УДК 622.277:621.927 DOI: 10.21440/0536-1028-2020-1-97-104

Analysis of connections and restrictions occurring due to the oscillations of vibrating screen working member and cantilevered screen deck

Arkadii V. Iudin[1], Viktor S. Shestakov[1]*
[1] Ural State Mining University, Ekaterinburg, Russia
e-mail: shestakov.v.s@mail.ru

Abstract

Research relevance *is due to the need of mining companies in a more complete extraction of mineral resources, reduction of crude minerals losses, and extension of deposits' life. Some enterprises face the problem of complicated high-quality separation of hard-to-screen rock mass. Screen which are common in the industry often become clogged causing decline in the effectiveness of screening.*

Research aim *is to determine the links, build resonant curves and analyze the amplitude oscillations of the working member and cantilevered bars of the screen deck using numerical simulation.*

Methodology *includes the theoretical study of the dual mass oscillatory system with the use of numerical simulation.*

Results. *The motion of the vibrating screen with circular oscillations of the working member, which includes the screen deck of a cascade type with cantilevered bars, is regarded as oscillation of a dual mass system. The amplitudes of oscillations of both masses make in-phase and anti-phase movements relative to the driving force of the drive. The analysis showed that the amplitudes of the working member are practically independent of the screen deck parameters, and bars oscillation amplitudes vary over a wide range. The expressions are given calculating the parameters of cantilevered bars and the values of the initial data of the GIT-51 screen. Resonance curves of amplitude and frequency relations are constructed. The conditions are established under which the oscillation amplitudes of the cantilevered bars take on hyperadmissible values; it is shown that for a particular oscillating system there is a transitional resonance value.*

Summary. *The natural oscillation frequency ratio ranges are established of the entire system with the frequency of forced oscillations in which the oscillation amplitudes of cantilevered bars reach the specified parameters. It is shown that by changing the screen deck parameters at the design stage, it is possible to adjust the inter-frequency range and establish the operating mode of the screen.*

Key words: *screen; screen deck; oscillations; amplitude; frequency; cantilever beam; cascade.*

Introduction. Screens with various screen decks (SD) are applied internationally [1–4]. Krupp Fordertechnik and Mogensen (Germany) companies assimilate the production of screens with cascade SD in the form of fingered bars [5, 6]. The screens have circular or linear oscillations and are regarded as single mass or dual mass oscillating systems. Works [7–9] are dedicated to the study of the workflow; they consider a screen as a single mass system and study the impact of various factors on the technological parameters, vibrotransportation speed, and the effectiveness of screening.

Research problem solving. Works [10, 11] were the first to regard the movement of the vibrating screen with the inertial drive which includes SD in the form of cascade cantilevered bars as the oscillation of a dual mass system; the system of differential equations has been presented together with its solution, the numerical simulation of

the workflow has been carried out, and primary analysis of the results has been given as well as the recommendations on SD design. The screen consists of a frame, which supports the working member with mass M through resilient supports with elastic coefficient C_s. It is the first mass. The working member includes a box with side plates and a cascade screen deck with cantilevered bars interposed between the side plates. Bars represent trapezium-shaped cantilevers with individual bending stiffness; when the screen operates, they make oscillatory movements under the action of vibratory drive perturbing force which changes with frequency ω. Bars together represent the second mass m.

Oscillatory movement of cantilevered SD depends on the following parameters: M – the mass of the working member; m – the mass of bars; l – the lengths of the cantilever; b, h – the width and the thickness of a bar; m_w – the mass of the additional weight at the end of the bar; C_b – the elastic coefficient of the bar; ω_b – the frequency of natural oscillations of a bar; C_s – the elastic coefficient of basic elastic links.

Here are the expressions calculating the parameters of the cantilevered SD.

Total mass of SD cantilevered bars is determined by the formula

$$m = \frac{N_b}{g}\left(0,5G_w + 31,8\mu hl\right), \tag{1}$$

where N_b – the number of bars in SD; μ – the reduction coefficient of the mass of bars, μ = 33/140 [12].

Total stiffness of bars [13]

$$C_s = \frac{3EJ_x}{l^3}N_b = 2\cdot 10^8\frac{h^3}{l^3}N_b, \tag{2}$$

where E – the elasticity modulus; J_x – the moment of inertia of a bar cross section.

The natural frequency of the working member is determined by the formula:

$$\omega_s = \frac{30}{\pi}\sqrt{\frac{C_s}{M}}.$$

The natural frequency of bars is determined by the formula:

$$\omega_b = \frac{30}{\pi}\sqrt{\frac{C_b}{m}}.$$

For a screen of GIT type, the frequency of natural oscillations of the working member (WM) ω_s is calculated by the coefficient of system's detuning, the value of which is assigned between 5 and 6. With vibratory drive forced oscillations frequency ω = 80–90 s^{-1}, value ω_s is between 15 and 19 s^{-1}. The research has been carried out under the following values of the parameters: ω_s = 16.5 s^{-1}; M = 460 kg.

The value of stiffness for screen resilient supports is determined by the expression:

$$C_s = \pi\omega_s^2 M / 30 = \pi\cdot 16.5^2\cdot 460/30 = 12.5\cdot 10^4 \text{ N/m}.$$

With the purpose of analyzing and determining the links and restrictions occurring due to the oscillations of the system, the following dimensionless quantities have been

introduced: $\beta = A_m / A_M$ – the coefficient of exceeding the permissible amplitude of bar A_m relative to the amplitude of WM A_M; $\gamma = \omega_{sh} / \omega_{sl}$ – the ratio of the highest ω_{sh} and the lowest ω_{sl} frequencies of natural oscillations of the entire system; $\mu = \omega/\omega_{sh}$ – the ratio of forced oscillations frequency ω and the highest frequency; $\alpha = m/M$ – mass reduction coefficient.

Fig. 1 shows the graphs of screen SD parameters' (l, m, C_s, m_w) impact on the dimensionless quantities (β, γ). The connection of particular parameters of SD and their impact on system oscillations are contained in [11]. The graph shows the integrated arrangement of a range of estimates of variables. The analysis has shown that under constant values of parameters M and C_s the amplitude of WM oscillations A_M changes within a small band – from 3.9 to 4.2 mm (over the module), and the amplitude of a bar oscillations A_m changes within a wide range – from 0.1 to 55 mm (over the module).

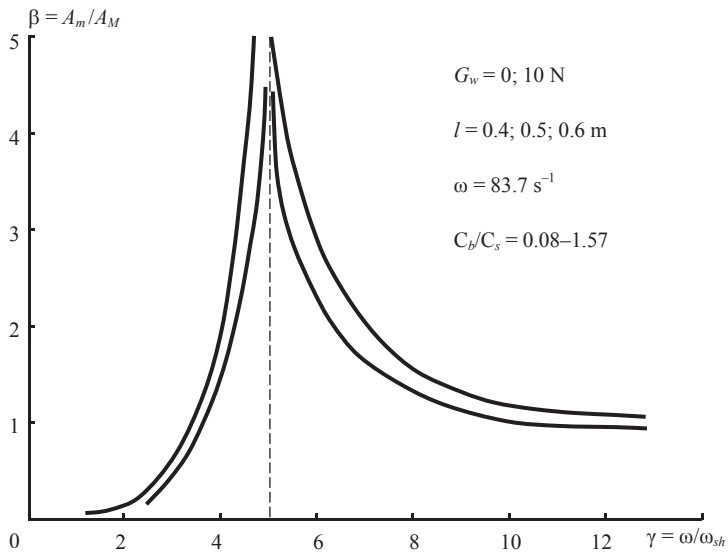

Fig. 1. The connection of the ratios of system's natural oscillation amplitudes and frequencies
Рис. 1. Связь отношений амплитуд и частот собственных колебаний системы

It is important to determine the connections and conditions under which Am rises sharply and exceeds the permissible value. Two resonances manifest under of the dual mass system oscillations: the lowest resonance with the frequency of natural oscillations of the entire system ω_{sl} and the highest frequency ω_{sh}. For the screen with the considered values of the parameters, ω_{sh} is between 16.4 and 16.9 s^{-1}, and the lowest resonance occurs under the frequency $\omega = \omega_{sh}$. At fig. 1 this state does not manifest as soon as the ratio $\omega_{sh} = \omega_{sl} = 1$ is hardly realized. Investigation has shown that when the values of γ verge towards 1, the value of the amplitudes β decreases and falls within 0.023–0.16, i. e. the value of amplitude A_m verges towards the value of amplitude A_M. Two masses move in-phase, which bears out the conclusions of the theory for of the dual mass system oscillations [12, 14]. At fig. 1 the arrangement of resonance under $\gamma = 5$ is shown, it basically depends on ω_{sh}. The position of the highest resonance at axis γ (under constant ω_{sl} = 16.5 s–1) is connected with frequency ω_{sh} coincidence with the frequency of screen forced oscillations $\omega = 83.7$ s^{-1}. Value $\gamma = 5$ is associated only with this particular system under consideration. Resonant curve (under constant ω_{sl})

resembles the outline of the curve of dynamic coefficient under the oscillations of single mass conservative systems with one resonance under $\gamma = 5$. Assuming that permissible in practice ratios $A_m / A_M \leq 1{,}5$, it may be noted that acceptable amplitudes are reached under ratios $\gamma < 4$ and $\gamma > 7$. The range of frequencies ratio from 4 to 7 determines the hyperadmissible oscillations of SD bars.

It is shown at fig. 1 that by changing parameters l, m, b, h, G_s it is possible to control the value of γ, correspondingly influence the value of ratio β and shift resonant ratio γ (in the system under consideration $\gamma = 5$). Resonant ratio γ may be referred to as "transitional". Value $\gamma < 4$ shows that the lowest and the highest resonances with frequency ω_{sl} and ω_{sh} are within the zone prior to forced oscillations frequency $\omega = 83.71$ s^{-1}, value $\gamma > 7$ is in the zone behind this frequency. By changing the structural parameters of the system (M, m, l, C_s, C_b, m_w) and forced oscillations frequency ω it is possible to establish the operation modes at the stage of design: afterresonant ($\omega_{sl} < \omega$, $\omega_{sh} < \omega$) and interresonant ($\omega_{sl} < \omega$, $\omega < \omega_{sh}$).

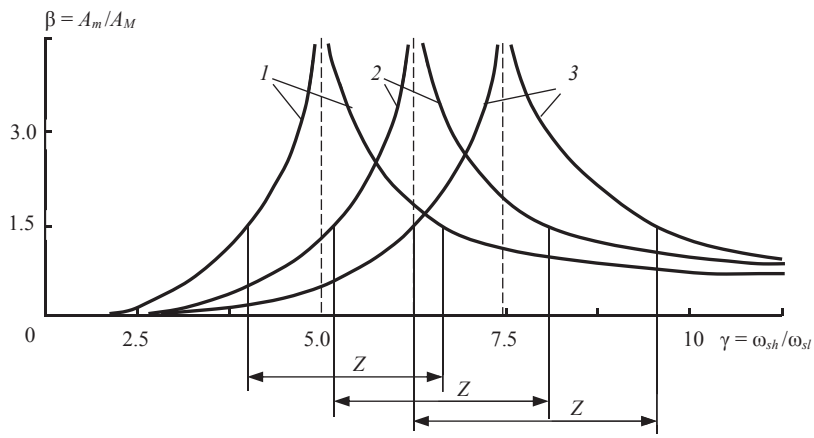

Fig. 2. Resonant curves of oscillation amplitudes and frequencies ratios if the frequencies of screen drive forced oscillations are changed:
1, *2*, *3* – under $\omega = 83.7$; 104 and 125 s^{-1} correspondingly

Рис. 2. Резонансные кривые отношений амплитуд и частот колебаний при изменении частоты вынужденных колебаний привода грохота:
1, *2*, *3* – при $\omega = 83{,}7$; 104 и 125 с$^{-1}$ соответственно

As the frequency of forced oscillations ω grows up to 104.125 s^{-1}, "transitional" value of ratio γ shifts towards larger values and will make up 6.25 and 7.5 correspondingly (fig. 2). The range of hyperadmissible values β within resonant curves Z grows as well. The growth of Z is proportional to the growth of ω and described by the regression equation $Z = 190 + 0.3\omega$. The growth of parameter Z shows that the range of frequency values of bars oscillations ($\omega_b \approx \omega_{sh}$), under which the amplitudes exceed the admissible values, extends. Curves at fig. 2 are asymmetric relative to resonant frequencies ω_r. The ascending branches of curves are steeper, and under the accepted ratio $\beta \leq 1.5$ the left asymmetry is $0.2\omega_r$ on average, and the right one is $0.3\omega_r$. Under the accepted value of ω_r, bars natural oscillation frequency should be lower than $0.2\omega_r$ and higher than $0.3\omega_r$.

Research results analysis has shown that the operational mode of the screen with bar SD and the amplitude of oscillations A_m and A_M depends on the ratio of parameters $\mu = \omega/\omega_{sh}$. At fig. 3, *a* there is a graph by means of which the conditions of exceeding bar oscillation amplitudes A_m relative to the amplitude of WM oscillations A_M under

$\omega = 83.71\ s^{-1}$. At graph $\beta = f(\mu)$ the results have been filtered, when $A_m / A_M > 1.5$. It has been predetermined that the ratio of amplitudes of β almost does not depend on the ratio ω/ω_{sl} and is distributed within the range of 4.93–5.03 under the stated ratio of amplitudes. Under μ = 1 resonant oscillations of cantilevered bars manifest, ratio β

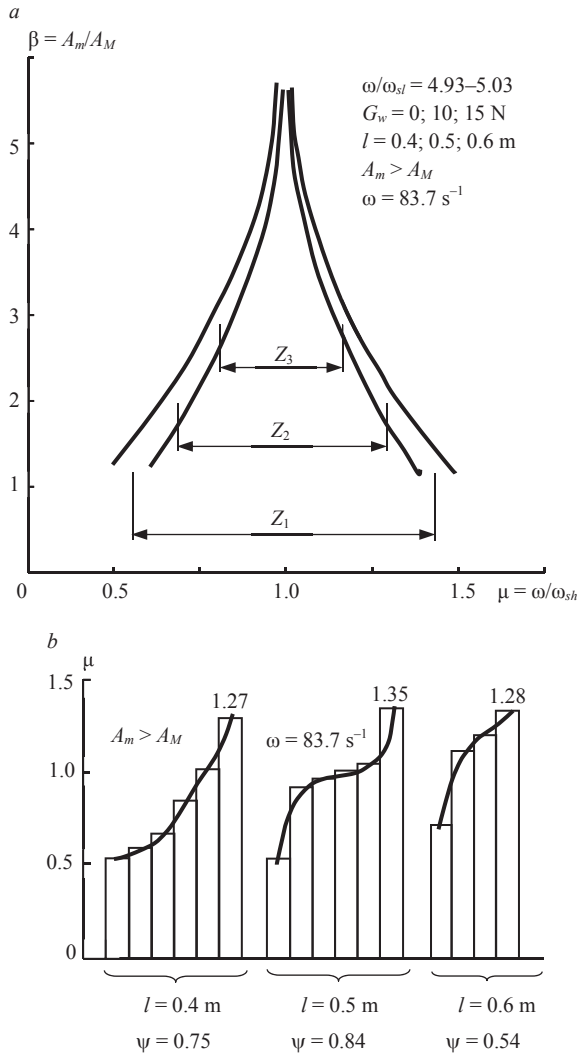

Fig. 3. System's parameters impact on the amplitude of bars oscillations:
a – resonance characteristics and the area of manifestation of hyperadmissible oscillation amplitudes; *b* – histogram of hyperadmissible oscillation amplitudes distribution under various lengths of bars

Рис. 3. Влияние параметров системы на амплитуду колебаний колосников:
a – резонансная характеристика и область проявления сверхдопустимых амплитуд колебаний; *b* – гистограмма распределения сверхдопустимых амплитуд колебаний при различной длине колосников

under the conservative system tends toward infinity. In the range of frequency ratios μ = 0.5–1.45 (parameter Z_1 at fig. 3, *a*), the ratio of amplitudes β increases sharply. When the ratio of frequencies falling into the range of μ = 0.5–1.45 (Z_3 at fig. 3, *a*),

amplitude A_m is three time as high as amplitude A_M. At can be seen from fig. 3 that it is not advisable to plan the parameters of the resilient system of bars with the ratios of frequencies μ in the range higher than Z_1. Rational range of frequencies ratios μ is within the limits prior to 0.5 and behind 1.45 under bar lengths of l = 0.4; 0.5; 0.6.

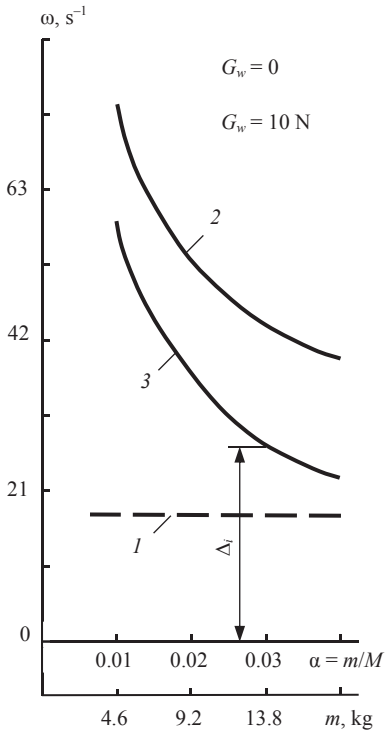

Fig. 4. Variation of system's natural oscillations frequencies depending on the mass of the cantilevered bars
Рис. 4. Изменение частот собственных колебаний системы в зависимости от массы консольных колосников

It has been stated that under identical parameters of bars the growth of bar length l reduces the coefficient of stiffness C_s by 1.72–1.95 times. At fig. 3, *b* there is a histogram of frequencies ratios μ distribution under various lengths of bars. It has been shown that the widest range of variation ψ of ratios μ is at the length of bars of l = 0.5 m, and with the length of l = 0.6 m the growth of ratios β is most apparent.

It has been stated that the range between resonances Δ largely depends on the lengths of the bars l and the mass of the bars m. In the studied system of parameters of GIT-51 screen, the operation mode may be basically defined as afterresonant. By changing the parameters of SD system it is possible to control (pull together or apart) the values of Δ bringing the system to afterresonant or interresonant modes.

As the ratio of masses $\alpha = m/M$ grows up to 0.04 (fig. 4) the lowest frequencies of natural oscillation of the entire system ω_{sl} change practically linearly and do not depend on α (*curve 1* at fig. 4). The highest frequency of natural oscillations of the entire system ω_{sh} is most dependent on the parameters of SD bars, l, m, C_s, m_w, and reduces nonlinearly (*curve 2*) with the growth of ratio α. Interresonant zone Δ largely depends on the ratio α and the mass of cantilevered bars (*curve 3*). The decline of Δ makes the start of the screen easier in the afterresonant mode of operation. In table 1 the results of numerical simulation data array processing for the studied system are shown

Table 1. The results of the system's numerical simulation depending on the mass of bars

Таблица 1. Результаты численного моделирования системы в зависимости от массы колосников

The number of curve at fig. 4	Regression equation	Correlation ratio	Fisher's criterion	Confidence estimate
1	$\omega_{sl} = 170–0.59m$	0.99	48.4	5.77
2	$\omega_{sh} = 1485m^{-0.49}$	0.95	557.7	0.14
3	$\Delta = 1620m^{-0.71}$	0.99	957.3	0.15

depending on the mass of bars. By changing l, m, C_s, m_w, et al it is possible to establish the required mass of bars and, consequently, control the values of ω_{sh} and Δ.

Conclusion. The movement of the screen with the cascade screen deck and cantilevered bars is considered as the oscillation of the dual mass oscillating system.

The presented results of numerical simulation of the process show the dependences of the working member and bar SD oscillation amplitude on the natural oscillation frequencies of the entire system and screen drive forced oscillation frequency. It has been shown that by changing the parameters of the cantilevered SD at the stage of design it is possible to control the interfrequency range and establish the operational mode of a screen. The range of ratio of system resonant frequencies has been determined under various frequencies of forced oscillations, under which the setting of bars natural oscillations determines the hyperadmissible amplitudes of oscillations.

REFERENCES

1. Vaisberg L. A., Kartavyi A. N., Korovnikov A. N. *Screen decks of screens. Design, materials, and experience*. St. Petersburg: VSEGEI Publishing; 2005. (In Russ.)

2. Vaisberg L. A. *Engineering and design of vibrating screens*. Moscow: Nedra Publishing, 1986. (In Russ.)

3. Iudin A. V., Maltsev V. A., Kosolapov A. N. H*eavy vibrating feeders and screen feeders for mine reloading systems*. Ekaterinburg: Uralskii tsentr akademicheskogo obsluzhivaniia Publishing; 2009. (In Russ.)

4. Kartavyi A. N. *Vibrating sets for mineral and technogenic crude processing. Simulation and the components of design according to the criteria of energy and resource efficiency*. Moscow: MSMU Publishing; 2013. (In Russ.)

5. Meinel A. Zu den Grundlagen der Klassierung siebschwieriger Materialien. *Aufbereitungstechnik*. 1999; 7: 313–325.

6. Meinel A. Zur Klassierung siebschwieriger Schuttueter – Enige Siebklassiererbeispiele. *Aufbereitungstechnik*. 2001; 11: 533–539.

7. Vaisberg L. A., Rubisoff D. H. Mathematische Beschreibung der Vibrationssiebung. *Aufbereitungs Technik*. 1990; 7: 378–386.

8. Ferrara G., Preti U., Schena G. D. Modelling of screening operations. *Intern. J. of Mineral Processing*. 1988; 22 (1–4): 193–222.

9. Wodzinski P. Classification and systematics of screens. *Powder*. 2007; 19 (3): 160–173.

10. Iudin A. V., Shestakov V. S., Abdulkarimov M. K. Computational scheme development and movement description of a screen with the cascade screening surface on the basis of two-masses oscillatory system. *Izvestiya vysshikh uchebnykh zavedenii. Gornyi zhurnal = News of the Higher Institutions. Mining Journal*. 2018; 4: 81–86. (In Russ.)

11. Iudin A. V., Shestakov V. S., Abdulkarimov M. K. Numerical analysis of two-masses conservative system oscillations of vibrating screen for the rock mass difficult to screen. *Izvestiya vysshikh uchebnykh zavedenii. Gornyi zhurnal = News of the Higher Institutions. Mining Journal*. 2018; 5: 93–101. (In Russ.)

12. Timoshenko S. P. *Oscillations in engineering*. Moscow: Fizmatizdat Publishing, 1959. (In Russ.)

13. Babakov I. M. *Oscillation theory*. Moscow: Nauka Publishing; 1965. (In Russ.)

14. W. Ker Wilson. *Vibration engineering*. Moscow: Izmashgiz Publishing; 1963. (In Russ.)

Received 17 April 2019

Information about authors:

Arkadii V. Iudin – DSc (Engineering), professor of the Department of Mining Machines and Complexes, Ural State Mining University. E-mail: gpt2004@mail.ru

Viktor S. Shestakov – PhD (Engineering), professor of the Department of Mining Machines and Complexes, Ural State Mining University. E-mail: shestakov.v.s@mail.ru

DOI: 10.21440/0536-1028-2020-1-97-104

Анализ связей и ограничений при колебаниях рабочего органа и консольной просеивающей поверхности вибрационного грохота

Юдин А. В.[1], Шестаков В. С.[1]

[1] Уральский государственный горный университет, Екатеринбург, Россия

Реферат

Актуальность работы *обусловлена потребностью горных предприятий в более полном извлечении полезного ископаемого, снижении потерь сырья и продлении срока службы месторождений. Проблемой, с которой сталкиваются некоторые предприятия, является сложность качественного разделения трудногрохотимой горной массы. Освоенные промышленностью грохоты часто забиваются, эффективность грохочения понижается.*

Цель работы: *используя численное моделирование, установить связи, построить резонансные кривые и выполнить анализ амплитудных колебаний рабочего органа и консольных колосников просеивающей поверхности грохота.*

Методология: *теоретические исследования двухмассной колебательной системы с использованием численного моделирования.*
Результаты. *Движение вибрационного грохота с круговыми колебаниями рабочего органа, включающего просеивающую поверхность каскадного типа с консольными колосниками, рассмотрено как колебание двухмассной системы. Амплитуды колебаний обеих масс совершают синфазные и противофазные движения относительно вынуждающей силы привода. Анализ показал, что амплитуды рабочего органа практически не зависят от параметров просеивающей поверхности, а амплитуды колебаний колосников изменяются в широком диапазоне. Приведены выражения для расчета параметров консольных колосников и значения исходных данных, принятых к анализу на примере грохота ГИТ-51. Построены резонансные кривые амплитудных и частотных отношений. Установлены условия, при которых амплитуды колебаний консольных колосников принимают сверхдопустимые значения, показано, что для конкретной колебательной системы существует переходное значение резонанса.*
Выводы. *Установлены диапазоны соотношения частот собственных колебаний всей системы в увязке с частотой вынужденных колебаний, при которых амплитуды колебаний консольных колосников достигают заданных параметров. Показано, что изменяя параметры просеивающей поверхности на этапе проектирования, можно регулировать межчастотный диапазон и определять режим работы грохота.*

Ключевые слова: *грохот; просеивающая поверхность; колебания; амплитуда; частота; консоль; каскад.*

БИБЛИОГРАФИЧЕСКИЙ СПИСОК

1. Вайсберг Л. А., Картавый А. Н., Коровников А. Н. Просеивающие поверхности грохотов. Конструкции, материалы, опыт применения. СПб: ВСЕГЕИ, 2005. 252 с.
2. Вайсберг Л. А. Проектирование и расчет вибрационных грохотов. М.: Недра, 1986. 144 с.
3. Юдин А. В., Мальцев В. А., Косолапов А. Н. Тяжелые вибропитатели и питатели-грохоты для горных перегрузочных систем. Екатеринбург: Уральский центр академического обслуживания, 2009. 402 с.
4. Картавый А. Н. Вибрационные агрегаты для переработки минерального и техногенного сырья. Моделирование и элементы расчета по критериям энерго- и ресурсоэффективности. М.: МГГУ, 2013. 328 с.
5. Meinel A. Zu den Grundlagen der Klassierung siebschwieriger Materialien //Aufbereitungstechnik. 1999. Nr. 7. S. 313–325.
6. Meinel A. Zur Klassierung siebschwieriger Schuttueter – Enige Siebklassiererbeispiele // Aufbereitungstechnik. 2001. Nr.11. S. 533–539.
7. Vaisberg L. A., Rubisoff D. H. Mathematische Beschreibung der Vibrationssiebung // Aufbereitungs Technik. 1990. Nr. 7. S. 378–386.
8. Ferrara G., Preti U., Schena G. D. Modelling of screening operations // Intern. J. of Mineral Processing. 1988. Vol. 22. No. 1–4. P. 193–222.
9. Wodzinski P. Classication and systematics of screens // Powder. 2007. Vol. 19. No. 3. P. 160–173.
10. Юдин А. В., Шестаков В. С., Абдулкаримов М. К. Разработка расчетной схемы и описание движения грохота с каскадной просеивающей поверхностью на основе двухмассной колебательной системы // Известия вузов. Горный журнал. 2018. № 4. С. 81–86.
11. Юдин А. В., Шестаков В. С., Абдулкаримов М. К. Численный анализ колебаний двухмассной консервативной колебательной системы вибрационного грохота для труднограхотимой горной массы // Известия вузов. Горный журнал. 2018. № 5. С. 93–101.
12. Тимошенко С. П. Колебания в инженерном деле. М.: Физматиздат, 1959. 439 с.
13. Бабаков И. М. Теория колебаний. М.: Наука, 1965. 559 с.
14. У. Кер Вильсон. Вибрационная техника. М.: Измашгиз, 1963. 415 с.

Поступила в редакцию 17 апреля 2019 года

Сведения об авторах:

Юдин Аркадий Васильевич – доктор технических наук, профессор кафедры горных машин и комплексов Уральского государственного горного университета. E-mail: gpt2004@mail.ru
Шестаков Виктор Степанович – кандидат технических наук, профессор кафедры горных машин и комплексов Уральского государственного горного университета. E-mail: shestakov.v.s@mail.ru

Для цитирования: Юдин А. В., Шестаков В. С. Анализ связей и ограничений при колебаниях рабочего органа и консольной просеивающей поверхности вибрационного грохота // Известия вузов. Горный журнал. 2020. № 1. С. 97–104 (In Eng.). DOI: 10.21440/0536-1028-2020-1-97-104
For citation: Iudin A. V., Shestakov V. S. Analysis of connections and restrictions occurring due to the oscillations of vibrating screen working member and cantilevered screen deck. *Izvestiya vysshikh uchebnykh zavedenii. Gornyi zhurnal = News of the Higher Institutions. Mining Journal*. 2020; 1: 97–104. DOI: 10.21440/0536-1028-2020-1-97-104