UDC 620.17.05
DOI: 10.15587/1729-4061.2021.226697

DESIGNING A SHOCK TEST SYSTEM PROTOTYPE BASED ON A HYDROELASTIC DRIVE

Oleksii Sheremet
Doctor of Technical Sciences, Professor*
E-mail: sheremet-oleksii@ukr.net
Tetiana Kiriienko
Assistant*
E-mail: kiritan1@rambler.ru
Andrii Besh
Senior Lecturer*
E-mail: ray_mele@ukr.net
Kateryna Sheremet
Laboratory Assistant
Department of Intelligent Decision Support Systems**
E-mail: artks@ukr.net
*Department of Electromechanical Systems of Automation and Electric Drive**
**Donbass State Engineering Academy
Akademichna str., 72, Kramatorsk, Ukraine, 84313

Laboratory shock tests involve the reproduction of simple one-time and repeated pulses of a certain waveform. In practice, such mechanical impacts on an object are implemented at specialized testing equipment – shock systems.

A promising direction in the development of shock machines includes the structures that operate on the energy of elastic deformation of the compressed liquid and the shell of the vessel that contains it. Such systems make it possible to improve the versatility, manageability, and accuracy of impact tests.

Underlying this study is the use of a hydroelastic drive to design a prototype of the automated electro-hydraulic system for a shock test system.

The proposed shock test system prototype makes it possible to expand the functionality of the installations to perform impact tests with a series of pulses, as well as improve manageability and increase the level of automation. The main feature of the proposed structural scheme is that the reconfiguration for a new impact pulse occurs very quickly. Owing to the presence of a driven rotary drum with braking devices, the bench makes it possible to generate a shock pulse repetition frequency of 1–2 Hz.

The constructed mathematical model of the shock machine takes into consideration the inertia of moving masses, the rigidity of the liquid or "one-way" spring of the charging chamber, as well as the influence of dampers on which the test platform rests. The variables in the mathematical model are linked by differential equations describing two periods within a shock system work cycle: charging and pulse generation. The model's practical value is to determine the dynamic characteristics of the test installation, as well as to calculate the required structural and technological parameters.

The differential equations describing the movements at the shock machine have been solved in a numerical way. The study results have established the optimal value (in terms of minimizing the overload on an article on the return stroke of the rod) for the damping factor of the braking device, which is 13,000 kg/s. In this setting, the ratio of the amplitude of acceleration on the reverse stroke to the amplitude of effective acceleration during tests is reduced to a minimum of 0.195

Keywords: shock test system, hydroelastic drive, damping factor, impact acceleration

Received date 21.01.2021
Accepted date 12.03.2021
Published date 26.04.2021

***How to Cite:** Sheremet, O., Kiriienko, T., Besh, A., Sheremet, K. (2021). Designing a shock test system prototype based on a hydroelastic drive. Eastern-European Journal of Enterprise Technologies, 2 (7 (110)), 58–65. doi: https://doi.org/10.15587/1729-4061.2021.226697*

1. Introduction

Many devices and assemblies that operate under actual conditions can be exposed to impact influences exerted by the surrounding objects. As a result of the impact influence on an object, its mechanical strength may be affected or certain functional deviations in its operational performance may occur. Given this, the types of impact tests are mainly determined by two factors such as impact strength and impact resistance.

There are the following types of impact tests [1]: for impact strength under repeated forcing; for impact resistance under repeated forcing; for the influence exerted by high-intensity single strikes; for strength during transportation, operation, and fall. In addition to these types of impact tests in the laboratory setting, numerous model tests of those objects can be carried out whose size and weight do not make it possible to design the required shock machine.

Timely laboratory diagnosing and forecasting the condition of a technical facility make it possible to reduce the time, the volume of repair operations, and the number of spare parts; to reduce the number of sudden failures; to bring down lost profits caused by equipment outages.

As practice has shown, it is possible, in mechanics and electromechanics, to effectively diagnose machines mainly based on the vibration and impact due to the following:

– disturbance forces arise directly at the site of the defect;

– the spectrum of a vibration signal contains a large amount of diagnostic information;

– one can explore both a separate element of the structure and the entire machine or assembly in general.

The testing methodology is devised on the basis of the initial requirements for the performance of the test object or the strength of individual elements of the object, taking into consideration the possibility of cost-effective implementation of testing methods at the available test equipment. In the testing of articles, external influences are simulated in such a way that the shock pulses correspond to the impacts exerted during real-world operations.

Copyright © 2021, Authors. This is an open access article under the Creative Commons CC BY license

Laboratory impact tests imply the reproduction of simple single and repeated impact pulses, as well as impact effects, which are complex fading transition processes.

The tests aim to ensure that the test conditions are as close as possible to the conditions of an actual impact on the object. To this end, technologists determine the type, form, duration of impact, the maximum impact acceleration, the direction of impact influence, the number of impacts acting on the object during operation. In addition, it is necessary to establish the characteristics of the tested object (the size, mass, location of impact application, article's operating conditions).

The relevance of this work aimed to address laboratory impact tests is to expand the functionality of shock systems and improve the overall level of test equipment automation.

2. Literature review and problem statement

In accordance with the categorization of external influences on a tested object [2], the impact load can be initiated in the following way: through individual impacts, a series of impacts, by a set change in the impact pulse, and vibration.

Work [3] gives the following classification of test systems by the type of impact load carried out:

– for single impact pulses;
– for multiple impact loads;
– for complex types of impact load;
– for specific conditions of collision.

However, the common disadvantage of many of the approaches reported in papers [2, 3] is the need to design a separate shock system for each type of testing. In most cases, there is no possibility for an operational change in the shape or frequency of impact pulses.

Multiple impact loads can be executed by mechanical-type systems that implement the braking principle. Preliminary acceleration of the impact platform before falling on the braking device is carried out either at free or in the forced (using elastic elements) fall. This is the approach that has been implemented in work [4]. The disadvantage of test systems built in this way is the impossibility of quickly changing the shape of impact pulses. In addition, free-fall machines often have significant limitations on the amplitude of impact acceleration.

More controlled is the process of generating a recurring dynamic load on an article using electric motors. This approach is considered in work [5], which proposes to create a dynamic load on the product by mechanically connecting two induction motors on a common shaft.

Paper [6] reports the results of studying a laboratory bench for the testing of electromechanical transmissions. Similarly to work [5], the forcing is generated by electric motors but the speed of their rotation is controlled by a neural network algorithm.

The general drawbacks of the test systems, which use induction motors for the immediate formation of impact load, are the low speed and complexity of a rapid change in the shape of pulses. In addition, the issue of structural limitation on the amplitude of impact acceleration remains unresolved in works [5, 6]. Since significant impact accelerations lead to the destruction of the mechanical part of electric machines, such solutions have only a limited, highly specialized application.

Work [7] proposes a laboratory sample of the test bench designed to simulate a mechanical load on an electric motor. However, such a bench is practically unsuitable for the reproduction of impact pulses.

The authors of work [8] have solved a highly specialized task – the reproduction of the road load exerted on an article when transporting it by car. The conditional profile of the road is formed by changing the speed of the electric motor. It is well known that electric motors are not designed to operate under a mechanical load of a pulsed nature. Thus, the possibilities of applying the solution reported in [8] for impact tests are extremely limited.

Work [9] solved another highly specialized task related to a bench simulation of the movement of a bogie over a rail track. In general, the cited work has the same shortcomings as paper [8].

A slightly different approach to the technical implementation of the impact testing bench was implemented in work [10]. An electromagnetic vibrator with an appropriate control system is used as an impact pulse shaper. The solution proposed in [10] can be used to study the vibration only of those articles whose dimensions and mass are small. In addition, due to the significant electromagnetic inertia of the coils of inductivity, it is fundamentally impossible to build a fast-acting electromagnetic vibrator to reproduce the high-amplitude impact load.

The topics addressed in some works imply a whole set of tests for a particular product at once. For example, the authors of work [11] offer a test bench to test the front axle of cars for durability and dynamic durability. Study [12] proposes the design of a test bench to analyze the tense state of bolts. It is obvious that works such as [11, 12] are highly specialized and are only indirectly related to impact tests.

It is not always that a shock test system is a highly specialized structure. Some models of shock test machines are industrially produced. The world's leading manufacturers of shock systems are Lansmont Corporation (USA), ELSTAR Elektronik (Switzerland), and STI (China). Most shock machines produced by these companies operate on the principle of a table with a tested object dropped on an anvil with pads that determine the waveform and duration of impact. These are typically single-action systems, thereby failing to reproduce a series of pulses.

Thus, in most industrially produced shock test systems the necessary pulse waveform is created by braking devices. In some cases, the free drop of a table is used, less often – a table with an article is accelerated by a pneumatic drive [13] or an electromagnetic drive. However, the pneumatic drive does not make it possible to obtain pulses of short duration at high peak values of accelerations because of the long acceleration path. Electromagnetic-driven systems are only effective for low-mass objects.

The author of work [14] proposed a fundamentally new structure of shock systems that operate on the energy of elastic deformation of the compressed liquid and the shell of the vessel that contains it. Such machines make it possible to improve the versatility, manageability, and accuracy of impact tests. The main advantage of shock machines with a hydroelastic drive is the simplicity and reliability in repeating their work cycles. However, the level of automation of shock test systems based on a hydroelastic drive is low. Also unresolved are the issues related to the algorithmization of the process of shock test system operation and its pairing with modern microprocessor devices.

All this suggests that there is an unresolved task to design a universal test machine that could meet the following requirements:

– conducting impact tests with a series of pulses with the possibility of rapid change of waveform and frequency;
– no significant structural restrictions on the amplitude of impact acceleration;

– high-speed performance;

– the easy automation of shock tests.

The implementation of the study results would enhance the functionality of installations to conduct shock tests with a series of pulses, as well as improve the technological process manageability and increase the level of automation.

3. The aim and objectives of the study

The aim of this work is to design a shock test system prototype based on a hydroelastic drive that could enable intensive multiple impact influences on articles with a series of mechanical pulses of variable frequency, amplitude, and waveform.

To accomplish the aim, the following tasks have been set:

– to build a diagram of the projected shock system based on a hydroelastic drive and describe the operational algorithm to reproduce a series of impact pulses of various waveforms;

– to define the basic estimation parameters and construct a mathematical model of the proposed shock system with a hydroelastic drive;

– to build a transitional platform and impact acceleration functions and determine the damping factor optimal value for a braking device that minimizes the negative "emission" of the impact pulse.

4. The study materials and methods

Certain provisions used in the current research are borrowed from the theory of hydroelastic drive, which employs both changes in the volume of working chambers and the interaction of the liquid with the machine's working body to transmit energy. Years of research conducted at the Donbas State Engineering Academy (Kramatorsk, Ukraine) have confirmed the effectiveness of utilizing the energy of compressed liquid for impact action machines.

The principle of structure formation and the algorithm of shock test system operation are based on the theoretical and experimental studies of machines and processes for treating metals with pressure based on a hydroelastic drive.

The dynamic modes of the shock test system operation were studied using the mathematical engineering software Mathcad by Parametric Technology Corporation (USA).

The Runge-Kutta method with automatic step choice was applied to solve differential equations numerically. This method was implemented using the Rkadapt feature embedded into Mathcad.

5. The scheme and basic principles of a shock system's operation

Fig. 1 shows the scheme of the projected shock system based on a hydroelastic drive. The system includes a platform to host a tested article. The platform is connected to the rods of damping hydraulic cylinders.

The acceleration device or a charger chamber is a hydraulic cylinder filled with a working liquid, designed so that when the rod moves "down" the liquid is compressed under the piston. The working liquid filling the charging chamber in the hydroelastic drive is the mineral oil AMG-10 whose volumetric elasticity modulus E_0=1,680 MPa at a temperature of 20 °C. Other mineral oils with an elasticity module of up to 2,000 MPa can also be used.

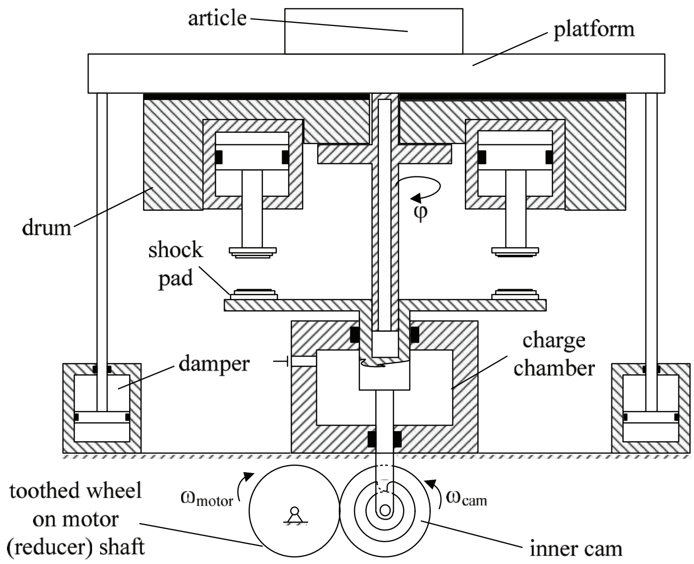

Fig. 1. Schematic of the shock test system

Reducing the volume of the charge chamber when the rod moves "down" at distance h (Fig. 2) deforms the liquid and vessel walls elastically. Thus, the hydroelastic spring is compressed or the chamber is charged.

The movement law of the charging chamber rod is formed by a special design cam (Fig. 3). The profile of the cam should ensure the smooth movement of the rod in line with the sinusoidal law during charging. The hydroelastic spring unclenching, on the contrary, demonstrates an impulsive character, that is, when the chamber is discharged, there is a sharp movement of the rod "up", which acquires at the highest point, determined by the run h, the speed ϑ_{max} (Fig. 2).

Thus, the cam tightens the rod of the brake device by squeezing the liquid according to law (1). Next, the wheel of the rod, rolling around the profile of the cam, breaks off, and the rod moves up under the influence of the force created by the expanding fluid.

$$f(\phi)=\frac{h}{2}\left(1-\cos(2\phi)\right), \qquad (1)$$

where $f(\varphi)$ is the law of rod movement in the charging process; h is the rod stroke; φ is the cam rotation angle.

The further movement continues until the liquid is fully expanded, that is until the rod is raised to the initial height h. This height is also the distance at which the piston is pulled inside the chamber, and is determined by the ledge on the cam profile. The rest of the cam's inner surface is in the shape of a circle and does not affect the further process of pulse formation.

The drum with interchangeable braking devices (Fig. 4) can rotate around its axis, assigning the impact position to a pair of braking devices with the same power characteristics required by the test program. Moreover, during the battery charging the drum manages to turn at 60°, assigning the impact position (in Fig. 4, it corresponds to 0°) to the braking device that is required by the program.

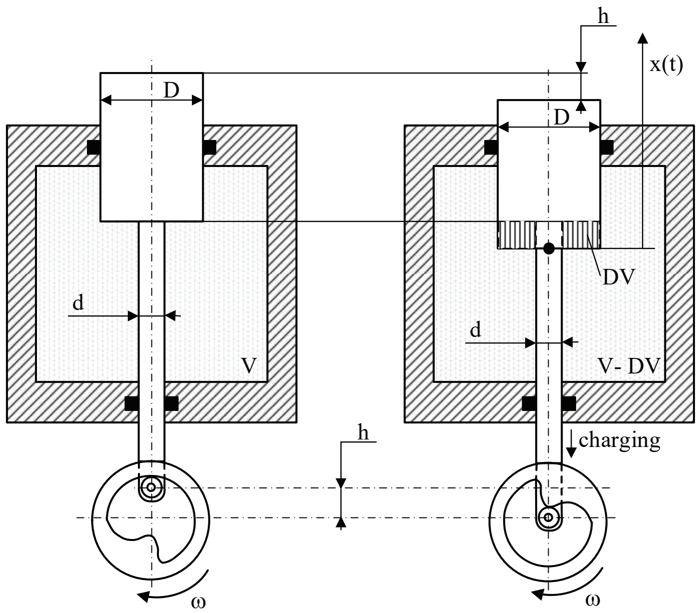

Fig. 2. Charge chamber before and after charging

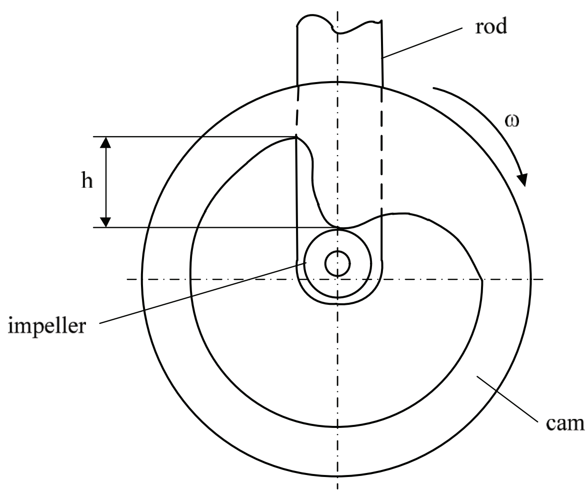

Fig. 3. Charger cam profile shape

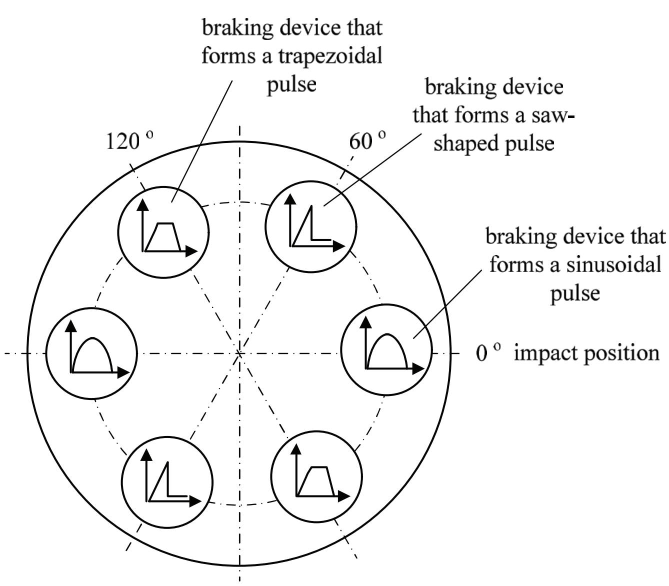

Fig. 4. Schematic of the drum with braking devices

The impact pulse is formed at each specific moment with the help of a pair of identical braking devices of a special design, which affects the waveform of the resulting pulse of impact acceleration. By collecting several brake hydraulic cylinders (devices) of different designs and combining them in a cassette (drum), we have the possibility to reproduce a series of impact pulses of different waveforms. This series of pulses is formed by turning the cassette at a discrete angle (for example, at 60°).

The shock system operates in the following way:

1. In the initial position, the charge chamber rod is stationary; the liquid is expanded.

2. The cam tightens the rod, squeezing the liquid under the piston of the charge chamber.

3. The wheel of the rod breaks off the ledge of the cam and the liquid is expanded.

4. The charging chamber rod, under the influence of fluid elasticity, moves "up" along with the charging platform and impact pads.

5. When the impact pads of the platform collide with the impact pads of brake devices, the rods of the braking devices, in line with the law of the preservation of the amount of movement, acquire the initial speed.

6. There is a process of forming an impact pulse by the braking devices, during which the drum with the braking devices moves together with the platform that hosts a tested article.

7. As a result of the influence of damping hydraulic cylinders, the oscillatory process fades rapidly, and the platform returns to its initial position.

8. The drum is reconfigured to a new impact pulse, turning at a certain angle according to the program.

9. The process is repeated from point 2 to point 8.

The main feature of the proposed structural scheme is that the reconfiguration of the new impact pulse occurs very quickly. That is, while the charge chamber cam executes a full rotation, the drum is reconfigured to a new impact pulse set by the test program. The approximate estimated frequency of the charge chamber cam is 60–120 rpm; the drum – about 300–400 rpm. Thus, a given shock system makes it possible to obtain a shock pulse repetition frequency of 1–2 Hz.

6. Constructing a mathematical model of the shock system

To draw up an estimation scheme of the projected shock system, consider a sketch in Fig. 1. By representing the shock system with some degree of idealization, one can conclude that it consists of the following elements:

– a large mass M, which is the mass of the platform, drum, brake devices' hulls, the rods of damping hydraulic cylinders, and an article;

– a small mass m, which is the mass of moving parts of brake devices (pistons with rods);

– a liquid or "one-way" spring of rigidity C_m, which links the brake device's rod and the platform;

– a damper installed between the brake device's rod and the platform, with a damping factor K_m, which is throttle grooves in the piston of the braking device;

– two mechanical or "two-side" springs of rigidity C_M, installed inside the damping hydraulic cylinders (between the platform and the base of the shock system);

– two dampers with a damping factor K_M, set between the platform and the base of the shock system.

When drawing up an estimation scheme of the shock system at the stage of pulse formation, the impact of the charge chamber was taken into consideration indirectly (through the initial speed of the brake devices' rods). Taking the above components into consideration, the estimation scheme of the projected shock system takes the form shown in Fig. 5.

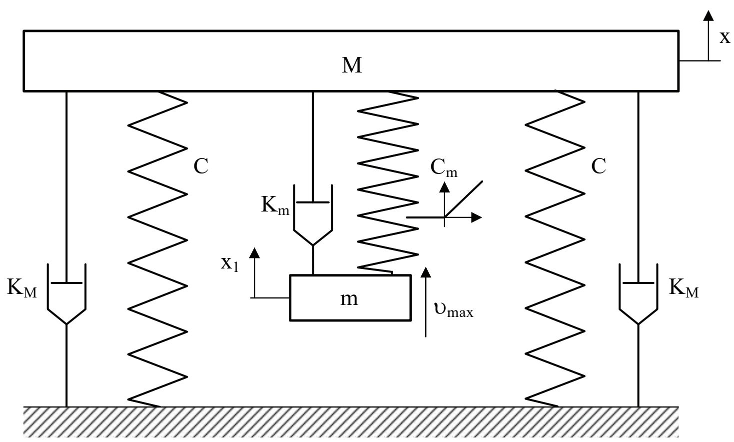

Fig. 5. Estimation scheme of the projected shock system

In order to construct motion equations, we applied those variables that fully describe the accepted estimation scheme. A working cycle of the projected shock system includes two periods: charging and pulse formation. Each period needs a separate mathematical notation.

At the charging stage, the most important parameter is the final value of the speed of the charge chamber rod. This value is determined by solving differential equation (2) that describes the movement process of moving parts of the charge chamber.

$$m_{cc}\ddot{x}_{cc} = p_0 S - \frac{E \cdot S^2}{V_1} x_{cc}, \tag{2}$$

where m_{cc} is the mass of moving parts (the rod and piston of the charge chamber); p_0 is the initial pressure in the charge chamber; S is an effective area; x_{cc} is the displacement of the rod of the charge chamber (zero is the uncharged state of the system, that is, the point of equilibrium); E is the fluid elasticity module; V_1 is the volume of the charge chamber.

Solving the above equation produced an expression to determine the initial velocity of the brake devices' rods (3).

$$\vartheta_{\max} = \xi \cdot \Omega \sqrt{\left(h + \frac{p_0 V_1}{ES}\right)^2 - \left(\frac{p_0 V_1}{ES}\right)^2}, \tag{3}$$

where Ω is the frequency of piston oscillations (4); ξ is the mass multiplicity factor (the ratio of the mass of the moving parts of the charge chamber m_{cc} to the mass m of the moving parts of the brake device).

$$\Omega = S\sqrt{\frac{E}{mV_1}}. \tag{4}$$

The displacement of the platform with a tested article is considered as the movement of an absolutely solid body exposed to the influence of the elasticity force $F_{el.mech}$ of mechanical springs, the elasticity force $F_{el.liq}$ of the working environment (liquid), the force of viscous friction $F_{V.FM}$, and the gravity force of the moving parts G_M. We constructed a differential motion equation mapped onto the x axis, directed vertically upwards along the movement of the platform with an article after assigning the initial speed to the brake devices' rods.

$$M\ddot{x} = F_{el.mech} - G_M - F_{el.liq} - F_{V.FM}. \tag{5}$$

The moving parts of the brake device (a piston with a rod) are under the influence of the force of viscous friction $F_{V.Fm}$, the elasticity force $F_{el.liq}$ of the working environment (liquid), and the gravity force G_M. We built a differential equation of the piston movement of the brake device with a rod after assigning the initial speed to the brake devices' rods mapped onto the x_1 axis, directed vertically upwards in the direction of the movement of the drum with the brake devices.

$$m\ddot{x}_1 = F_{el.liq} - F_{V.Fm} - G_m. \tag{6}$$

The elasticity force $F_{el.liq}$ of the mechanical springs of rigidity C_M

$$F_{el.mech} = 2C_M(\lambda_{st} - x), \tag{7}$$

where λ_{st} is the static deformation of mechanical springs under the influence of gravity force G_M and G_m.

The elasticity force $F_{el.mech}$ of the liquid spring of rigidity C_m

$$F_{el.liq} = C_m(\lambda_{st1} + x - x_1), \tag{8}$$

where λ_{st1} is the liquid spring static deformation under the force of gravity G_m.

A liquid spring is one-sided, that is, when compressed, the rigidity is C_m, and, when stretched, the rigidity increases by several orders of magnitude.

The force of viscous friction $F_{V.FM}$ acting on mass M

$$F_{V.FM} = K_M\dot{x}, \tag{9}$$

where $\dot{x}$ is the speed of movement of the platform with a tested article.

The force of viscous friction $F_{V.Fm}$ acting on mass m

$$F_{V.Fm} = K_m\dot{x}_1, \tag{10}$$

where $\dot{x}_1$ is the movement speed of the brake device's piston with a rod.

The gravity force acting on the body of mass M

$$G_M = Mg, \tag{11}$$

where g is the acceleration of free fall.

The gravity force acting on the body of mass m

$$G_m = mg. \tag{12}$$

Considering expressions (7) to (12), equations (5), (6) take the following form:

$$M\ddot{x} = 2C_M\left(\lambda_{st} - x\right) - Mg - C_m\left(\lambda_{st1} + x - x_1\right) - K_M\dot{x}. \quad (13)$$

$$m\ddot{x}_1 = C_m\left(\lambda_{st1} + x - x_1\right) - mg - K_m\dot{x}_1. \quad (14)$$

The values of static deformations λ_{st} and λ_{st1} have been determined. In the position of equilibrium, the following expressions hold: $2C_M\lambda_{st}=Mg+mg$ and $C_m\lambda_{st1}=mg$. Taking into consideration the values of static deformations, equations (13) and (14) take the following form:

$$M\ddot{x} = -2C_M x - C_m\left(x - x_1\right) - K_M\dot{x}. \quad (15)$$

$$m\ddot{x}_1 = C_m\left(x - x_1\right) - K_m\dot{x}_1. \quad (16)$$

The resulting equations (15) and (16) have made it possible to determine the trajectory of the platform and calculate the curve of the transition process by impact acceleration. In addition, they helped derive the optimal value of the damping factor for the brake device.

7. Determining the optimal damping factor value for the brake device

The dynamic mode of shock test system operation was calculated for the following initial data:

– the weight of the platform, drum, brake devices' hulls, the rods of damping hydraulic cylinders, and an article is 180 kg;

– the weight of the piston and rod of the brake device is 10 kg;

– the initial speed of the brake device rod is 10 m/s;

– the volume of the chamber of the brake device is 10^{-2} m^3;

– the liquid's elasticity module is 2,000 MPa;

– the rigidity of mechanical springs is 25,000 N/m;

– a damping factor for damping hydraulic cylinders is 100 kg/s;

– a damping factor for a brake device is 10,000 kg/s;

– the effective area of the brake device piston is $1.257 \cdot 10^{-3}$, which is necessary to calculate the rigidity of the liquid spring (17).

$$C_m = \frac{ES^2}{V}. \quad (17)$$

The platform's movement diagram based on our modeling is shown in Fig. 6. The effective part of the law of change in the impact acceleration is depicted in Fig. 7.

The resulting dependences illustrate the following acceptable results: the duration of transition processes is no more than 1 s, the platform's movements do not exceed 10 mm, the amplitude of the impact acceleration is 200 m/s^2.

An important parameter when designing a shock system is the magnitude of the negative "emission" of acceleration during the testing process. This value is important because an article on the reverse stroke may experience a larger overload than that on the direct stroke. When designing hydroelastic drives, it was found that the maximum value of the negative "emission" of acceleration should not exceed a quarter of the amplitude [15].

The issue of choosing the optimal damping factor value for braking devices was investigated using a model. At the same time, other estimation parameters were taken to be constant. The study results are given in Table 1 and shown in Fig. 8.

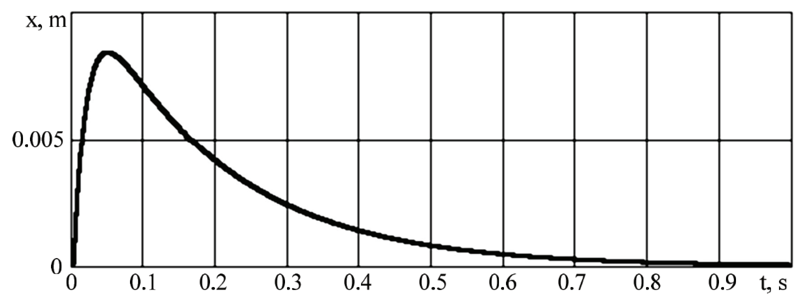

Fig. 6. Platform displacement

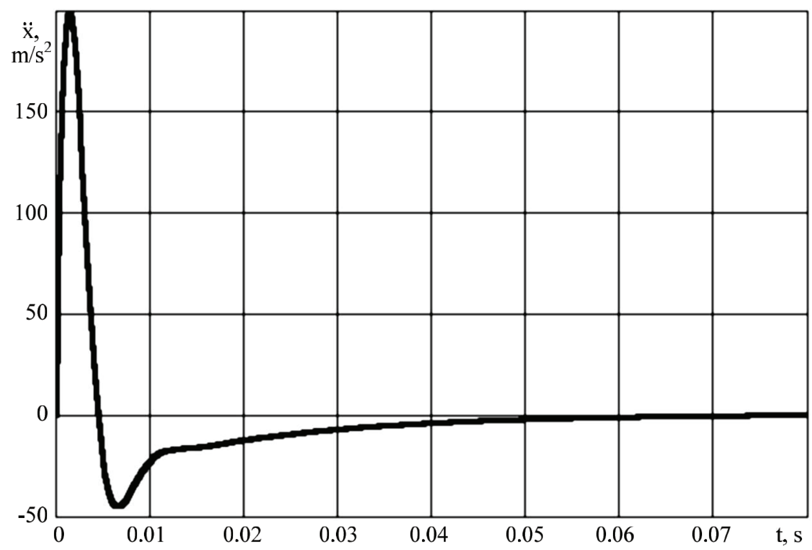

Fig. 7. The transition process for impact acceleration

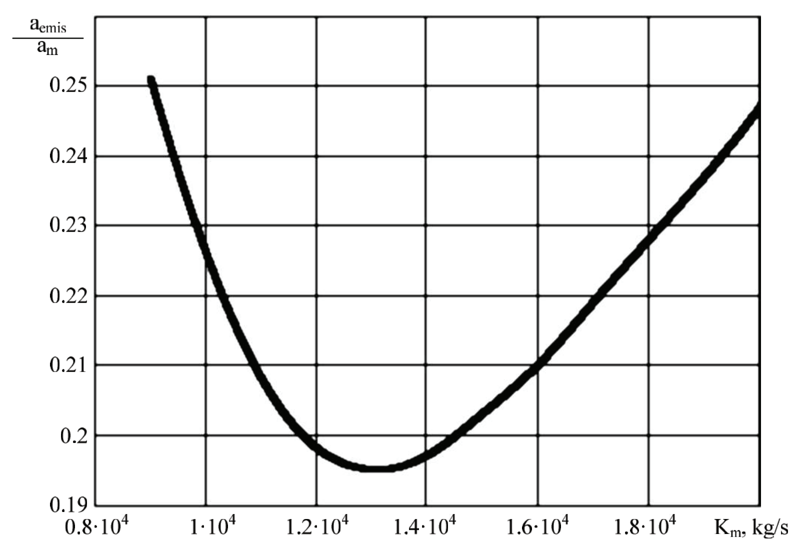

Fig. 8. Dependence of the negative "emission" to the amplitude of effective acceleration on K_m

Table 1

Experimental results involving a model

No. of entry	Damping factor for a brake device, kg/f	The ratio of negative «emission» to the amplitude of effective acceleration	The duration of the transition process (t.p.) before entering the zone of 1 %, s
1	0	1.037	diverging t.p.
2	100	1.016	diverging t.p.
3	200	0.998	diverging t.p.
4	300	0.981	0.4
5	400	0.964	0.35
6	500	0.947	0.28
7	600	0.931	0.26
8	700	0.915	0.24
9	800	0.899	0.22
10	900	0.883	0.2
11	1,000	0.868	0.18
12	2,000	0.731	0.12
13	3,000	0.617	0.096
14	4,000	0.524	0.081
15	5,000	0.445	0.077
16	6,000	0.381	0.069
17	7,000	0.328	0.062
18	8,000	0.285	0.057
19	9,000	0.251	0.054
20	9,500	0.237	0.053
21	10,000	0.226	0.052
22	10,500	0.216	0.051
23	11,000	0.208	0.05
24	12,000	0.198	0.047
25	13,000	0.195	0.045
26	14,000	0.197	0.043
27	15,000	0.203	0.042
28	16,000	0.21	0.039
29	17,000	0.219	0.037
30	18,000	0.228	0.035
31	19,000	0.237	0.033
32	20,000	0.247	0.032
33	21,000	0.257	0.03

Based on the dependence shown in Fig. 8, the optimal damping factor value for a brake device is 13,000 kg/s. In this case, a minimal value of the ratio a_{emis}/a_m=0.195<0.25 is achieved.

The practical value of the developed model is to determine the dynamic characteristics of a test installation, as well as to calculate the required structural and technological parameters. In particular, the variable parameters in the design of the physical model of the proposed shock test system prototype are the damping factors of brake devices K_m.

Thus, the interchangeable brake devices that form impact pulses should be selected based on the specifications for other structural elements of the test system (the mass, rigidity, and damping factors for the supports that hold a platform). For example, when designing a physical model of the shock system with the numerical values of the parameters presented above, the optimal value K_m=13,000 kg/s (according to the study results using a model).

8. Discussion of the study results and recommendations on the application of the proposed shock test system prototype

The proposed scheme of a universal shock test system has a series of structural advantages over known solutions [2–4]:

– the presence of a drum with special-design brake devices (Fig. 4) makes it possible to quickly change the waveform of impact pulses or alternate a series of pulses of different waveforms (trapezoidal, saw-shaped, sinusoidal);

– owing to the replacement of brake devices (Fig. 4) it is possible to expand the functionality of the shock system, as well as to modernize it without downtime and changes in the overall design;

– the rotation of the drum with brake devices (Fig. 1) and the charge chamber cam (Fig. 2) is performed by separate independent electric drives, which allows for complex impact testing programs with pulses of variable frequency, waveform, and amplitude;

– the high-speed performance is due to the speed of rotation of the charge chamber cam, restricted by the law of rod movement in the charging process (1).

Compared to the systems that use a consolidated drop [4] or electric motors to form an impact load [5, 6], the proposed solution has no significant limitations on the amplitude of impact acceleration. In the proposed shock system prototype, the acceleration amplitude is limited by the specifications of the charge chamber (the volume, wall material, working liquid) and the stroke of the rod during charging (1). Like other shock machines that employ a hydroelastic drive [14, 15], the proposed solution makes it possible to obtain huge impact accelerations (up to 8,000 m/s^2). The result of our modeling is the established amplitude of acceleration of 200 m/s^2 (Fig. 7).

The procedure of reproducing impact pulses takes into consideration the need to rotate the drum at 60° each time before striking another blow (Fig. 4). The proposed algorithm of shock system operation assumes the cyclical nature of technological operations. Thus, it can be easily interpreted as a software code for an industrial microcontroller that manages the impact test process.

The estimation scheme (Fig. 5) contains all the calculation parameters necessary for the mathematical notation of the shock system and the fabrication of a physical model.

Differential equations describing the movement of inertial masses (15), (16), as well as the differential equation of the charge chamber (2), form the basis of the mathematical model of the shock test system. In general, the developed model makes it possible to determine the following dynamic characteristics of the system: platform displacement (Fig. 6), the maximum speed of brake devices (3), impact acceleration (Fig. 7).

When fabricating a physical model of the proposed shock system, the mathematical model can be used to determine the optimal values of certain parameters. The structural parameters of interchangeable brake devices are variable (the most significant is the damping factor K_m).

Our study involving a model has established the dependence of the ratio of the negative "emission" of acceleration (on the return stroke of the platform) to the amplitude of effective acceleration on K_m (Table 1, Fig. 8). It is obvious that this dependence (Fig. 8) has a pronounced minimum, setting which would improve the efficiency of vibration treatment. The reduction in energy losses can be indirectly estimated by the reduced area under the acceleration curve in the $\ddot{x}$<0 region (Fig. 7). Thus, the mathematical model constructed could make it possible to design the shock system with a minimum negative "emission" of the acceleration of a hydraulic spring.

In general, the results of our studies allow them to be considered as a solution to the task of creating a universal shock test system prototype that could ensure intense multiple impact influences with a series of mechanical pulses of variable frequency, amplitude, and waveform.

The disadvantages of the proposed solution include:

– the interchangeable brake devices that provide for the required pulse waveform have the structural features and limitations regarding possible braking laws;

– the contact areas of the impact pads, which transmit the pulse from the charge chamber to the brake device, as well as the charge chamber cam, will be exposed to increased wear;

– in practice, it is necessary to accurately synchronize the operation of the charge chamber and the rotary drum, otherwise, it is possible that emergency may occur when the impact pulses are applied outside the impact pads;

In the practical implementation of a shock test system prototype, there may occur restrictions imposed by the electric drive of drum rotation and the electric drive of cam rotation (maximum moment, speed, current). In addition, to accurately synchronize the operation of the charge chamber and the rotary drum, the electric drive must be adjustable and must be equipped with an appropriate microcontroller control system.

The design of an automated electric drive for the drum and charge chamber cam may be a promising direction of further research, which is not considered in this work. This area is the most important as the structural benefits of the proposed solution cannot be implemented without an automated electric drive and a microcontroller control system.

References

1. Sisemore, C., Babuska, V. (2020). The Science and Engineering of Mechanical Shock. Switzerland: Springer Nature Switzerland AG, 362. doi: https://doi.org/10.1007/978-3-030-12103-7
2. Engel, C., Herald, S., Davis, S., Dean, S. (2006). Mechanical Impact Testing: Data Review and Analysis. Journal of ASTM International, 3 (8), 13538. doi: https://doi.org/10.1520/jai13538
3. Iskovych-Lototskyi, R. D., Obertiukh, R. R., Sevostianov, I. V. (2006). Protsesy ta mashyny vibratsiynykh i vibroudarnykh tekhnolohiy. Vinnytsia: UNIVERSUM-Vinnytsia, 291.
4. Wang, J., Zhang, J. (2019). Research on High-Power and High-Speed Hydraulic Impact Testing Machine for Mine Anti-Impact Support Equipment. Shock and Vibration, 2019, 1–12. doi: https://doi.org/10.1155/2019/6545980
5. Echevarria, I., Lasa, J., Casado, P., Dominguez, A., Eguizabal, I., Lizeaga, M. et. al. (2015). Test bench for helicopter electro mechanical actuation system validation: Design and validation of dedicated test bench for aeronautical electromechanical actuators. 2015 IEEE International Conference on Industrial Technology (ICIT). doi: https://doi.org/10.1109/icit.2015.7125151
6. Zhao, W., Song, Q., Liu, W., Ahmad, M., Li, Y. (2019). Distributed Electric Powertrain Test Bench With Dynamic Load Controlled by Neuron PI Speed-Tracking Method. IEEE Transactions on Transportation Electrification, 5 (2), 433–443. doi: https://doi.org/10.1109/tte.2019.2904652
7. Syrigos, S. P., Karatzaferis, I. C., Tatakis, E. C. (2013). Four-quadrant fully controlled mechanical load simulator. 2013 15th European Conference on Power Electronics and Applications (EPE). doi: https://doi.org/10.1109/epe.2013.6634640
8. Song, Q., Liu, W., Zhao, W., Li, Y., Ahmad, M., Zhao, L. (2019). Road Load Simulation Algorithms Evaluation Using A Motor-in-the-loop Test Bench. 2019 IEEE Transportation Electrification Conference and Expo, Asia-Pacific (ITEC Asia-Pacific). doi: https://doi.org/10.1109/itec-ap.2019.8903852
9. Achour, T., Pietrzak-David, M. (2012). An experimental test bench for a distributed railway traction mechanical load emulator. IECON 2012 - 38th Annual Conference on IEEE Industrial Electronics Society. doi: https://doi.org/10.1109/iecon.2012.6388725
10. Ruan, J. J., Lockhart, R. A., Janphuang, P., Quintero, A. V., Briand, D., de Rooij, N. (2013). An Automatic Test Bench for Complete Characterization of Vibration-Energy Harvesters. IEEE Transactions on Instrumentation and Measurement, 62 (11), 2966–2973. doi: https://doi.org/10.1109/tim.2013.2265452
11. Shuai, Z., Zhaokun, X., Haisong, J. (2011). The Development of a Test Bench for the Dynamic Strength and Durability of Auto Front Axle Rocker. 2011 Third International Conference on Measuring Technology and Mechatronics Automation. doi: https://doi.org/10.1109/icmtma.2011.765
12. Yan, T. (2012). Construction of cylinder head vibration bolts test bench and stress analysis of the bolts. 2012 2nd International Conference on Consumer Electronics, Communications and Networks (CECNet). doi: https://doi.org/10.1109/cecnet.2012.6201582
13. Chivu, C. (2014). Pneumatic Driving in Material Handling Systems. Recent, 15 (3 (43)), 155–159.
14. Roganov, L. L. (2011). Sovershenstvovanie tehnologiy i mashin dlya raznyh otrasley mashinostroeniya na osnove razvitiya gidrouprugih i klinosharnirnyh mehanizmov. Obrabotka materialov davleniem, 2 (27), 163–168.
15. Rohanov, L. L., Rohanov, M. L., Hranovskyi, A. Ye. (2013). Udarni stendy na bazi hidropruzhnoho pryvodu. Kramatorsk: DDMA, 161.