Hammer crusher - influences of design and execution of vibroprotection and machine properties on vibration intensity

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Abstract

On an example of a hammer crusher the author analyzes the factors participating in the proper operation of any machine. Their number includes the estimate of load magnitude, the safety reserves in its determination, the influence of material characteristics of the machine, vibration isolation and subgrade, the influence of the execution of vibroprotection arising from installation inaccuracies and, last but not least, the influence of the machine design. The individual factors are assessed quantitatively and qualitatively on the example of a comparison of response produced by a theoretical model and experimental results.

Keywords: hammer crusher, vibroprotection, dynamic analysis

1. Description of the machine / foundation system

Elastic supporting on foundations was designed for a hammer crusher of rotor speed of 24.7 Hz, mill weight of 6.78 t and electric motor drive weight of 2.12 t. The foundation block with the firmly connected machine and its motor drive was sprung against the foundation tank structure by 8 elastic elements of 2 viscous dampers. The inert mass of the R.C. foundation together with the machine represented a system of total weight of 40 t.

To assure the minimum dimensions of the foundation tank and to prevent its interference with the foundations of the hall columns the foundation block was not supported at the bottom, as is usually the case. Instead of that 5 pairs of steel cantilever beams consisting of two plate-connected 180 mm channels were used projecting from the foundation block near its upper surface. The elastic damping elements were placed under the steel cantilevers.

2. Tuning of the system

With regard to operating speed the lowest 6 natural frequencies of the foundation block/machine system were tuned so as not to influence the machine within the range of its operating speed. These basic natural frequencies of the foundation block with the machine on springs were designed in the frequency range of 1.8 – 4.4 Hz. Further higher natural frequencies of the foundation mounting (foundation block bending, etc.) were high above the operating speed of the machine.

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Frequency No.	Design	Inaccurate installation of spring elements	Locked spring elements
1	1,793	1,789	6,075
2	2,065	2,185	6,299
3	2,456	2,449	8,358
4	3,053	3,007	13,34
5	3,480	3,501	16,58
6	4,403	4,437	18,19
7	92,81	92,78	94,65

Tab. 1. Computed natural frequencies [Hz] of machine / foundation system.

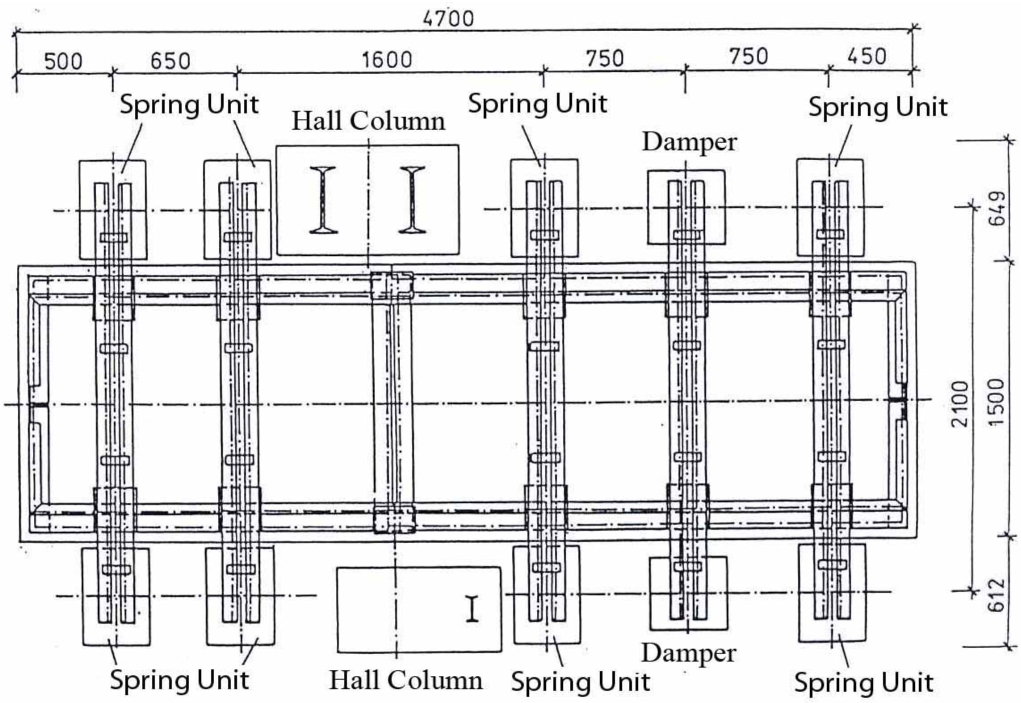

Fig. 1. Plan view of machine foundation with spring and damping units.

For reasons of economy the viscous vibration absorbers of the foundation block were not installed during actual construction. Moreover, the contractor placed the steel channel cantilevers very inaccurately. Due to greater span (by 70 mm) caused by inaccurate installation of spring elements the flexural rigidity of cantilevers supports was lower than designed, which was the reason for successive cantilever strengthening with welded vertical and horizontal plates until ten times the initial flexural rigidity of twin channel cantilevers has been achieved. When the machine was started while the spring elements were concealed, it was ascertained that the spring elements had been installed under the block, but were not released (unlocked), as a result of which the block vibrated on steel channel cantilevers only – naturally in higher frequencies.

These phenomena were discovered by measurements and a detailed dynamic analysis (Table 1). It was ascertained that further cantilever strengthening had practically no significant influence on the final behaviour of the machine in operation. A change of cantilever length (span), possibly a change of its rigidity by the welding of further plate splices would change the tuning of the foundation, but in the range very distant from that of operating speed of the machine. Both the computations and the measurements have proved the insensibility of the system in the operating speed range to afore mentioned modifications.

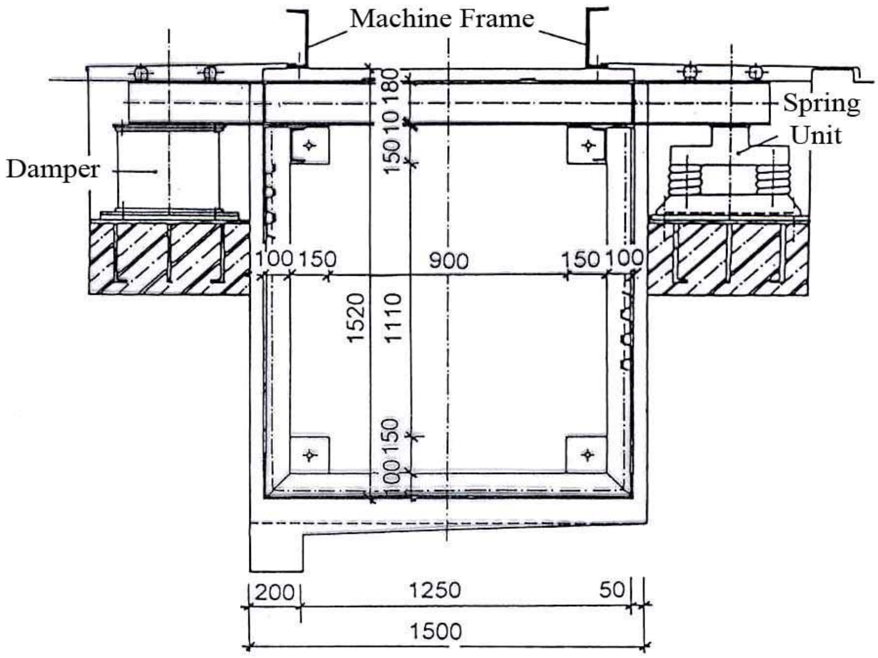

Fig. 2. Cross section of machine foundation.

3. Forced vibrations of the system

The magnitude of excitation forces of the hammer crusher was not available for the design of machine springing. Therefore this magnitude was estimated on the basis of rotor mass (M = 1.913 t).

According to ČSN 73 0032 this estimate is based on the possible unbalance of the machine, unless the manufacturer specifies the excitation forces more accurately. In this particular case the possible unbalance of the crusher rotor for the speed of 24.7 Hz was determined at $\varepsilon_p\,\omega^2 = 0.105$ ms^{-1}, where ε_p is the specific unbalance and ω is the circular operating excitation frequency.

The determination of excitation force according to ČSN 73 0032 used a safety reserve corresponding to the coefficient of 6.5 (determined with regard to the possible wear of the machine) and the reserve with a coefficient of 1.15 (indeterminateness of unbalance determination) for the estimate of the standard value of excitation force amplitude

$$F = 1.15 \cdot 6.5 \cdot M \cdot \varepsilon_p\,\omega^2 = 1.501 \text{ kN}. \qquad (1)$$

A comparison of the requirements of the afore mentioned standards with other sources has revealed that the authors of earlier works (V. Koloušek, E. Rausch, A. Major and others) used higher excitation force values specified as 20% to 30% of rotor weight, i.e.

$$F = (0.2 \text{ to } 0.3) \cdot 19.13 = 3.8 \text{ to } 5.7 \text{ kN}. \qquad (2)$$

The design used the conservative value of excitation force, i.e. 5.7 kN, which corresponded with the maximum response of (idle run) stationary vibrations:

41.5 μm vertically during the machine speed passage through natural frequencies of crusher mounting, while considering the damping by the logarithmic damping decrement of 0.1 for R.C. block and 0.01 for steel and spring elements,

9.7 μm for operating speed.

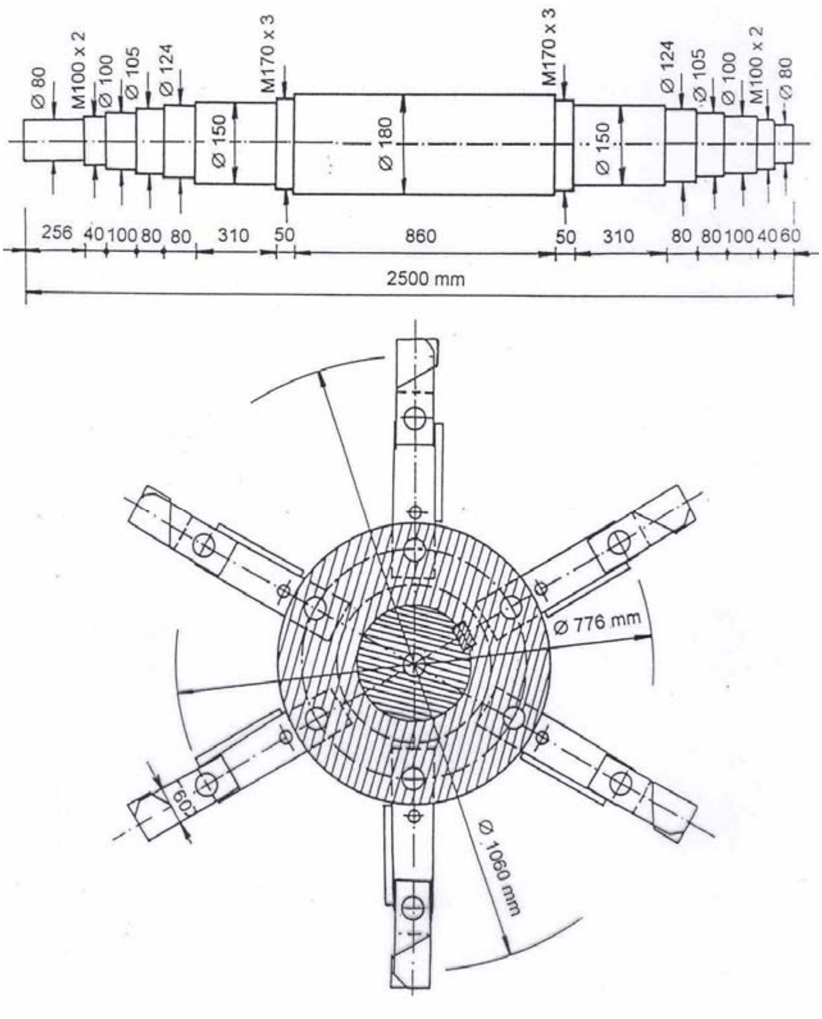

Fig. 3. Machine rotor plan.

4. Start of machine operation

The machine starting was accompanied with repeated dynamic measurements of the system in several phases. These measurements have made it possible to discover that the concealed spring insulators were not released which was relatively easy to remedy. More complicated was the fact that even during the idle run of the machine at the operating speed of 24.7 Hz the amplitudes of the vibrations of the system increased as if the crusher ran in resonance (Fig. 4). This probable resonance manifested itself not only by a considerable increase of crusher vibrations on elastic elements, but also by the rattling of all parts and plates connected with mill foundations. Vibration measurements confirmed that the excitation force of the rotor was not simply harmonic, but that higher harmonic components were superimposed on the basic 24.7 Hz frequency, depending on the quality of rotor balancing (In foundations higher, possibly also lower harmonic modes do not become dominant, as a rule).

The analysis of the system response records made during starting and unbraked termination of the machine run has revealed that the speed corresponding with half, third and quarter frequency of operating speed of the rotor system generates resonance vibrations of the system (Fig. 4). This significant influence of the fractions of the speed frequency on the magnitude of excitation force was caused by an incorrect method of balancing the rotor with attached hammers. The operator balanced the crusher not by assuring practically the same weight of the crusher hammers about the whole rotor circumference, i.e. on all six arms, but merely by endeavouring the axially symmetrical opposite hammers to be of the same weight.

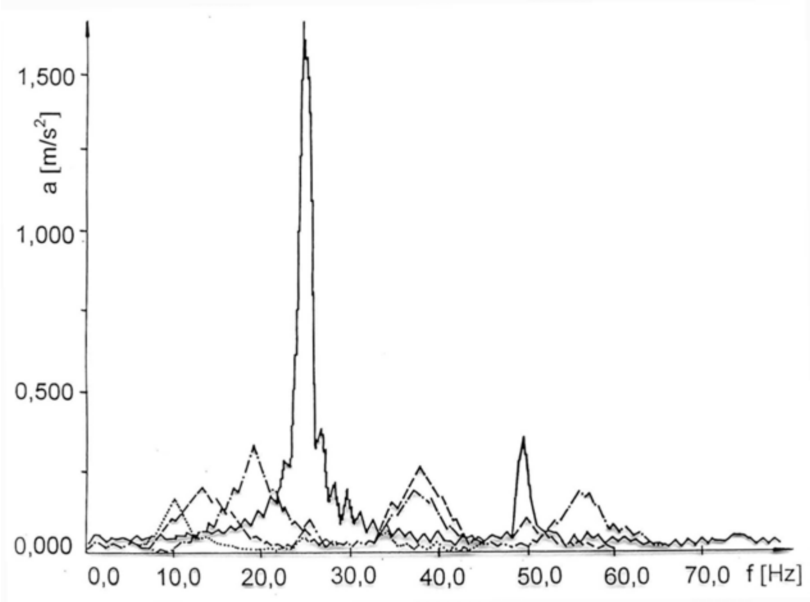

Fig. 4. Frequency spectra of foundation vibration (for various parts of foundation block).

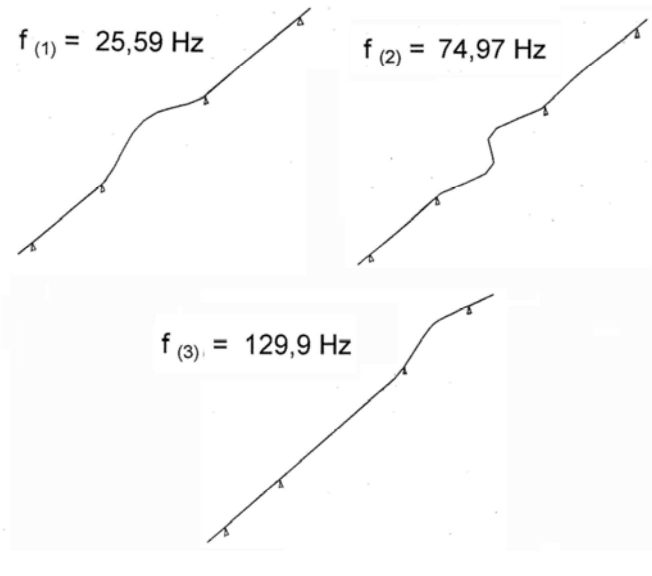

Fig. 5. The first three natural modes of rotor axis.

So it happened that in one case simulating the unbalancing of the mill by the removal of one hammer, i.e. by an artificial unbalance of several kg, the vibrations of the machine were calmer than in complete state and the whole mill seemed better balanced.

This situation resulted in the search for the causes of considerable machine vibrations during operating speed in the very characteristics of the rotor system of the machine. The next phase was devoted to the verification of the possibility of machine rotor resonance (Fig. 3) with the operating frequency of the machine.

The simplified rotor natural calculation made for this purpose did not include the influence of eccentricity of rotor unbalance and the gyroscopic effect on the spectrum of natural frequencies and vibration modes of the rotor. For this reason the results were considered merely informative. The computation has proved very good agreement of the first natural frequency of the rotor with the operating speed of the machine (Fig. 5). This agreement was the source of excessive vibrations of the system caused by an enormous increase of the order of excitation forces with the natural frequency of the crusher rotor.

Moreover, the incorrect balancing of the whole rotating system increased the magnitude of excitation forces even further. This has made it possible to explain the resonance behaviour of the system in the region of 12.5 Hz (half the resonance frequency of 24.7 Hz) and its second and third multiple of 50 Hz and 75 Hz. The machine manufacturer has not acknowledged this shortcoming and the machine was replaced with another type with modified mounting on foundations.

5. Conclusion

If the machine rotor is in resonance with rotor operating frequency, the assumptions of standards and other similar rules concerning the magnitude of excitation forces need not apply. The magnitude of excitation forces currently varies within decimal multiples of rotor mass and lower values. The effect of crushing represents an increase of excitation forces of some + 30% as compared with idle run. In resonance, due to whatever reason, progressive increase of excitation force can be expected, which will manifest itself, naturally, by excessive vibrations of the system. Sprung foundations cannot be designed with sufficient reserve to satisfy the afore mentioned multiple ”overload” produced by the incorrect design of the machine, unless this state has been made known by the manufacturer or user of the machine. Even then this fact results in considerable ineffective oversizing of the foundation block of the system. The paper presents an example of errors which can occur during the installation and starting of operation of the machine.

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