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Original Research Article

Two stage vibration isolation of vibratory shake-out conveyor

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ARTICLE INFO

Article history:
Received 15 July 2014
Accepted 2 October 2016
Available online 12 November 2016

Keywords:
Vibratory conveyor
Dynamical behavior
Vibration isolation

ABSTRACT

In this paper the effectiveness of two stage vibration isolation on example of a vibratory conveyor has been shown. Vibratory conveyors are used for separating the casting from the mold in foundries. In the considered case the shake-out conveyor has been supported directly on the foundation. Due to that the high amplitudes of vibrations on the foundation has been observed, which are transmitted to the building structure. To reduce the vibration transmission from the conveyor to the foundation, two-stage vibration isolation has been applied. The mathematical model of two stage vibration isolation has been shown. Based on the results of calculations, simulations and measurements a significant reduction of vibrations at the building structure has been achieved.

1. Introduction

The vibratory machines and devices are mounted in the industrial buildings directly on foundations or a metal structure supported on the ceiling. In both cases the vibration is transmitted to the foundation and the supporting structure. To reduce the transmission of vibrations from the vibrating machine to the supporting structures a proper vibration isolation will be required [1–6]. Active vibration isolation systems [7] provide sophisticated solutions for vibration problems mainly in fields of precise technologies such as metrology, optics, manufacturing etc. A multiple-degree-of-freedom active vibration isolation systems was presented in [8]. Active vibration control can also be applied in civil structures [9]. Various types of methods can be used in active vibration control, however the main limitation is the limited amplitude and frequency range of vibration [9]. Due to the operation of shake-out conveyors, high amplitudes of vibrations and forces generated by the accelerated masses, application of an active vibration isolation system is not justified. Another approach to the problem of vibration in case of heavy machinery and structures is the research of various types of passive vibration isolation systems [10].

This paper presents the theoretical background and experimental investigation results regarding the vibration isolation effects obtained by using a two-stage vibration isolation of the vibratory conveyor. Introduction of an additional, frequency tuned ballast mass to a single stage vibration isolation should decrease the vibration outside resonance frequency, resulting in a decrease of forces transmitted to the foundation of the device.

2. Description of the conveyor

Fig. 1 presents the vibratory conveyor (1), mounted to four supporting elements (4) which are fastened directly to the

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http://dx.doi.org/10.1016/j.acme.2016.10.001

Nomenclature

a_1	vibration amplitude of mass 1 (m)
a_2	vibration amplitude of mass 2 (m)
m_1	conveyor's mass (kg)
m_2	frame mass with additional mass attached (kg)
c_1	stiffness coefficient of the conveyor support (N/m)
c_2	stiffness coefficient of the second stage of vibration isolation (N/m)
c_{rel}	relative stiffness coefficient c_2/c_1
b_1	damping coefficient of the conveyor support (Ns/m)
b_2	damping coefficient of the second stage of vibration isolation (Ns/m)
m_{rel}	relative mass coefficient m_2/m_1
F	exciting force (N)
F_0	amplitude of excitation force (N)
F_T	transmitted force (N)
ω	angular frequency (1/s)
Ω	angular frequency of exciting force (1/s)
φ	phase angle (rad)
η	ratio of frequencies Ω/ω
SSVI	single stage vibration isolation system
DSVI	double stage vibration isolation system

foundation (5) of the first floor inside the building. The conveyor is excited by a set of two rotational shafts with eccentric masses (2). Each shaft is driven by an electric motor at the rotational frequency of 25 Hz. The electrical motors are electronically synchronized in order to control the phase shift between the drives and change the impact angle of the conveyor. This frequency provides optimal conditions for castings transportation through the chute. Vibrations caused by the conveyor were measured according to the PN-90/N-01357 standard, at the base frame of the conveyor and its foundation. It was established that the vibration exceeded human perceptibility threshold value sixteen times. Such vibrations are a health hazard for people and may damage the foundation, supporting elements and building structure.

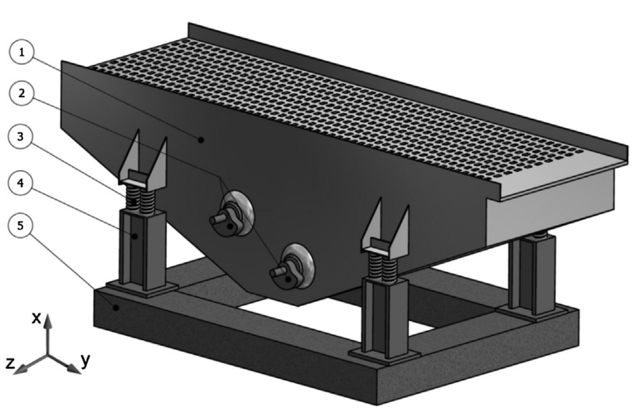

Fig. 1 – Shake out conveyor – factory setup (SSVI).

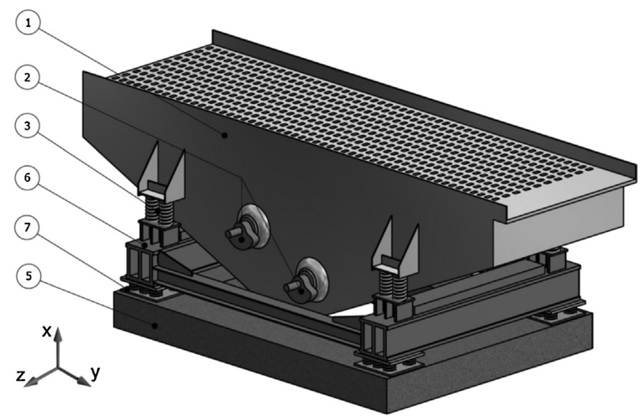

Fig. 2 – Shake out conveyor with second stage of vibration isolation. Frame supported on spring elements (DSVI).

The shake-out conveyor was connected to the mounting frame by four sets of spring elements (3), which forms the first stage of vibration isolation. After the modernization (Fig. 2) instead of rigid connection of the frame to the foundation, four additional spring elements (7) were used. Together with frame (ballast mass) (6) they form a second stage of vibration isolation.

3. Mathematical model

The scheme of the single-stage and double stage vibration isolation has been shown in Fig. 3. Mass m_1 represents conveyor's mass, c_1 is the stiffness of the system connecting conveyor to the base. The b_1 coefficient represents damping of the system. Mass m_2 is the ballast mass included into the conveyor in form of a frame, c_2 and b_2 represent stiffness and damping coefficients of implemented vibroisolators, placed between the frame and the foundation.

According to the technical data of the conveyor mass $m_1 = 11{,}000$ [kg] and $m_2 = 6600$ [kg]. The spring stiffness coefficients were set to $c_1 = 2116$ [kN/m] and $c_2 = 5500$ [kN/m].

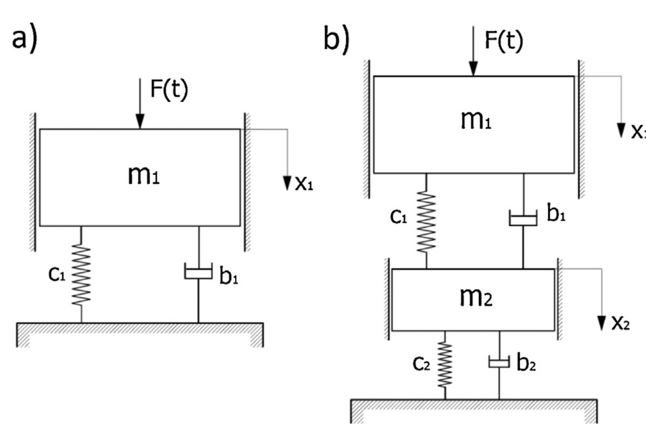

Fig. 3 – Scheme of the single-stage (SSVI) and double-stage (DSVI) conveyor's vibration isolation system.

The excitation force amplitude, generated by the rotation of drive shafts at 25 [Hz] was set to 360 [kN].

It can be seen that mathematical models presented in Fig. 3 are simplified to the vertical motion along the X-axis. The real device posesses multiple degrees of freedom, but for theoretical purpose, it can be assuemed, that the conveyor is loaded mostly symetrically so no rotations around the X and Y and translation along Z-axis occurs. The horizontal forces acting on the system are defined by the phase shift between the two shafts. The phase shift can be adjusted whithin the range of 0–35°, what influencess the velocity of the transported material.

The forces transmited to the foundation are higher when the vertical component of the excitation force is increased [11–13]. The worst load case is present when no horizontal forces are taken into acount as in models presented in Fig. 3.

Equation of motion for a SSVI system is following:

$$m_1\ddot{x}_1 + c_1x_1 + b_1\dot{x}_1 = F \tag{1}$$

where $F = F_0\sin(\omega t)$

The influence of the damping of steel springs, resulting mainly from the material damping ratio, which is below 0.005, can be neglected therefore the transmissibility for SSVI system can be calculated from the following formula [7–9].

$$V = \left|\frac{F_T}{F_0}\right| = \left|\frac{1}{\sqrt{(1-\eta^2)^2}}\right| \tag{2}$$

Equations of motion of DSVI system are following (Fig. 3):

$$\begin{aligned} &m_1\ddot{x}_1 + c_1(x_1-x_2) + b_1(\dot{x}_1-\dot{x}_2) = F \\ &m_2\ddot{x}_2 + c_2x_2 - c_1(x_1-x_2) + b_2\dot{x}_2 - b_1(\dot{x}_1-\dot{x}_2) = 0 \end{aligned} \tag{3}$$

General solutions of these equations by neglecting of damping are as follows:

$$x_1 = a_1\sin(\omega t + \varphi)$$

$$x_2 = a_2\sin(\omega t + \varphi)$$

After introduction of these solution into Eq. (2), the determinant can be written as:

$$\begin{vmatrix} c_1 - m_1\omega^2 & -c_1 \\ -c_1 & c_1 + c_2 - m_2\omega^2 \end{vmatrix} = 0$$

Characteristic equation is following:

$$\omega^4 - \frac{(c_1+c_2)m_1 + c_1m_2}{m_1m_2}\omega^2 + \frac{c_1c_2}{m_1m_2} = 0 \tag{4}$$

Solution of these equation, which are natural frequencies of DSVI are following:

$$\omega_{1/2} = \sqrt{\frac{1}{2}\frac{c_1(m_1+m_2)+c_2m_1}{m_1m_2} \mp \sqrt{\left[\frac{1}{2}\frac{c_1(m_1+m_2)+c_2m_1}{m_1m_2}\right]^2 - \frac{c_1c_2}{m_1m_2}}} \tag{5}$$

The transmissibility of DSVI system without including the damping can be calculated from:

$$V = \left|\frac{F_T}{F_0}\right| = \left|\frac{c_2x_2}{F_0}\right| = \left|\frac{-c_1c_2}{(c_1-m_1\omega^2)(c_1+c_2-m_2\omega^2)-c_1^2}\right| \tag{6}$$

In Fig. 4 the comparison of transmissibility for a single and two stage vibration isolation systems has been shown.

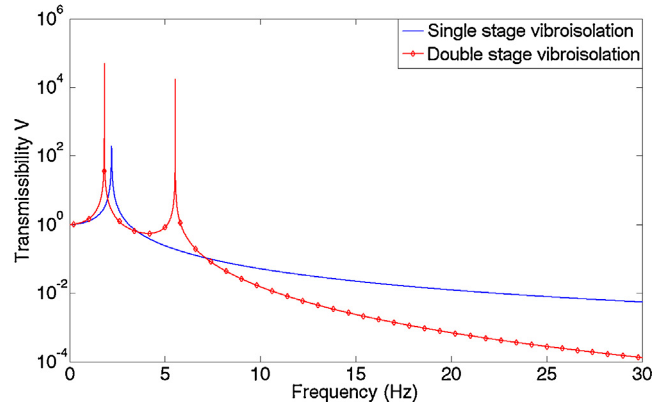

Fig. 4 – Comparison of transmissibility for single and double stage vibration insulation of the shake-out conveyor.

Comparing transmissibility in the low frequency range, shows that the vibration transmissibility of both SSVI and DSVI is similar. No noticeable reduction of transmissibility can be observed in case of the DSVI, furthermore an additional resonance frequency appears. Nevertheless, a strong reduction of vibration transmissibility can be observed mainly in the over-resonance frequency range, so in the operational frequency of the conveyor. The selected values of stiffness and mass should ensure a lowest natural frequency of the vibration isolation system and the smallest weight of the ballast mass- from economical point of view.

Fig. 5 presents the influence of the frame mass m_2 on the transmissibility of DSVI system. It can be seen, that the increasing of the mass m_2 causes the moving of the second natural frequency of DSVI system to lower frequencies. The efficiency of DSVI increases slightly with the increasing of the ballast mass.

Fig. 6 presents the influence of the ballast mass m_2 on the transmissibility around the excitation frequency of 25 Hz. Increasing of the value of ballast mass m_2 reduces slightly the transmissibility, so the vibration isolation is more effective. A correlation between ballast mass and transmissibility can be stated as follows: increase of ballast mass by the factor of two will result in a decrease of transmissibility by the same factor.

4. FEM modal analysis

In order to perform a modal analysis FEM models of the conveyor and the vibration isolation systems have been created. In case of the SSVI the conveyor body was connected to the foundation directly via spring elements. In case of DSVI system, the main body was connected similarly as above to the frame ballast mass. The frame was then supported at its corners to the foundation with four spring elements. The modal analysis was conducted for a narrow range of frequencies from 0–10 Hz, where the conveyor body and frame vibrate as rigid bodies.

The conveyor body and the mass frame can translate along three directions and rotate around three axes. In case of the SSVI system 6DOF and for DSVI 12 DOF models have been used

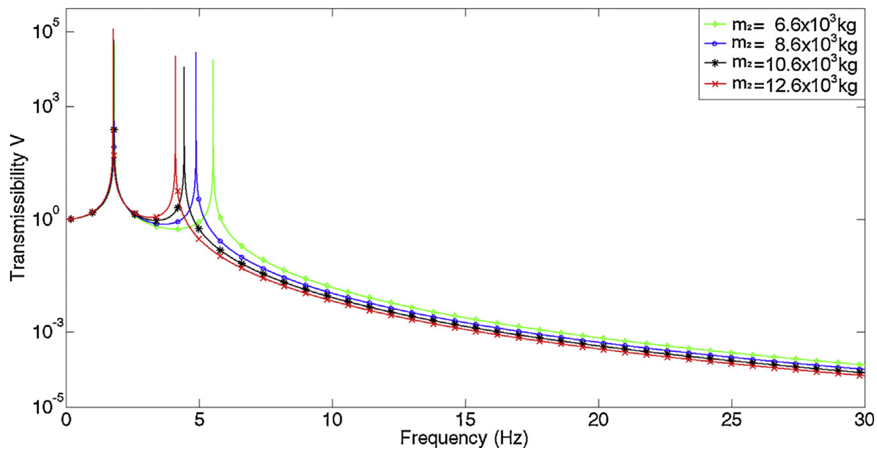

Fig. 5 – Influence of mass m_2 on the transmissibility of DSVI system.

for the simulation. Due to the increased degrees of freedom, a larger number of natural frequencies and mode shapes were identified. For further comparison only mode shapes were considered, that correspond to the mathematical model presented in Fig. 4, and which are essential for the conveyor operation.

Fig. 7 presents the first mode shape acquired from modal analysis. Fig 7a and b presents two extremes of the displacement. At the frequency of 1.831 Hz the displacement of conveyor body along the vertical axis, some minor displacement along the vertical axis can be observed.

Fig. 8a and b presents two extremes of the displacement at the second mode. At the frequency of 5.486 Hz the main displacement of the frame along the vertical axis can be observed.

Results acquired from FEM modal analysis ware compared in Table 1 with the analytical results. It can be seen that natural frequencies calculated analytically are in agreement with natural frequencies from the FEM simulation.

5. Dynamical simulation

To investigate the dynamical displacement of the conveyor a multibody dynamical simulation was conducted. Two setups

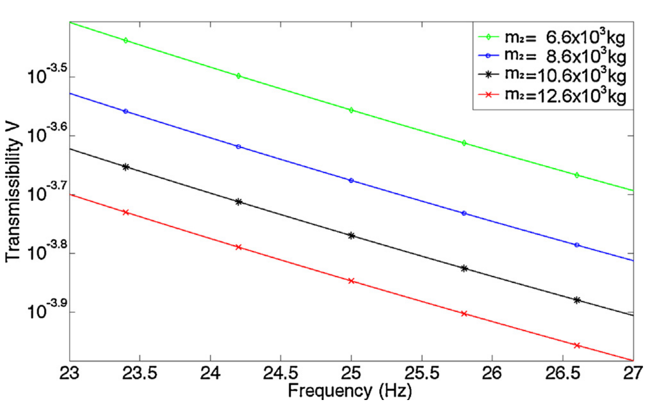

Fig. 6 – Influence of the mass m_2 on the transmissibility of DSVI system around its operation frequency of 25 Hz.

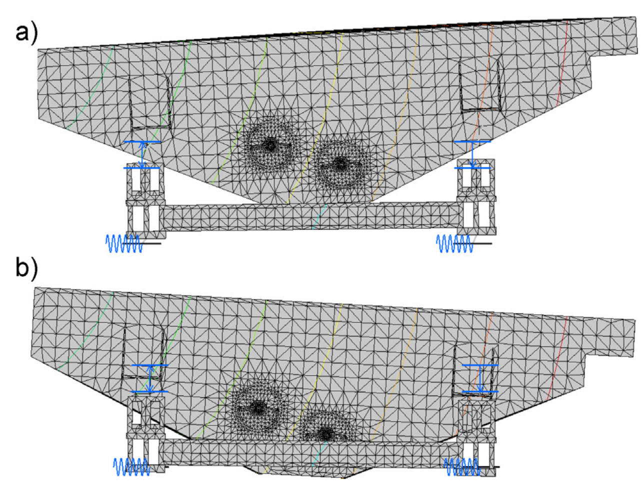

Fig. 7 – FEM modal analysis results of DSVI system – Mode 1: 1.831 Hz.

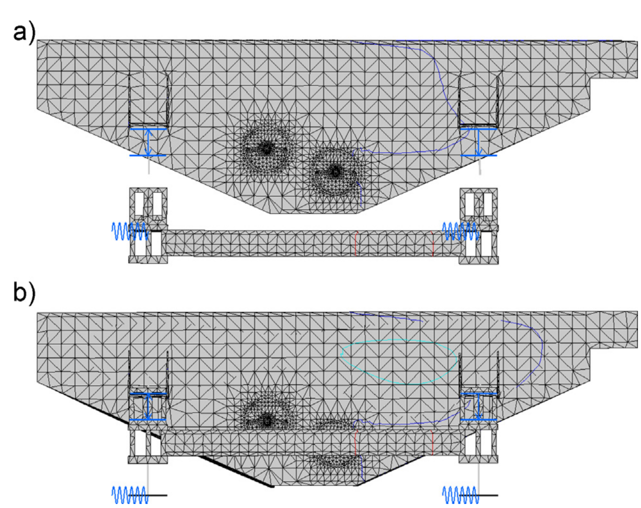

Fig. 8 – FEM modal analysis results of DSVI system – Mode 2: 5.486 Hz.

Table 1 – Natural frequencies – theoretical vs. FEM simulation results.

Results	SSVI	DSVI	
	f_0 [Hz]	f_{01} [Hz]	f_{02} [Hz]
Analytical solution	2.211	1.828	5.472
FEM analysis	2.205	1.831	5.486

Table 2 – Dynamic simulation results: excitation at 25 Hz.

Type of vibroisolation	Conveyor vertical displacement [mm]	Force transmitted to foundation [N]
Single stage	2.352	4974
Two stage	2.165	156

Table 3 – Measurements of acceleration for single and two stage vibration insulation.

Meas.pt.	Acceleration RMS value at Shake-out [m/s^2]		Acceleration RMS value at Foundation [m/s^2]	
	Single stage	Two stage	Single stage	Two stage
1	31.004	9.990	0.978	0.049
2	34.720	10.060	0.489	0.059
3	31.297	9.610	0.782	0.033
4	33.449	10.000	1.956	0.020

of the conveyor were simulated. The first one corresponds to the SSVI system provided by the manufacturer (Fig. 1). The second setup includes the DSVI system with the frame mass m_2 (Fig. 2).

The conveyor, supports, frame and additional masses have been modeled as rigid bodies. Masses have been connected with spring elements to the foundation. The excitation was introduced at the frequency of 25 Hz.

As mentioned above the main forces transmitted to the foundation are generated by the displacement of the conveyor in the vertical direction. The information about the spring parameters (stiffness and damping coefficients) allows to calculate the force transmitted to the foundation. In order to investigate the influence of the DSVI system, additional simulations have been carried out. It can be observed in Table 3, that the displacement of the conveyor in vertical direction is similar in both cases of vibration isolation.

It can be clearly seen, that in case of DSVI system a strong reduction of the amplitude of the dynamic force transmitted to the foundation occurs. The effectiveness of the DSVI system can then be established by comparing the final values shown in Table 2. In case of the SSVI system the dynamic force amplitude reached a value around 5 kN. Implementation of the frame and spring elements caused a strong reduction of the dynamic force transmitted to the foundation. The transmitted force reaches the value of 156 N, which is roughly 30 times smaller as in case of a SSVI system.

6. Experimental verification

In order to verify the theoretical investigation, the industrial two stage vibration isolation system has been investigated. Measurements of acceleration in the vertical direction were taken for a SSVI (Fig. 1) and DSVI systems (Fig. 2). Measurement points were placed at four support points on the container of the conveyor and four corresponding points on the foundation.

Table 3 presents the results of measurements from SSVI vs. DSVI system. It can be seen that in case of the DSVI system the acceleration values can be significantly reduced, both at the conveyor and at foundation.

7. Summary

In case of vibratory conveyors, the transmission of vibrations to the foundation is often too high. In such case a single stage vibration isolation of the conveyor can be ineffective. Significant reduction of vibration transmission can be achieved by implementation of a passive two stage vibration isolation system. To achieve a two stage vibration isolation an additional ballast mass between the conveyor and foundation is necessary.

The superiority of DSVI to SSVI system was confirmed by analytical, simulation and experimental approach. Results obtained from theoretical considerations were compared with simulation results. Results from simulations show, that in case of a DSVI system a strong reduction of the force transmitted from the vibratory conveyor to the foundation can be observed. According to the diagrams presented in Fig. 4 the force transmitted to the foundation by excitation at 25 Hz, is around 30 times smaller in case of the DSVI system. Similar reduction of force transmitted to foundation was found based on dynamic simulation. Measurements taken at the support points of the conveyor show also a strong reduction of acceleration values in case of the DSVI system. With increasing of the ballast mass in DSVI system the efficiency of vibration isolation slightly increases. According to DIN 4024-1 [14], it is also important to ensure, that the stiffness of the supporting structure is minimum 10 times higher than the total stiffness of the vibration isolation system in order to achieve high effectivity of the isolation.

Acknowledgement

This work was funded by Wrocław University Technology Faculty of Mechanical Engineering.

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