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CREATION OF DAMPING ALLOYS WITH OPTIMUM PHYSICAL-MECHANICAL PROPERTIES FOR GEOLOGICAL EXPLORATION EQUIPMENT PARTS

Abstract. One of the effective methods of reducing noise is to quench it at the source of occurrence. Such methods include replacement of percussion mechanisms with without impact ones, replacement of gears with V-belts, etc. The most acceptable method of reducing noise at the source of occurrence is the use of damping materials. Non-metals (plastics, wood, polyethylene, etc.), non-ferrous metals can be used. However, the most relevant to reduce noise at the source of occurrence in geo-exploration production is the use of iron-based damping metallic materials. In this work, the task was to assess the acoustic and vibration characteristics of standard steels 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25Х2ГНТА, which are used to manufacture parts of geo-exploration equipment, new damping alloys GGR-1, GGR-2, GGR-3. The choice of the chemical compound of the GGR-1 alloy made it possible to obtain an anti-vibration damping alloy having a quenching martensite structure. It is recommended to use it for exploration equipment parts (drilling rigs, drill rod, drilling blade, adapters for drilling head, drill pipe, etc.).

Keywords: noise, vibration, damping, geological exploration, mechanical properties, acoustic and vibration characteristics, chemical compound, equipment, experimental alloys.

Introduction. The noise of impact origin is the most common and harmful industrial factor of industry.

One of the high-performance and efficient production is geological exploration production. The equipment for geological exploration production is characterized by intense vibration and increased radiated noise (drilling rigs, drill rod, drilling blade, adapters to the drill head, drilling pipe, etc.). Industrial noise and sound vibration worsen working conditions, negatively affect the health of workers. Intense vibration is the cause of damage to the structures of machines and mechanisms and reducing their service life. All these problems pose to designers and technologists the task of reducing the parameters of noise and vibration.

Frequently, geological explosive production is dominated by percussion and final noise, characterized as the most harmful to workers health. At short impulses the likelihood of hearing loss increases.

Very often, intense noise is emitted by parts made of 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА steels (standard steels) (gear wheels, gear rims, connecting rods, shaped castings and other parts of geological explosive equipment) [1, 13].

One of the effective methods of reducing noise is to quench it at the source of occurrence. Such methods include replacement of percussion mechanisms with without impact ones, replacement of gears with V-belts, etc. The most acceptable method of reducing noise at the source of occurrence is the use of damping materials. Non-metals (plastics, wood, polyethylene, etc.), non-ferrous metals can be used.

However, the most relevant to reduce noise at the source of occurrence in geo-exploration production is the use of iron-based damping metallic materials.

The aim of the work is the development and research of new grades of damping steels for castings, reducing the noise of impact origin, generated in parts and assemblies during the operation of geological prospecting equipment.

Research objectives:

- to evaluate the vibration and physical-mechanical properties of well-known steels (15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА) used for parts subjected to shock loads;

- to develop new alloys for parts subjected to shock loads, differing in chemical composition, but not inferior in terms of mechanical and technological characteristics to known standard grades of alloyed steels.

As an object, both standard and newly smelted alloys were considered. The purpose of these steels is given in table 1. Table 2 presents the chemical compounds of the investigated steels. The acoustic characteristics (sound level, level of sound pressure) and vibration (vibration acceleration level, overall vibration acceleration level) characteristics of the alloys were investigated.

For the study, standard alloy steels were selected for castings of grades 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА and melted alloyed alloys GGR-1, GGR-2 and GGR-3, whose mechanical characteristics are shown in table 3.

In this work, the task was to assess the acoustic and vibration characteristics of standard steels 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25Х2ГНТА, which are used to manufacture parts of geo-exploration equipment, new damping alloys GGR-1, GGR-2, GGR-3.

Standard alloyed casting steels 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА in the form of a plate were investigated.

The damping ability of metallic materials is characterized by a combination of vibration and physical-mechanical characteristics, such as vibration acceleration level, internal friction, electrical resistivity, density, shear modulus, Young's modulus and a number of metallographic features. In the present study, a series of experiments were aimed at establishing patterns that determine the relationship of structurally sensitive factors and microstructure with the optimization parameter – the level of vibration acceleration of low-alloy structural steels, the composition of which was specified by the experiment planning matrix.

Experimental alloys were smelted in a crucible induction furnace with a capacity of 12 kg with the main lining. The source material was sheet metal of steel 10. Doping was carried out with 97,6% metallic manganese, 77,5% FeSi and 99,98% metallic nickel. Carbonaceous additive served as synthetic cast iron with a carbon content of 3,9%. Steel was cast into a metal mold with dimensions of 210x115x115 mm.

Samples before forging were heated in a laboratory oven to a temperature of 1200 °C with a holding time of 1 hour. Ingots were forged using a forged hammer to stripes with final dimensions of 700x90x10 (12) mm. After each pass, the strips were placed in an oven to achieve a temperature of 1200 °C.

One of the objectives of this work is the development of new damping metallic materials based on iron. In this regard, by adding alloying elements to the chemical compound of standard steel grades, new alloys with enhanced damping properties were obtained. The principles of alloying of alloys in the work are based on the study of the phase diagrams of Fe – C, Fe – Si, Fe – Mn, Fe – Cr, Fe – La, Fe – Ca, Fe – V, Fe – Ni. State diagrams determine in equilibrium the phase composition of the alloy depending on the temperature and concentration of the components and allow qualitatively characterizing many physical-chemical, mechanical and technological properties of the alloys.

Casting was made in the chill mold. Casting in the chill mold compared with the sand form has several advantages: the relative durability of the form and accelerated cooling of the casting in it, a sharp reduction or almost complete elimination of the consumption of molding materials; an increase in the removal from the molding site by 2-6 times, an increase in labor productivity by 1.5-6 times, a decrease in surface roughness, an increase in the accuracy of castings, an increase in the density of castings, a reduction in profit margins and often even their elimination [15].

Based on the analysis of installations for the study of vibration (level of vibration acceleration, total level of vibration acceleration) properties of the alloys, the device "KazNTU" -2 was selected for a comprehensive study of the vibration properties of plate steel samples [6] (figure 1).

Table 1 – Purpose and general characteristics of standard steels [7]

Steel	Purpose
15ХГН2ТА 15Х2ГН2ТА 15Х2ГН2ТРА	Disks, sprockets, gears, connecting rods, crosses, forks, fingers, gears, shafts, cam couplings, covers and other parts of geological exploration techniques.
20ХГНТР 25 Х2ГНТА	Responsible details of exploration equipment, gears, crosses, levers, etc.

Table 2 – The chemical compound of the investigated steels

Mark of steels, alloys	Chemical compound, %								
	C	Si	Mn	Cr	Ni	Ti	S	P	Fe
							no more		
15ХГН2ТА	00,13-0,18	00,17-0,37	10,7-1,0	≤0,7-1,0	–	–	0,035	00,035	The rest
20ХГНТР	00,18-0,24	00,17-0,37	10,8-1,1	≤0,4-0,7	–	–	0,035	00,035	The rest
GGR-1	00,2	00,2	00,8	11,0	00,35	00,15	0,045	00,04	The rest
GGR-2	00,3	00,3	00,75	11,0	00,45	00,18	0,045	00,04	The rest
GGR-3	,0,3	00,3	11,0	11,0	00,55	00,20	0,045	00,04	The rest

Table 3 – The mechanical properties of the investigated steels

№	Mark of steel	$\sigma_в$	Impact strength KCU, J/cm^2	δ_5	ψ	$\sigma_т$, MPa
				%		
				not less		
1	15ХГН2ТА	1330	127	11,5	59,5	1180
2	20ХГНТР	1200	80	9	50	1000
3	15Х2ГН2ТА	1380	127	12	58	1190
4	15Х2ГН2ТРА	1320	120	14	62	1190
5	GGR-1	1400	140	13	25	1300
6	GGR-2	1350	145	14	30	1250
7	GGR-3	1400	140	12	35	1320

The installation works as follows. The ball-drummer 6 was installed on an inclined plane 5. The ball-drummer 6 rolls down from the inclined plane 5 and makes a free fall into the geometric center of the plate sample 3. The ball-drummer 6 rebounds from it and enters the receiver of the balls 11. The noise from the impact of the striker ball 6 and sample 3 is recorded by the OCTAVE-101A sound level meter 12. Sample (plate) 3, oscillating in the interweaving of nylon yarns 1 creates a vibration, which is estimated by the device model "Bruel & Kjer" model 22048. The tension of the sample nylon threads 1 is always constant, since the load 10 controls this tension. The height of the fall of the ball can be changed using the screw mounting rack drummer 15. The entire system of mounting the sample 3 and the ball-striker 6 is installed on the frame 2, which with the help of the uprights 13 is located at a certain height above the floor.

In the measurements, steel (ШХ15) impact balls of the following diameters were used: 9,5 mm; 12,7 mm; 15,2 mm; 15,8 mm and 18,3 mm (mass of balls, shock, respectively: 2,5 g; 5 g; 9 g and 25 g).

At the installation, steel lamellar (50x50x5 mm) specimens were examined.

The vibration acceleration levels were measured in the range of 31,5-31500 Hz, the total vibration acceleration level - according to the "Lin" characteristic.

The mass of the ball, the density of the sample, the distance from the point of impact to the sample, the thickness of the sample are interrelated according to [8]:

$$m \langle 4{,}6 \cdot \rho \cdot l \cdot h^2, \tag{1}$$

where m – mass of lamellar-sample, g; ρ – density of material lamellar-sample, g/cm^3; l – distance from impact point to nearest edge of sample plate, cm; h – thickness of material lamellar-sample, cm.

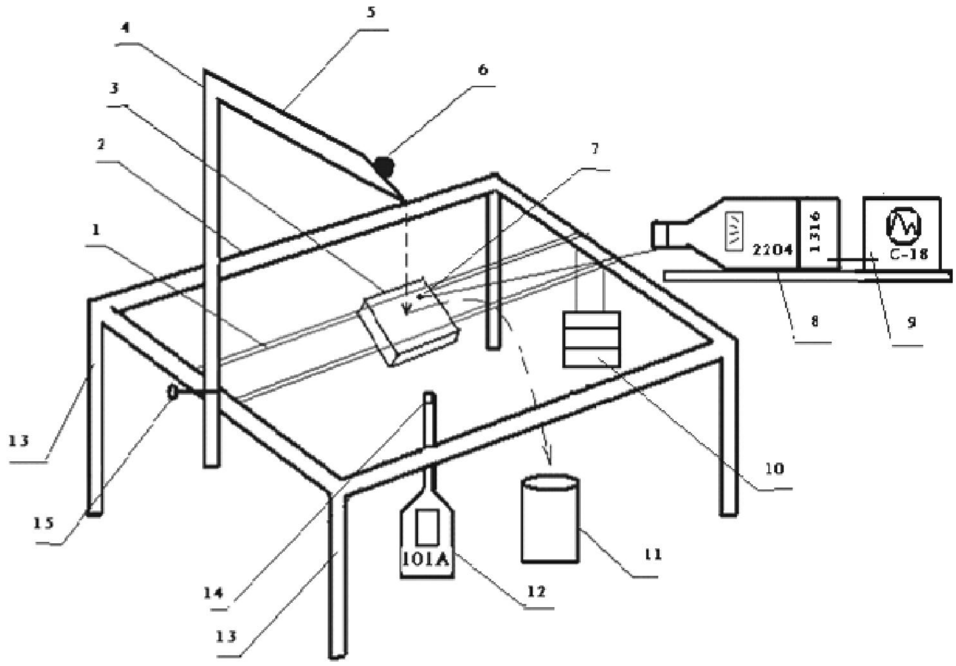

Figure 1 – Device "KazNTU-2" for the study of the vibration properties of solid plate (lamellar) samples [6, 18]:
1 – nylon thread; 2 – frame; 3 – lamellar (50x50x5 mm) sample; 4 – frame stand; 5 – inclined plane; 6 – ball-drummer;
7 – Vibration sensor of "Bruel&Kjer" vibrometer model 2204; 8 – vibrometer "Bruel&Kjer" model 2204;
9 – oscilloscope C-18; 10 – load; 11 – ball receiver; 12 – noisemeter "OCTAVE -101A"; 13 – frame stands;
14 – microphone of noise meter "OCTAVE -101A"; 15 – fastening pin rack screw

The correction for the change in the vibration signal from atmospheric pressure was carried out using a pistonphone of the brand PF-101. The air temperature and humidity in the laboratory were kept constant. Vibration measurements were found as the average of five measurements.

We also carried out mathematical processing of the experimental results and the determination of confidence intervals in accordance with the method [9]. Before starting, the adjustment of the measuring path was carried out by checking the sound pressure levels of the reference sample.

Vibration characteristics of the investigated standard steels 15ХГН2ТА, 20ХГНТР, 15Х2ГН2ТА and new damping alloys GGR-1, GGR-2 and GGR-3 are presented in tables 4, 5 and in figures 2, 3.

Table 4 presents the vibration characteristics of the samples (plates with size 50x50x5 mm) from standard steel 15ХГН2ТА, 20ХГНТР, 15Х2ГН2ТА, 15ХГН2ТРА after collision with impact balls with diameters d = 9,5 mm, d = 12,7 mm, d = 15,5 mm and d = 18,3 mm of steel ШХ15.

The nature of vibration acceleration levels (VAL) of standard steel 15ХГН2ТА, 20ХГНТР, 15Х2ГН2ТА, 15Х2ГН2ТРА has the following features:

- vibration acceleration levels of the samples studied vary in the range of 61-128 dB;
- maximums of vibration acceleration levels are observed at frequencies of 31,5 Hz, 63 Hz and 125 Hz;
- minimum levels of vibration accelerations of samples are typical for frequencies of 250-31500 Hz (61-68 dB);
- maximum values of the vibration accelerations of the compared samples are characteristic in collisions with a hammer-ball with a diameter of d = 18,3 mm;
- minimum values of the vibration acceleration levels of the compared samples are typical in collisions with impact balls with diameters d = 9,5 mm and d = 15,2 mm;
- maximum levels of vibration acceleration according to the "Lin" characteristic for samples 15ХГН2ТА, 20ХГНТР, 15Х2ГН2ТА, 15Х2ГН2ТРА are observed during collision with impact balls with diameters d = 12,7 mm and d = 18,3 mm (125-129 dB).

In the study of the sound emission characteristics of alloys, amplitude-dependent damping of vibration acceleration was found. Amplitude-dependent damping of vibration acceleration (ADDV) consists in the fact that when a ball striking a larger mass hits a sample, it generates a level of vibration acceleration of a smaller value than when a ball striking a smaller mass collides.

Table 4 – Vibration characteristics of standard steels (plates 50x50x5 mm) after casting

Mark of steel	Diameter of ball-drummer, d, mm	Vibration acceleration levels, dB, in octave bands with geometric average frequencies, Hz											TVAL, dB
		31,5	63	125	250	500	1000	2000	4000	8000	16000	31500	
15ХГН2ТА	9,5	85	104	86	65	64	63	61	64	65	70	68	105
	12,7	90	106	88	68	69	65	65	63	68	69	70	107
	15,2	88	106	90	67	75	68	67	62	67	68	71	108
	18,3	91	109	88	68	77	69	68	63	68	69	72	110
20ХГНТР	9,5	91	108	85	71	75	68	65	64	61	62	72	110
	12,7	91	114	82	73	76	69	69	65	62	64	77	115
	15,2	93	116	88	75	77	70	70	66	64	68	78	118
	18,3	94	122	85	78	75	72	72	67	68	70	80	124
15Х2Н2ТА	9,5	88	104	78	62	69	70	62	67	66	65	64	106
	12,7	85	109	89	72	71	69	66	68	67	64	66	110
	15,2	86	117	91	66	74	65	68	68	69	66	68	118
	18,3	90	120	96	67	73	64	69	70	70	66	71	122
15Х2ГН2ТРА	9,5	91	117	92	70	66	71	60	62	66	71	72	118
	12,7	97	122	96	71	72	68	64	69	67	72	73	125
	15,2	98	126	98	72	77	66	65	70	67	73	74	128
	18,3	98	128	100	74	78	65	69	72	68	74	77	129

In steel 15ХГН2ТА ADDV it is observed at frequencies of 31,5 Hz (drummers 12,7 mm and 15,2 mm); 63 Hz (drummers 12,7 mm and 15,2 mm); 125 Hz (drummers 15,2 mm and 18,3 mm); 250 Hz (drummers 12,7 mm and 15,2 mm); 4000 Hz (drummers 9,5 mm and 12,7 mm); 8000 Hz (drummers 12,7 mm and 15,2 mm); 16,000 Hz drummer 9,5 mm; 12,7 mm; 15,2 mm; 18,3 mm).

In steel 20ХГНТР ADDV is observed at frequencies of 31,5 Hz (drummers 9,5 mm and 12,7 mm); 125 Hz (drummers 9,5 mm; 12,7 mm; 15,2 mm and 18,3 mm); 500 Hz (drummers 15,2 mm and 18,3 mm).

In steel 15Х2ГН2ТА ADDV it is observed at frequencies of 1000 Hz (drummers 9,5 mm and 12,7 mm); 500 Hz (drummers 15,2 mm and 18,3 mm); 4000 Hz (drummers 12,7 mm and 15,2 mm); 16,000 Hz (drummers 9,5 mm; 12,7 mm; 15,2 mm and 18,3 mm).

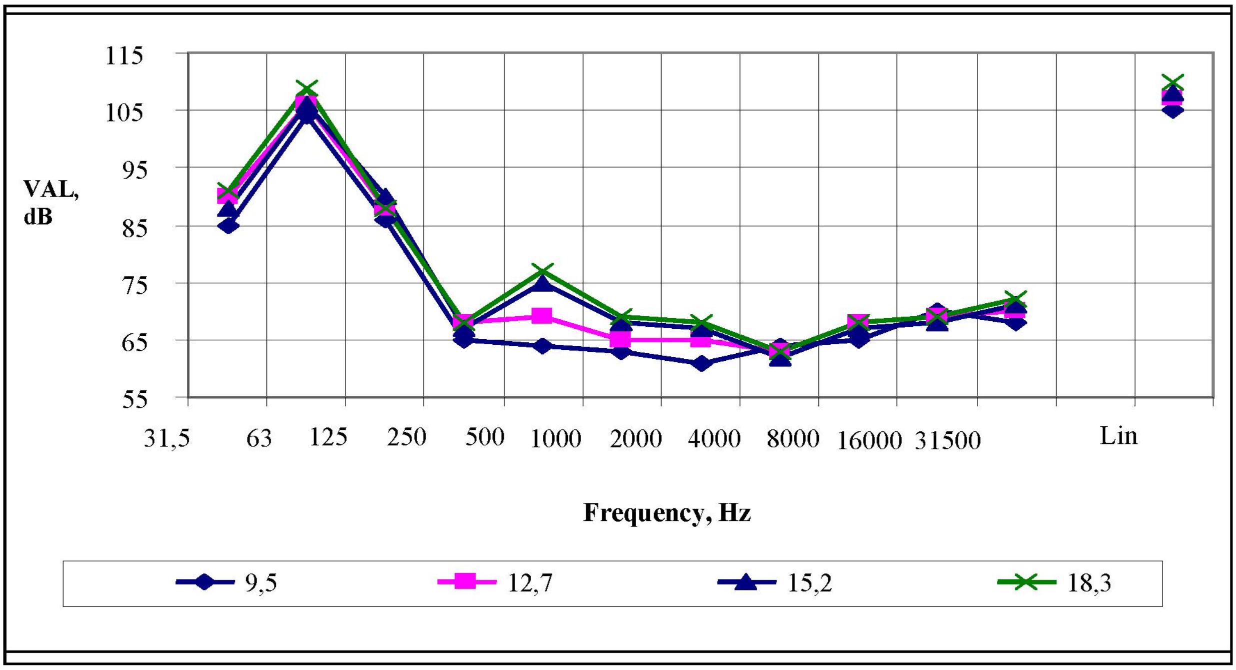

Figure 2 – Characteristics of vibration accelerations of the sample 15ХГН2ТА at impact

In steel 15Х2ГН2ТРА ADDV it is observed at frequencies of 31,5 Hz (drummers 15,2 mm and 18,3 mm); 1000 Hz (drummers 12,7 mm; 15,2 mm and 18,3 mm); 8000 Hz (drummers 12,7 mm and 15,2 mm).

Figure 2 shows the characteristics of the vibration accelerations of the 15ХГН2ТА sample at impact. In accordance with figure 2, it is maximum at a frequency of 31,5 Hz, 63 Hz, 125 Hz at impact 15ХГН2ТА sample with a hammer-ball with a diameter of d = 15,2 mm and 18,3 mm VAL = 90-109 dB, and minimal with a 15ХГН2ТА impact with a ball-drummer with a diameter of d = 15,2 mm VAL = 62 dB.

In figure 3 shows the characteristics of the vibration accelerations of the developed alloy sample GGR-1 at impact.

Table 5 – Vibration characteristics of the developed steels (plates 50x50x5 mm) after casting

Mark of steel	Diameter of ball-drummer, d, mm	Vibration acceleration levels, dB, in octave bands with geometric average frequencies, Hz											TVAL, dB
		31,5	63	125	250	500	1000	2000	4000	8000	16000	31500	
GGR-1	9,5	80	99	81	65	65	62	62	65	66	68	64	100
	12,7	82	100	79	66	68	64	65	68	69	70	65	102
	15,2	83	104	82	68	70	66	67	69	71	71	66	105
	18,3	85	107	83	70	72	67	68	71	72	73	68	108
GGR-2	9,5	81	100	80	66	66	64	65	64	68	70	67	102
	12,7	83	102	82	68	69	65	67	66	70	71	68	103
	15,2	85	106	83	69	71	66	67	66	70	71	69	108
	18,3	86	108	85	70	73	68	69	67	71	78	70	110
GGR-3	9,5	80	109	79	67	65	63	66	66	69	71	68	110
	12,7	79	111	79	69	66	65	67	68	70	73	69	112
	15,2	82	113	80	70	67	68	69	70	72	74	70	115
	18,3	85	117	82	72	69	70	71	71	73	75	72	119

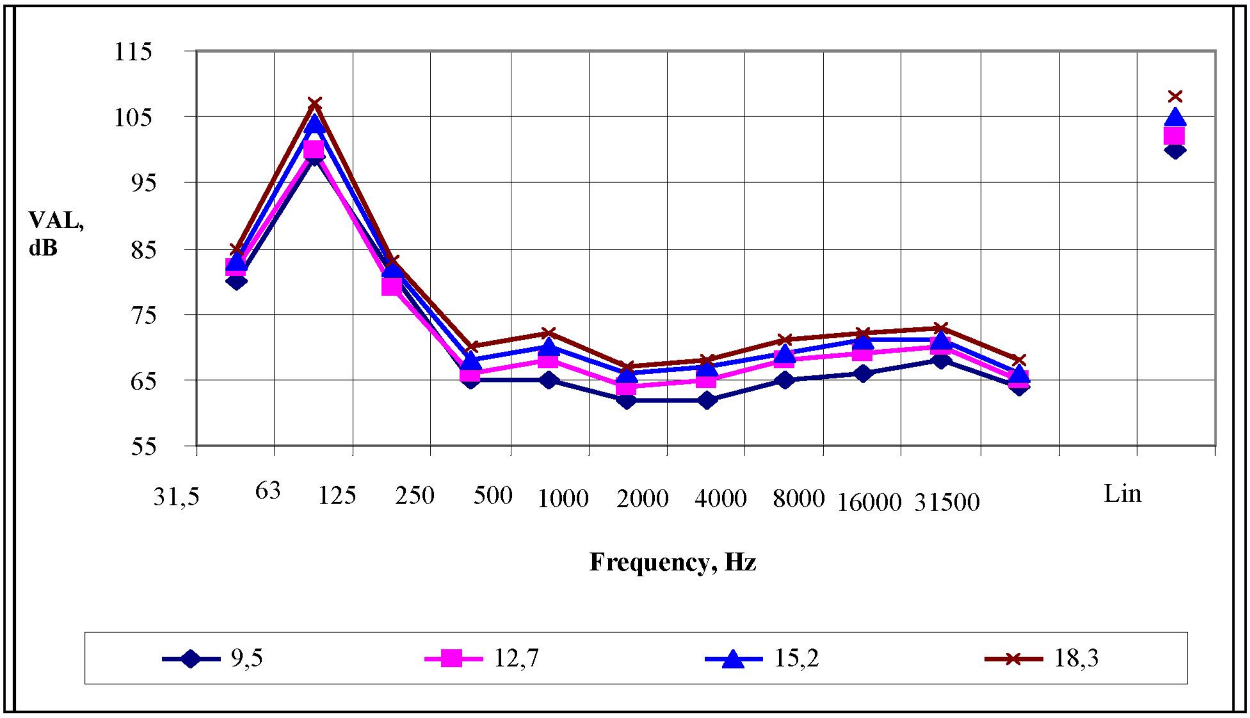

Figure 3 – Characteristics of vibration accelerations of the sample GGR-1 at impact

Conclusion. The choice of the chemical compound of the GGR-1 alloy made it possible to obtain an anti-vibration damping alloy having a quenching martensite structure. It is recommended to use it for exploration equipment parts (drilling rigs, drill rod, drilling blade, adapters for drilling head, drill pipe, etc.).

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ГЕОЛОГИЯЛЫҚ БАРЛАУ ЖАБДЫҚТАРЫНЫҢ БӨЛШЕКТЕРІНЕ АРНАЛҒАН ТИІМДІ ФИЗИКАЛЫҚ-МЕХАНИКАЛЫҚ ҚАСИЕТТЕРГЕ ИЕ ДЕМПФЕРЛІК ҚОРЫТПАЛАРДЫ ДАЙЫНДАУ

Шуды бәсеңдетудің ең тиімді жолдарының бірі ол шуды пайда болу көзінде төмендету. Мұндай әдістерге соққылы механизмдерді соққысыз түрге алмастыру, тісті берілістерді ременді берілістерге ауыстыруды жатқызуға болады. Шуды пайда болу көзінде бәсеңдету тәсілдерінде демпферлік материалдарды қолдану ұтымды болып табылады. Бейметалдар да (пластмассалар, ағаш, полиэтилен және т.б.), түсті металдар қолданылуы мүмкін.

Бірақ та, геологиялық барлау саласында шуды пайда болу көзінде бәсеңдету үшін темір негізіндегі демпферлік металды материалдарды қолдану өзекті болып табылады.

Жұмыста келесі негізгі міндеттер қойылды: геологиялық барлау жабдықтарының бөлшектерін дайындауда қолданылатын 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА стандартты болаттардың, сонымен қатар ГГР-1, ГГР-2, ГГР-3 жаңа демпферлі қорытпаларының акустикалық және дірілдік қасиеттерін бағалау көзделді.

ГГР-1 қорытпасының химиялық құрамы шынықтыру арқылы мартенситті құрылыммен дірілге қарсы демпферлік қорытпа алуға мүмкіндік берді. Аталған қорытпаны геологиялық барлау жабдықтары бөлшектеріне (бұрғылау құрылғылары, бұрғылау штангасы, бұрғылау күрегі, бұрғылау басының өткізгіштері, бұрғылау құбыры және т.б.) қолдануға ұсынылады.

Түйін сөздер: шу, діріл, демпферлеу, геологиялық жабдықтар, механикалық қасиеттер, акустикалық және дірілдік қасиеттер, химиялық құрам, жабдық, тәжірибелік қорытпалар.

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СОЗДАНИЕ ДЕМПФИРУЮЩИХ СПЛАВОВ С ОПТИМАЛЬНЫМИ ФИЗИКО-МЕХАНИЧЕСКИМИ СВОЙСТВАМИ ДЛЯ ДЕТАЛЕЙ ГЕОЛОГОРАЗВЕДОЧНОГО ОБОРУДОВАНИЯ

Аннотация. Одним из эффективных методов снижения шума является гашение его в источнике возникновения. К таким методам следует отнести замену ударных механизмов на безударные, замену зубчатых передач на клиноременные и т.п. Наиболее приемлемым методом снижения шума в источнике возникновения является использование демпфирующих материалов. Могут быть использованы неметаллы (пластмассы, древесина, полиэтилен и др.), цветные металлы.

Однако наиболее актуальным для снижения шума в источнике возникновения в геологоразведочном отрасле является применение демпфирующих металлических материалов на железной основе.

В настоящей работе была поставлена задача – оценить акустические и вибрационные характеристики стандартных сталей 15ХГН2ТА, 15Х2Н2ТА, 15Х2ГН2ТРА, 20ХГНТР, 25 Х2ГНТА , которые используются для изготовления деталей геологоразведочного оборудования, новых демпфирующих сплавов ГГР-1, ГГР-2, ГГР-3.

Выбор химического состава сплава ГГР-1 позволил получить демпфирующий антивибрационный сплав, имеющий структуру мартенсита закалки. Его рекомендуется использовать для деталей геологоразведочного оборудования (буровые установки, буровая штанга, буровая лопатка, переходники на буровую головку, труба буровая и др.).

Ключевые слова: шум, вибрация, демпфирование, геологоразведочные оборудования, механическое свойства, акстические и вибрационные характеристики, химический состав, установка, экспериментальные сплавы.

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DYNAMIC ANALYSES OF A CLUTCH OF CRANK PRESS

Abstract. The paper studies the dynamic of a clutch of crank press. At present, the dynamic study of clutch of the crank presses, with account of interaction with other blocks, is a priority. The crank press contains movable parts and assemblies, the mass of which is from one hundred kilograms to several tons. These parts and assemblies are connected cyclically by clutch of crank press with high speeds and they are subject to large dynamic loads. To simulate and analyze the movement of crank press with clutch, a software package: SimulationX is used. SimulationX is a software package for modeling and analyzing the dynamics and kinematics of cars, industrial equipment, electric, pneumatic and hydraulic drives, hybrid engines, etc. As a result of dynamic calculation, important dynamic parameters of the crank press clutch and working slide are determined. It is shown that dynamic loads sharply increase almost in all blocks of the crank press when the clutch is switched on.

Key words: dynamics, crank press, clutch, slide, moment, oscillations, SimulationX.

Introduction. Crank press is a machine with a slide-crank mechanism, designed for stamping various parts [1-5]. During the work of crank press, significant dynamic loads occur in blocks and mechanisms, especially when it is turned on. These dynamic loads are associated with operational feature of the crank press, which includes shock cyclic loads with sudden, almost immediate stops. In this connection, the study of the dynamics of clutch of the crank presses, is of great interest. Figure 1 shows block diagram of the press [1].

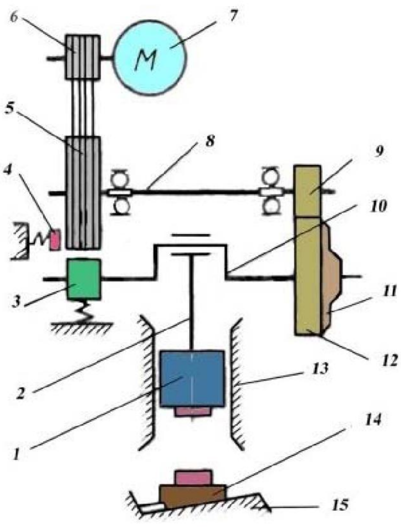


Figure 1 –
Block diagram of crank press:
1 - slide, 2 - crank rod,
3 - brake of crank mechanism,
4 - flywheel brake, 5 - flywheel,
6 - sheave, 7 - electric motor,
8 - drive shaft, 9 - drive gear,
10 - crank shaft, 11 - clutch,
12 - driven gear, 13 - crosshead guide,
14 - wedge-type platten,
15 - press board

Operating principle of the crank press (see figure 1): the crankshaft 10 rotates about an axis and activates through the crank rod 2 a slide 1 with punch. The press drive consists of an electric motor 7, a V-belt drive and a flywheel 5. The press clutch 11 is located on the end of the crankshaft 10. Brake 4 serves to stop the press. Brake 3 serves to stop the crank mechanism of the press.

The drive of the press is carried out from an electric motor with a flywheel. Since the parameters of motion of the actuating link – tool slide, depend only on the kinematic links of the main working mechanism, crank presses are referred to uncontrolled machines with limited movement of the tool slide, equal to the double radius of the crank or double eccentricity of the eccentric. Asynchronous electric drive accelerates with power flywheel and all the guide links with the corresponding moment of inertia to the steady angular velocity during the technological cycle and dispatches kinetic energy of the rotational motion of the flywheel to it. In this case, the crank shaft and all driven members of the crank-slider mechanism are fixed, the slide is in the upwardmost (initial) position. When the clutch 11 is turned on, the crank shaft (cranked axel 10) is rotated, driving and driven members move together, the slide with fixed upper die make a working stroke. After completion of the working stroke, the slide makes a return stroke. If the press works by single stokes, then when the slide reaches its initial position, the clutch 11 is turned off and at the same time the brake 3 is turned on. The slide stops in the upper (initial) position and the work cycle is completed.

Feed clutch of the crank press. Coupling clutches and brakes are provided in the press drive system, which make it possible to transmit motion to the actuator (operating mechanism) from the drive, but at the right time, vice versa, to stop the slide of mechanism without turning off the electric motor [1].

Switching on and off, and interlocking of the clutch and brake are performed using the control system [1]. The clutch, brake and control system form the so-called press start system, on the performance of which the reliability and safety of operation of the press as a whole is depended. The press start system works under difficult conditions – a large number of turning on per unit of time, on-off limited time (<0.1c), absolute security in operation.

Crank press's clutches should transmit moment of rotation up to 16 MNm and at the same time ensure the life of the structure and dampen vibrations, arising during the coupling [2]. The presses start systems and friction disc clutches most fully meet these requirements.

Disc clutches may be – single and multiple disc. Single-disc compact clutches with friction inserts are widely used, manufactured from retinax ФК-16А, ФК-24А or ferodo (figure 2). In pair with inserts, discs of steel 5, cast iron СЧ25 are working.

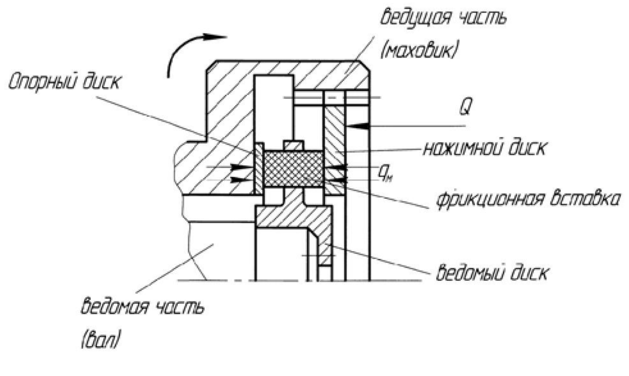


Figure 2 – Scheme of a single-disc friction clutch

The moment transferred by the one-disk frictional clutch (figure 2):

$$M_m = 2f \cdot q_m \cdot R_{cr} \cdot n \cdot F_{bc},$$

where f – friction coefficient $f = 0.35$; q_m – friction surface pressure q_m = 0,6-1,2 Мпа; R_{cr} – average insert radius; n – number of inserts; F_{bc} – work surface area of insert. Single-disc clutches transfer moment up to 140000 Нм, multiple-disc – up to 100 MNм with ferodo coverings.

The scheme of the multi-disc friction clutch is shown in figure 3.

Moment, transferred by the clutch:

$$M_m = \int_{R_2}^{R_1} 2\pi f \cdot m\rho \cdot d\rho \cdot \rho q_m = \frac{3}{2}\pi(R_1^3 - R_2^3) f q_m m$$

where q_m=0.4-0.6 Мпа (at n_m<180 об/min); q_m=0.3Мпа (at n_m>180 об/min); f =0.35; m

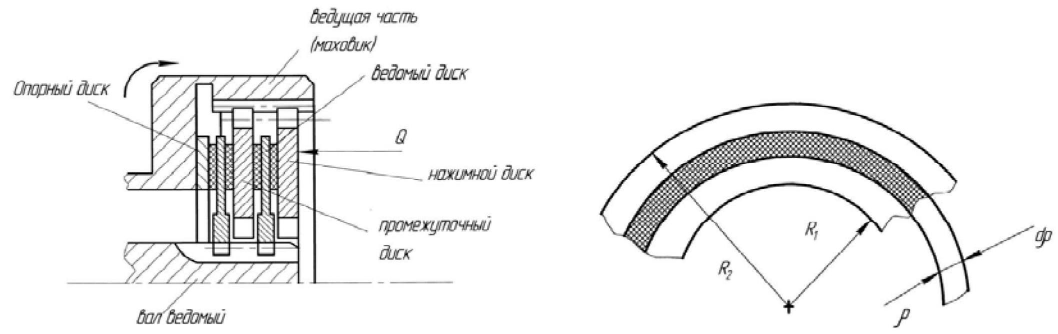


Figure 3 – Scheme of the multi-disc friction clutch

Calculation of disc friction clutches. Calculation of disc friction clutches made on the transmitted torque, specific force on the rubbing surfaces and the value of the wear rate (calculation for heating).

Original for the calculation is the nominal moment M_K^H , operating on the main shaft, which is driven to the clutch shaft. The required torque transmitted by the clutch [1]:

$$M_m^{mp} = \frac{\beta M_K^H}{i_m \eta_m}$$

where β = 1.1-1.3 is a factor of safety, taking into account the inertia and oscillation of the coefficient of friction;

$$M_K^H = m_K^H \cdot P_H,$$

m_K^H - reduced (modified) shoulder corresponding to the nominal angle α_H ; P_H - nominal force; $i_m \eta_m$ - gear ratio and transmission efficiency from the coupling shaft to the main shaft (when installing the coupling on the main shaft $i_m \eta_m$ = 1).

It is necessary that the torque transmitted by the coupling $M_m \geq M_m^{mp}$. Based on the moment (torque) transmitted by the clutch (coupling) press M_m, determine the force (effort) on the slider:

$$P_m = \frac{M_m}{m_K}$$

The results of the calculation P_m are entered in the table and build a graph of efforts on the slider based on the moment transmitted by the clutch $P_m = f(\alpha)$. When the clutch is turned on, part of expended energy goes into heat, causing heating of parts, friction inserts and linings. As an indirect thermal calculation, a performance calculation is applied in terms of wear. For this, the work balance is made when the clutch is turned on (at the initial moment, the speed of the driven disks due to slippage is somewhat less). Over a period of turn-on of the coupling t - the leading part transmits the moment M_m, moreover, during this time it turns for angle α_1. The balance of work when the clutch is turned on is as follows:

$$M_m \alpha_1 = \frac{I_{bm} \omega_H^2}{2} + M_c \alpha_2 + A_{mp},$$

where I_{bm} – moment of inertia of clutch's driven parts, reduced to clutch (coupling) shaft; M_c – moment of resistance of the driven parts; α_2 – the angle of rotation of the driven parts in a time t; A_{mp} – work spent on friction in the clutch (coupling). Angles α_1 and α_2 can be determined from the equation of dynamics for the driven parts.

$$I_{bm}\frac{d\omega}{dt}+M_m-M_c$$

take up I_{bm}, M_m, M_c as a constant for the period *t* (turn-on):

$$I_{bm}\omega_M=(M_m-M_c)t$$

from where:

$$t=\frac{(M_m-M_c)}{I_{bm}\omega_M}$$

Angular rotational rate of the driven part (at a known time *t*):

$$\omega_M=\frac{(M_m-M_c)t}{I_{bm}}$$

Turning angle of driven disks (substituting instead *t* its value):

$$\alpha_1=\omega_M t=\frac{I_{bm}\omega_M^2}{M_m-M_c}$$

Turning angle of driven part:

$$\alpha_2=\int_0^t\omega_M t=\int_0^t\frac{M_m-M_c}{I_{bm}}\cdot tdt=\frac{M_m-M_c}{I_{bm}}\cdot\frac{t^2}{2}=\frac{I_{bm}\omega_M^2}{2(M_m-M_c)}$$

substituting in the equation of the balance of work α_1 и α_2 will get:

$$A_{mp}=\frac{M_m}{M_m-M_c}\cdot\frac{I_{bm}\omega_M^2}{2}$$

Taking $\frac{M_m}{M_m-M_c}=\alpha_M$ = 1,05-1,16 (coupling on the main shaft) α_M =1,25-1,35 (coupling on a transmission shaft).

If the friction work is divided by the area of friction surfaces *F* [m^2] and multiplied by the actual number of turn-ons per minute, we get the wear rate [1]:

$$K_{izn}=\alpha_M\cdot\frac{I_{bm}\omega_M^2}{2F}\rho\cdot n_H\le[K_{izn}],$$

where ρ - use factor of number of moves, n_H - calculated speed of the press, the values are given in table 1.

Table 1

Type of equipment	n_H	ρ
1. Sheet-plate stamping press, ventilating, bending, high capacity cutoff	<15	0.7-0.85
2. Also of average capacity	20-40	0,50-0,65
3. Horizontal forging machine, plate cutter (sheet metal shears), shearing and multi-function, sheet metal press of average capacity	25-60	0,55-0,70
4. Hot forging crank driven press, embossing, high capacity varietal shears	70-110	0,30-0,45
5. Also of average capacity	40-70	0,45-055
6. Multi-operated sheet-plate stamping and shearing presses, high speed	90-120	0,20-0,45

[K_{izn}] = 0.7-0.8 M J/м^2 min – single disk clutches with retinax inserts.
[K_{izn}] = 0.4-0.5 M J/м^2 min – multi disk clutches with ferodo plates.

Formula shows that with decreasing ω the wear is reduced (but the moment is growing). After checking the wear rate, the piston diameter is selected based on the pressure in the pneumatic cylinder. p_c = 0,3-0,4MPa.

To improve the operation of the clutch, two-stage feed is made, and the pressure increases until the time of the working operation. [1].

Dynamic model of a crank press with feed clutch. When modeling the dynamics of a crank press with the feed clutch and simulation of the operation of the feed clutch, various software systems are used [6-17].

To simulate and analyze the movement of a crank press with the feed clutch, this work uses a software package: SimulationX [18].

SimulationX – is software for modeling and analyzing the dynamics and kinematics of automobiles, industrial equipment, electric, pneumatic and hydraulic actuators, hybrid engines, etc. It is used for the design, modeling, simulation, analysis and virtual testing of complex mechatronic systems. It simulates the behavior and interaction of various physical objects of mechanics (1D and 3D), driving equipment, electrical, hydraulic, pneumatic and thermodynamic systems, as well as magnetism and analog and digital control systems. It performs the following tasks: system modeling in the time and frequency domains; simulation of transient processes in linear and nonlinear systems or stationary simulation to calculate a model in a periodic state (nonlinear or linear). Model libraries are divided by simulated physical applications. Tools and interfaces complement SimulationX for integrated analysis of systems and structure.

Figure 4 shows dynamic model of a crank press with feed clutch on the SimulationX software package [19-22].

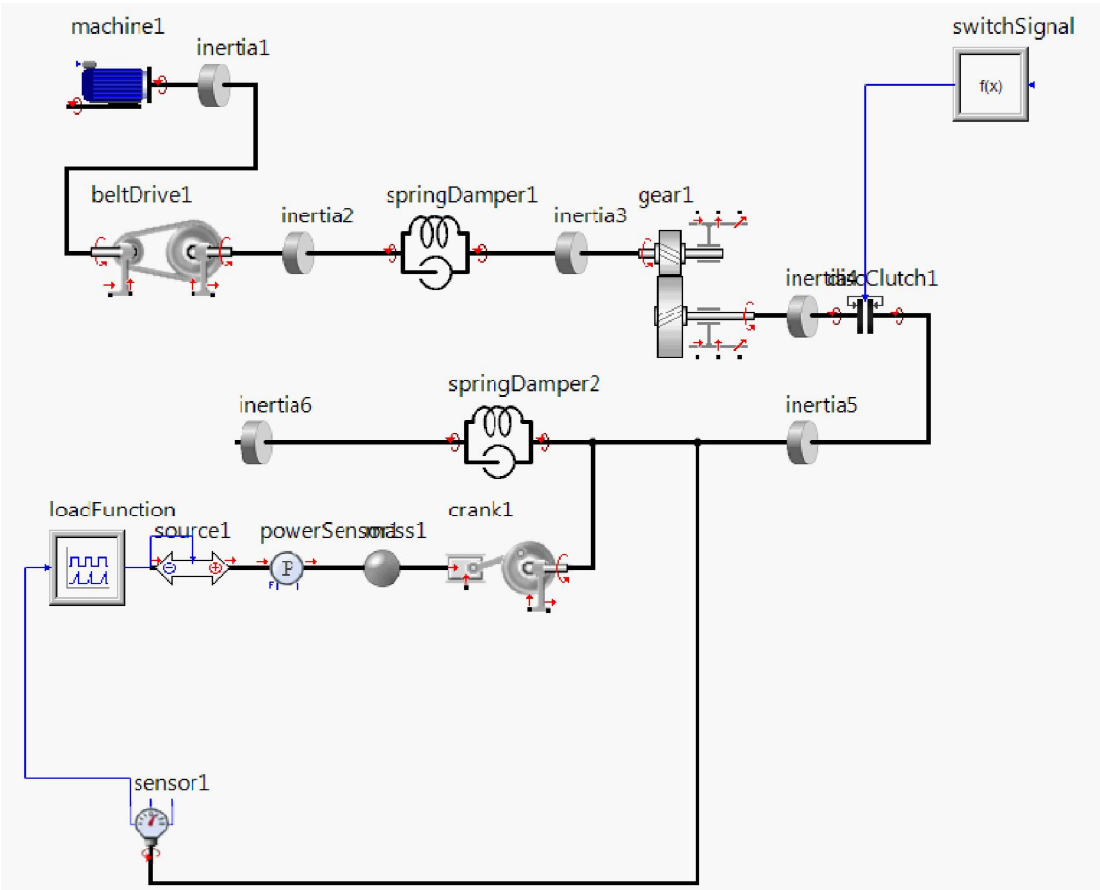


Figure 4 – Dynamic model of a crank press with feed clutch on the SimulationX software package

The elements of the SimulationX library that were used to compile the model are shown in figure 5.

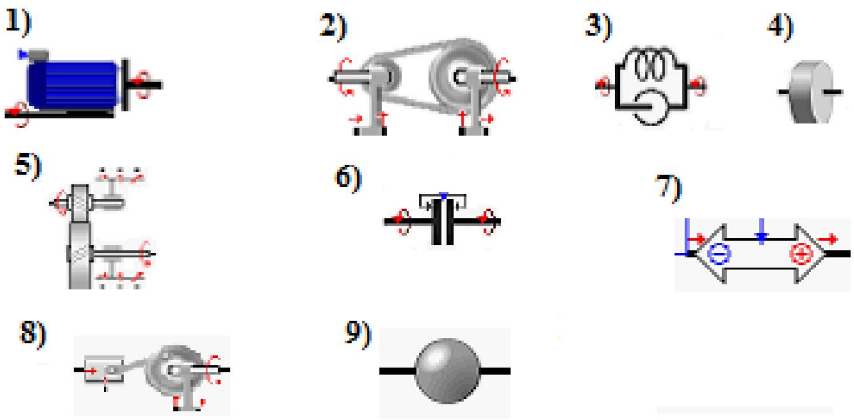


Figure 5 – The elements of the SimulationX library

List of symbols (see figure 5) and description of the elements of the SimulationX library:

1 – Asynchronous motor. This element simulates the simple asynchronous motor. The model is based on the stationary characteristics of the motor. This element models the asynchronous motor with sufficient accuracy when simulating the drive of a machine. It simulates engine starting, transient and steady-state processes, depending on the load and the speed of rotation of the shaft.

2 – Belt drive. This element models the operation of belt transmission, with the account of elastic-dissipative characteristics. The model takes into account the reactions and movements in the bearings of the pulleys of the belt drive that allows you to simulate the interaction of the transmission with the base.

3 – Spring - Damper – backlash. The model represents elastic and / or damped behavior between the rotational links, with the possibility of taking into account backlash. Springs always acts in parallel with the dampers.

4 - Inertia. This element models the moment of inertia of a rotary link. It is also possible to simulate a variable moment of inertia.

5 – Gear. The Transmission element is an ideal converter of rotational movements and forces operating between two components in a rotating mechanical system. It works as an ideal converter without taking into account dissipation and, fulfills the specified gear ratio or the conditions of power balance in input and output. The Transmission element allows you to model fixed and variable ratios for angles or velocities in input and output.

6 – Disc clutch. The Model Disc Clutch is a component that turns on or interrupts the flow of torque (and therefore power transmission) between the drive components. The model can be used to simulate multi-plate clutch of machines or gearboxes. In addition, it is possible to simulate the friction of the brakes (for example, an automatic transmission). Elasticity, damping and clutch friction parameters can be considered. In transmission of the models, the clutch can be activated by a signal from the switch.

7 – External force. This type of element allows you to simulate the forces between two components, or only on one component of the mechanical model. It provides universal, functional power transfer in the mechanical model.

8 – Crank mechanism. The element models a slide-crank mechanism, taking into account the backlash in the hinges, the elastic-dissipative properties of the connecting rod.

9 – Mass. This element models the mass of a linear link. Variable mass modeling is also possible.

Initial parameters of the model:

Crank press motor power W=0.5 квт, rated engine speed n=450 rpm. The numerical values of the dimensions, moments of inertia of the nodes of the crank press and the stiffness of the shafts are taken from [1].

The nominal force developed by the slide of the slide-crank working mechanism in the area before the extreme low point of the slide's stroke is modeled by a sine-wave signal generator (loadFunction) and linear force (load) (figure 6). This load force depends on the angle of the crank. The maximum force is reached at the lower point of the slide’s stroke and is equal to 4000n.

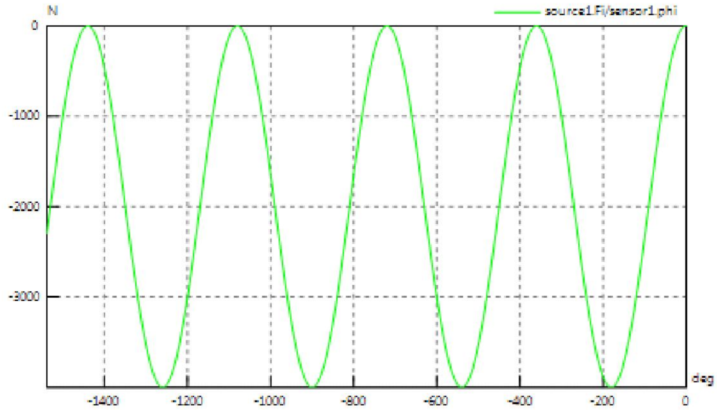


Figure 6 – The nominal force, developed by the slide of the slide-crank working mechanism

Parameters of the feed clutch of the crank press are shown in figure 7.

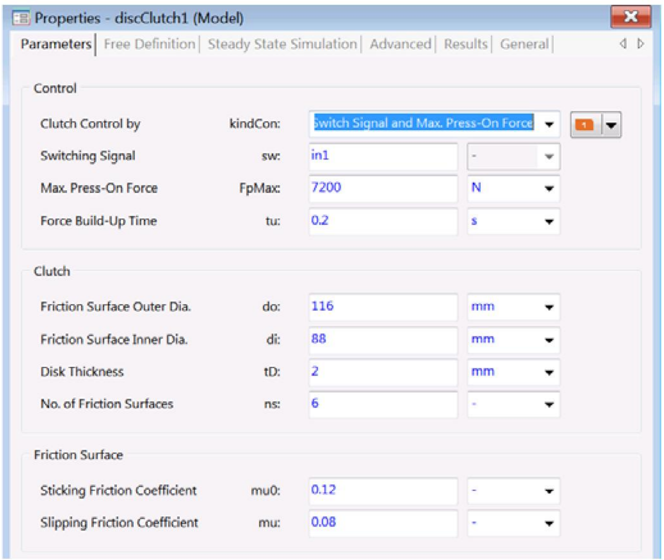


Figure 7 – Parameters of the feed clutch of the crank press

Simulation results: Feed clutch of the crank press is activated on the 10th second and connects the moving flywheel to the actuator. Figure 8 shows the moment transmitted by the feed clutch.

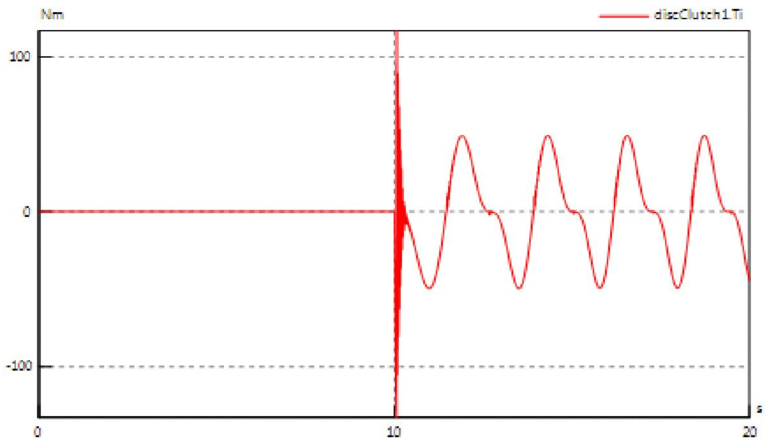
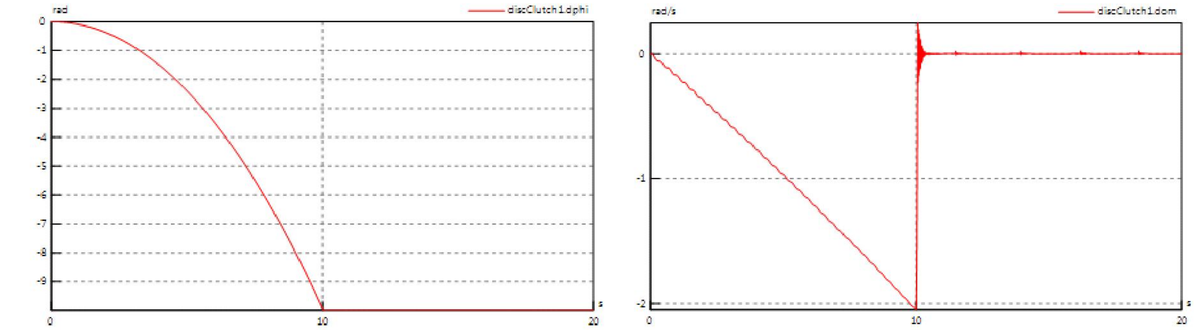


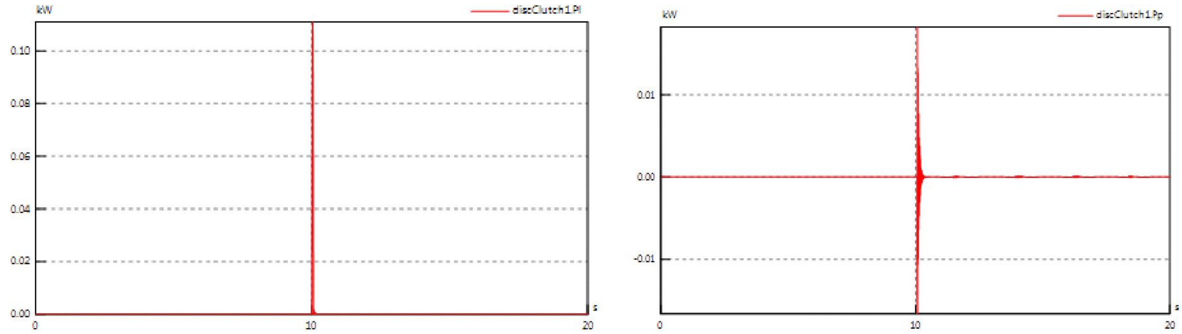
Figure 8 – moment transmitted by the feed clutch

Figures 9 and 10 show the various data of the feed clutch of the crank press



(a) (б)

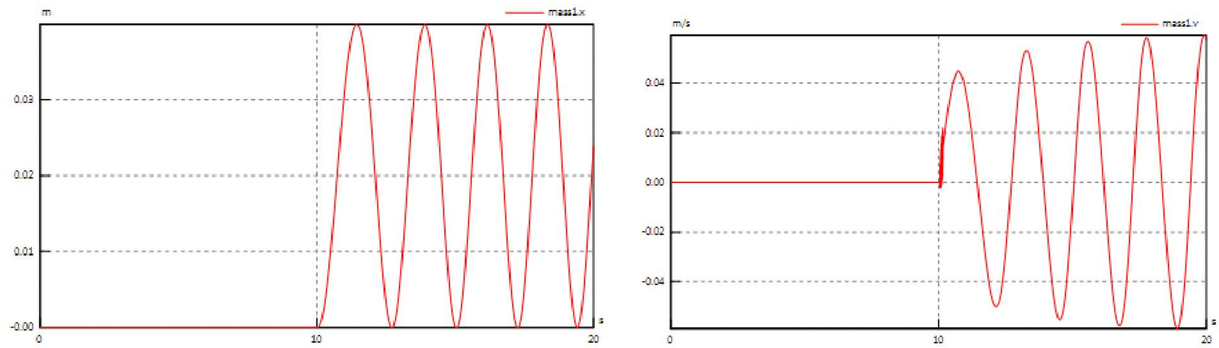
Figure 9 – Calculated data of the feed clutch a) relative angular displacement of the discs; b) relative angular velocity of the disks



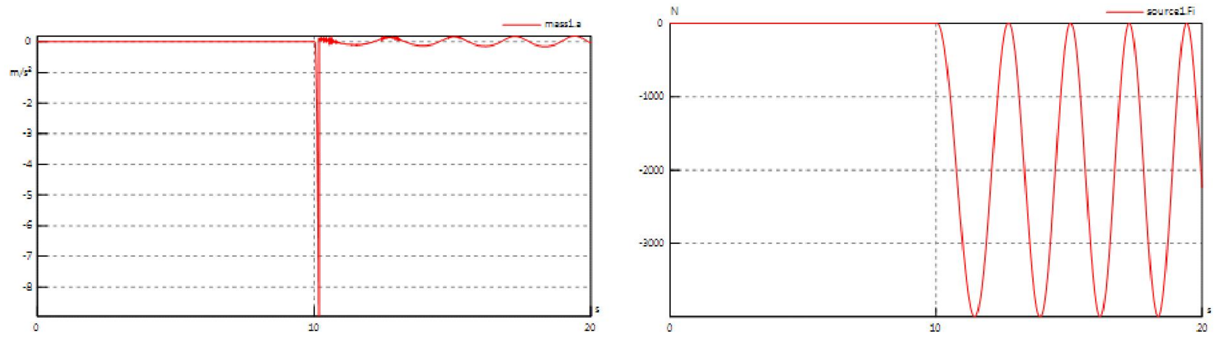
(a) (б)

Figure 10 – Calculated data of the feed clutch a) turn-on power loss; б) potential energy change of the clutch

Figure 11 a, b, c shows movement, speed, acceleration and load of the press slide.



(а) (б)



(в) (г)

Figure 11 – Estimated data of the press slide a) movement; б) speed; в) acceleration; г) load

Conclusion.

1. The SimulationX software package allows one to simulate the dynamics of the feed clutch of the crank press, taking into account its design parameters as part of the crank press, and the interaction with all its nodes.
2. As the result of the dynamic calculation following is determined; moment transmitted by the feed clutch, relative angular displacement of disks, relative angular velocity (speed) of the disks, turn-on power loss and potential energy change of the clutch. The displacement, speed, acceleration, and load of the slide of the crank press are determined at the moment of clutch turn-on and after movement.
3. Dynamic loads in the nodes of the crank press sharply increase at the moment of clutch turn-on.
4. When studying the dynamics of a crank press, it is necessary to take into account the design features of the feed clutch, especially plate wear and adjustment, which requires further research of this unit.
5. Visibility of the models and graphical results are especially useful for students and engineers in the study of the feed clutches of the crank presses.

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ҚОСИІНДІ БАСПАҚҚА ҚОСЫЛҒАН ЖАЛҒАСТЫРҒЫШТЫҢ ДИНАМИКАЛЫҚ ЗЕРТТЕУІ

Аннотация. Жұмыста қос иінді баспаққа қосылған жалғатырғышты динамикалық түрде зертелді. Қазіргі кезде басқа түйіндердің өзара әсерін ескеріп, қос иінді баспаққа қосылған жалғастырғышты динамикалық зерттеу маңызды мәселе болып отыр. Қос иінді баспақ массасы 100 килограммнан бірнеше тоннаға дейін болатын қозғалмалы бөлшектер мен түйіндерден тұрады. Үлкен динамикалық жүктемемен үлкен жылдамдықпен әсер ететін қос иінді баспаққа қосылылған жалғастырғышқа осы бөлшектермен түйінде циклді түрде қосылып отырады. Қосылылған жалғастырғышты қос иінді баспақтың қозғалысын моделдеп және талдау үшін SimulationX бағдарламалық комплекс қолданылды. SimulationX бағдарламалық комплексі - өнеркәсіптік жабдықтар, электр,- пневмо,- гидрожетектер, гибридті қозғалтқыш, автокөліктердің кинематикасы мен динамикасын талдап және моделдеу үшін пайдаланылды. Динамикалық есептеу нәтижесінде жұмысшы бұлғақтың және қос иінді баспаққа қосылылған жалғастырғыштың маңызды динамикалық параметрлері анықталды. Жалғастырғышты қосқан сәтте қос иінді баспақтың барлық түйіндерде динамикалық жүктеме ұлғая бастағаны көрсетілген.

Түйін сөздер: динамика, қосиінді баспақ, жалғастырғыш, бұлғақ, момент, тербеліс, SimulationX.

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ДИНАМИЧЕСКОЕ ИССЛЕДОВАНИЕ МУФТЫ ВКЛЮЧЕНИЯ КРИВОШИПНОГО ПРЕССА

Аннотация. В работе исследуется динамика муфты включения кривошипного пресса. В настоящее время динамическое исследования муфт включения кривошипных прессов с учетом взаимодействия с остальными узлами является актуальной задачей. Кривошипный пресс содержит подвижные детали и узлы, масса которых от ста килограмм до нескольких тонн. Данные детали и узлы циклически подключаются муфтой включения кривошипного пресса с большими скоростями и на них действуют большие динамические нагрузки. Для моделирования и анализа движения кривошипного пресса с муфтой включения используется программный комплекс: SimulationX. SimulationX – программный комплекс для моделирования и анализа динамики и кинематики автомобилей, индустриального оборудования, электро-, пневмо- и гидроприводов, гибридных двигателей и т.д. В результате динамического расчета определены важные динамические параметры муфты включения кривошипного пресса и рабочего ползуна. Показано, что динамические нагрузки практически во всех узлах кривошипного пресса резко возрастают в момент включения муфты.

Ключевые слова: динамика, кривошипный пресс, муфта, ползун, момент, колебания, SimulationX.

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