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COMPUTER MODELING AND INVESTIGATION OF DYNAMICS OF SYSTEM “VESSEL–REINFORCEMENT” IN SHAFTS WITH BROKEN GEOMETRY

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КОМПЬЮТЕРНОЕ МОДЕЛИРОВАНИЕ И ИССЛЕДОВАНИЕ ДИНАМИКИ СИСТЕМ «СОСУД–АРМИРОВКА» В СТВОЛАХ С НАРУШЕННОЙ ГЕОМЕТРИЕЙ

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mine shaft, skip, cage, mine shaft reinforcement, mine winding plants' dynamics, “vessel – reinforcement” system, protective runner, box guide, roller directors, bunton.

The article describes results of mathematical modeling and analysis of instrumental measurements of the dynamic interaction between lifting vessels and reinforcement of mine shafts. Based on the analysis of instrumental measurements data of contact loads between the protective runner and the guides shown that the process of long-term operation of the dynamic interaction of vessel with guides becomes substantially uniform in the shafts depth. There are sites with different parameters of guides profiles curvature, varying degrees of wear of guides and buntons. The specified factors lead to the fact that during movement the vessel experiences dynamic indignations, various on level and character, from guides. And vessel creates reciprocal dynamic loads on reinforcement, various on level.

It is shown that constant-sign deviation from vertical of guides on long sections of shafts during the motion of the vessel at operating speeds result in a one-sided pressing of working faces of the runner to the guides and the excitation of shock interaction with steps at the junctions of guides, which can not be prevented by the elasticity of the roller directors.

Finite element models of the dynamic interaction of vessel with a reinforcement are developed. They take into account the lifting vessels inertia parameters, shifting the mass center of cargo in the vessel, actual spatial profiles of the guides, the support rigidity and wear of the reinforcement with guides, charts hoist speed, cinematic gaps in pairs of “runner – guide”.

Researches of spectral characteristics of revolting impacts on lifting vessel from guides profiles according to the actual surveying measurements of their deviations from a vertical in the operating shafts are executed.

It is shown that the shafts sections, where the frequency of the external perturbation by reinforcement are close to the natural frequency of the vessel with the elastic roller directors, in the “vessel – reinforcement” systems appears an effect of resonant excitation of horizontal fluctuations in angular and translational degrees of freedom with a high level of contact loads on the reinforcement.

The dependences of the contact loads level on reinforcement from magnitude of cargo mass center displacement in the vessel are received.

Ключевые слова:

шахтный ствол, скип, клеть, армировка шахтного ствола, динамика шахтных подъемных установок, система «сосуд–армировка», предохранительный башмак, коробчатый проводник, роликовые направляющие, расстрел.

Представлены результаты математического моделирования и анализа аппаратных измерений динамического взаимодействия подъемных сосудов с жесткой армировкой шахтных стволов. На основании анализа данных значительного числа аппаратных измерений контактных нагрузок между предохранительными башмаками скольжения и проводниками показано, что при длительной эксплуатации процесс динамического взаимодействия сосуда с проводниками становится существенно неоднородным по глубине стволов. В них появляются участки с различными параметрами кривизны профилей проводников, различной степени износа проводников и расстрелов. Показано, что указанные факторы приводят к тому, что в процессе движения сосуд испытывает различные по уровню и характеру динамические возмущения со стороны проводников и создает различные по уровню ответные динамические нагрузки на армировку по участкам ствола.

Показано, что знакопостоянные отклонения проводников от вертикали на протяженных участках ствола при движении сосуда на рабочей скорости приводят к одностороннему прижатию рабочих граней башмаков к проводникам и возбуждению ударного взаимодействия с уступами на стыках проводников, которое невозможно предотвратить за счет упругости роликовых направляющих.

Разработаны конечно-элементные модели динамического взаимодействия сосуда с армировкой, учитывающие инерционные параметры подъемных сосудов, смещение центра масс груза в сосуде, фактические пространственные профили проводников, опорную жесткость армировки проводников и их износ, диаграммы скорости подъемной машины, кинематические зазоры в парах «башмак–проводник».

Выполнены исследования спектральных характеристик возмущающих воздействий на подъемный сосуд профилей проводников по данным фактических маркшейдерских измерений их отклонений от вертикали в действующих стволах.

Показано, что на участках ствола, на которых частоты внешнего возмущения со стороны армировки оказываются близкими к частотам собственных колебаний сосудов с упругими роликовыми направляющими, в системах «сосуд–армировка» возникает эффект резонансного возбуждения горизонтальных колебаний по угловым и поступательным степеням свободы с повышенным уровнем контактных нагрузок на армировку.

Получены зависимости уровня контактных нагрузок на армировку от величины смещения центра масс груза в сосуде.

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Introduction

The problem of stabilization of dynamic interaction between skips and mine shaft steelwork guides is quite sophisticated. Its solution depends on the relation between parameters of ‘skip-steelwork’ system and, particularly, the relation between skips’ natural frequency of oscillation on roller supports and external multi-frequency perturbations on the part of the guides upon condition that the system is stable against the parametric perturbations caused by variable stiffness at the shaft steelwork pitch. Skips in coal and ore winding plants of mining production sites operate at a rate of 8–12 m/s, normally performing the trip using two box guides via elastic roller supports and stiff steel slide runners. Every skip is suspended on one rope.

Many shafts are located in the area of subsurface rock displacement and therefore have significant guide verticality faults in certain areas. In the course of dynamic testing at 10–11 m/s elevation rate, these areas manifest guide contact overload by conveyance vehicles.

During movement along the mine shaft, the skip experiences forced horizontal oscillations in the frontal and lateral planes within the limits of kinematic clearances between the protective runners working surfaces and the guides. Amplitudes and frequencies of these oscillations are determined by inertia parameters of the skip, stiffness of roller supports, curvature parameters of each guide and skip vertical speed.

The mine shaft contains guide sections with varying curvature parameters. Therefore the amplitude-frequency parameters of skip horizontal perturbation caused by the guides and transferred via elastic-dissipative roller support units are widely variable across different sections of the mine shaft.

In certain sections of the shaft the vertical deviations of the guides are negligible, not exceeding 3–5 mm, and are distributed across the long span of the shaft, constituting 10–30 lengths of the conveyance vehicle. The skip moves through such section smoothly, performing low frequency oscillations and resting on the guides only via spring-assisted roller supports within the limits of the spring working stroke, i.e. functions according to the designed smooth model of interaction with the steelwork.

Besides, in certain sections of the shaft, local vertical deviations of the profile are fluctuating or have significant one-sided deviations from the vertical line. If the amplitudes of fluctuating or local vertical deviations of the guides profile reach 20–30 mm within the skip length (i.e. 3–4 spans of steelwork tiers), the forced oscillation amplitudes go beyond the kinematic clearance (as per safety requirements, maximum clearance shall not exceed 23 mm per side), causing the working surfaces of protective runners to come into hard contact with the guides.

It was experimentally established that the maximum contact force in case of cyclic impact interaction between the vehicle and the guides in these conditions reaches 50–60 kN [1]. High operational load results in premature wear of the guides, buntons and their mounting assemblies below acceptable level. High dynamic contact loads result in rapid spread of fatigue cracks and guides welding seams disruption. Therefore the dynamic load level on the guides produced by conveyance vehicles is the main contributor to the reliability and safety of the “vehicle-steelwork” system.

This load is determined by two constituents: load received by the roller damper, and load received by stiff protective runner of the vehicle. The maximum level of these loads is mainly determined by curvature of guides within a local section of the mine shaft, vertical speed and vehicle weight.

Roller spring-assisted dampers of the skip are designed to hold it in a centered position relative to the box guides within the limits of kinematic clearances and to prevent hard contact between the runners and the guides during oscillations of the vehicle.

The main property that defines operation of the conveyance vehicle roller support is the type of dependence between horizontal displacement of the contact point between the roller and the guide and resistance force, which is defined by the design, parameters and elastic-dissipative properties of the guiding system.

In fact, it is necessary to ensure that the conveyance vehicle moves relative to the steelwork along the entire shaft (in frontal and lateral planes separately) in such a way that all horizontal perturbations from the guides would be absorbed by the roller damper system preventing

hard impact contact between the runners and the guides, i.e. to ensure the movement pattern with maximum 10 kN cumulative load on the guide [2].

In order to solve this problem, it is possible to employ a number of mathematical modeling systems, including direct inference of differential equations for the system movement with their further analytic and numeric solving by classical methods, or finite-element modeling in standard programming systems, such as SolidMotion.

Computer modeling of dynamic processes in “vehicle-steelwork” system in mine shafts with disturbed geometry

A vast array of studies has been dedicated to the mathematical modeling of “vehicle-steelwork” system. The most comprehensive review is provided in the works of Prof. V.I. Dvornikov [3–5]. The core of his studies is focus on the problems of dynamics and parameterization of guiding system elements for shafts with designed parameters of guide profiles. The disturbed geometry shafts manifest guide profile geometrical parameters that are significantly different from designed parameters; their correction may be complicated due to the challenging technical and geological conditions at the mine shaft.

SolidMotion system application, on the other hand, is the most convenient option, which, however, has a number of limitations as regards the real-life system modeling capacity. Therefore at the first stage of the research it was decided to use the capability of SolidMotion computation system for modeling and study of interaction between skips and steelwork in real-life conditions of the mine shafts with traditional design roller supports, equipped with rubber spring-assisted dampers with permanent experimentally defined stiffness of damping unit within the limits of kinematic clearances. The guides design, their type and results of stiffness parameters evaluation are described in [6].

In order to build a computational finite-element model in SolidMotion system, the conveyance vehicle is simulated by a solid non-deformable body with overall dimensions and weight equal to dimensions and weights of skips operating in ore mining shafts.

Parameters of winding plants used for calculations are listed in Table.

Skip winding parameters

Parameter	Value
Lifting height, m	940 m
Lifting speed (operating), m/s	10.9 m/s (operating)
Conveying vehicle type	Skips: SO-9.5-174-0.12.000-FO
Vehicles own weight, kg	17 000
Lifting capacity, kg	Up to 23 000

In the course of the skip horizontal movement modeling, it is considered to be fixed in vertical direction relative to the static space, while the guides are assumed to be perfectly rigid bodies that move vertically relative to the skip pursuant to the desired laws, interacting with it by means of four elastic-dissipative connections. The connections characteristics are defined in a standard way in the modeling system. The skip performs horizontal movement under the effect of the four coercing mechanical forces in frontal plane and eight forces in lateral plane per desired functions of time as a system with five degrees of freedom.

Solid body model of the skip interacts with models of guides sections by way of elastic-dissipative connections in the points of installation of real roller supports on the skip (Figure 1).

Horizontal movements of guides relative to the skip are modeled by functions of time which are defined according to a specific procedure using a mathematical processor based on real survey data for vertical deviations of guides at each steelwork tier, data of tire numbers distribution across the depth marks in the shaft and diagram of the skip vertical speed measured by the winding plant digital control unit.

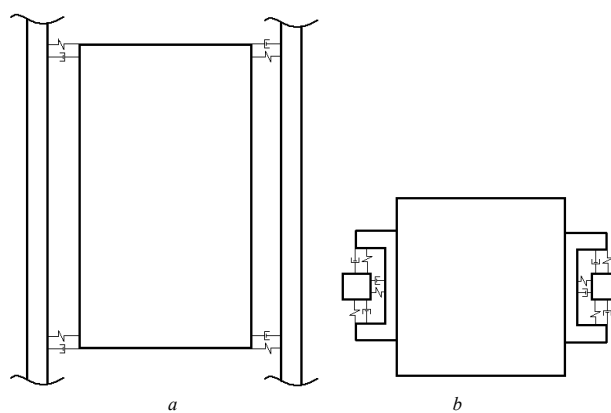


Figure 1. Solid body model scheme of “skip – roller supports – guides” system:
a – side view; *b* – top view

The guides are set at the desired distances from the skip, equal to the real kinematic side clearances with limit values (15–23 mm) defined by the safety rules. During the simulation experiment, the clearances vary within and outside of these limits, since in the shafts with disturbed geometry clearances may reach 30–40 mm to avoid vehicles jamming at the curved sections, and in the narrow sections may go down to 10 mm per side.

Figure 2 shows guide profiles in the skip compartments of the shafts.

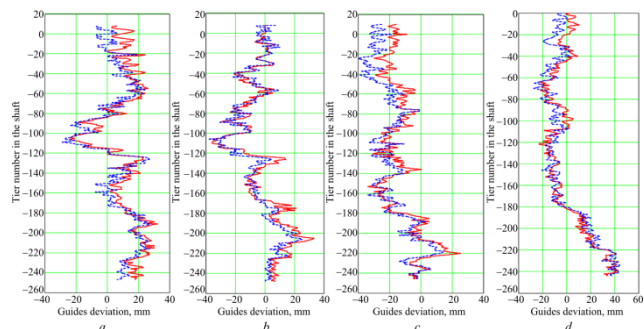


Figure 2. Guide profiles:
a – 7/8 shaft No. 2; *b* – 9/10 shaft No. 2;
c – 1/2 shaft No. 1; *d* – 5/6 shaft No. 1

The skip moves along the rails, interacting with each section intermittently by the upper and lower roller supports, separately on either side. To construct functions defining instantaneous values of horizontal deviation for each guide from vertical line in the upper and lower belt of the skip time-wise within the cycle, additional calculations are performed in Mathcad software system.

For the lower part of the vehicle, the moment of time when the horizontal movement of the guide section is defined, is calculated with a shift equal to the time in which the vehicle covers the section equal to its height with the current speed value.

Based on the results of a respective program unit, files with guide horizontal movement matrices are constructed evolving in cycle time, to be transferred to the SolidWorks-SolidMotion system for skip dynamic analysis.

Figure 3 shows actual diagrams for linear speeds of winding drums rotation (skips movement speed) in the respective shafts.

It is apparent that actual diagrams significantly differ from the traditional five-period speed diagram. Below are the results of numeric research of “skip-steelwork” dynamics in SolidMotion system [7].

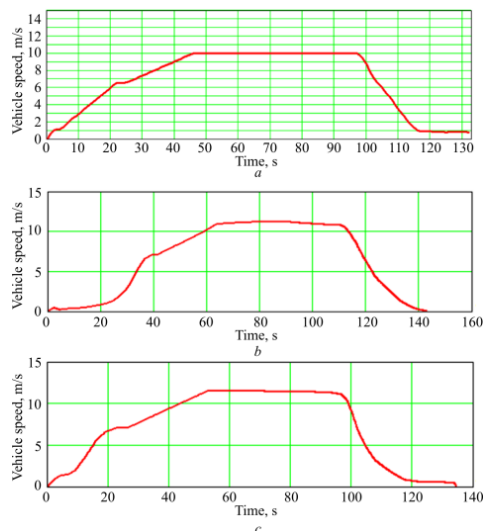


Figure 3. Diagrams of lifting speed in skip compartments:
a – southern and northern compartment of shaft No. 1;
b – hard compartment of shaft No. 1; *c* – southern and northern compartment of shaft No. 2

We study the dependence of dynamic loads between skip and guides in the appointed skip compartments of shafts on the values of kinematic clearance and stiffness of the operating types of guides. In the course of the research, energy dissipation will be neglected to receive a clearer picture of interaction (contact loads will be inherently higher than when the energy dissipation is taken into account).

Figure 4, *a* shows frontal oscillation charts for the southern skip between the guides in the frontal plane at initial clearance of 15mm per side in the base mark area with stiffness of each roller support equal to 150 kN/m.

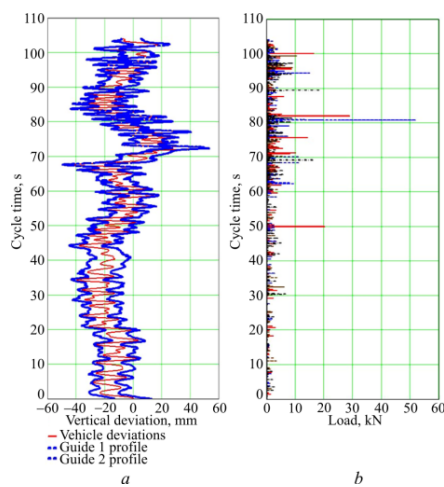


Figure 4. Charts: *a* – movement of guide upper belt runners in the guides along the shaft; *b* – contact force between guides and skip along the shaft

It is clear that within some of the sections the skip moves smoothly, interacting with the guides only via the roller supports; in other sections, it comes into a hard contact with them. Oscillation frequency is higher in the hard contact sections than in the smooth sections.

Figure 4, *b* shows contact load charts for interaction of the southern shaft No. 2 skip with the guides in the same lifting cycle at the stages of acceleration, steady motion and beginning of deceleration. It is clear that the substantially curved local section of the guides cause impact loads between the skip and upper and lower guide belts.

Figure 5, *a, b* shows enlarged fragments of the same charts at the section of intense cyclic impact interaction between skip and guides from 70th to 100th second of upward trip. It is clear that the frequency of impacts between the skip and guides, including all four runners, reaches 4–5 Hz. Besides, in sections with different profile parameters, load values differ by a factor of several times. In spite of the prevailing number of interactions with low level of contact load, in certain parts of the cycle individual hard impacts of significant magnitude occur.

Figures 5, *c* and *d* shows charts of skip movement and contact loads in the event when one of the guides thrusts against the bunton and, thus reinforced, has double stiffness compared to the other one. The chart in Figure 5, *d* shows that in this case the contact loads on the stiffer bunton are approximately twice higher than on the second one.

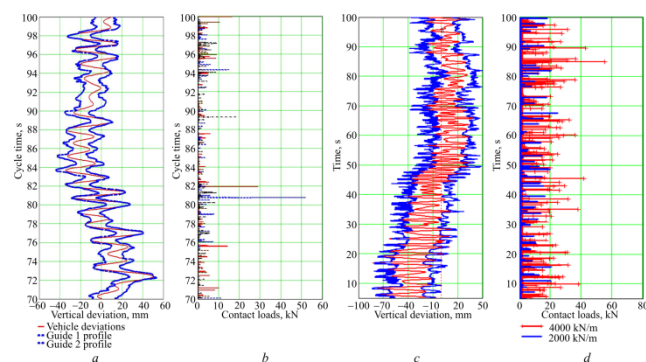


Figure 5. Charts: *a* – skip upper belt runners motion in the guides along the cyclic impact section; *b* – contact forces between guides and skip along the cyclic load section; *c* – skip upper belt runners motion in the guides with varying support stiffness; *d* – contact forces at varying support stiffness of guides

At each moment of time the vehicle dynamics is determined by the profile parameters in the current section of movement. The profile parameters for each individual guide are a random set of numbers possessing their own statistic characteristics. The vehicle dynamics is formed by both individual profile parameters of either of the two guides and their combination.

Research by means of the calculation model has shown that in case of skip movement at the same speed diagram and same stiffness of guides, the kinematic clearances variation creates an overall tendency of contact load level decrease in the system. However, at the same time their peak values occur at different points of the shaft length within the limit of the general section with increased level of cyclic impact interaction.

At varying stiffness of guides, the vehicle manifests varying behavior when entering each next section of the shaft. Instantaneous values of kinematic clearances at the entrance to each subsequent section form at the end of the previous section and depend on the stiffness of all guides (frontal and lateral). Therefore the impact between the vehicle and the same fault in the profile at various initial clearances at the faulty section entrance results in varying values of contact loads, since for each roller support the contact with the slanted section of the guide occurs with varying horizontal collision speeds.

Figure 6 shows resonance frequency charts from the finite-element model of the “skip – roller support – guides” system by five degrees of freedom, built in the SolidSimulation programming system depending on the guides stiffness in the contact points between the roller and the guide.

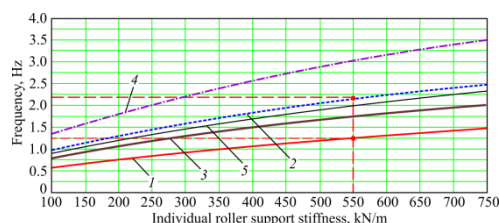


Figure 6. Individual frequencies of “skip – roller supports” system: 1 – progressive frequency in the frontal plane; 2 – angular frequency in the frontal plane around the lateral inertia axis; 3 – progressive frequency in the lateral plane; 4 – angular frequency in the lateral plane around the frontal inertia axis; 5 – angular frequency in the lateral plane around the rope axis

It is clear that as the stiffness of each roller support increases up to a factor of 7, oscillation frequencies in frontal and lateral planes for each degree of freedom increase by $\sqrt{7} = 2.6$ times. Simultaneously, since in the frontal plane the vehicle's oscillations are defined by the stiffness values of the four identical roller supports, and in the lateral plane – of the eight, the progressive and angular oscillation frequencies in the lateral plane are higher than the respective frequencies in the frontal plane (curves 2 and 4, 1 and 3) by $\sqrt{2} = 1.4$ times. This is fully concordant with the physical entity of the researched process and confirms the correctness of the developed calculation model.

Figure 7 show spectra of geometrical parameters in the profiles of guide system 7/8 of the southern skip compartment of shaft No. 2 (see Figure 2, *a*) in the section of increased collision level: progressive skip movement and slope angle in the frontal plane at the upward motion interval 60–100 s according to the speed diagram shown in Figure 3, *c*.

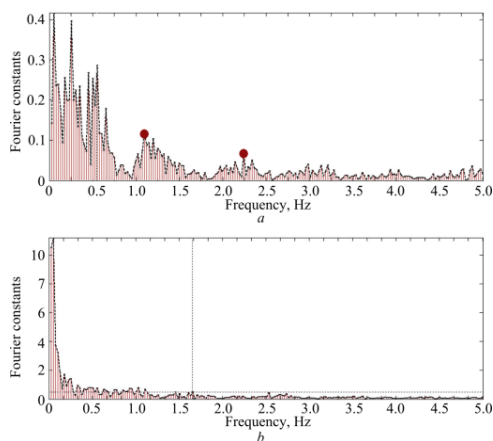


Figure 7. Skip-guides perturbation spectrum:
a – by the skip axis angle;
b – by the skip progressive movement

Data in Figure 7 indicate that the profile of guide system 7/8 in the frontal plane at upward motion diagram (see Figure 3, *c*) in the upward motion interval 60–100 s creates perturbation to the skip slanted oscillations with frequencies in the intervals: 0.1–0.7; 1.0–1.5; 2.0–2.5 Hz. In the progressive frontal movements of the skip, significant perturbations occur only at very low frequencies (from 0.1 to 0.25 Hz).

Comparing these spectra to the resonance frequency charts in Figure 7, we can see that the most evident dynamic reaction of the skip to the

guide data profile occurs during its slanted oscillations in the frontal plane (curve 2, resonance frequency range 0.5–2.5 Hz). This is concordant to the collision picture shown in figure 5, *b*, where the runners successively come into frontal impact contact with the guides.

Figure 8 shows the skip oscillation spectra at roller spring stiffness values 150 and 550 kN/m in the section of intense collision with guides (see Figure 5). Clearly, in the first case 0.5–0.65 and 0.8–1.2 Hz frequencies dominate. They match the skip resonance frequencies at this spring stiffness (Figure 6, curves 1 and 2).

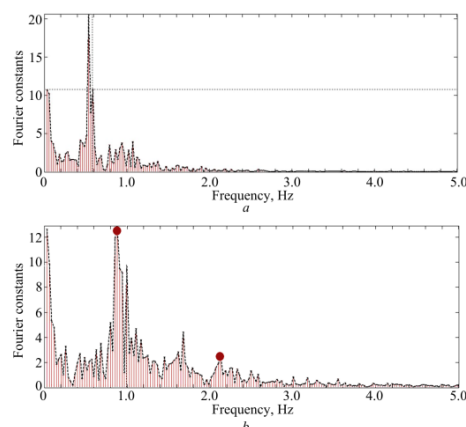


Figure 8. Skip oscillation spectrum at spring stiffness: *a* – 150 kN/m; *b* – 550 kN/m

In the second case, the values of domineering frequencies increase up to 0.8–1.25; 1.5–1.8; 2.1–2.3 Hz. They also stay within the resonance frequency range at this magnitude of roller spring stiffness.

Numeric research of dependencies between the maximum contact loads in the skip upward motion cycle and the roller supports stiffness at various values of kinematic clearances per side at the base mark of the shaft has shown the inverse relation between the clearance and maximum load values.

This is qualitatively concordant with the results of load measurement data processing using AKN equipment discussed in [8] for Tashtagol mine vertical shaft, and corresponds to the physical essence of the studied process (systemic absence of parametrical resonance caused by variability of the guides frequency over the steelwork pitch length) [9–20].

Similar tendency can be observed at increasing stiffness of roller supports with unchanged values of kinematic clearances at base mark. Contact loads between runners and guides go down as guides

stiffness increases, within the limit of 100–700 kN/m up to 3 times for guide profiles shown in Figure 2, *a* in case of a loaded skip upward motion according to the speed diagram shown in Figure 3, *c*. Their peak loads occur in various moments of time at various modeled sections according to the same speed diagrams.

Charts in Figure 9 show mean loads for the time of the skip crossing the section of the shaft with increased collision between the runners and the guides in the interval within 60–100 s of upward motion depending on the guide stiffness for different magnitudes of basic clearances. Obviously the

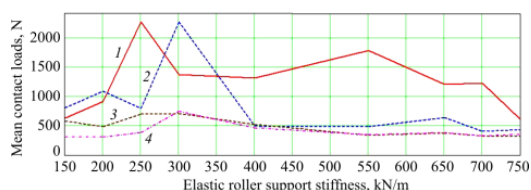


Figure 9. Dependence of mean loads for the period of skip movement time within the upward motion cycle of 60–100 s on the roller supports stiffness: 1, 2 – upper belt roller supports on opposite guides; 3, 4 – lower belt roller supports on opposite guides

contact loads for all clearance values are at maximum in the 250–350 kN/m stiffness interval. Comparative analysis of these results and curves in Figures 7 and 8 shows that perturbation frequencies from guide profiles in this shaft section are approaching the skip natural oscillation frequencies at these roller support stiffness values, which creates precursors for resonance increase of its oscillations in this section of the shaft.

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Summary

1. In the pre-resonance mode (by parametric perturbation criterion caused by periodic stiffness of guides) of interaction between the conveyance vehicles and the steelwork, a conglomerate of parameters produces a dominating influence on the level of dynamic loads: guides curvature in local sections of the mine shaft equal to 2–3 lengths of the vehicle, load mass center deviation from the rope suspension axis, cumulative kinematic clearances in the “runner-guide” pair.

2. Spectral parameters of fluctuating vertical deviations of guides depending on the speed of vertical movement of the conveyance vehicle have multifrequency nature and at certain combinations of inertial parameters of the vehicle, elasticity of roller dumners, stiffness of steelwork, may approximate the frequencies of natural oscillations of the “vehicle-steelwork” system and cause the increase of dynamic load on the steelwork.

3. Solid-body mathematical modeling of interaction between the conveyance vehicles and the steelwork enables determining the level and nature of dynamic force variation as vehicles and guides interact in the lifting process according to the operating speed diagram.

4. In combination with the equipment wear data for guides and buntons, the proposed methods enable determining the dependence between the dynamic interaction level and the parameters of guide profile correction while the rest of the parameters remain unchanged.

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