483. Investigating Possibilities of a Table Vibrator with Controlled Damping

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(Received 30 September 2010; accepted 9 December 2010)

Abstract. This paper considers the possibility of ensuring a desired regime of vibrations of the table vibrator with a support, where a damping element with a magnetorheological fluid is used. A feedback system is employed for control of the damping element. The system receives signals from the accelerometer that measures the vibrations of the table vibrator. Solution of the equations describing vibrations of the table vibrator demonstrated that double integration of signals of the accelerometer ensures a stable regime of vibrations for varying mass of a product. If the signal of the accelerometer is integrated once and then passes through a frequency detector, a dangerous increase of the vibrations in the resonance zone is avoided.

Keywords: table vibrator, magnetorheological fluid, feedback, amplitude-frequency response.

Introduction

Large-block constructions are being increasingly used in recent years. Application of blocks and individual details produced in enterprise or on the ground leads to a considerable reduction of costs of construction works, improvement of their quality and shortening of duration of operations.

Table vibrators constitute one of the means of mechanization of production of blocks for civil or industrial construction. They also are used for making hollow floor decking, ribbed floor decking, breezeblocks, architectural details, and ferroconcrete crossties.

Although table vibrators of various bearing capacity are widely used in formation of concrete products, nevertheless a number of problems are present. In addition to other factors, they are also caused by the circumstance that vibrations are generated by equipment having certain power and frequency. Besides, depending on raw materials used for production of a specific article, certain parameters of the vibrations are optimum (Martynov V. *et al*, 2008). On the other hand, they depend on the construction of the table vibrator, first of all, on the elasticity of the suspension and the characteristic damping forces. In addition, the amplitude of the vibrations depends on the mass of a product. For this reason, table vibrators with adjustable tension of springs of the suspension (Murray A. R. *et al* 1982) or the length of the elastic elements (Charles D. H. 1993) have been developed. For example, constructions are available where the length of the crank fixed to the vibrator engine is adjusted, thus changing the force generating the vibrations. However, these constitute a passive means of regulation, i.e. the adjusting actions are carried out before the engine of the table vibrator is switched on, and so the chosen parameters do not change during the operation. However, on variations of the stiffness of the mix under processing as well as upon the impact of casual factors, changes of the parameters of the vibrations are possible. In addition, manual readjustment of a vibrator takes a lot of time. For facilitation of the adjustment procedures, steel springs are replaced with pneumatic supports and the stiffness of the latter is varied by adjusting the pressure of the compressed air supplied to them.

However, the highest efficiency is achieved when active vibration control systems are implemented. In this case, the parameters of the suspension of the table vibrator are changed according to the signals of sensors involved in measuring vibrations of the table vibrator. Such systems are already used with a purpose to ensure the desired regime of vibrations of platforms that are employed for various purposes. In this case it is important to provide a rapid change of a certain parameter of the suspension. Instead of a vibrator, a controllable magnetostrictive drive (Gao Yean Lei *et al* 2009) or special servodrives (Yuan Lu *et al* 2009) can be used.

When using an active system for control of the magnitude of stiffness or damping, it is important to ensure a convenient and rapid adjustment of the parameters. Because electric signals are used in the feedback system, the parameters of the suspension should be adjusted in electric way as well. One of the approaches of generation of controllable damping forces is implementation of dampers with magnetorheological (MR) fluid (David J. C. *et al* 2000). In these dampers, a gap between surfaces moving in relation to each other is filled with the MR fluid, i.e. a suspension of fine iron powder in lube oil. If a magnetic field crosses the gap, a force impeding a shift of the surfaces in respect of each other appears. Experimental tests showed that the said force:

$$\mathrm{H} = \tau_{\mathrm{H}} S, \qquad (1)$$

where: τ_H - the relative resistance force (N/m^2), S – the area of the surfaces forming the gap.

The relative resistance force is in direct ratio with the induction B of the magnetic field that crosses the gap and the induction, in its turn, is in direct ratio with the number of ampere-turns Iw of the electromagnet that generates the magnetic field. It was determined that the relative resistance force τ_H does not depend on the relative velocity of the surfaces that form the gap, i.e. the generated resistance force can be analytically described in the same way, as the dry friction (Coulomb) force:

$$\tau_{\mathrm{H}} = K sign(v), \qquad (2)$$

where: K – the coefficient that depends on the physical properties of the magnetorheological fluid.

Thus, in MR damper the generated damping force is controlled by adjusting the current in windings of the electromagnet.

The scheme and the mathematical model of the table vibrator

The scheme of the table vibrator with integrated MR damper and the feedback system is presented in Fig. 1.

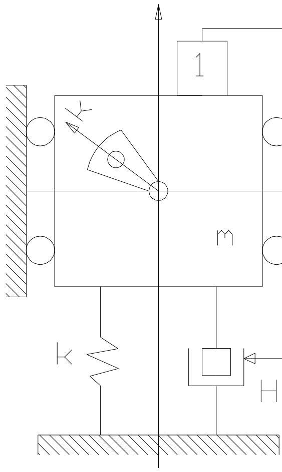


Fig. 1. The structural scheme of the considered table vibrator

On the scheme, *m* is the joint mass of the table vibrator and the mix for producing the product; *k* – the stiffness of the suspension; and *H* – the damper that generates the force corresponding to (1). The seismic accelerometer *1* measures the acceleration and the alternating voltage generated in it is transferred to the integrator *2*. Then this signal is transferred to the voltmeter *3*. In the output of the voltmeter, a direct current voltage equal to the effective value of the input voltage of the voltmeter is formed. Then the voltage is amplified in the amplifier and transferred to the electromagnet mounted in the damper *H*. Depending on the effective value the voltage applied to the winding is proportional to, i.e. the number of integrations of the signal in the integrator (double integration, single integration or no integration), we obtain the shift feedback system, the velocity feedback system or the acceleration feedback system respectively.

If the vibrator generates the force *F*, and its rotational frequency is ω, the dynamics of the system upon the steady-state conditions shall be described by the following equation:

$$m\ddot{x} + H \cdot sign\,\dot{x} + kx = F \sin \omega t, \qquad (3)$$

where: *x* - the shift of the mass *m*.

The equation (3) can be approximately solved by applying the method of harmonic linearization. After linearization of the non-linear member of the equation (3), we obtain the following equation:

$$m\ddot{x} + \frac{4H}{\pi a \omega}\dot{x} + kx = F \sin \omega t, \qquad (4)$$

where: *a* - the amplitude of the mass *m*.

Depending on the number of integrations of the signal from the seismic accelerometer:

$$H = Aa_e, \quad H = Ba_e\omega, \quad H = Ca_e\omega^2, \qquad (5)$$

where: *A, B, C* – the transfer factor for the shift, velocity or acceleration feedback systems, respectively; $a_e = 0,707a,$ (the effective value of the amplitude).

The results of analytic investigation

When one of the expressions (5) is inserted into (4), we find the following solutions (their dimension-free expressions are provided below):

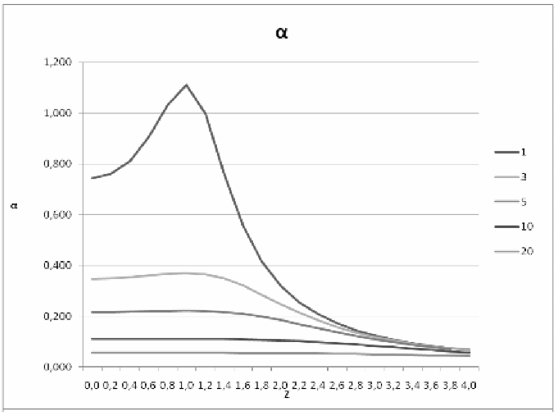
$$\alpha = \frac{\pi}{\sqrt{\pi^2(1-z^2)^2 + 8S^2}}, \qquad (6)$$

$$\alpha = \frac{\pi}{\sqrt{\pi^2(1-z^2)^2 + 8R^2z^2}},$$

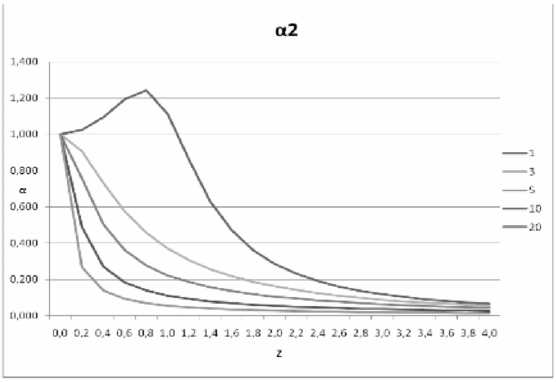
$$\alpha = \frac{\pi}{\sqrt{\pi^2(1-z^2)^2 + 8T^2z^4}},$$

where: $\alpha = \frac{ak}{F}, \; z^2 = \frac{\omega^2 m}{k}, \; S = \frac{A}{k}, \; R = \frac{B}{m\omega_0}, \; T = \frac{C}{m}, \; \omega_0^2 = \frac{k}{m}.$

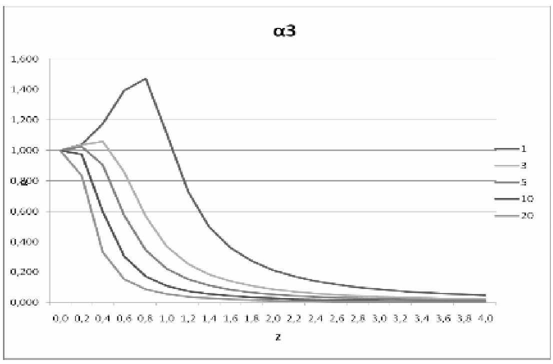
Fig. 2 presents amplitude-frequency responses for various types of feedback. After their analysis, it may be concluded that the system is the most efficient when the shift feedback system is used. If the value of the transfer factor *S* is sufficient, constant amplitude of the vibrations is maintained in a broad frequency band. In addition, the value of the dimension-free transfer factor *S* does not depend on the natural frequency ω_0 of the system and the amplitude-frequency response will not depend on the total mass, i.e. no changes of it will take place on using the table vibrator for different products. When the velocity feedback system is used, the impact of the controlled damper is close to the one of a liquid friction damper, because the damping force is proportional to the velocity of the moving mass. However, the said force increases in the resonance zone and in such a way, the undesirable phenomena bound with increase of the vibration during the racing are removed. If the signal of the accelerometer is not integrated, i.e. in case of the acceleration feedback, a higher dependence of the amplitude of the vibrations on the excitation frequency is found. So, using an integrating circuit in a feedback system is usable for keeping the chosen regimes on varying the mass of the product.



a)



b)



c)

Fig. 2. Amplitude-frequency response of the table vibrator: *a* – the shift feedback, when *S=1; 3; 5; 10; 20; b* - the velocity feedback, when *R=1; 3; 5; 10; 20; c* – the acceleration feedback, when *T=1; 3; 5; 10; 20*

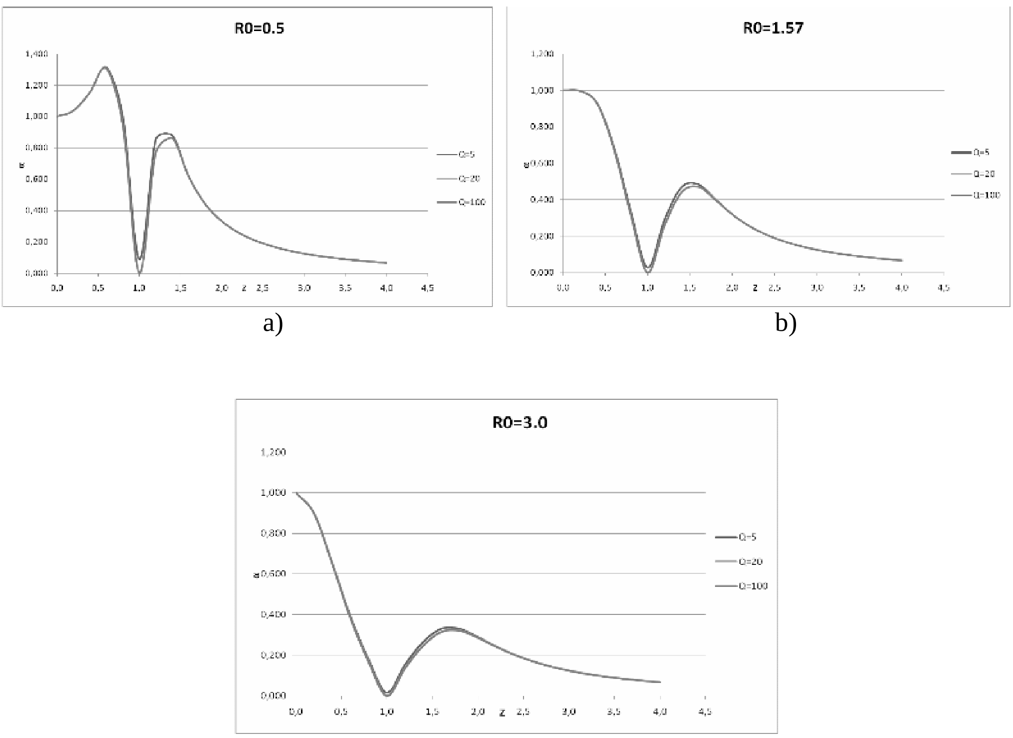
The feedback system with a frequency detector

The best operational parameters are ensured in a case of the shift feedback. However, twice integration of the signal of the accelerometer increases the duration of its transfer and this circumstance can cause a negative impact upon the transitional processes. The velocity feedback system is less inert and the impact of the damper is close to the one of a liquid friction damper. If the transfer factor is lower, in such a case vibrations in the resonance zone can increase. In case of this type of feedback, the operational parameters of the table vibrator can be improved by using a frequency detector. In such a case, in the velocity feedback system (shown in Fig. 1), a frequency detector of Q quality is inserted between the integrator *2* and the voltmeter *3*. It is necessary to avoid increase of the amplitude of the vibrations in the resonance zone, i.e. when $\omega \approx \omega_0$, or $z \approx 1$. Because of this, if we adjust the frequency detector for the frequency ω_0, we will have the value of the transfer factor of the velocity feedback that is a function of frequency instead of constant value of it, i.e.:

$$R = \frac{R_0}{\sqrt{(1-z^2)^2 + z^2/Q^2}}, \qquad (7)$$

where: R_0 – the value of the transfer factor of the integrator.

Using (5) and (7), the amplitude-frequency responses provided in Fig. 3 were produced. It can be observed that vibrations of the platform are reduced in the resonance zone and a stable vibration regime is ensured in the zone of higher frequencies.



a) b)

c)

Fig. 3. Aamplitude-frequency response of the table vibrator with the velocity feedback system equipped with a frequency detector, when R_0*=0.5; 1.57; 3.0*

Conclusions

1. If a damping element with a magnetorheological fluid is used in the support of vibration table, the parameters of the support are easily adjustable in response to material and mass of the article.

2. When the damping element is controlled by means of the signal of the accelerometer that measures the vibrations of the table vibrator and the signal is integrated twice (the shift feedback), a stable regime of vibrations of the table vibrator is ensured, which is not dependent on the mass of the product.

3. When the velocity feedback with a frequency detector is used, vibrations of the table vibrator are reduced in the resonance zone upon keeping the desired level of the vibrations in the zone of operational frequencies of the table vibrator.

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