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Investigation of Screening Machine Operation in the "Beating" Mode as Vibration Exciter Shafts with a Rigid Kinematical Connection Rotate

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The basic investigation results of screening machine that operates in the "beating" mode with the use of rigid kinematical connection between vibration exciter shafts are presented in the paper.

Keywords: RIGID KINEMATICAL CONNECTION, OPERATING DEVICE, SCREENING MACHINE, BEATING, VIBRATION EXCITER, AMPLITUDE OF OSCILLATION, CASSINIAN OVAL

Introduction

Movement of screening member in the "beating" mode when vibration exciter shafts are not connected with each other by kinematical connection is investigated in works [1, 2]. In some cases, it is reasonable to use vibration exciters with a rigid kinematical connection between shafts for screening machine drives operating in the "beating" mode. Sharp change of vibration exciter shafts loading during their operation essentially affects the kinematic parameters of screening member. Application of the drive with a rigid connection between shafts enables to increase the ratio of angular rates of shafts up to 0.99, which is impossible in a self-synchronizing drive since this ratio is limited to 0.95 [1].

Results and Discussion

If take the angular rate of the first shaft rotation ω_1 as a constant ω, the angular rate of the second shaft rotation will be $\omega_2 = u_r \cdot \omega$, where u_r is the reduction ratio between shafts $u_r = \omega_2/\omega_1$. In this case, for a loading diagram of screening machine presented in **Figure 1**, centrifugal force vector due to unbalances on the vibration exciter shafts will come to:

$$\overline{P_1} = m \cdot e \cdot \omega^2 \begin{bmatrix} \sin(\varphi_{01} + \omega \cdot t) \\ 0 \\ \cos(\varphi_{01} + \omega \cdot t) \end{bmatrix},$$

$$\overline{P_2} = m \cdot e \cdot \omega^2 \cdot u_r^2 \begin{bmatrix} \sin(\varphi_{02} + \omega \cdot u_r \cdot t) \\ 0 \\ \cos(\varphi_{02} + \omega \cdot u_r \cdot t) \end{bmatrix}, \quad \text{(Eq. 1)}$$

where *m* – the mass of unbalance; *e* – the eccentricity of unbalance; φ_{0i} – the initial angle of unbalance position.

We will take the coordinates of mass centre *H* of process material x_H, y_H, z_H – in the bound coordinate system of screen box *xyz*. If the material is distributed symmetrically on the working surface of the screen x_H=0, the centre of mass *CH* of the system accordingly will move by value:

$$\Delta y_C = \frac{m_H}{M} \cdot y_H, \; \Delta z_C = \frac{m_H}{M} \cdot z_H, \quad \text{(Eq. 2)}$$

and the points of exciting forces application from vibration exciters in view of technological loading will have new coordinates:

$$r_{CAi} = \begin{bmatrix} x_i - \Delta x_C \\ y_i - \Delta y_C \\ z_i - \Delta z_C \end{bmatrix} = \begin{bmatrix} x_i \\ y_i - \Delta y_C \\ z_i - \Delta z_C \end{bmatrix}. \quad \text{(Eq.3)}$$

Exposed to vibration, the dissipation force $\overline{P}_H$ acts on the process material surface *H.* This dissipation force is applied in the centre of mass and depends on viscous friction in the array, which can be taken into account by volumetric coefficient of viscous friction $K_{spec.x}$, $K_{spec.y}$, $K_{spec.z}$ ($N \cdot s/m^4$) along the respective coordinates determined according to [3] by experiment. Further dissipation force applied to the material with

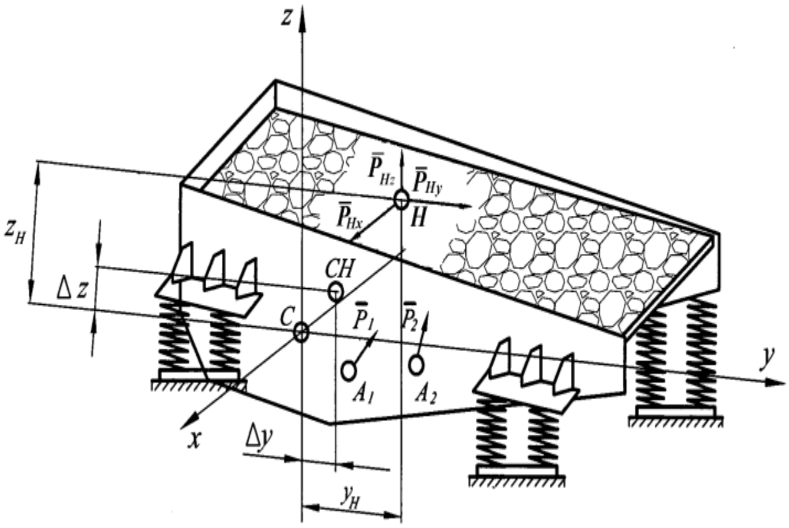

Figure 1. Loading diagram of vibration machine

volume V_H (m^3) and expressed in terms of projections of velocity vector of point H and of viscous friction coefficients (kg/s) along the corresponding directions (χ_{Hx}=K$_{spec.x}$·V$_H$; χ_{Hy}=K$_{spec.y}$·V$_H$, χ_{Hz}=K$_{spec.z}$·V$_H$) is determined by the formula:

$$\bar{P}_H = \begin{bmatrix} -\chi_{Hx} \cdot \dfrac{dx_C}{dt} \\ -\chi_{Hy} \cdot \dfrac{dy_C}{dt} \\ -\chi_{Hz} \cdot \dfrac{dz_C}{dt} \end{bmatrix} \quad \text{(Eq. 4)}$$

We call the mass of vibration machine box without technological loading M and associated mass of the material m_H, which will change inertia of vibration machine rotation and, as result, change its inertia tenser as follows:

$$[I_{CH}] = \begin{bmatrix} I_{xx} & -I_{xy} & -I_{xz} \\ -I_{yx} & I_{yy} & -I_{yz} \\ -I_{zx} & -I_{zy} & I_{zz} \end{bmatrix}, \quad \text{(Eq. 5)}$$

where

$I_{xx} = I_{xx} + m_H \cdot (x_H)^2$;

$I_{yy} = I_{yy} + M \cdot (\Delta y_C)^2 + m_H (y_H - \Delta y_C)^2$;

$-I_{zy} = -I_{zy} - M \cdot (\Delta z_C) \cdot (\Delta y_C) - m_H \cdot (z_H - \Delta z_C) \cdot (y_H - \Delta y_C)$;

$-I_{yz} = -I_{yz} - M \cdot (\Delta z_C) \cdot (\Delta y_C) - m_H \cdot (z_H - \Delta z_C) \cdot (y_H - \Delta y_C)^2$;

$I_{zz} = I_{zz} + M \cdot (\Delta z_C)^2 + m_H \cdot (z_H - \Delta z_C)^2$.

We introduce the dissipative force into the basic dynamic equation and obtain the law of screening member motion that operates in above-resonance mode under technological loading:

$$\begin{cases} \dfrac{d\bar{V}_{CH}}{dt} = \dfrac{1}{M + m_H} \cdot \sum\limits_{i=1}^{n} \bar{P}_i + \dfrac{1}{M + m_H} \cdot \sum\limits_{i=1}^{n} \bar{G}_i + \dfrac{\bar{P}_H}{M + m_H} \\ \dfrac{d\bar{\omega}_{CH}}{dt} = [I_{CH}]^{-1} \cdot \left\{ \sum\limits_{i=1}^{n} (\bar{r}_{CAi} \times \bar{P}_i) + \sum\limits_{i=1}^{n} [(\bar{r}_{CAi} + \bar{r}_i) \times \bar{G}_i] + (\bar{r}_{CH} \times \bar{P}_H) \right\} \end{cases} \quad \text{(Eq. 6)}$$

where $\bar{V}_{CH}$ – the velocity vector of mass centre of the system (points CH); $\bar{\omega}_{CH}$ – the angular rate vector of the system in mass centre CH rotation ; $\bar{r}_I$ – the radius-vector of mass center position of i unbalance in relation to point A_i;

$\bar{G}_i = \begin{bmatrix} 0 \\ 0 \\ -m_i g \end{bmatrix}$ – the gravitational vector of i unbalance attached to the centre of box mass;

$\bar{\omega}_C$ – the angular rate of body in mass centre rotation.

Solution of set of equations (6) at the change of centrifugal force vector of unbalances enables to analyze the vibration machine performance in the range of reduction ratios between shafts from 0.95 to 1, since only this range is unstable for vibration machines with self-synchronizing drive [1, 2].

The contour of screening member GS 3.5x1 (installed in the stock-conveying system of BF No. 8 of blast-furnace shop No. 1 at JSC "Arselor Mittal Kryvyy Rih" for sinter separation) at reduction ratio between shafts u$_r$=0.99 and productivity Q=500 t/h is shown in **Figure 2**. It is clear from **Figure 2** that each point of vibration member moves in elliptic path with time-varying latera recta and angles of axes location so that the curves of security have a shape of Cassinian ovals, sizes and orientation of which vary lengthwise the operating device. This contour promotes active loosening of material in charging section and improves the contact between small particles and screen. And inclination of ovals in the middle and offloading section of screening machine towards material transportation ensures necessary productivity.

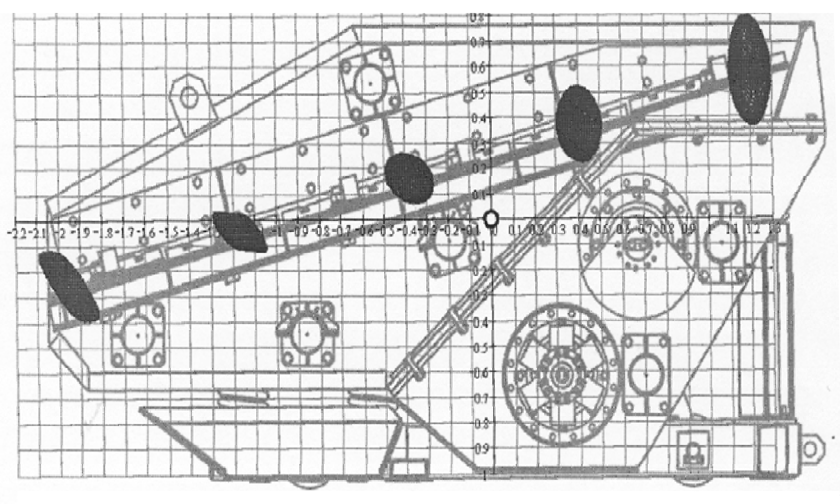

Figure 2. Distribution of contours along the length of screen GS 3.5x1 that operates in the "beating" mode (full scale)

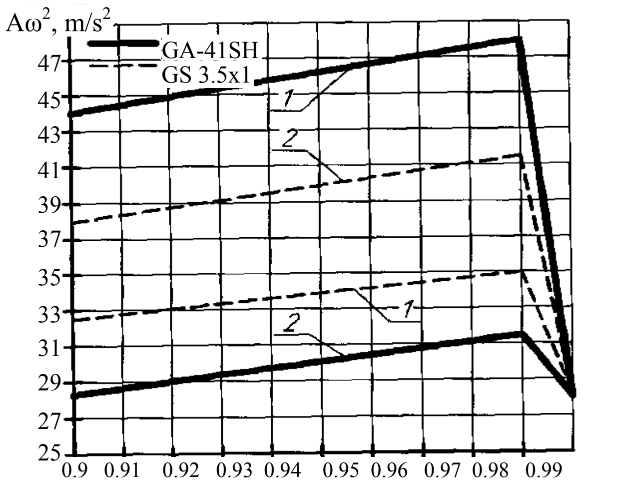

Figure 3. Interrelation of vertical component of acceleration amplitude on reduction ratio: 1 – for poir of offloading section of the screen; 2 – for points of loading section of the screen

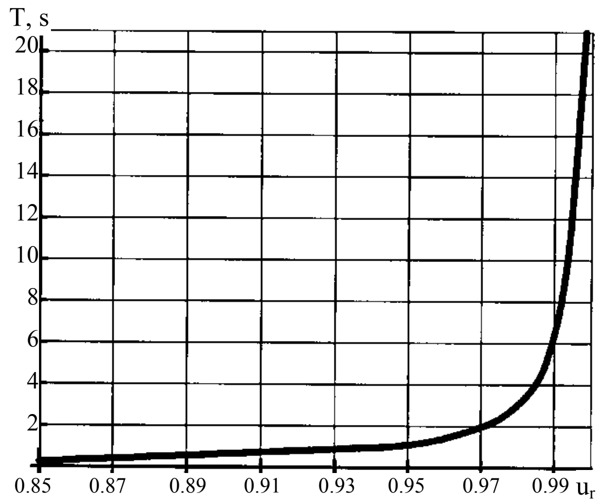

Figure 4. Interrelation of "beats" period on reduction ratio with a rigid connection between shafts of unbalances

The dependences for vertical component of acceleration amplitude of points in offloading and loading sections of screen boxes GA-41SH (installed in the stock-conveying system of BF No. 8 of blast-furnace shop No. 1 at JSC "Arselor Mittal Kryvyy Rih" for sinter separation) and GS 3.5x1 (at productivity Q=350 t/h and Q=500 t/h, respectively) on reduction ratio at the rigid connection between shafts are presented in **Figure 3**.

It follows from **Figure 3** that the vertical component of acceleration amplitude of operating device points depends on reduction ratio between shafts and grows in proportion to the ratio of angular rates of vibration exciter in the range of 0.9-0.99. For heavy loaded machines, its maximum values are 1.1-1.7 times more than the similar component of synchronous vibration mode.

Analysis of established interrelations shows that, unlike the self-synchronizing drive, the vibration machine with a rigid kinematical connection of shafts enables to choose the rational "beats" period in the area of reduction ratios 0.95-0.99 in the range from 1.2 up to 6.5 s (on the basis of process requirements).

The interrelation of "beats" period and reduction ratio with rigid connection between shafts of unbalances, typical for investigated screens GA-41SH and GS 3.5x1, is presented in **Figure 4**. It is obvious that the period increases linearly up to ratios u_r=0.95, further it reaches the maximum value of 1.2 s, after that the function grows hyperbolically and the period is T_b=6.5 s in the area of reduction ratios 0.99.

Conclusions

The period of screening member "beats" rises hyperbolically in the field of ratios of angular velocities of inertia drive shafts with a rigid kinematical connection 0.95-0.99, and the vertical components of amplitudes as compared to the similar component of synchronous mode increase in a direct proportion to this ratio. Thus, each point of operating device moves in elliptic paths with time-varying latera recta and angles of axes location so that the curves of security have a shape of Cassinian ovals, which allows intensifying the screening process.

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Исследования работы грохота в режиме «биений» при вращении валов вибровозбудителя с жесткой кинематической связью

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В статье приведены основные результаты исследований грохота, работающего в режиме «биений» при использовании жесткой кинематической связи между валами вибровозбудителя.