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Influence of dash-pot with controllable damping coefficient on damping efficiency of TMDI

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Abstract. In this paper we study the dynamics of two degree freedom system, which is consist of main body and tuned mass damper with inerter (TMDI). We add the dash-pot with variable damping coefficient to TMDI to study the overall efficiency of the device. The shape of non-linear characteristic of the dash-pot is dependent on one control parameter which governs the steepness of the function and the value of damping coefficient changes according to the relative displacement or velocity between main mass and tuned mass damper. We show the two parameters diagrams showing the maximum amplitude of the main body versus frequency of excitation of main body and controlling parameter.

1 Introduction

All mechanical systems are susceptible to oscillate according to its natural frequency [1]. Therefore, matching of the external excitation forcing frequency with the natural frequency can lead to resonance phenomenon what is one of the reasons of structure's destruction [2]. Assuming inability to remove external excitation one can move the work point far from the natural frequency or use am external devices which absorb the energy. Limitation of frequencies' spectra corresponding to the first alternative is useful mainly for systems working under known and restricted conditions. In a more general case with low predictability of external forcing frequency, additional devices become the best solution [3]. In 1909 Frahm patented the classical tuned mass damper (TMD) [4]. The TMD is consists of a body of mass m and spring with stiffness k where the mass m is smaller than the mass of damped system but their natural frequencies are the same. Frahm device is extremely effective in reducing the response of the damped structure only in resonance, but outside the resonance it causes the amplification of oscillations' magnitude. To overcome this problem Den Hartog [3] proposed to add a viscous damper to Frahm's system design.

One of possible solution to extend the efficiency of TMD is addition of an inerter [6]. Inerter is a mechanical device generating force proportional to acceleration between its ends [5]. Initially inerters were successfully applied in sports cars' suspensions [7]. Now we observe growing number of studies on other possible applications [8–12].

This work is devoted to the examination of the efficiency of a tuned mass damper with ineter (TMDI) with a non-linear dash-pot. We use the non-linear dash-pot which damping function is dependent on one control parameter and relative displacement or velocity between the TMDI

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and main mass. We show how such dash-pot is influencing the mitigation properties of TMDI using different shapes of damping functions.

2 The model of the tuned mass damper with inerter and non-linear dash-pot

Let us describe the physical model of the system. It has two degrees of freedom and consists of two coupled oscillators that can move in the vertical direction. The first one serves as a model of the main oscillator which motion we are going to mitigate. It is connected with the support and forced via kinematic excitation realized with the crank mechanism. The second oscillator is connected to the first one and represents the TMDI. It is connected with the main oscillator via linear spring, non-linear viscous damper and inerter. The motion of the system is described by two generalized coordinates: the position of the main oscillator by coordinate x, while the displacement of the TMDI by coordinate y. The main oscillator is characterized with the following parameters: M is its mass, K is the stiffness of the single spring that connects the main mass to the ground and C is the viscous damping coefficient of dash-pot that links mass M and the support. Formula $A\cos\left(\frac{s\pi}{30}t\right)$ expresses the kinematic excitation of the main oscillator with the frequency $\frac{s\pi}{30}$ that corresponds to the speed $s\,[rpm]$ of the forcing servomotor and the crank on the exciting plate set to A. During the investigation we analyse the response of the structure for fixed crank $A = 0.02955\,[\mathrm{m}]$ and different frequencies of excitation that corresponds to the following range of rotational speeds $s\epsilon\langle 150,\, 250\rangle$ [rpm]. To characterize TMDI we use the following parameters values: the moving mass is given by m, the stiffness of the spring that connects it to the main oscillator is described with parameter k. The viscous damping function of the non-linear dash-pot that connect mass m with M is given by c. Values of parameters are following: $M = 102.66\,[\mathrm{kg}]$, $K = 8181.0\,[\frac{\mathrm{N}}{\mathrm{m}}]$, $C = 20.0\,[\frac{\mathrm{Ns}}{\mathrm{m}}]$, $A = 0.02955\,[\mathrm{m}]$, $s\epsilon\langle 150,\, 300\rangle$ [rpm], $m = 12.81\,[\mathrm{kg}]$, $k = 10985.1\,[\frac{\mathrm{N}}{\mathrm{m}}]$ and $I = 6.877\,[\mathrm{kg}]$. Damping function c is variable and we introduce it in each subsection separately with corresponding equations of system's motion .

3 Influence of a linear viscous dash-pot on the response of the system

In this section we study the response of a linear system with varying of linear viscous damping coefficient. The system is given by the following equations of motion:

$$M\ddot{x} + 7Kx + C\dot{x} + I\left(\ddot{x} - \ddot{y}\right) + k\left(x - y\right) + \\ + c\left(\dot{x} - \dot{y}\right) = KA\cos\left(\frac{s\pi}{30}t\right), \quad (1)$$

$$m\ddot{y} - I\left(\ddot{x} - \ddot{y}\right) - k\left(x - y\right) - c\left(\dot{x} - \dot{y}\right) = 0, \quad (2)$$

where the c is varied in the following range $c\epsilon\langle 30,\, 330\rangle$ [Nm/s]. As aforementioned, all our investigations are possible to realize experimentally and the chosen values of all parameters are based on the characteristics of real devices. We present obtained results in Fig. 2(a) as a colour diagram. The plot is calculated numerically with Auto07p, we divide the range of parameter c in 100 steps and for each we compute the FRC. The blue colour corresponds to low amplitude and it dominates in the plot. The increase of amplitude is observed in neighbourhood of resonances. For low value of damping $c = 30\,[\mathrm{Nm/s}]$ two resonances occur for $s = 194\,[\mathrm{rpm}]$ and $s = 257\,[\mathrm{rpm}]$. For maximum value of damping coefficient we observe one resonance for $s = 208\,[\mathrm{rpm}]$. The transition is caused by increase of value of damping coefficient which cause the stiffening of connection between main body and damper. Hence, the independent motion of the TMD is no more possible and the system is reduced to one DoF.

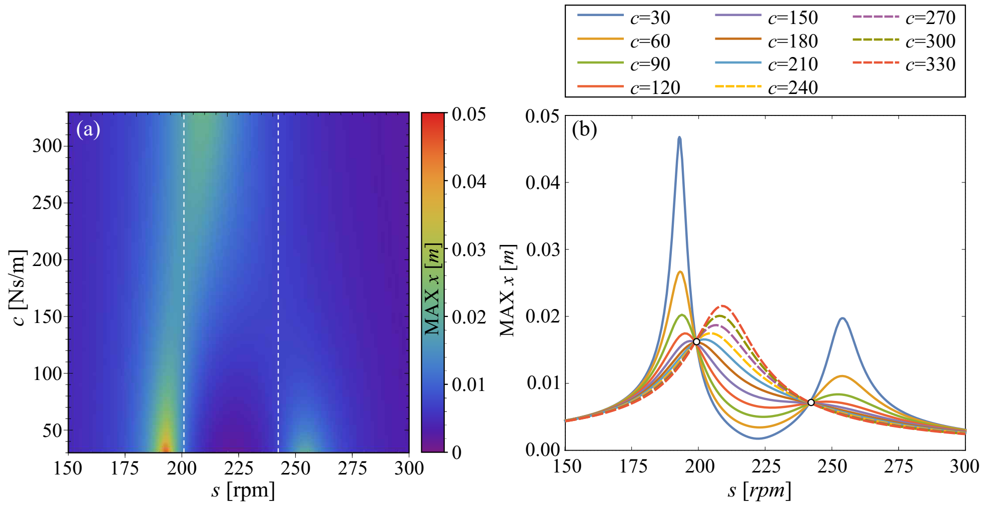

Figure 2. In panel (a) we show two dimensional plot of maximum amplitude of damped system (max x[m]) in two parameters space: c and s. Plot shows influence of linear viscus damper with constant damping coefficient (Eqs (1,2)) on system response. Panel (b) shows eleven exemplary FRCs of damped system. Points P and Q are marked as circles in panel (b).

The corresponding FRCs are shown in Fig. 2(b) for eleven values of damping coefficients starting with $c = 30\,[\mathrm{Nm/s}]$ and step $\Delta c = 30\,[\mathrm{Nm/s}]$. The amplitude of second resonance is deceasing rapidly and finally for $c = 150\,[\mathrm{Nm/s}]$ the increase of amplitude is not more present. As it is well known all FRCs are crossing two points - P and Q where the response of system is nearly independent on value of damping coefficient.

4 Influence of a non-linear viscous dash-pot on the response of the system

In this subsection we show the influence of non-linear damping on the response of the main system. The equations of system are as follow:

$$M\ddot{x} + 7Kx + C\dot{x} + I(\ddot{x} - \ddot{y}) + k(x - y) + \\ + c(x, y, \dot{x}, \dot{y})(\dot{x} - \dot{y}) = KA\cos\left(\frac{s\pi}{30}t\right), \quad (3)$$

$$m\ddot{y} - I(\ddot{x} - \ddot{y}) - k(x - y) + \\ - c(x, y, \dot{x}, \dot{y})(\dot{x} - \dot{y}) = 0, \quad (4)$$

where $c(x, y, \dot{x}, \dot{y})$ is a continuous damping function which depends on the retaliative displacement or velocity between the main system and the TMD. For sake of generality, we include all state variables in the general form of damping function. However, we consider two types of damping function, i.e., one is dependent on the retaliative displacement and second one on the relative velocity. The system is smooth, hence we perform all calculations in Auto07p.

The first case corresponds to the damping dependent on the retaliative displacement between the main mass and the TMD. For $(x - y) = 0.0$ the damping function has minimum value: $c_{MIN} = 30\,[\mathrm{Nm/s}]$ and it is increasing according to following formula:

$$c(x, y, \dot{x}, \dot{y}) = c(x, y) = \\ = c_{MIN}\left(1 + 10\tanh\left(a(x-y)^2\right)\right), \quad (5)$$

to its maximum value: $c_{MAX} = 330$ [Nm/s]. The controlling parameter of the damping function is a and it varies in range $a \in \langle 10, 30000\rangle$ [m^{-2}] which let us change the value of relative displacement for which function achieves its maximum. Ten exemplary plots of the damping functions are presented in Fig. 3(a), we restricted the range of relative displacement to $|x - y| < 0.15$ [m].

Results are shown in Fig. 3(b, c). In panel (b) we present a two dimensional colour plot of maximum amplitude of main system as a function of rotational speed s (frequency of excitation) and parameter of damping function a. White dash lines indicate the location of points $P = 205$ [rpm] and $Q = 240$ [rpm] for system with linear viscous damping. In Fig. 3(c) we show eleven exemplary FRCs for chosen values of parameter a within the considered range. As it is easy to see points P and Q do not change their locations with change of shape of damping function, while FRCs are significantly different. For low value of a we observe two resonances and for maximum value of a the main mass and TMD behave like one DoF system.

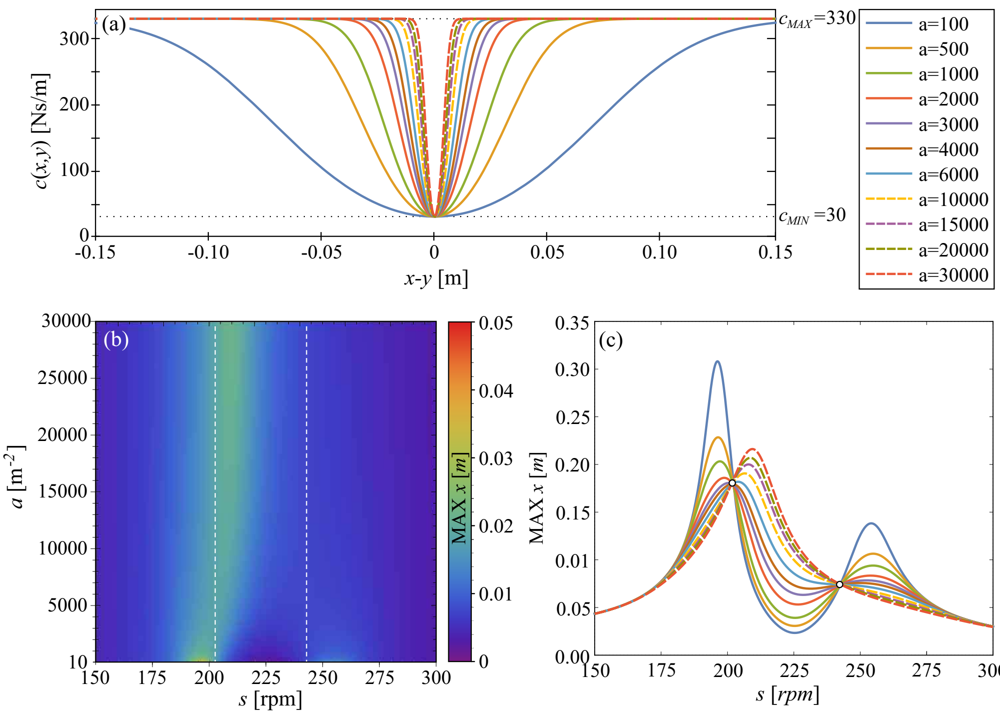

Figure 3. In panel (a) we show the damping functions $c(x, y)$ (Eq. 5) for selected values of a. In panel (b) we show two dimensional plot of maximum amplitude of damped system (max x[m]) in two parameters space: a and s. The last panel (c) shows eleven exemplary FRCs of damped system. Points P and Q are marked as circles in panel (b).

In second case we consider the same shape of damping function as in previous case but it is dependent on retaliative velocity:

$$c\left(x, y, \dot{x}, \dot{y}\right) = c\left(\dot{x}, \dot{y}\right) = \\ = c_{MIN}\left(1 + 10\tanh\left(a\left(\dot{x} - \dot{y}\right)^2\right)\right), \quad (6)$$

where parameter a is in the range $a \epsilon \langle 0.1,\ 7\rangle$ [s^2/m^2]. Eleven exemplary plots are shown in Fig. 4(a) and as in previous case the value of a is governing the slope of function.

The corresponding two dimensional colour plots of amplitude versus rotational speed of servomotor and parameter a are shown in Fig. 4(b). The dashed white lines indicate the position of pointspoints $P = 205$ [rpm] and $Q = 240$ [rpm]. With increase of parameter a the system is behaving as one DoF system. All periodic solutions are stable and location of points P and Q along the FRC is not affected by non-linear damping (see Fig. 4(c)).

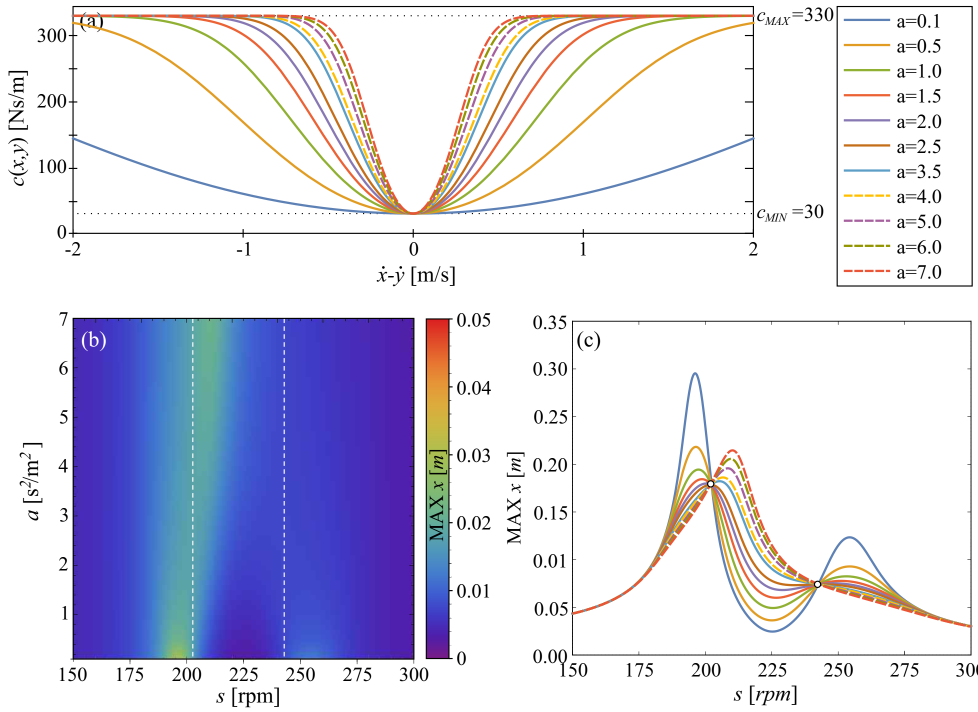

Figure 4. In panel (a) we show the damping functions $c\left(\dot{x}, \dot{y}\right)$ (Eq. 6) for selected values of a. In panel (b) we show two dimensional plot of maximum amplitude of damped system (max x[m]) in two parameters space: a and s. The last panel (c) shows eleven exemplary FRCs of damped system. Points P and Q are marked as circles in panel (b).

5 Conclusions

In this paper we present the description of the influence of a non-linear dash-pot with variable coefficient of damping placed between the main mass and the TMD. In case of linear dashpot there are two points along the FRC that do not dependent on the value damping coefficient known as points P and Q. We study how using the non-linear dash-pot and the inerter we can decrease the maximum amplitude of damped body. Analysis proves that suspicion of stationary character of this points is true even with non-linear damper. However, we find ranges where one can observe the decrease of amplitude of main system.

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