Modelling and Numerical Simulation of Parametric Resonance Phenomenon in Vibrating Screen

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Abstract

In this paper the numerical studies of parametric resonance phenomenon in vibrating screen are presented. Numerical simulations are performed in Ansys Workbench software. Modal analysis is carried out to find the natural frequencies and mode shapes of the sieve. The effect of the excitation frequency on the sieve vibrations in parametric resonance conditions was investigated using the transient analysis. The comparison of numerical and experimental results is presented. It is shown, that two mode shapes of sieve vibrations occur close to the screen operation frequency. Linear dependence between excitation frequency and sieve vibration frequency is obtained. The most stable transient response and the highest vibration amplitude of the sieve is obtained for excitation frequency 47.07 Hz. The range of excitation of parametric resonance is nearly the same as for experimental data.

Keywords: screen, parametric resonance, forced vibrations

1. Introduction

Screening operations are very important part of processing mineral materials. Screens are fundamental instrumentation for minerals separation in order to produce final mineral products for customers. Vibrating screens are one of the most extensively used tools in screening processes. Rapid evolution of vibrating screens occurred in 19th and 20th centuries. Nowadays the level of screens development is stabilized and machine building companies often produce similar screens, and their construction differs in details [10].

The screening process of the naturally wet mineral materials is generally more difficult in comparison to screening of the dry mineral materials. Here, particles of the material combine to form aggregates, that significantly increase the time of the screening process [11]. Therefore, the water supply need to be applied for material particles disintegration. The other solution of this problem is to generate high impact energy by increasing vibration amplitude, that can crush glutted grains of material and degrease adhesion forces between the material and the sieve. The large mass of the conventional screens connected with large amplitude vibration results in reduced life of a machine and increases the energy consumption [9]. Exciting of sieve parametric vibrations in the screen results in the large amplitude vibrations with relatively low energy consumption and can be suitable for screening wet materials.

The first screen with parametrically excited sieve was designed by Slepyan et al. [5]. They also found the mathematical model of the simplified vibrating screen system, where the sieve is modelled as a string connected with two masses [6-7]. The dynamic analysis of vibrating screen was presented in work [1]. The analytical and numerical methods were used to find the sieve natural frequencies and mode shapes. In work [2] the experimental analysis of vibrating screen operation in parametric resonance conditions was presented. In this paper the full plate was used instead of the perforated sieve. Complex dynamic analysis of the large vibrating screen was presented by Zhao et al. [12]. They found optimal dynamic design of the screen by performing structural optimization. Li and Song [4, 8] presented the dynamic analysis of chaotic vibrating screen. Another dynamic analysis of vibrating screen with variable elliptical trace was presented by He and Liu [3]. They analyzed characteristics of the screen by applying multi-degree-of-freedom theory. The kinematic parameters for different motion traces were also determined.

The present paper concerns the dynamic analysis of vibrating screen system with parametrically excited sieve. Numerical simulations were performed to find the effect of excitation frequency on sieve parametric oscillations.

2. Numerical modelling of parametric resonance screen system

As shown in Fig. 1 the model of parametric resonance screen system was prepared according to laboratory parametric resonance screen in Ansys Workbench software [2]. It is simplified to two beams, that are connected with a sieve (plate with rectangular cut-outs). The sieve is fixed inside the beams between the rubber pads. The whole system is suspended by springs with stiffness k equal to 275 N/mm and preload equal to quarter of sieve preload - $F_t/4$.

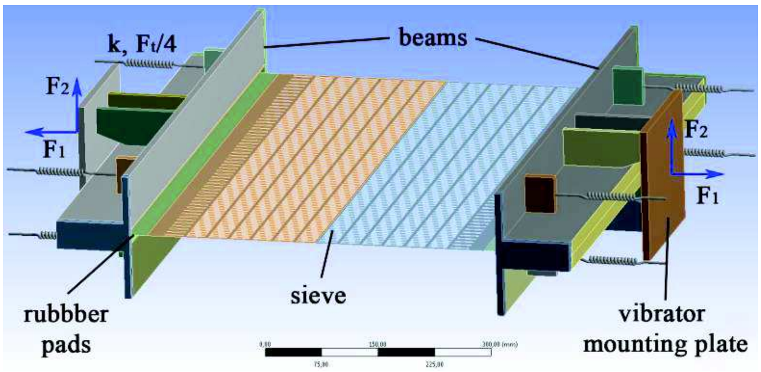

Figure 1. The simplified screen system prepared in Ansys Workbench Design Modeler

The material models of all components was assumed as elastic. In the model, density of external vibrators mounting plates was increased to compensate a mass of electrical vibrators. Contact in the whole model is defined as bonded, except the interactions between the rounded part of the rubber pads and the sieve, where frictionless contact was applied. The finite element model is composed of 131 990 elements. The sieve is modeled by using 4-nodes shell elements. For the beams and the rubber pads meshing the 3-dimentional 20-nodes hexahedron elements and 10-nodes tetrahedron elements were used.

Three steps of numerical analysis were performed in Ansys Workbench software to find the dynamic response of the screen system. In the first step the static structural analysis was carried out in order to apply the sieve preload (F_t) with value 1000 N. For this tension value the natural frequency of the screen system is close to 25 Hz, what was verified experimentally. Then the modal analysis was realized to obtain the natural frequencies of the screen. Afterwards the transient analysis was performed. The time of the analysis equal to 0.4 s was established. This is the minimal time, where the sieve vibrations is being stabilized. Two sinusoidal phase shifted forces (F_1 and F_2) were applied to the beams to simulate the excitation force, which in the real model is generated by rotating eccentric masses (Fig. 1). Excitation frequencies, close to double natural frequency of the screen system were applied with different excitation forces that correspond to the parameters from laboratory parametric resonance screen (Table 1).

Table 1. Excitation parameters used in transient analysis

Excitation frequency, Hz	Magnitude of excitation force, N	Sieve preload, N
41.23	833.6	1000
43.7	936.1	1000
47.07	1086.4	1000
51.13	1282.2	1000
55.82	1528.1	1000
58.74	1692	1000

3. Results and discussion

Two vibration mode shapes occurred close to the screen operation frequency equal to 25 Hz (Fig. 2). The first mode - one side sieve bending, is determined for the natural frequency equal to 25.564 Hz, while the second mode - double side sieve bending is observed for the frequency value of 25.607 Hz. These very close natural frequencies may cause appearance of different mode shapes and lead to unstable vibration amplitude level during the screen operation.

Screen system excitation with frequency close to double natural frequency of the system results in fast vibration amplitude grow (Fig. 3). This phenomenon is observed for all respected cases of excitation frequencies. The most stable transient response of the sieve is obtained for excitation frequency 47.07 Hz. This value is lower than double natural frequency of the screen system obtained in the modal analysis. This is the effect of numerical dumping, which is applied in Ansys Workbench during the problem solving. Numerical dumping eliminates the high frequency modes and stabilizes the numerical integration schemes, but it also affects in lower modes. For all values of excitation force, except 47.07 Hz and 58.74 Hz, the beat phenomenon is observed.

Vibration excitation with a frequency of 58.74 Hz is characterized by unstable sieve motion and the lowest amplitude. Therefore, to obtain stable sieve motion the excitation frequency need to be very close to double natural frequency of the system. The first vibration mode shape is observed for all cases, even when additional loads on the sieve surface were applied to excite the second mode.

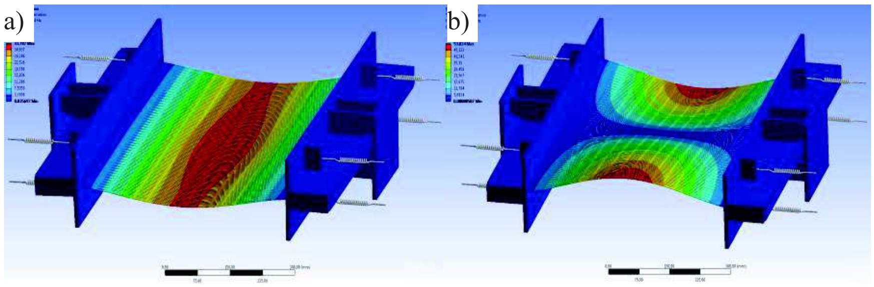

Figure 2. Free vibrations mode shapes of screen system: a) first mode; b) second mode

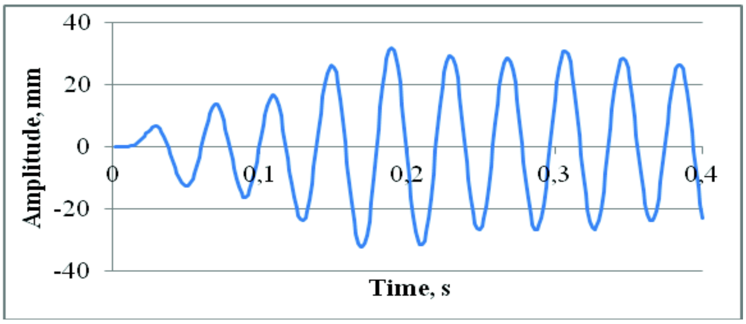

Figure 3. Transient response of the sieve for excitation frequency: 51.13 Hz

The effect of excitation frequency on sieve vibration frequency is presented in Fig. 4. The value of vibration frequency increased with the excitation frequency level. This dependence is nearly linear. The linear character of these relations is confirmed experimentally.

The value of excitation frequency has a significant impact on sieve vibration amplitude (Fig. 5). The vibration amplitude increases together with an increase of excitation frequency level, until its maximal value is obtained. This takes place when the excitation frequency is equal to 47.07 Hz. Further increasing of the excitation frequency results in amplitude decrease. The range of the excitation frequency, where the parametric resonance was observed, is nearly the same for both numerical and experimental data. However, the maximum value of vibration amplitude obtained numerically is almost two times larger than in the experiment. Moreover, the amplitude value in a function of excitation force from experimental data exhibits two local maximums, what cannot be observed in the numerical analysis. This could be the structural damping effect, which was not taken into consideration in the numerical analysis.

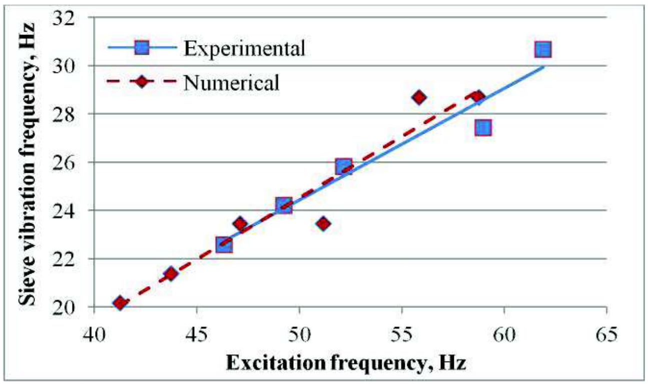

Figure 4. Effect of excitation force on sieve vibration frequency

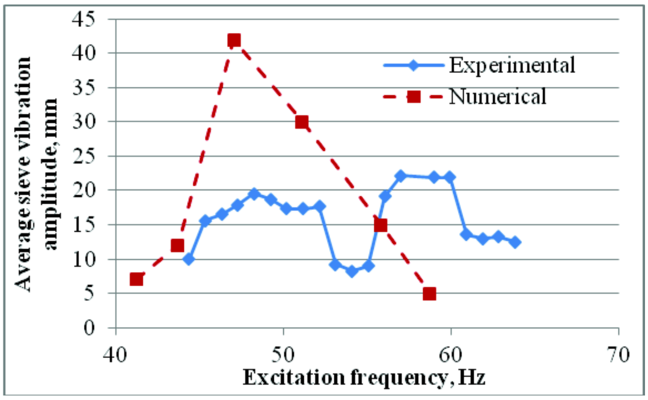

Figure 5. Effect of excitation force on sieve vibration amplitude for different excitation frequencies

4. Conclusions

Two vibration mode shapes occurred close to the screen operation frequency - one side bending and double side bending. This may cause appearance of different mode shapes and lead to unstable vibration amplitude level during the screen operation.

The value of vibration frequency increased with the excitation frequency level. The linear character of this dependence is observed in both numerical and experimental results.

The most stable transient response and the highest vibration amplitude of the sieve is obtained for the excitation frequency of 47.07 Hz. The range of excitation of parametric resonance is nearly the same as for experimental data. Differences of the amplitude level between numerical and experimental results are observed. It is an effect of damping, which is not considered in the numerical simulation.

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