APPLICATION OF NUMERICAL ANALYSIS IN DYNAMIC STATE DIAGNOSIS OF THE MACHINE WITH A SHOCK CHARACTER OF OPERATION

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Summary

Numerical analysis is a basic tool for the engineer. Over the years many methods have been developed which turned out to be more or less universal in applications. In the approach to the problem of machine vibration damping based on the machine model, many approximate calculation methods must be used or combined so as to benefit from their advantages and eliminate possible weak points. The hybrid method, developed as a combination of a deformable finite elements method and the dynamics of multi-link systems, was used in the study as one of the tools for vibrating screen's dynamic state diagnostic. The exemplary application of numerical modelling to the issue of multidirectional optimization of an effective machine vibration isolation at the design stage is presented.

Keywords: numerical methods, diagnostics, resonance frequency, structure vibrations, vibration isolation

ANALIZA NUMERYCZNA W DIAGNOZOWANIU STANU DYNAMICZNEGO MASZYN O UDAROWYM CHARAKTERZE PRACY

Streszczenie

Analiza numeryczna stanowi podstawowe narzędzie inżyniera. Na przestrzeni lat powstało wiele metod, które okazały się w zastosowaniach mniej lub bardziej uniwersalne. Podejście do zagadnienie tłumienia drgań maszyny w oparciu o jej model wymaga stosowania wielu metod obliczeń przybliżonych lub ich łączenia w sposób umożliwiający wykorzystanie zalet każdej z nich oraz wyeliminowanie ewentualnych wad. Metoda hybrydowa powstała z połączenia metody odkształcalnych elementów skończonych i dynamiki układów wieloczłonowych została wykorzystana w pracy jako jedno z narzędzi diagnozowania stanu dynamicznego przesiewacza wibracyjnego. Pokazany został przykład zastosowania modelowania numerycznego do zagadnienie wielokierunkowej optymalizacji przy opracowaniu skutecznej wibroizolacji maszyny.

Słowa kluczowe: model numeryczny, diagnozowanie, częstotliwość rezonansowa, drgania konstrukcji, wibroizolacja

1. INTRODUCTION

In case of a machine with the impact type of operation, the diagnostics of dynamic state is much more difficult than a similar analysis for equipment and devices in which vibrations occur as the undesirable factor associated with their operation. In case of facilities like mills, sieves and shaker conveyors, vibrations of working elements constitute a principle of their operation [1, 2].

A shaking sieve consists of a working part which vibrates with a speed as high as 100 mm·s^{-1}; the travel is even of several millimetres, and the supporting structure is permanently fixed to the foundation. Therefore, the vibrations of this part of machine structure should be as low as possible. Thus, we are facing a situation in which one of machine elements vibrates and these vibrations cannot be avoided by eliminating their origin [3].

High positive forces acting on the working elements are favourable for the examination of their resonance characteristics. On the other hand, if the dynamic state of the machine is incorrect, possible structural changes must be implemented with great caution; otherwise they may cause dangerous effects to the surroundings [4, 5].

2. VIBRATION CHARACTER EXAMINATION FOR VIBRATING SCREEN

The sieve, which is the main part of the screen, has a mass of several tonnes and the instantaneous value of its absolute acceleration reaches 30-40 m·s^{-2}. In such circumstances the force applied by such mass to the supporting structure exceeds the value of 100 kN.

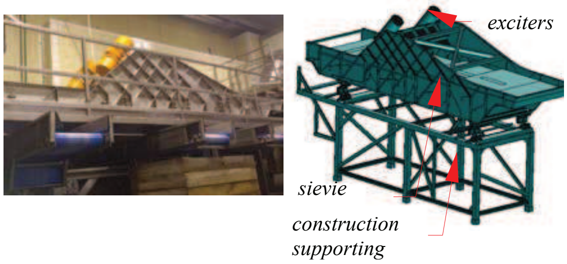

Fig. 1. ROSTA vibrating screen produced: a) general view, b) structural diagram

In case of screens the sieve vibration frequency is particularly important. This frequency may reach the value between ten and twenty Hz, significantly higher than for mills or eccentric hammers with a large mass of the connecting rod/beater system. The inertial force resulting from the sieve acceleration and mass transfers to the sieve supporting structure. With properly selected vibration isolators the forces transferred to supports are less than the positive input value, nevertheless they can also reach a significant level. This leads to the formation of cracks and breaks in the sieve supporting structure and causes subsidence of foundations. Both the foundation and the sieve supporting structure resting on it should be properly designed to ensure sufficient strength of elements and, moreover, they should be feature vibration-damping properties.

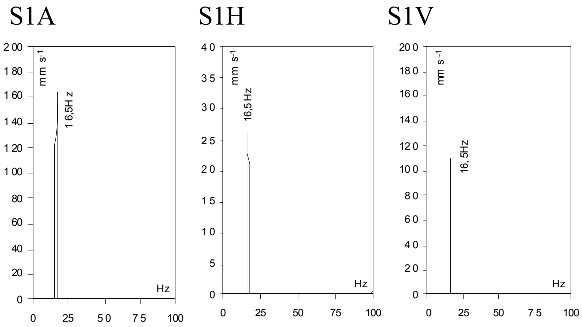

Fig. 2. Amplitude-frequency curves for the vibration speed of the sieve in the following directions: lon gitudinal (A), transversal (B), and vertical (C) with respect to the machine axis

The amplitude-frequency curves for vibration speed of the front part of the sieve in the direction of the raw material movement are presented on the graphs (Fig. 2).

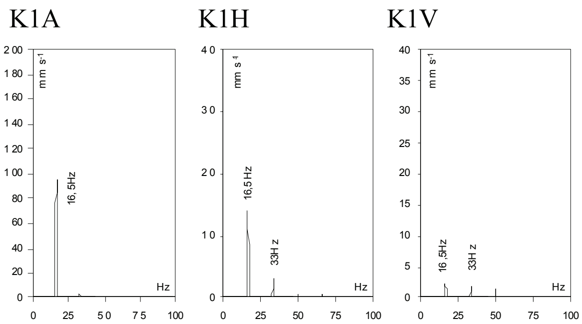

Fig. 3. Amplitude-frequency curves for the vibration speed of the screen structure in the following directions: longitudinal (A), transversal (B), and vertical (C) with respect to the machine axis

The vibration speed amplitude in the direction of the screen axis reaches the value of 100 mm·s^{-1}. In the direction perpendicular to the sieve axis the vibration speeds are also high, reaching the level of 15 mm·s^{-1} in the front part and up to 20 mm·s^{-1} in the rear part. Much smaller values for vibration speed amplitude were measured in the vertical direction. This is understandable, because the structure shows the least deformation in this direction.

The highest sieve vibration speeds are to be found in the direction perpendicular to its axis (~160 mm·s^{-1}), however vibration speed amplitudes in the vertical direction reach similar values. This results from the design of vibration isolators used in the vibrating screen.

The resonance frequencies were determined using the short-time Fourier transformation of the vibration speed vs. time curves determined during machine coasting. The measurements were taken in three directions, in the points situated in the front and rear part of the sieve.

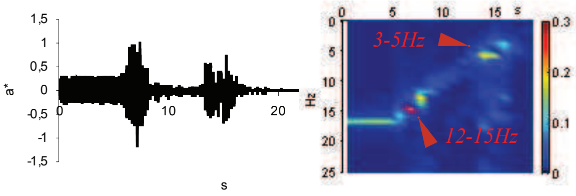

Fig. 4.Time waveforms and short-time Fourier transform of the acceleration of structural vibrations during sieve coasting

The shape of the transform presented in Fig. 4 suggests unfavourable distribution of frequency ranges in which free vibrations of the structure are excited. Free vibrations occur within the range of 3-5 Hz and 12-15 Hz. While the lower frequency vibrations unfavourably affect the structure life during machine start-up and coasting phases, the higher frequency range is located near the shaker excitation frequency.

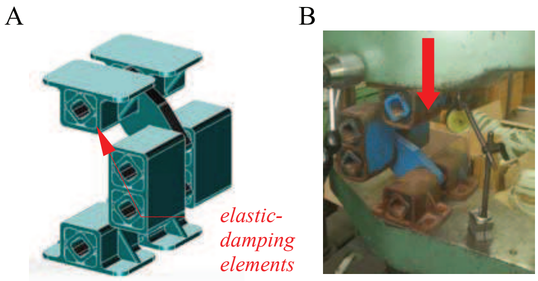

Fig. 5. Design of ROSTA vibration isolator (A) and its static characteristic curve determination method (B)

The sieve part of the vibrating screen rests on the vibration isolators manufactured by Spanish ROSTA (Fig. 5). They are isolators with plastic inserts used as the elastic-damping elements. Their design allows deflection in both vertical and horizontal directions.

Table 1. Features of the vibration isolator type AB 50 TWIN

Load [N]	Free vibrations frequency [Hz]	Horizontal rigidity [N mm^{-1}]	Vertical rigidity [N mm^{-1}]
5,000 – 12,000	2.1 – 2.4	170	340

The rigidity values of this vibration isolator are shown in Table 1. The manufacturer guarantees its correct operation under the load within the range of 5.000 – 12.000 N.

A very important parameter characterizing the vibration isolator is its free vibrations frequency. This parameter should be understood as the resonant frequency of the system composed of the vibration isolator itself and the loading mass ensuring the static deflection value recommended for this type of a vibration isolator [6].

For the ratio of the excitation frequency to the free vibrations frequency the value of mistuning factor is:

$$\varepsilon = \frac{\omega}{\omega_0} = \frac{16.5\cdot 2\pi}{2.4\cdot 2\pi} = 6.8 > \sqrt{2} \qquad (1)$$

Therefore the criterion of correct isolation is met.

In order to verify the data published by ROSTA, the characteristic curve of the vibration isolator's horizontal rigidity was determined. The determination method is shown in Fig. 5. Based on the measurement results of deformation as a function of applied force, it was possible to determine the curves presented in Fig. 6.

A B

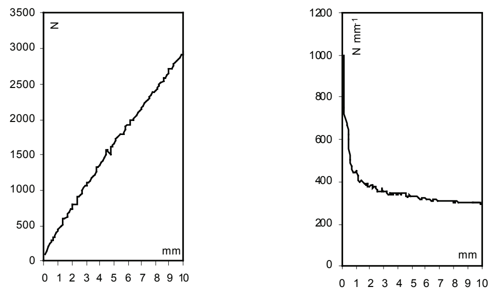

Fig. 6. Static characteristic curves for the vibration isolator: (A) force-deformation, (B) rigidity-deformation

The graph clearly shows the linear relationship between the isolator deflection and the loading force, therefore a wide range of deflection exists for which the rigidity values vary only within a narrow range of values.

A B

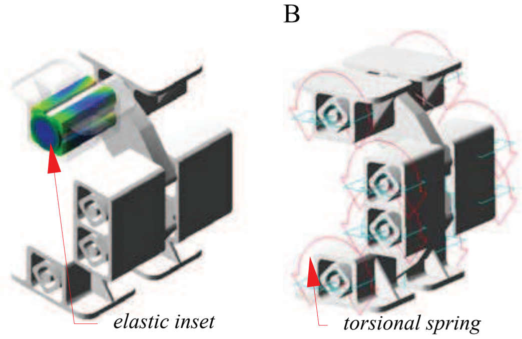

Fig. 7. Models of vibration isolator: with an elastic insert (A), and with elastic-damping elements (B)

The modelling of vibration isolator with elastic inserts made of highly deformable plastic (Fig. 7A) is a quite complicated task, because the interactions between the insert and the surfaces of the nest are of contact nature [7]. For the purpose of correct representation of these interactions the values of parameters valid in the contact zone, first of all the rigidity, must be known. Consideration of the contact nature of interactions, even in the rigid body – deformable body system, significantly extends the duration of calculations. Therefore, for the purpose of numerical analysis the vibrating isolator model was assumed composed of rigid solids and elastic-damping elements in the form of torsion springs with the rigidity factor of 3.3 N·mm·deg^{-1} (Fig. 7B).

A B

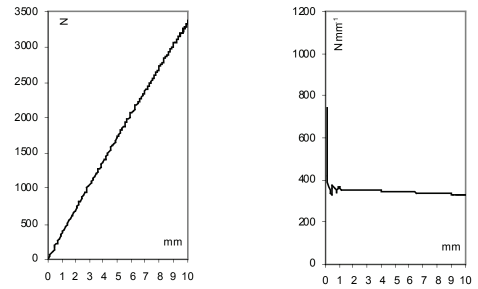

Fig. 8. Static characteristic curves for the vibration isolator: (A) -force-deformation, , (B) - rigidity-deformation

The static characteristics for the vibration isolator model with assumed rigidity of springs is close to static characteristics determined for the examined isolator. It can be seen in Fig. 8(A) and, first of all, in Fig. 8(B).

In the numerical analysis the vibration isolator was statically loaded with the force of 8 750 N equal to the fraction of sieve weight acting on each of the four supporting elements. The assumed value of the dynamic load was 32 000 N, which corresponds to the shaker excitation force.

A B

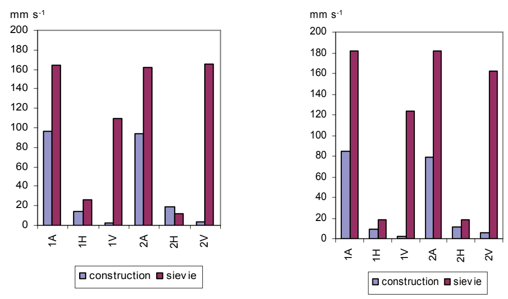

Fig. 9. Sieve and supporting structure vibration speed amplitudes: (A) for the real object, (B) for the model

The efficiency of the vibration isolator in the aspect of vibration damping can be determined based on the ratio of the sieve and the support vibration speeds. The figures 9(A) and 9(B) show respectively the amplitudes of the sieve vibration speed (brown bar) and the supporting structure (blue bar) for the vibrating sieve and its numerical representation. Based on these figures, the conclusion may be drawn that responses of both systems on identical excitation are almost the same. This is a substantial confirmation of compliance between the dynamical features of the model and a real object. In each case, the ratio of sieve and support vibration velocity is less than one for both horizontal and vertical vibration direction.

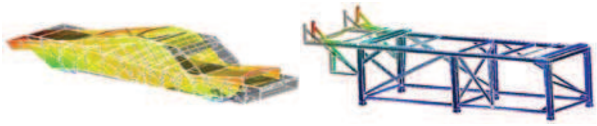

Fig. 10. First forms and free vibration frequencies for the sieve and supporting structure

In designing the model for numerical calculations, the following rules were observed:

– The sieving part of the vibrating screen constitutes a rigid structure. It can be concluded from the natural vibration frequency (Fig. 10). The free vibration frequencies of the sieve and the supporting structure do not differ dramatically because the sieve mass is much greater than the mass of the supporting structure.
– The sieve, as the rigid element, affects the screen dynamics through its inertia only.
– The sieve is connected with a compliant part, i.e. the frame, through elastic-damping elements, i.e. vibration isolators.

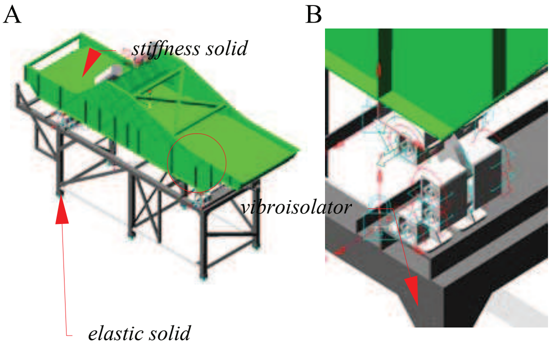

Fig. 11. The hybrid model of vibrating screen (A) and the connection of a rigid solid with a deformable solid by the elastic-damping element in the hybrid model (B)

Such an arrangement is shown in Fig. 11 and is very well suited to modelling by using the method of multi-body systems (MSD - *multibody system dynamics*) [8, 9.

The model built for the purpose of numerical analysis consists of a rigid solid corresponding to the sieve with respect to the mass and moments of inertia. The supporting structure consists of solid elements in the form of profiles divided into deformable finite elements. Such a combination of the multi-mass dynamics method with the method of deformable finite elements leads to the hybrid model having this advantage that it allows to determine system's response to excitation, along with determination of stresses, for the elements subjected to the analysis only. Consequently, the calculation duration is much shorter. In the model of the vibrating screen the previous model of vibration isolator was used.

Fig. 12 shows two basic frequencies and corresponding forms of free vibrations of the vibrating screen. Since these frequencies are both lower than the excitation frequency, they are distinctly visible on the short-time Fourier transform (*STFT*). The results obtained are consistent with the results of object's examinations, during which free vibrations were found with the frequency of 3 – 5 Hz (as compared to 4.9 Hz occurring in the model) and 12 - 15 Hz (14.7 Hz in the model). The next calculated free vibration frequency is 22.2 Hz. Since it exceeds the excitation frequency by almost 6 Hz, is not visible on the vibration spectrum even with small damping occurring in the system.

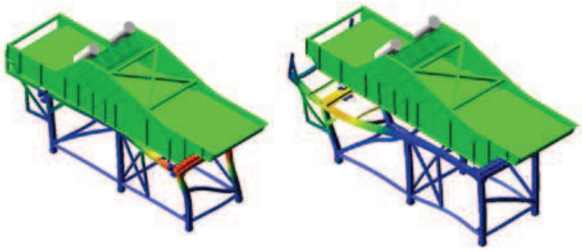

Fig. 12. Basic frequencies and corresponding forms of free vibrations of the vibrating screen

The character of system vibrations determined for the model is similar to the character of the real object's vibrations.

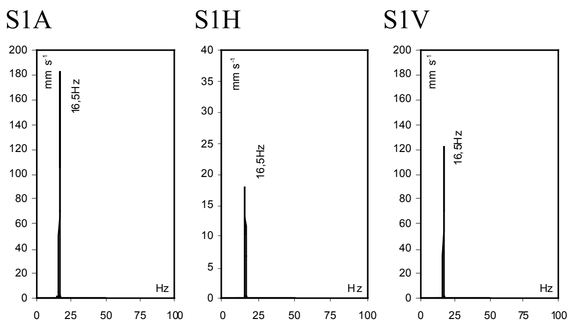

Fig. 13. Amplitude-frequency characteristics for vibration speed in the model of sieve

This conclusion can be drawn by comparing speed spectra presented in Figure 2 & 3 with amplitude-frequency characteristics shown in Figures 13 & 14.

Having the model verified, we can consider the modification of supporting structure in such a way that the vibration level is reduced while the required parameters of sieve vibration are maintained. An exemplary solution is shown in Fig. 15.

K1A K1H K1V

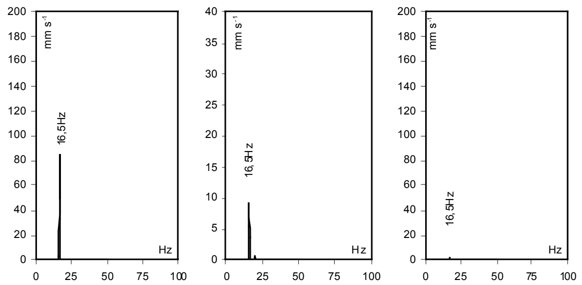

Fig. 14. Amplitude-frequency characteristics for vibration speed in the model of the vibrating screen supporting structure

A B

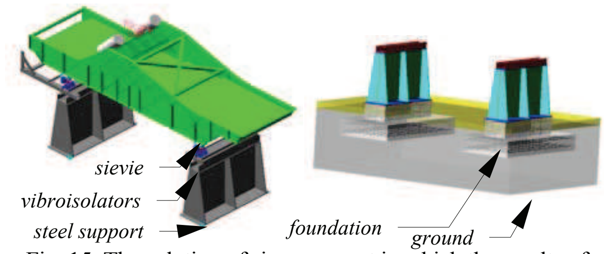

Fig. 15. The solution of sieve support in which the results of system vibrations numerical analysis are taken into account: A – supports and sieve, B – supports and foundation

Admittedly, in this solution the resonance frequency is lower than excitation frequency and during machine start-up and coasting the amplitude of vibrations will be high, but this effect cannot be eliminated due to the requirements specified for vibration isolators.

3,9 Hz 20,8 Hz

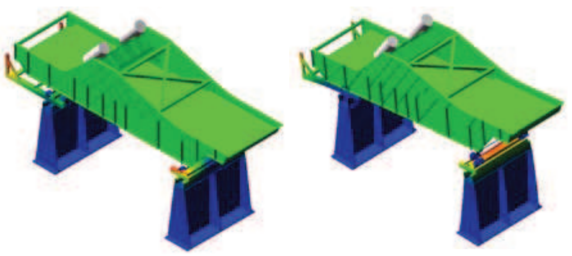

Fig. 16. Forms and the lowest frequencies of free vibrations of the vibrating screen following foundation design modification

Based on the analysis of vibration forms it can be seen that at the resonance frequency of 3.9 Hz the vibrations of the sieve will occur rather than vibrations of the supporting structure and the foundation. The next free vibrations frequency is 20.8 Hz and, being higher than the excitation frequency (16.5 Hz), it does not create a risk for the lifetime of the screen's supporting structure.

S1A S1H S1V

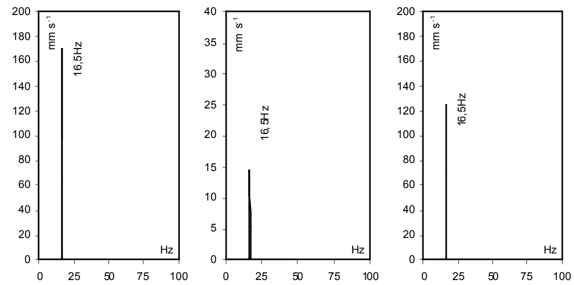

Fig. 17. Amplitude-frequency characteristics for the vibration speed of the sieve part of the vibrating screen after modification of the sieve supporting structure

K1A K1H K1V

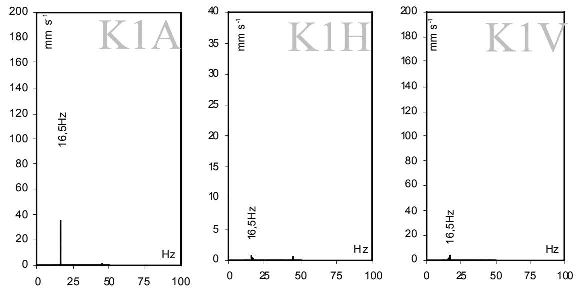

Fig. 18. Amplitude-frequency characteristics for vibration speed of modified sieve supporting structure

From the structure durability point of view, the foundation mass and strength are important. Stresses occurring in both foundation blocks and steel members of the supporting structure can be determined numerically based on the model.

Table 2. Loads exerted on the vibrating screen support

Item	Load type	Load value, N	
		total	per support
1	Vibrating screen mass	36,297	18,149
2	Shaker force, vertical	128,000	64,000
3	Shaker force, horizontal	128,000	64,000
4	Steel support mass	29,640	14,820
5	Foundation & reinforcement mass	227,920	11,3960

Table 2 presents the loads acting on the supports, i.e. sieve weight, force exerted on the sieve by shakers, support weight, and foundation weight. For these loads the values of deformations and stresses occurring in the cross-sections of the supports and the foundations, and also in the ground should be determined.

A B

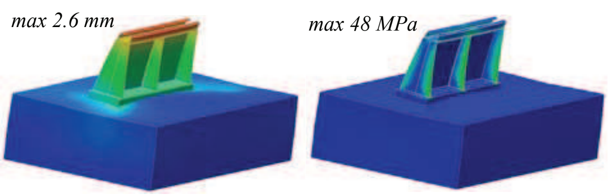

Fig. 19. The value and character of deformation of the support (A); and stress distribution in steel part and ground (B)

As it follows from calculations, the maximum deformation of the metal part of the support should not exceed the value of 2.6 mm, whereas the maximum reduced stress will occur in cross-sections of the support plate reinforcing elements and it will reach the value of 48 MPa. These results are satisfactory in the aspect of support strength. The stresses in the foundation concrete blocks are small.

3. CONCLUSIONS

It was the purpose of this study to demonstrate the usefulness of numerical methods to determination of dynamic properties of the examined system based on its numerical model. In order to obtain reliable results, the representation used for this purpose should be very close, if not identical, to the real object with respect to basic properties like mass, rigidity and damping. The most difficult issue is to make a reliable estimation of two last quantities. As it was pointed out, such an estimation is not necessary for each member of a system which was constructed as the model of the analysed machine. The examined case of two massive members, i.e. the sieve and the sieve supporting structure, joined together by elastic-damping elements is a good example of application of the hybrid method for machine dynamic state analysis. The sieve itself, with its strength properties and vibration eigenforms which are not essential for the problem, is modelled as a rigid solid. The situation is different in case of the sieve supporting structure which, similar to the foundation or ground, may not be treated as a rigid structure. The dynamic properties of such a model should be specified on the basis of real object examination results. The structure mass is usually known with sufficient accuracy. The rigidity of the system can be assumed based on the determined resonance characteristics. The most difficult problem results from the insufficient knowledge of damping in the system [10]. Such damping can be determined based on the response of the system to a specified excitation.

The problem of structure vibration damping can be approached differently than presented in this study, i.e. by using a pure experimental method. However, such a method has one obvious disadvantage, namely the suitability of a given solution can be evaluated only when the solution is fully implemented. Regrettably, it can turn out that the problem is not eliminated at all. If the concept is verified at the earlier stage based on the model, the risk associated with such "hit and hope" method is eliminated, because already the shape of amplitude-frequency characteristics of vibration speed for the model of the vibrating screen with modified supporting structure indicates that the modification of the vibrating screen mounting structure, basically consisting in making it more rigid, will cause dramatic reduction of vibrations of these machine elements which are required to ensure the smallest vibration amplitudes while maintaining high amplitudes of sieve oscillations, necessary for correct segregation of individual fractions of the material being sieved.

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