IOP Conference Series: Materials Science and Engineering

Optimization of dynamic absorber parameters for protecting the machines against the vibrations transmitted through their base

To cite this article: R Ibănescu and M Ibănescu 2021 *IOP Conf. Ser.: Mater. Sci. Eng.* **1037** 012054

IOP Conf. Series: Materials Science and Engineering 1037 (2021) 012054 doi:10.1088/1757-899X/1037/1/012054

Optimization of dynamic absorber parameters for protecting the machines against the vibrations transmitted through their base

R Ibănescu[1] and M Ibănescu[2]

[1]Department of Theoretical Mechanics, "Gheorghe Asachi" Technical University of Iași, Bd. Dimitrie Mangeron nr. 67, 700050, Iași, Romania
[2]Department of Structural Mechanics, "Gheorghe Asachi" Technical University of Iași, Bd. Dimitrie Mangeron nr. 67, 700050, Iași, Romania

*Corresponding author's e-mail: ribanesc@yhoo.com

Abstract. The correct functioning of machines could be affected by the vibrations transmitted through their base from other devices. The first possible protection measure is to support these machines on elastic elements with damping. The present work presents a protection solution which consists in the attachment of a dynamic absorber with viscous friction to the considered machine, in order to diminish the amplitudes of the vibrations caused by base movement. The mathematical model is developed for the real case when the base vibrations are not transmitted to the machine, but to the elastic supporting elements. The analytical expression of machine vibration amplitude in steady state is obtained. This amplitude could be diminished by an accurate choice of the three dynamic absorber parameters (mass, elastic constant, coefficient of viscous friction). An optimization program conceived in MATLAB provides the values of these parameters which lead to the minimum amplitude of machine vibrations.

1. Introduction

The normal functioning of machine tools could be affected by the vibrations coming from outer sources. For diminishing these effects and ensuring the machine tools functioning with no perturbations, the amplitude of these vibrations must be reduced, as much as possible. There are several possibilities for reaching this goal. One of the most used procedure is represented by the use of devices for the passive control of vibrations. Between them, the dynamic vibration absorber has been intensively studied, because of its wide range of application in different engineering fields, as civil engineering, automotive, shipbuilding [1], [2], [3], [4]. This wide use is explained by its high effectiveness relative to its constructive simplicity. A lot of recent studies have been focused on the optimization of the dynamic absorber parameters, the absorber being considered not only in the classical variant [5], but also in the new variant, with inverter and negative stiffness [6], [7], [8], [9], [10]. The present paper presents the model of a machine tool subjected to a sinusoidal displacement, transmitted through its support base. For mitigating the vibrations induced to the considered machine, a classical dynamic absorber is attached to it. The parameters of this dynamic absorber are optimized by using a software dedicated to MATLAB facilities. The approach in this paper is a lot more user friendly compared to the theoretical approaches of other works [11], [12], because it answers quickly at the practical problem of finding the absorber optimum parameters for a given vibrating system, by using three very explicit diagrams.

2. The mechanical system model

The model of the mechanical system contains a mass M, which is the mass of the machine tool supported on the ground through a spring, whose stiffness is k and a damper with the damping coefficient c, as shown in figure 1.

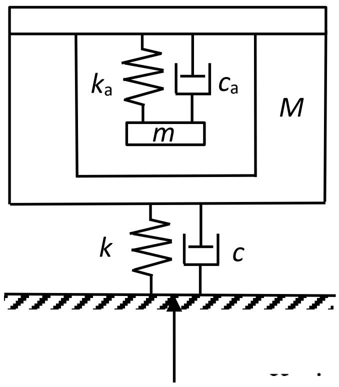

Figure 1. The model of the mechanical device.

It is considered that the supporting surface oscillates according to a sinusoidal law:

$$x = X_0 \sin(\omega_0 t) \tag{1}$$

A classical vibration dynamic absorber (C-DVA) is attached to mass M. This absorber consists of a mass m, a spring of stiffness k_a and a damper having the damping coefficient c_a, as it can be seen in figure 1.

This model differs from those presented in other works because it contains the damper of damping coefficient c_a between mass M and the ground, and that the excitatory vibration is induced by the displacement of the supporting surface, which is transmitted to the spring and damper, on which mass M is supported.

3. Mathematical model

The mechanical system, without the vibration dynamic absorber, is considered. By denoting the displacement of the mass M by x_1, the motion differential equation for this mass is:

$$M\ddot{x}_1 + c\dot{x}_1 + kx_1 = kx + c\dot{x} \tag{2}$$

By substituting equation (1) in equation (2), it results:

$$\ddot{x}_1 + \frac{c}{M}\dot{x}_1 + \frac{k}{M}x_1 = \frac{k}{M}X_0\sin(\omega_0 t) + \frac{c}{M}\omega_0 X_0\cos(\omega_0 t) \tag{3}$$

The steady state response will be the particular solution of a non-homogeneous equation, presumed of the following form:

$$x_{1p} = A\sin(\omega_0 t) + B\cos(\omega_0 t) \tag{4}$$

It is imposed the condition for x_{1p} to satisfy equation (3) and so, the values of coefficients A and B are determined. After performing the calculus, by identification, the amplitude of mass M oscillation, denoted by A_M, is determined:

$$A_M = X_0(k^2 + c^2\omega_0^2)^{1/2} \Big/ \left[(k^2 - M\omega_0^2)^2 + c^2\omega_0^2\right]^{1/2} \tag{5}$$

The angular velocity corresponding to the maximum amplitude is obtained by differentiating relation (5) and finding the roots of the derived equation. The root corresponding to the maximum amplitude is:

$$\omega_0 = \left[(Q - Mk^2) \big/ (Mc^2) \right]^{1/2} \tag{6}$$

where

$$Q = (M^2 k^4 + 2Mc^2 k^3)^{1/2} \tag{7}$$

There are considered the following numerical values: M = 5000kg, X_0 = 0.0001m, c = 2000Ns/m, k = 100000N/m. The resulted maximum amplitude of mass M is 1.124x10^{-3}m, corresponding to an angular velocity ω_0=4.461s^{-1}. The graph which describes the variation of the amplitude in terms of angular velocity, in case of no dynamic absorber, is pictured in figure 2.

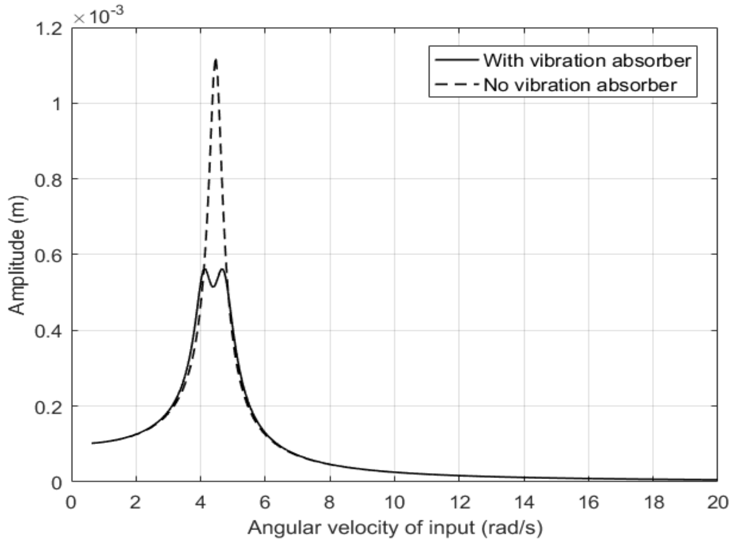

Figure 2. The amplitude of the mass *M*.

For diminishing the maximum value of the amplitude, a vibration dynamic absorber is attached to mass M. In this new case, denoting the displacement of mass m by x_2, the system of differential equations that describes the dynamics of the mechanical system shown in figure 1 is:

$$M\ddot{x}_1 + k(x_1 - x) + k_a(x_1 - x_2) + c(\dot{x}_1 - \dot{x}) + c_a(\dot{x}_1 - \dot{x}_2) = 0 \tag{8}$$

$$m\ddot{x}_2 - k_a(x_1 - x_2) - c_a(\dot{x}_1 - \dot{x}_2) = 0 \tag{9}$$

The system of state equations is determined based on equations (8) and (9)and it has the shape:

$$\dot{v}_1 + v_1(c + c_a)/M - v_2 c_a/M + x_1(k + k_a)/M - x_2 k_a/M = xk/M + \dot{x}c_1/M \tag{10}$$

$$\dot{v}_2 - v_1 c_a/m + v_2 c_2/m - x_1 k_2/m + x_2 k_2/m = 0 \tag{11}$$

$$\dot{x}_1 = v_1 \tag{12}$$

$$\dot{x}_2 = v_2 \tag{13}$$

where v_1 and v_2 are the velocities of mass M and m, respectively. The steady state response is given by a particular solution, having the form:

$$x_1 = a_1 \sin(\omega_0 t) + b_1 \cos(\omega_0 t) \tag{14}$$

$$x_2 = c_1 \sin(\omega_0 t) + d_1 \cos(\omega_0 t) \tag{15}$$

$$v_1 = e_1 \sin(\omega_0 t) + f_1 \cos(\omega_0 t) \tag{16}$$

$$v_2 = g_1 \sin(\omega_0 t) + h_1 \cos(\omega_0 t) \tag{17}$$

where a_1,b_1, c_1, d_1, e_1, f_1, g_1 and h_1 are unknown coefficients, which result by identification, after replacing equations (14), (15), (16) and (17) and their derivatives in equations (10), (11), (12) and (13).There are eliminated the unknowns e_1, f_1, g_1 and h_1, by expressing them in terms of a_1,b_1, c_1 and d_1. The following system of algebraic equations results:

$$Aa_1 - Bb_1 - Cc_1 + Dd_1 = kX_0 / M \tag{18}$$

$$Ba_1 + Ab_1 - Dc_1 - Cd_1 = c\omega_0 X_0 / M \tag{19}$$

$$-Ea_1 + Fb_1 + Gc_1 - Hd_1 = 0 \tag{20}$$

$$-Fa_1 - Eb_1 + Hc_1 + Gd_1 = 0 \tag{21}$$

where:

$$\begin{array}{ll} A = (k + k_a) / M - \omega_0^2 & E = k_a / m \\ B = (c + c_a)\omega_0 / M & F = c_a \omega_0 / M \\ C = k_a / M & G = k_a / m - \omega_0^2 \\ D = c_a \omega_0 / M & \mathrm{H} = \mathrm{c}_a \omega_0 / m \end{array} \tag{22}$$

The system of equations is solved and the amplitude x_1 of mass M is obtained according to the formula:

$$x_1 = (a_1^2 + b_1^2)^{1/2} \tag{23}$$

4. Optimization of vibration dynamic absorber parameters

This optimization of dynamic absorber parameters consists in the assessment of m, c_a and k_a values, which correspond to the minimum amplitude of mass M.

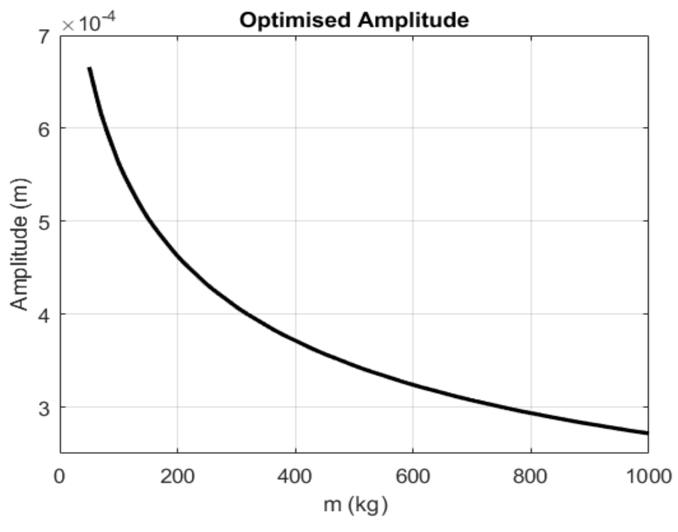

Figure 3. The amplitude with respect to the absorber mass m.

The original values of these parameters are: m=100kg, c_a=1000Ns/m, k_a=100000N/m.

The calculus of the optimum values is performed by using a software which has been created by the authors, based on function *fmincon* from MATLAB.

The optimum values have been computed when $\omega_0 \in [\pi / 5,\ 16\pi]$ for the following intervals of the considered parameters $c_a \in [1,\ 10000]$, Ns/m $k_a \in [100,\ 200000]$N/m and $m \in [1,\ 100]$kg.

The resulted optimum values of absorber parameters are: m=100 kg, c_a=79.1 Ns/m and k_a=1882.2 N/m.

For these values, the maximum amplitude is A=5.635x10^{-4}m, for an angular velocity ω_0 = 4.147s^{-1}.

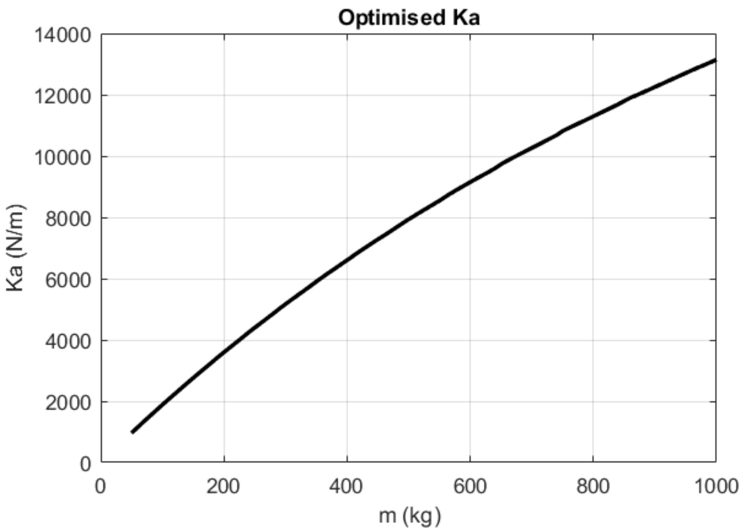

Figure 4. The stiffness k_a with respect to the absorber mass m.

The graph which shows the relation amplitude versus angular velocity is pictured in figure 2. It can be seen that the maximum amplitude decreases with 50% compared with the maximum amplitude of 11.24x10^{-4}m, corresponding to the case without absorber.

It has been noticed that in case of changing the maximum value for mass m in the previously mentioned interval, the software provides as optimum value that maximum value. For this reason, the optimization of this parameter, mass m, has been abandoned.

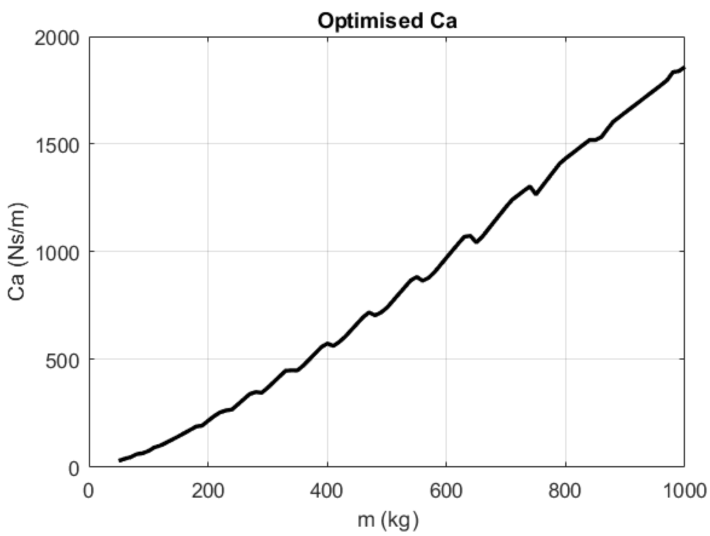

Figure 5. The damping coefficient c_a with respect to the absorber mass m.

In these circumstances, another optimization software has been conceived, which provides the minimum amplitude and the values of parameters c_a and k_a, for each considered value for mass *m*. The results are presented in figure 3, figure 4 and figure 5. From figure 3, the constructive accepted couple mass-minimum amplitude is chosen. The parameter k_a results from figure 4, while c_a results from figure 5, both corresponding to a selected mass *m*.

As an example, *for* m=400kg, the resulted optimum parameters are: the minimum amplitude is A_{min}=3.715x10^{-4} m, k_a=6600N/m and c_a=574.9Ns/m.

It has been noticed that the amplitude mitigates less and less when the value of mass *m* increases, as shown in figure 3. So, when the mass *m* increases from 100kg to 500kg, the amplitude decreases from 5.635x10^{-4}m to 3.446x10^{-4}m, which means a decrease of 38.84%. When mass m=1000kg, the corresponding amplitude is 2.717x10^{-4}m. So, compared to the value which results for m=500kg, the amplitude decreases with 21.15%.

5. Conclusions

The goal of this paper is to solve the problem of selection for classical dynamic vibration absorber (C-DVA) in terms of their effectiveness for a wide frequency range, around the maximum frequency.

The solution provided by the present work is based on an optimization software created by the authors, which provides the optimum values for the parameters of the dynamic absorber, attached to a vibrating system with damping.

Because the absorber mass has always the maximum value in the considered interval, the conceived software offers the possibility of selecting the best solution for the couple absorber mass–minimum amplitude. Then, for this solution, the optimum corresponding parameters, that is the spring stiffness and the absorber damping coefficient, can be immediately determined.

In this manner, each user can conceive an optimum dynamic absorber, based on the absorber mass and the corresponding optimum amplitude.

6. References

[1] Magdaleno A and Lorenzana A 2020 *Mech. Sist. Sign. Proc.* **135** 106378
[2] Love J S and Taylor Z J 2019 *J. of Wind Eng. & Ind. Aerodyn.* **194** 103977
[3] Huang X, Su Z and Hua H 2018 *Ocean Eng.* **155** pp 131-143
[4] Tehrani G G and Dardel M 2019 *Commun. Nonlin. Sci. Numer. Simulat.* **69** pp 1-30
[5] Shen Y, Chen L, Yang X, Shi D and Yang J 2016 *J. Sound. Vibr.* **361** pp 148-158
[6] Chun S, Lee Y and Kim T H 2015 *J. Sound. Vibr.* **335** pp 55-65
[7] Hu Y and Chen M Z Q 2015 *Int. J. Mech. Sci.* **99** pp 297-307
[8] Zhou S, Jean-Mistral C and Chesne S 2019 *Int. J. Mech. Sci.* **157-158** pp 528-541
[9] Shen Y, Peng H, Li X and Yang S 2017 *Mech. Sist. Sign. Proc.* **85** pp 193-203
[10] Shen Y, Xing Z, Yang S and Sun J 2019 *Mech. Sist. Sign. Proc.* **133** 106282
[11] Brown B and Singh T 2011 *J. Sound. Vibr.* **330** pp 2437-2448
[12] Tigli O F 2012 *J. Sound. Vibr.* **331** pp 3035-3049