Vibrations in Physical Systems 2020, 31, 2020314 https://doi.org/10.21008/j.0860-6897.2020.3.14

Optimal Configuration and Parameters of Translational Dynamic Vibration Absorbers in Vibroisolation of Mechanical Press

Waldemar ŁATAS
Institute of Applied Mechanics, Faculty of Mechanical Engineering, Cracow University of Technology, 31-864 Kraków, Al. Jana Pawła II 37, Poland
waldemar.latas@pk.edu.pl

Zygmunt DZIECHCIOWSKI
Laboratory of Techno-Climatic Research and Heavy Duty Machines, Cracow University of Technology, 31-864 Kraków, Al. Jana Pawła II 37, Poland
zygmunt.dziechciowski@mech.pk.edu.pl

Abstract

The paper deals with the problem of optimizing the positions and parameters of dynamic vibration absorbers for a mechanical press subjected to polyharmonic inertial excitation. Under the assumption of small vibrations a linear dynamic model of a rigid body performing a planar motion, on viscoelastic supports, with an attached system of translational tuned mass dampers is constructed. The problem of vibroisolation of the machine is presented, considering selected harmonics of the force transmitted to the ground, the solutions with a single DVA are proposed.

Keywords: Dynamic Vibration Absorber, discrete system vibration, machine vibration isolation

1. Introduction

Tuned mass dampers are used in a variety of problems regarding vibrations reduction. Due to the simplicity of construction and their effectiveness, tuned mass dampers can be used in structures such as offshore platforms or motorcycle handlebars, they are efficient devices for high-speed machining chatter suppression [1–3].

In civil engineering, TMDs are employed in large structures with low internal damping and low natural frequencies exposed to wind or earthquake forces, such as suspension and cable-stayed bridges, high-rise buildings, antennas, masts, chimneys and wind power towers [4–7].

Many theoretical works were devoted to optimization of the parameters of tuned mass dampers for different types of excitations (including random ones) in both linear and non-linear problems [8–12].

In the different branches of industry, during the operation of machinery such as vibrating screens, sieves, mills, textile machines, fans, or presses, occur dynamic forces transmitted to the surrounding environment. These forces may be the cause of noticeable undesirable vibrations and noise in the work area, in rooms located near working devices, or even in buildings (including residential ones) located near the manufacture or facility where a given machine is in use (the so-called structure borne vibrations and noise). For example, in the case of presses, strong vibrations are caused by changes in the speed of movable parts inside the press, impact of the ram, and, especially, by the pressing and punching processes. Larger presses generate vibrations of low frequencies which may coincide with the vibration frequencies of the building's own components, for instance floors. In many cases, it is therefore necessary to adequately safeguard people against these harmful or arduous factors.

In order to provide effective protection, it is most important to isolate the machine from the environment using supports with adequately selected elastic and damping properties. It may happen, however, that the physical properties of the supports are insufficient to ensure desired isolation. In this case, it may be helpful to use additional tuned mass dampers.

In the selection of parameters of tuned mass dampers, it is very important to determine their optimal positioning which can be limited due to technical considerations. In continuous systems, for example beams, usually the best positions of dampers are the concentrated forces application points and the location points of antinodes of the vibration modes to whose frequencies the dampers are tuned. In problems of „non-collocated" and „global vibration control" types and in the case of distributed excitation forces, determining the optimal positions of dampers is an additional problem to solve [13–14]. The correct positioning of tuned mass dampers also matters in discrete systems with many degrees of freedom, in which rigid bodies perform rotary and translational motions.

The present paper outlines the problem of vibroisolation of a press producing small metal elements which is subjected to inertial excitation resulting from the application of reciprocating motion of machine components. Dynamic forces of a large amplitude in the system do not result from the unbalance of the rotating shaft. They are related to the course of the technological process itself. Despite the originally used vibroisolation the working press was the source of vibrations transmitted to office rooms.

The aim of the work is to find the optimal positioning and determine the optimal parameters of dynamic vibration absorbers connected to the press, suppressing the selected harmonic components of force transmitted to the ground.

In the theoretical model, a planar motion of the machine modeled as a rigid body is assumed. The basic assumption is to apply the theory of linear vibrations, which is justified in the case studied and commonly accepted in solving the problems of vibroisolation.

2. Scheme of the press. Spectrum of the excitation force

Figure 1 shows the press diagram with basic dimensions and the direction of the resultant inertial force $F(t)$ acting on the system, arising from the slides movement at variable velocities. The press rests upon four supports, its mass equals $M = 11000\,\mathrm{kg}$, the moment of inertia of the press about the axis perpendicular to the plane of symmetry *xz* and passing through the center of mass equals $J_C = 4000\,\mathrm{kgm}^2$. The center of mass *C* lies at equal distance $b = 0.75\ \mathrm{m}$ from the vertical planes passing through the left and right supports and on the symmetry axis of the slide guide, i.e. on the action line of the resultant inertial force $F(t)$. The distance between the center of mass and the horizontal plane passing through the center points of the vibration absorbers equals $a = 0.90$ m, and the angle of inclination of the slide guide is $\beta_0 = 45^0$.

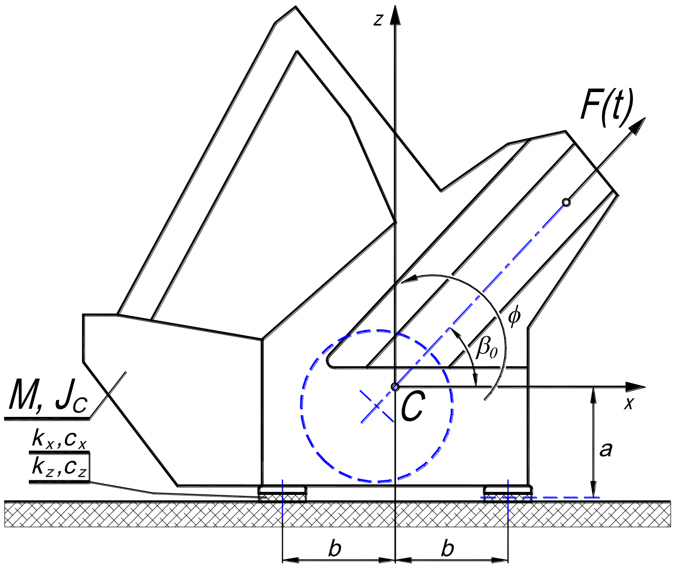

Figure 1. Mechanical press diagram without attached tuned mass dampers.

The resultant inertial force $F(t)$ is an effect of the movement of two slides (with masses of 850 kg and 820 kg, respectively) in opposite directions, at variable velocities along the guides attached to the frame. On the basis of the displacements of the press slides as a function of the angle of rotation of the drive shaft (movements of the slides result from the rotation of appropriately shaped cams), an approximation of the periodic force $F(t)$ was determined. Its harmonics are shown in Fig. 2. The dominance of the third and fifth harmonics of the excitation force is clearly visible. Since the rotational speed of the machine shaft is 190 rpm, these are components with frequencies of 9.50 Hz and 15.83 Hz, respectively.

Before starting calculations related to the selection of positions and parameters of dynamic vibration absorbers it is necessary to evaluate the effectiveness of the originally used vibroisolation.

The purpose of vibroisolation is to reduce the forces transmitted to the ground which are the source of vibrations transmitted by the subsoil and building's structural elements to nearby office spaces. It can be achieved through an appropriate selection of the parameters describing the supports on which the machine rests: stiffnesses k_x and k_z, damping coefficients c_x and c_z. The selection of the values of these parameters depends on the frequency range of the excitation force.

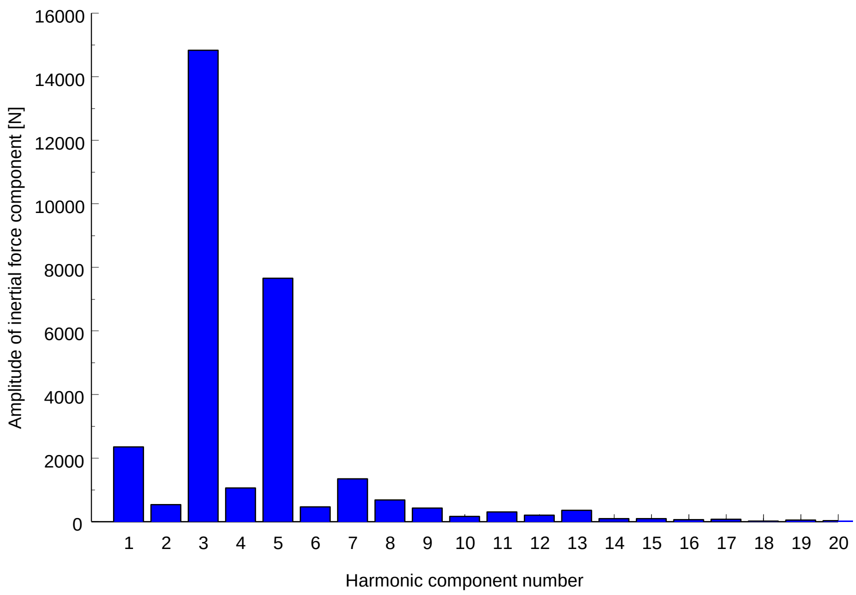

Figure 2. Amplitudes of components of polyharmonic excitation force.

3. Vibroisolation efficiency of the elastomer pads

The adopted assumptions and simplifications allow to write the equations of motion of the system presented in Fig. 1 by means of linear differential equations with constant coefficients:

$$M\ddot{x}_C + 4c_x\dot{x}_C + 4c_x a\dot{\varphi} + 4k_x x_C + 4k_x a\varphi = F(t)\cos\beta_0 \quad (1)$$

$$M\ddot{z}_C + 4c_z\dot{z}_C + 4k_z z_C = F(t)\sin\beta_0 \quad (2)$$

$$J_C\ddot{\varphi} + 4c_x a\dot{x}_C + 4(c_x a^2 + c_z b^2)\dot{\varphi} + 4k_x a x_C + 4(k_x a^2 + k_z b^2)\varphi = 0 \quad (3)$$

Due to the very low fundamental frequency of the excitation force, the solution based on the principle of supercritical vibroisolation (pneumatic absorbers) was abandoned due to high cost and relatively high unreliability, and polyurethane elastomer pads (chosen by the press manufacturer) were used. In the case of pads made of elastomer layers, both their stiffness and damping depend on the acting pressure. The parameters of the pads used in the press are: $k_x = 760\cdot10^5$ N/m, $c_x = 280\cdot10^3$ Ns/m, $k_z = 670\cdot10^6$ N/m, $c_z = 540\cdot10^3$ Ns/m. For the such values of stiffness and damping coefficients, the dimensionless damping coefficients in the horizontal and vertical directions are respectively $\zeta_x = 0.31$ and $\zeta_z = 0.20$. When ignoring damping, the three values of natural frequency of the system are approximately: $f_1 = 24.4$ Hz, $f_2 = 78.6$ Hz (the uncoupled vertical vibration mode), $f_3 = 105.9$ Hz.

In order to be able to estimate the effectiveness of vibroisolation with the given parameters, the amplification factors for each significant polyharmonic excitation force component should be determined. This requires solving the system of equations (1)–(3). In the examined case the force $F(t)$ is a periodic one and can be represented by the Fourier series:

$$F(t) = F_0 + \sum_{n=1}^{\infty}\left(P_n \cos n\nu t + Q_n \sin n\nu t\right) \tag{4}$$

The values of amplitudes of individual harmonic components of the excitation force are presented in Fig. 2. The solution of the system (1)–(3) is searched in the form of series:

$$x_C = x_0 + \sum_{n=1}^{\infty}\left(A_n \cos n\nu t + B_n \sin n\nu t\right) \tag{5}$$

$$z_C = z_0 + \sum_{n=1}^{\infty}\left(C_n \cos n\nu t + D_n \sin n\nu t\right) \tag{6}$$

$$\varphi = \varphi_0 + \sum_{n=1}^{\infty}\left(E_n \cos n\nu t + H_n \sin n\nu t\right) \tag{7}$$

Substitution of expressions (5)–(7) into Eqs. (1)– (3) leads to a system of linear algebraic equations for the unknowns: A_n, B_n, C_n, D_n, E_n, H_n.

The amplitudes of the harmonic components of the forces transmitted to the ground on the left and right side of the machine in the horizontal and vertical directions are given by:

$$\left|R_{nx}\right| = \left|R_{nx}\right|_L = \left|R_{nx}\right|_R = 2\sqrt{k_x^2 + c_x^2 n^2 \nu^2}\sqrt{A_n^2 + B_n^2} \tag{8}$$

$$\left|R_{nz}\right|_{L,R} = 2\sqrt{k_z^2 + c_z^2 n^2 \nu^2}\sqrt{C_n^2 + D_n^2 + b^2\left(E_n^2 + H_n^2\right) \mp 2b\left(C_n E_n + D_n H_n\right)} \tag{9}$$

Amplification factors are defined as the ratios of the amplitudes of horizontal or vertical harmonic force components, transferred to the support on the left and right side of the machine, to half of the amplitudes of individual harmonic components of the excitation force, projected onto the *x* and *z* directions:

$$g_{nxL} = g_{nxR} = \frac{2}{\cos\beta_0}\frac{\left|R_{nx}\right|}{\sqrt{P_n^2 + Q_n^2}} = 2\sqrt{2}\frac{\left|R_{nx}\right|}{\sqrt{P_n^2 + Q_n^2}} \tag{10}$$

$$g_{nzL,R} = \frac{2}{\sin\beta_0}\frac{\left|R_{nz}\right|_{L,R}}{\sqrt{P_n^2 + Q_n^2}} = 2\sqrt{2}\frac{\left|R_{nz}\right|_{L,R}}{\sqrt{P_n^2 + Q_n^2}} \tag{11}$$

Although the horizontal and vertical components do not have to be in the same phase, an equivalent amplification factor is defined by:

$$G_{nEQ} = \sqrt{g_{nx}^2 + g_{nz}^2} \tag{12}$$

where: $g_{nx} = g_{nxL} = g_{nxR}$, $g_{nz} = \max\left\{g_{nzL}, g_{nzR}\right\}$.

Figure 3 shows the amplification factors of individual harmonics of the forces in *x* and *z* directions and the equivalent amplification factor (expressions (10)–(12).

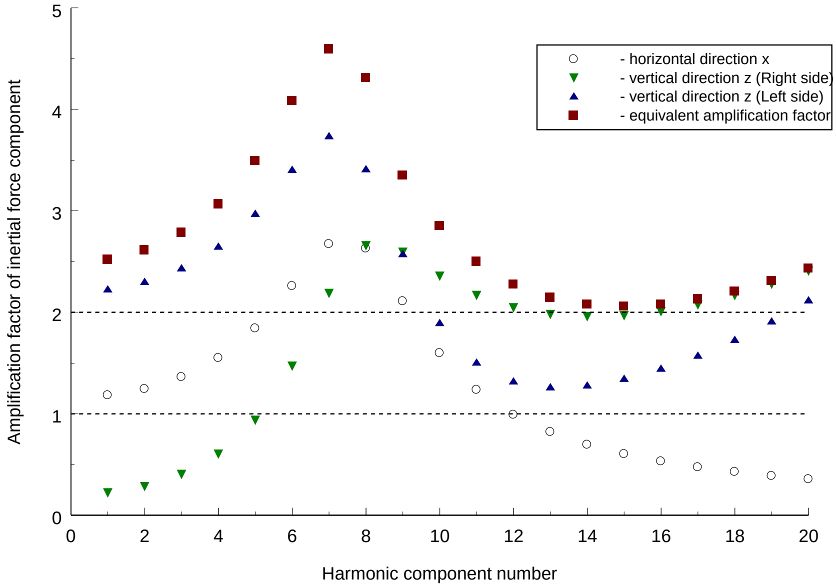

Figure 3. Amplification factors of excitation force components.

Considering the values of the amplitudes of the excitation force components (Fig. 2) and the amplification factors (Fig. 3) dependent on the properties of the elastomer pads used, it can be seen that additional vibration isolation for harmonic components with numbers *n*=3,5,7 is advisable.

4. Model of the machine with attached tuned mass dampers

Figure 4 shows a simplified press sketch with translational mass dampers, moving along the inclined directions, acting on the upper and right part of the machine (such distinction allows to conveniently describe the system with the only horizontal or only vertical dampers used). One damper in the picture means in fact two dampers mounted symmetrically with respect to the plane of symmetry on both sides of the machine, in order to avoid the rotational vibrations with respect to the vertical axis.

The positions, angles of inclinations and physical parameters of the TMDs are given by: d_i, h_i, α_i, m_i, k_i, c_i (upper side, $i = 1, 2, \ldots p$) and similarly by: d_j, h_j, α_j, m_j, k_i, c_j (right side, $j = p+1, p+2, \ldots p+s$). By using the positive and negative values

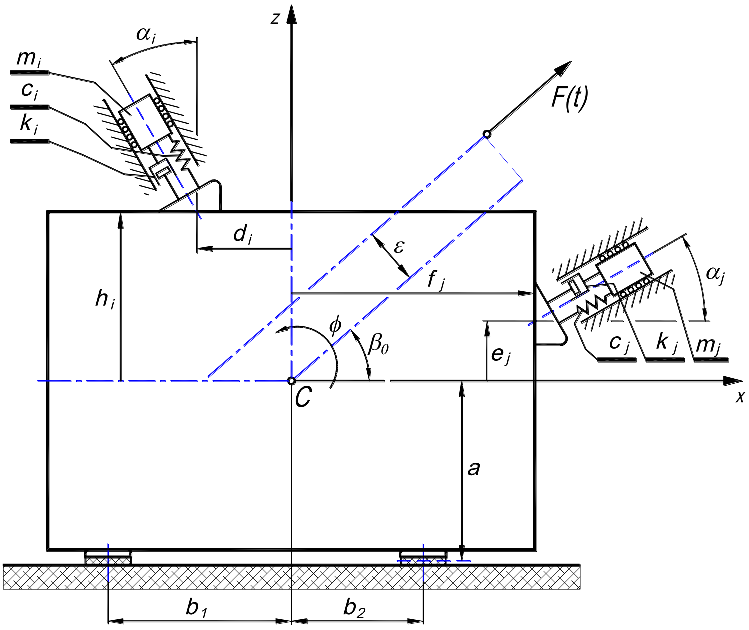

Figure 4. Mechanical press scheme with attached tuned mass dampers.

of the parameters: d_i, h_i or d_j, h_j formally any attachment point of the damper on the machine can be achieved (for example on the left side of the press).

The system of the equations of motion of the machine with the attached *s* tuned mass dampers on the right side of the machine (it is assumed that *p*=0) takes the form:

$$
\begin{aligned}
&M\ddot{x}_C + 4c_x\dot{x}_C + 4c_x a\dot{\varphi} + 4k_x x_C + 4k_x a\varphi \\
&-\sum_{j=1}^{s}\left(c_j\dot{y}_j - c_j\sin\alpha_j\dot{z}_C - c_j\cos\alpha_j\dot{x}_C + c_j(e_j\cos\alpha_j - f_j\sin\alpha_j)\dot{\varphi}\right)\cos\alpha_j \\
&-\sum_{j=1}^{s}\left(k_j y_j - k_j\sin\alpha_j z_C - k_j\cos\alpha_j x_C + k_j(e_j\cos\alpha_j - f_j\sin\alpha_j)\varphi\right)\cos\alpha_j \\
&= F(t)\cos\beta_0
\end{aligned}
\tag{13}
$$

$$
\begin{aligned}
&M\ddot{z}_C + 4c_z\dot{z}_C + 2c_z(b_2 - b_1)\dot{\varphi} + 4k_z z_C + 2k_z(b_2 - b_1)\varphi \\
&-\sum_{j=1}^{s}\left(c_j\dot{y}_j - c_j\sin\alpha_j\dot{z}_C - c_j\cos\alpha_j\dot{x}_C + c_j(e_j\cos\alpha_j - f_j\sin\alpha_j)\dot{\varphi}\right)\sin\alpha_j \\
&-\sum_{j=1}^{s}\left(k_j y_j - k_j\sin\alpha_j z_C - k_j\cos\alpha_j x_C + k_j(e_j\cos\alpha_j - f_j\sin\alpha_j)\varphi\right)\sin\alpha_j \\
&= F(t)\sin\beta_0
\end{aligned}
\tag{14}
$$

$$J_C\ddot{\varphi}+4c_x a\dot{x}_C+2\left(2c_x a^2+c_z(b_1^2+b_2^2)\right)\dot{\varphi}+4k_x a x_C+2\left(2k_x a^2+k_z(b_1^2+b_2^2)\right)\varphi$$
$$+\sum_{j=1}^{s}\left(c_j\dot{y}_j-c_j\sin\alpha_j\dot{z}_C-c_j\cos\alpha_j\dot{x}_C+c_j(e_j\cos\alpha_j-f_j\sin\alpha_j)\dot{\varphi}\right)(e_j\cos\alpha_j-f_j\sin\alpha_j)$$
$$+\sum_{j=1}^{s}\left(k_j y_j-k_j\sin\alpha_j z_C-k_j\cos\alpha_j x_C+k_j(e_j\cos\alpha_j-f_j\sin\alpha_j)\varphi\right)(e_j\cos\alpha_j-f_j\sin\alpha_j)$$
$$=-F(t)\varepsilon \tag{15}$$

$$\begin{aligned}&m_j\ddot{y}_j+c_j\dot{y}_j-c_j\sin\alpha_j\dot{z}_C-c_j\cos\alpha_j\dot{x}_C+c_j(e_j\cos\alpha_j-f_j\sin\alpha_j)\dot{\varphi}\\&\quad+k_j y_j-k_j\sin\alpha_j z_C-k_j\cos\alpha_j x_C+k_j(e_j\cos\alpha_j-f_j\sin\alpha_j)\varphi=0\end{aligned} \tag{16}$$

$j=1,2\ldots,s$

The solution of the system (13)–(16) is obtained similarly to the solution of the system (1)–(3), the additional series describing the displacements of TMDs (marked by y_j) should be added.

5. Numerical calculations – system with a single DVA

The effectiveness of using a single $(s=1)$ dynamic vibration absorber (i.e. TMD with no damping) will be presented. This choice is justified by the constant angular velocity of the drive shaft during press operation.

The aim is to find the position, inclination and tuning of the device to significantly reduce the selected harmonic components of the force transmitted to the ground. The equivalent amplification factor defined by (12) is taken as an objective function.

For the harmonic force $F(t)$ it turns out that a DVA tuned to a given excitation frequency, mounted in such a way that it can move along the line of action of the force, completely eliminates system vibrations (for any values of parameters β_0 and ε). Since installation of such an absorber would be difficult, a configuration easier to mount and ensuring sufficient vibration isolation should be found.

It was assumed that an absorber can be attached between the ground level and between the mass center level (Fig. 1, Fig. 2, $-0.9\text{ m}\le e_1\le 0.0\text{ m}$). On the right side of the machine $f_1=0.85\text{ m}$, whereas on the left side: $-2.35\text{ m}\le f_1\le -1.45\text{ m}$, and f_1 varies linearly with e_1 due to the slope of the machine edge. In the numerical calculations it was taken: $\beta_0=45^0$, $\varepsilon=0.0\text{ m}$, $b_1=b_2=0.75\text{ m}$.

The position e_{1OPT}, inclination α_{1OPT} and frequency ratio β_{nOPT} of the absorber are searched, which minimize, for a given mass m_1, the value of the G_{nEQ} (12). The frequency ratio is defined as: $\beta_n=\left(1/2\pi nf\right)\sqrt{k_1/m_1}$, where $f=3.17\text{ Hz}$, *n* denotes the harmonic number. Tables 1–3 show the calculations results for $m_1=300\text{ kg}$.

Table 1: $m_1 = 300$ kg, RIGHT SIDE

n	e_{1OPT} [m]	α_{1OPT}	β_{nOPT}	G_{nEQ}	g_{nx}	g_{nz}	Dissipation efficiency [%]
3	0.0	19	1.0005	0.7239	0.4774	0.5442	74.01
5	0.0	36	1.0010	0.7802	0.3819	0.6803	77.67
7	0.0	32	1.0020	0.8452	0.3627	0.7634	81.61

Table 2: $m_1 = 300$ kg, LEFT SIDE

n	e_{1OPT} [m]	α_{1OPT}	β_{nOPT}	G_{nEQ}	g_{nx}	g_{nz}	Dissipation efficiency [%]
3	-0.9	31	1.0000	0.3973	0.0271	0.3963	85.74
5	-0.9	35	0.9995	0.3901	0.1193	0.3714	88.83
7	-0.9	36	0.9990	0.3942	0.1405	0.3683	91.42

It is evident that for the considered harmonics the properly placed and inclined dynamic vibration absorber on the left side is much more effective than the absorber on the right side of the machine (the objective function values about two times lower). Despite this, the effectiveness of the absorbers mounted on the right is satisfactory, resulting in almost 75 percent (for n=3) or more reduction of the equivalent amplification factor. The optimal position of the absorber on right is at the level of the machine's mass center (the highest admissible position), whereas for the absorber on left – at the ground level (the lowest admissible position).

For comparison, Table 3 contains the calculations results for the horizontal absorber (i.e. $\alpha_1 = 0$), placed (on the left or right side) at the ground level ($e_1 = -0.9$ mm), as the easiest to mount from a technical point of view.

Table 3: $m_1 = 300$ kg, horizontal absorber ($\alpha_1 = 0$, $e_1 = -0.9$ mm)

n	β_{nOPT}	G_{nEQ}	Vibration efficiency [%]
3	1.0005	2.2037	20.89
5	1.0025	2.1232	39.22
7	1.0065	1.9720	57.09

From the results contained in Table 2 and Table 3 it can be seen, that for the absorber mounted near the ground on the left side of the machine the value of the objective function drops by more than 80 percent compared to the system with a horizontal absorber, the appropriate inclination angle is crucial here.

6. Conclusions

The problem of mechanical press vibroisolation, subjected to polyharmonic excitation force, with the attached tuned mass dampers is considered.

The results of numerical calculations carried out for a single dynamic vibration absorber (DVA), taking into account the restrictions imposed on its position, show a significant dependence of the effectiveness of the suppression device on the direction of its movement.

Although the system does a planar motion, it is shown that a translational-only absorber can significantly reduce the system vibration for all three degrees of freedom.

The results of calculations for a single absorber are presented, but in the problem considered the excitation force components frequencies are separable so significantly that an absorber tuned to one frequency has practically no effect on the effectiveness of the absorber tuned to another frequency.

References

1. Q. Yue, L. Zhang, W. Zhang, T. Kärnä, *Mitigating ice-induced jacket platform vibrations utilizing a TMD system*, Cold. Reg. Sci. Technol., 56 (2009) 84–89.
2. S. Agostoni, F. Cheli, E. Leo, M. Pezzola, *An innovative multi dof TMD system for motorcycle handlebars designed to reduce structural vibrations and human exposure*, Mech. Syst. Signal Process., 31 (2012) 298–315.
3. M. Wang,T. Zan, Y. Yang, R. Fei, *Design and implementation of nonlinear TMD for chatter suppression: An application in turning processes*, Int. J. Mach. Tool. Manu., 50 (2010) 474–479.
4. S.R. Chen, J. Wu, *Performance enhancement of bridge infrastructure systems: Long-span bridge, moving trucks and wind with tuned mass dampers*, Eng. Struct., 30 (2008) 3316–3324.
5. G. Bekdaş, S.M. Nigdel, *Estimating optimum parameters of tuned mass dampers using harmony search*, Eng. Struct., 33 (2011) 2716–2723.
6. J.M.W. Brownjohn, E.P. Carden, C.R. Goddard, G. Oudin, *Real-time performance monitoring of tuned mass damper system for a 183m reinforced concrete chimney*, J. Wind Eng. Ind. Aerodyn., 98 (2010) 169–179.
7. J. Snamina, P. Martynowicz, W. Łatas, *Dynamic Similarity of Wind Turbine's Tower-nacelle System and its Scaled Model*, Sol. St. Phen., 208 (2014) 29–39.
8. G.C. Marano, R. Greco, B. Chiaia, *A comparison between different optimization criteria for tuned mass dampers design*, J. Sound Vib., 329 (2010) 4880–4890.
9. F. Rüdinger, *Tuned mass damper with nonlinear viscous damping*, J. Sound Vib., 300 (2007) 932–948.
10. J.C. Ji, N. Zhang, *Suppression of the primary resonance vibrations of a forced nonlinear system using a dynamic vibration absorber*, J. Sound Vib., 329 (2010) 2044–2056.
11. G.C. Marano, R. Greco, S. Sgobba, *A comparison between different robust optimum design approaches: Application to tuned mass dampers*, Probabilistic Eng. Mech. 25 (2010) 108–118.
12. O.F. Tigli, *Optimum vibration absorber (tuned mass damper) design for linear damped systems subjected to random loads*, 331 (2012) 3035–3049.
13. J. Dayou, M.J. Brennan, *Global control of structural vibration using multiple-tuned tunable vibration neutralizers*, J. Sound Vib., 258(2), (2002) 345–357.
14. W. Łatas, *Optimal positions of the tunable translational-rotational dynamic absorbers in global vibration control problems in beams*, Journal of Theoretical and Applied Mechanics, 53(2) (2014) 467–476.