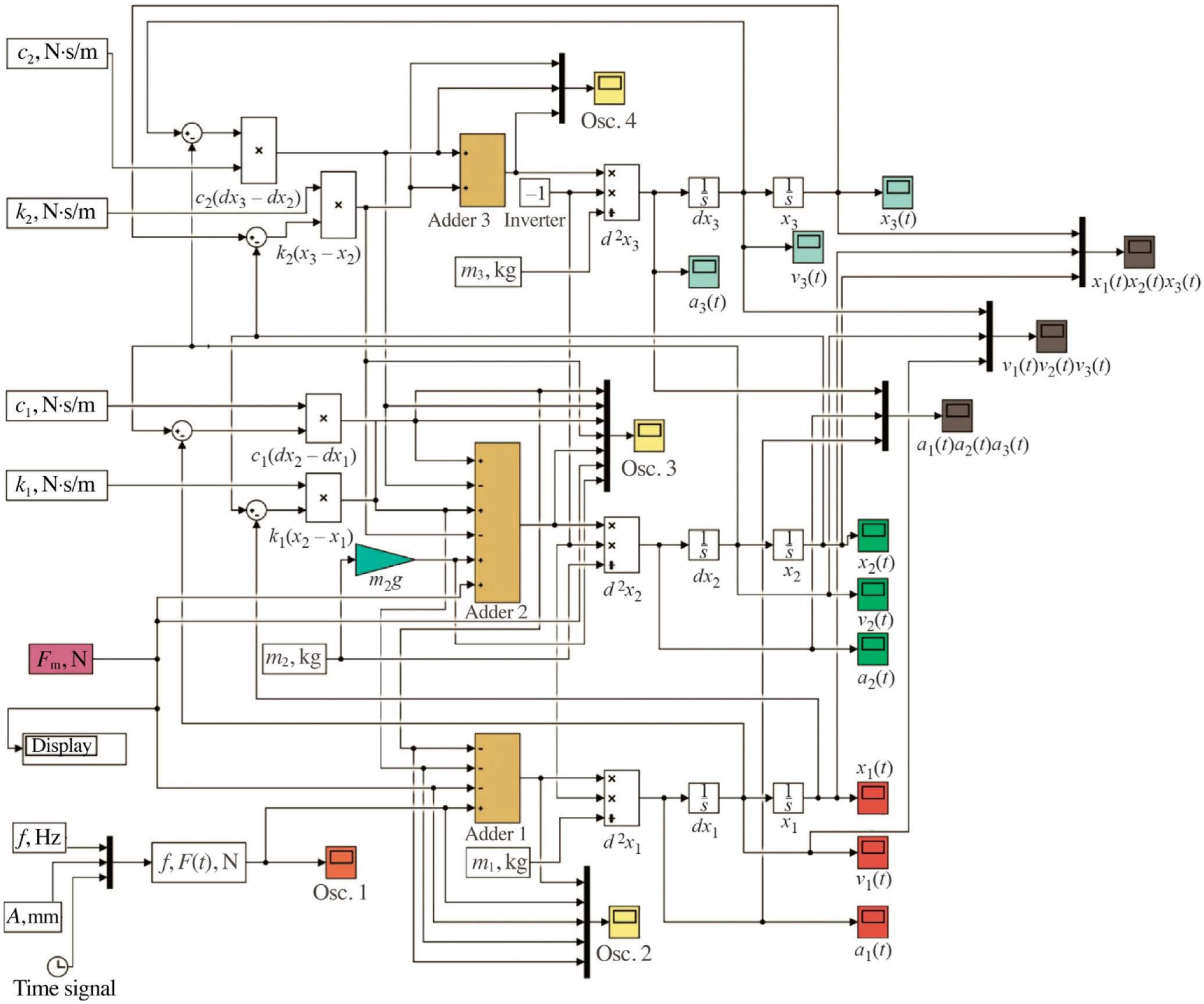

Fig. 4. Block diagram of magnetic vibration damper.

Let us transform the equation of motion to the form:

$$\begin{cases} \ddot{x}_3 = -\dfrac{1}{m_3}\left[c_2(\dot{x}_3 - \dot{x}_2) + k_2(x_3 - x_2) \right], \\ \ddot{x}_2 = -\dfrac{1}{m_2}\left[c_1(\dot{x}_2 - \dot{x}_1) + c_2(\dot{x}_2 - \dot{x}_3) + k_1(x_2 - x_1) + k_2(x_2 - x_3) + m_2 g + F_{\mathrm{m}} \right], \\ \ddot{x}_1 = -\dfrac{1}{m_1}\left[c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) - F(t) + F_{\mathrm{m}} \right]. \end{cases} \tag{2}$$

After expressing the accelerations in each of the three differential equations, a simulation model was constructed to analyze the efficiency of the vibration damping system having permanent magnets. The construction was carried out in the Matlab Simulink graphical simulation modeling environment, which was dictated by the presence of validated mathematical methods, clarity, and accuracy of presentation of the simulation data.

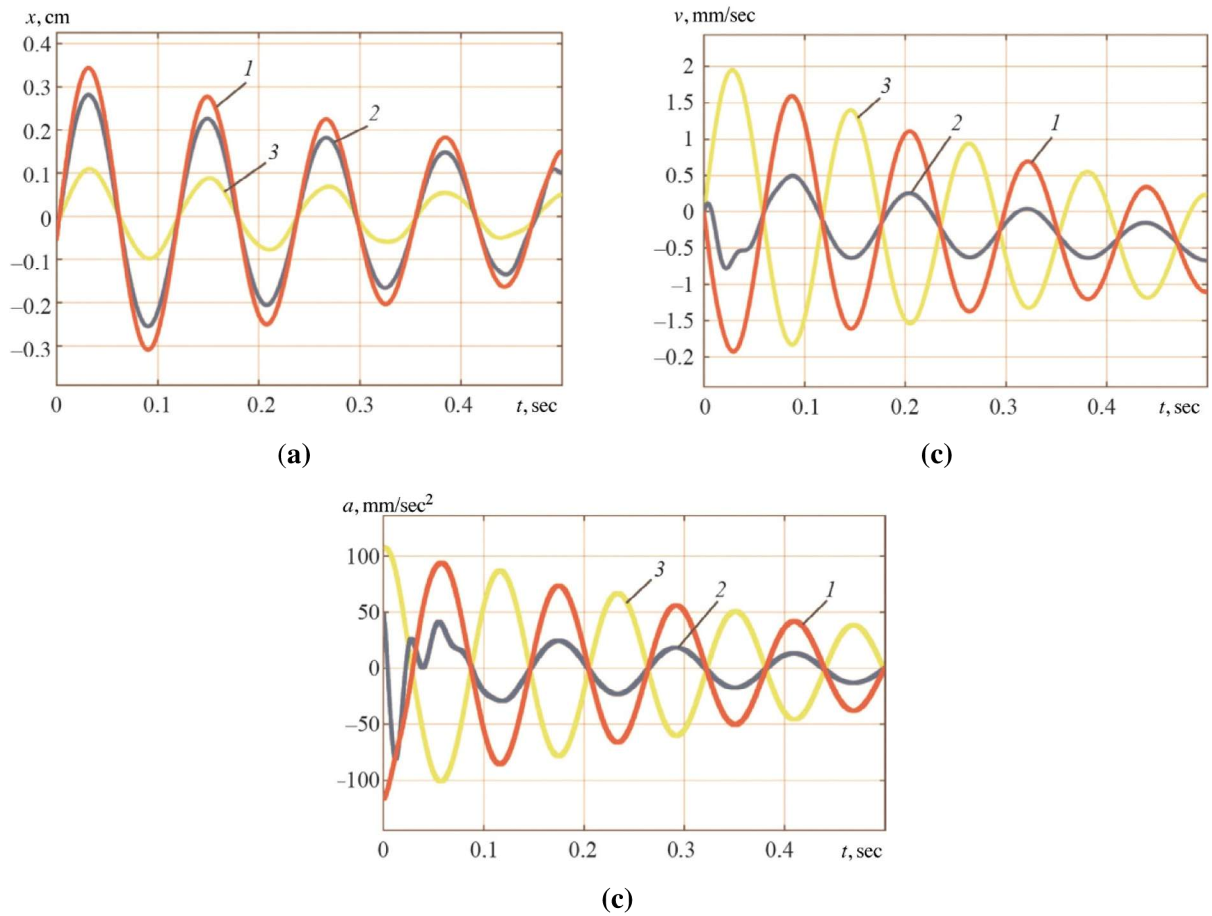

Fig. 5. Resulting graphs of displacements (*a*), velocities (*b*), and accelerations (*c*) for bottom limiter *1*, moving body *2*, and top limiter *3*.

The foundation of the obtained simulation model in the form of a block diagram (Fig. 4) consists of individual adders (Adder 1, Adder 2, Adder 3) with the number of inputs that equals the number of addends in the right-hand part of each of the equations (2), whereas the left-hand parts are outputs from the respective adders. For a quantitative description of vibrations, we shall first divide the results at the outputs of the adders (values of accelerations a_1, a_2, a_3) by m_i and then integrate: after the first integration we will get the values of the velocities v_1, v_2, v_3 and after the second integration, we will get the displacements x_1, x_2, x_3.

As a result of simulation modeling (in which the block diagram is realized by subsystems of the Matlab Simulink program), graphs of displacements, velocities, and accelerations were obtained (Fig. 5) for the monitored points, namely, bottom limiter, moving body, and top limiter.

According to the displacement graph (Fig. 5a), displacement of the bottom limiter is the maximum because this component is subjected first to the impact of the perturbing force $F(t)$. Displacement of the moving body is less because it lies between the limiters and is an intermediate component.

Displacement of the top limiter is the minimum because only vibrations “left” after vibration damping act on it. The velocity graph (Fig. 5b) illustrates diminution of the rate of displacement of the monitored points along the measurement axis, which also indicates diminution of the influence of vibration. This is evident also from the acceleration graph (Fig. 5c): diminution of the influence of vibration is expressed in steady diminution of dynamic interaction between the monitored points.

CONCLUSIONS

Analysis of the main methods of DS vibration control has enabled the authors to propose and investigate a vibration damper based on semiactive principle of action.

Dependencies of displacements, velocities, and accelerations on time are constructed.

The stability and efficiency of operation of the vibration damping system are confirmed by the obtained data.

REFERENCES

1. É. A. Petrovskii, V. V. Bukhtoyarov, and E. O. Starostina, "Assessment of risks of functioning of components of technological equipment," *Khim. Neftegaz. Mashinostr.* No. 9, 42–45 (2016).
2. E. A. Petrovskii, K. A. Bashmur, and I. S. Nashivanov, "Technological possibilities of shock vibration dampers for increasing reliability of drilling equipment," *Stroit. Neft. Gaz. Skvazh. na Sushe i na More*, No. 2, 9–14 (2018).
3. M. F. Zalyaev, V. U. Yamaliev, E. M. Abutalipova, and A. N. Avrenyuk, "Necessity for taking account of vibration while constructing drill string components," *Khim. Neftegaz. Mashinostr.* No. 9, 45–48 (2016).
4. G. Dong and P. A. Chen, "A review of the evaluation, control, and application technologies for drill string vibrations and shocks in oil and gas wells," *Shock and Vibration*, **2016**, 1–34 (2016).
5. M. E. Wassell, M. E. Cobern, V. Saheta, A. Purwanto, and M. Cepeda, "Active vibration damper improves operation parameters and reduces drilling costs," *Neftegaz. Tekhnol.*, No. 1, 32–35 (2009).