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H-INFINITY OPTIMIZATION OF DYNAMIC VIBRATION ABSORBER WITH NEGATIVE STIFFNESS

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***Abstract**: In the present paper, a dynamic vibration absorber with negative stiffness is developed. This study aims to optimize absorber components by using the two-fixed point approach and H_∞ maximization criterion. Optimum values of the grounded stiffness and tuned mass damper components are derived to minimize resonance amplitude of an un-damped system under harmonic excitation. The two-fixed point theory is employed to minimize the tuning frequency, damping ratio and negative stiffness parameter. Optimum parameters of the absorber are then formulated and used to quantify the influence of each factor on the primary system response. Finally, the mitigation of resonance oscillation amplitude of the primary structure is compared with those of the traditional one. Thus it is shown that the pre-tensioned stiffness provides well attenuation at the resonant vibration range. Adding, this device can be also broadening the efficient frequency range of vibration*

***Key words:** Dynamic vibration absorber, pre-tensioned negative stiffness, H-infinity optimization, mitigation of the resonant vibration amplitude, control performance.*

1. INTRODUCTION

In many engineering domains, the tuned mass damper (TMD) or dynamic vibration absorber (DVA) is widely used in various fields of engineering and construction. The large use is due its reliability, efficiently and low cost. Really, DVA is among passive control device and one of the common procedures to control system vibrations. The optimization of DVA parameters plays an important concern to minimize maximum displacement magnitudes of the primary system under dynamic loads.

The optimization criterion is firstly applied to design traditional DVA. In literature, optimization criteria already developed in DVA design are regrouped in (1) stability maximization, (2) H_2 optimization and (3) H_∞ optimization. The last one of optimization was developed by Ormondroyd and Den Hartog [1] for damped TMD used for extending the resonance frequency range when the primary system is subjected to harmonic force. Optimal tuning factor and that of damping ratio were studied for traditional DVA by Hahnkamm [2] and later by Brock [3], respectively. Den Hartog [4] derived the optimum parameters of damped DVA using the fixed-point approach and proved that they are not closed due the introduction of hypotheses simplifying the mathematical problem. Based on bellow results, Nichihara and Asami [5] improved Den Hartog solutions and concluded that the fixed-point theory leads to well convergence of the exact solution of H_∞ optimization.

Then, various configurations of structural DVA have been proposed. In this domain, Asami [6] studied the optimization of the three-element device of DVA and concluded that this device had a better control performance with the same mass ratio. More to exhibit better control performance of DVA with negative stiffness, Ren [7] developed a DVA device with damping element connected directly to the ground. Based on previous results, Wang [8] coupled Asami and Ren models to obtain the optimum parameters of DVA with negative stiffness. The introduction of the inerter to three-element type of DVA is established and the effectiveness of vibration suppressing was investigated [9]. Once DVA was connected to a given structure, it is capable to absorb vibratory energy to protect the primary system to high excessive vibrations.

In this case, optimal choice of absorber parameters were suggested [10]. To reach pertinent results in the construction domain, Barredo et al. [11-12] integrated an inerter device to DVA to minimize displacement magnitudes of the primary system by using the extended fixed-points technique. Also, optimal parameters of DVA equipped with lightly and moderately linear damped rotary system under harmonic torque has been studied [13]. In this study, a closed formula of the optimum tuning coefficient is obtained based on the fixed point approach. This process is essentially used to protect equipment from steady state harmonic disturbances. The performance of a DVA using a magneto-rheological damper is then employed [14].

Currently, there are many studies aiming the pre-compressed load in stiffness element on vibration isolation system based on their mechanical characteristics [15-18] compared to those taken in the optimal design of TMD parameters. To improve the dynamic performance of TMDs, new devices of TMD with preload stiffness have been developed. In this subject, Hao et al. [19] have coupled the Maxwell model with viscoelastic material for multiples negative stiffness springs. In this case, TMD parameters are optimized with better reduction of oscillations both under harmonic and random excitations [20]. Most mechanical vibrations cause not only noise but also decrease of the service life and operating performance of equipments. To minimize these effects, a grounded stiffness and inerter have been employed to change the natural frequency of the system [21].

In this work, the H_∞ infinity optimization of a pre-tensioned tuned mass damper for passive control of un-damped linear systems is studied. When the pre-load is applied on spring, it tends to absorb a restoring energy in which the direction of the force is opposite to displacement one. The effect of pre-tensioned stiffness on the tuning frequency curve of the primary structure is investigated, thus an analytical solution of the optimum parameters is investigated based on the two-fixed points approach for Voigt-Kelvin DVA model under harmonic excitation.

The paper is organized as follow. Section II presents the equation of motion of DVA considering the pre-tensioned TMD. In section III, the formulation of optimum DVA parameters is described. The section IV shows the influence of pre-tensioned stiffness parameter, frequency tuning ratio and optimal damping coefficient. The section V shows the comparison between traditional DVA and pre-tensioned TMD. Conclusions of the study are drawn in section VI.

2. Pre-tensioned TMD model

In this study, the traditional TMD developed by Den Hartog (model A) shown in Fig. 1 is taken as a witness model to show the performance and ability of developed pre-tensioned TMD (model B) described in Fig. 2. In this modelling, the absorber is attached to the ground by a negative stiffness and damper according to the Voigt-Kelvin model.
The load is a harmonic force $F(t) = F_0 \cos(\omega t)$ acted on the primary structure, where F_0 and ω are the amplitude and frequency of the force excitation.

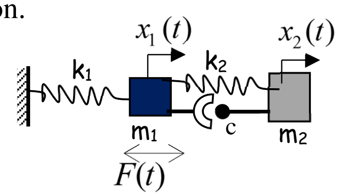

Fig. 1. Traditional model

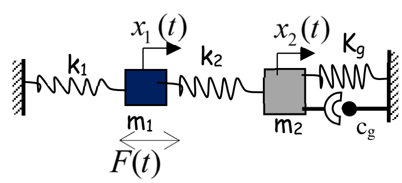

Fig. 2. DVA with negative stiffness

2.1. Analytical solution

Equations of motion of the primary system and DVA can be formulated using the second law of Newton.

$$m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2x_2 = F_0\, cos(\omega t)$$
$$m_2\ddot{x}_2 + c_g\dot{x}_2 + (k_g + k_2)x_2 - k_2x_1 = 0 \tag{1}$$

where m_1, m_2, k_1 and k_2 are the masses and stiffness coefficients of the primary system and absorber, respectively. k_g and c_g are the coefficient of pre-tensioned stiffness spring and damping parameter. x_1 and x_2 are the displacements of the primary system and absorber, respectively.
The Use of following dimensionless parameters is:

$$\omega_1^2 = \frac{k_1}{m_1}, \quad k_{eq} = k_2 + k_g, \quad \omega_2^2 = \frac{k_{eq}}{m_2}, \quad \xi = \frac{c_g}{2m_2\omega_2},$$

$$\mu = \frac{m_2}{m_1}, \; f = \frac{\omega_1}{\omega_2}, \; r = \frac{\omega}{\omega_2}, \; \lambda = \frac{k_2}{k_{eq}}.$$

Letting $F_0 \cos\omega t$ in (1) be represented by $F_0 e^{i\omega t}$, the steady-state solutions can take the forms:

$$x_1(t) = X_1 e^{i\omega t}$$
$$x_2(t) = X_2 e^{i\omega t} \tag{2}$$

Equation (1) can be rewritten in matrix form using the equation (2) with introducing dimensionless parameters.

$$\begin{pmatrix} f^2+\lambda\mu-r^2 & -\mu\lambda \\ -\lambda & 1-r^2+2i\xi r \end{pmatrix}\begin{pmatrix} X_1 \\ X_2 \end{pmatrix} = \begin{pmatrix} \frac{F_0}{k_1} f^2 \\ 0 \end{pmatrix} \tag{3}$$

The displacement of the primary structure is given by:

$$X_1 = \frac{F_0}{k_1}\frac{f^2\left(1-r^2+2i\xi r\right)}{\Delta} \tag{4}$$

where $\Delta = \left[\left(1-r^2+2i\xi r\right)\left(f^2+\lambda\mu-r^2\right)\right] - \mu\lambda^2$

The normalized amplitude amplification factor of the primary structure (model B) can be formulated using the equation (4).

$$H(r) = \left|\frac{X_1}{F_0/k_1}\right| = \frac{f^2\sqrt{\left(1-r^2\right)^2+\left(2\xi r\right)^2}}{\sqrt{\alpha^2+\beta^2}} \tag{5}$$

$\alpha = \left(1-r^2\right)\left(f^2+\lambda\mu-r^2\right) - \mu\lambda^2$ and

$\beta = 2\xi r\left(f^2+\lambda\mu-r^2\right)$

The amplitude of the frequency response function (5) can be written as

$$H(r) = \sqrt{\frac{A\xi^2+B}{C\xi^2+D}} \tag{6}$$

With; $A = 4f^4r^2$, $B = f^4\left(1-r^2\right)^2$, and $C = 4r^2\left(f^2+\lambda\mu-r^2\right)^2$

$$D = \left(\left(1-r^2\right)\left(f^2+\lambda\mu-r^2\right)-\mu\lambda^2\right)^2$$

It is seen that excited frequency-normalized amplitude curves pass by two-fixed points P and Q independently of the damping ratio (Fig. 3). The two-fixed point approach can be used to minimize the amplitude displacement of the primary based on DVA parameters.

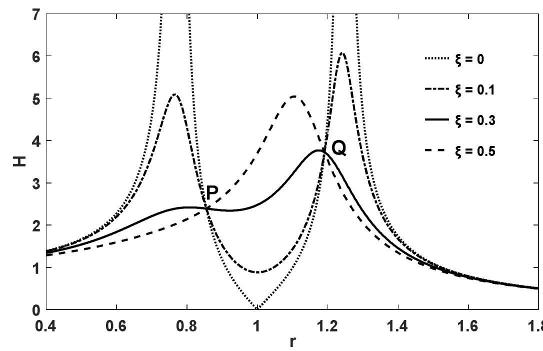

Fig. 3. Normalized amplitude-frequency curves under $f=1$, $\mu=0.10$, $\lambda=1.5$ for various damping values

2.2. Optimization of DVA parameters

2.2.1. Optimization of the relative natural frequency factor

An approach used to optimize damper parameters is based on the two fixed-points theory. This approach shows that displacement magnitudes of the primary masse pass through two specific points independently of the damping coefficient (Fig. 3).
The amplitude amplification factors for $\xi \to \infty$ in equation (6) becomes

$$H(r) = \frac{f^2}{f^2+\lambda\mu-r^2} \tag{7}$$

In the same manner, displacement magnitudes are the same applying the two-fixed point approach for over-damped system and un-damped absorber. One could write

$$\frac{A}{C} = \frac{B}{D} \tag{8}$$

Thus, the development of the equation (8) holds:

$$\frac{1}{f^2+\lambda\mu-r^2} = \pm\frac{\left(1-r^2\right)}{\left(1-r^2\right)\left(f^2+\lambda\mu-r^2\right)-\mu\lambda^2} \quad (9)$$

The negative part can be taken into account and the equation (9) becomes

$$2r^4-2\left(f^2+\lambda\mu+1\right)r^2+2\left(f^2+\lambda\mu\right)-\mu\lambda^2=0 \quad (10)$$

Roots of (10); relative frequency ratios at P and Q points are

$$r_{P,Q}=\sqrt{\frac{f^2+\lambda\mu+1\mp\sqrt{\lambda^2\mu^2+2\lambda^2\mu+2\lambda f^2\mu-2\lambda\mu+f^4-2f^2+1}}{2}} \quad (11)$$

The sum of root squares is

$$r_P{}^2+r_Q{}^2=f^2+\lambda\mu+1 \quad (12)$$

Applying the two-fixed point theory, it can be deduced that

$$\frac{1}{f^2+\lambda\mu-r_P{}^2}=-\frac{1}{f^2+\lambda\mu-r_Q{}^2} \quad (13)$$

That leads to

$$r_P{}^2+r_Q{}^2=2f^2+2\lambda\mu \quad (14)$$

The optimal relative frequency ratio can be obtained by using equations (12) and (14).

$$f^{opt}=\sqrt{1-\lambda\mu} \quad (15)$$

2.2.2. Optimization the damping factor

The substitution of equations (11) and (15) into (7), the common displacement amplitude of P and Q points become

$$H\left(r_P\right)=H\left(r_Q\right)=\frac{2\left(\lambda\mu-1\right)}{\lambda\sqrt{2\mu}} \quad (16)$$

The root of equation (5) in ξ^2 holds

$$\xi^2=\frac{f^4\left(1-r^2\right)^2-H(r_Q)^2\left[\left(1-r^2\right)\left(f^2+\lambda\mu-r^2\right)-\mu\lambda^2\right]^2}{4r^2\left[H(r_Q)^2\left(f^2+\lambda\mu-r^2\right)^2-f^4\right]} \quad (17)$$

According to [8], assuming $r^2=r_Q{}^2+\delta$ and $H(r)=\frac{2(\lambda\mu-1)}{\lambda\sqrt{2\mu}}$ with letting $\delta\to 0$, the relationship (17) can be written as

$$\xi^2=\frac{A_0+A_1\delta+A_2\delta^2+A_3\delta^3+\cdots}{B_0+B_1\delta+B_2\delta^2+B_3\delta^3+\cdots} \quad (18)$$

Neglecting the higher-order parameters, the equation (18) becomes

$$\xi^2=\frac{A_1}{B_1} \quad (19)$$

$$\begin{aligned}A_1 &= 2H^2\left[\left(f^2+\lambda\mu\right)^2+\left(f^2+\lambda\mu\right)\left(1-\lambda^2\mu\right)-\lambda^2\mu\right]-2f^4\\ &\quad -2\left[H^2\left(\left(f^2+\lambda\mu\right)^2+2\lambda\mu(2-\lambda)+4f^2+1\right)-f^4\right]r_Q{}^2\\ &\quad +6H^2\left(1+f^2+\lambda\mu\right)r_Q{}^4-4H^2r_Q{}^6\end{aligned}$$

$$B_1=4H^2\left(f^2+\lambda\mu\right)^2-4f^4-16H^2\left(f^2+\lambda\mu\right)r_Q{}^2+12H^2r_Q{}^4$$

Substituting equations (11), (15) and (16) in (19), the optimal damping coefficient relative to the Q-point is

$$\xi_Q^{opt}=\frac{1}{2}\sqrt{\frac{3\lambda^2\mu}{\sqrt{2\lambda^2\mu}+2}} \quad (20)$$

This procedure can be applied to the second P-point, the corresponding optimal damping coefficient is computed:

$$\xi_P^{opt}=\frac{1}{2}\sqrt{\frac{-3\lambda^2\mu}{\sqrt{2\lambda^2\mu}-2}} \quad (21)$$

Thus an average value of optimal damping coefficient can be evaluated

$$\xi^{opt}=\frac{1}{2}\sqrt{\frac{3\lambda^2\mu}{\left(2-\lambda^2\mu\right)}} \quad (22)$$

2.2.3. Optimization of the pre-tensioned stiffness factor

Optimal expressions of the relative frequency (15) and damping (22) are been developed as a function of the pre-tension stiffening . The two-fixed point displacement magnitudes are much lower meaning the best control performance of the absorber system. Pretension of the grounded stiffening will cause a pre-displacement of the primary structure; therefore an approximation is used leading to the value of the response of excitation at zero frequency equal to the values of the two invariant points, that is

$$H(r)\big|_{r=0} = H(r)\big|_{P,Q} \tag{23}$$

The equation (23) is applied to evaluate the value of pre-tension stiffness.

$$\begin{aligned} \lambda_{1,2} &= \pm\sqrt{\frac{2}{\mu}} \\ \lambda_{3,4} &= \pm\sqrt{\frac{1}{2\mu}} \end{aligned} \tag{24}$$

Values of the above equation are taken into the optimum natural frequency ratio, optimum damping ratio and the normalized amplitude of steady-state frequency response of the primary structure.

Obviously, only λ_3 this can guarantee the stability of the system. The optimum pre-tensioned stiffness parameter is

$$\lambda^{opt} = \frac{1}{\sqrt{2\mu}} \tag{25}$$

Finally, parameters of the tuned mass damper are then optimized using optimum values of the frequency of masses (15), the damper factor (22) and the pre-tension stiffness ratio (25).

3. Analysis of parameters optimization

In this study, parameters having an influence on the DVA response are the subject of this section. In this case, the influence of the stiffness, tuning frequency and damping coefficient is expressed based on the mass ratio because this last can be limited according to manufacture and of operational conditions.

So, eventually the mass ratio can take values of $0 < \mu \le 0.25$.

3.1. Influence of pre-tensioned stiffness parameter

Fig. 4 depicts the evolution of optimal pre-tensioned stiffness coefficient versus mass ratio. It is shown that λ^{opt} decreases with the increase of μ. This evolution is strongly emphasized for $\mu \le 0.25$. Adding, this study showed that the optimum pre-tension stiffness parameter is not more influenced by the mass ratio for $\mu > 0.25$.

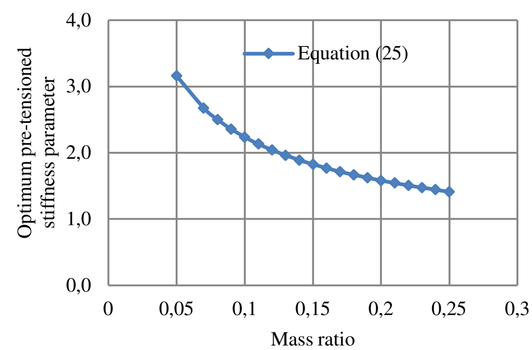

Fig. 4. Optimum pre-tensioned stiffness coefficient versus mass ratio

3.2. Influence of optimal tuning frequency ratio

The mass of DVA is based on the manufacturing cost and space conditions. It is seen that for feeble values of the pre-tensioned stiffness parameter. The optimal tuning factor f^{opt} has a linear relationship versus the mass ratio. The curve becomes nonlinear for $\lambda > 0.2$.

Variation of the optimum tuning frequency ratio is plotted against the mass ratio of the DVA. It is seen f^{opt} decreases with the increase of the mass ratio. Adding, the optimum tuning frequency ratio decreases with the increase of the pre-tensioned stiffness coefficient (Fig. 5).

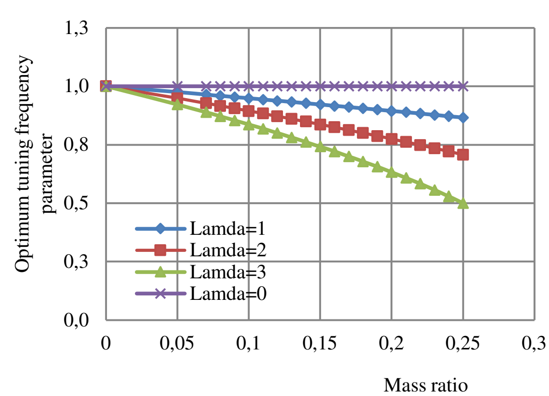

Fig. 5. Optimum tuning frequency ratio versus mass ratio

3.3. Influence of optimal damping coefficient

The influence of pre-tensioned stiffness parameter on optimal damping ratio is shown (Fig. 6). This effect is dispersed in a unified way for $\lambda \leq 2$ values. In this case, the optimum damping ratio increases with the increase of the mass ratio. Nonlinearity effect becomes very important for $\lambda > 0.2$. Thus the optimum damping ratio varies with a great value corresponding too little mass ratio variation. In addition, ξ^{opt} increases with the increase of the pre-tensioned stiffness factor.

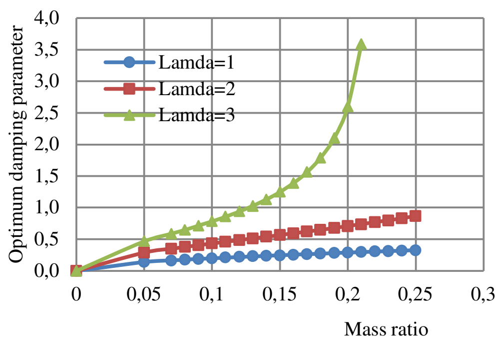

Fig. 6. Optimum dumping coefficient versus mass ratio

4. Comparison of the optimum tuning parameter

In this section, we will check the performance and the ability of the DVA with negative stiffness spring by comparison it with traditional DVA developed by Ormondroyd and Den Hartog [1]. Comparison of obtained results of the two types of DVAs is made. It is seen that DVA with negative stiffness spring can provide a better control performance than traditional DVA.

Figs. 7-9 describe the relative amplitude displacement versus frequency ratio for various mass ratios. This comparison is made between the pre-tensioned model and traditional DVA. It is seen that the pre-tensioned model presents two peaks, which are at r=0.50 and r=1.30, respectively. Regardless of the mass coefficient, the pre-tensioned model shows good vibration attenuation compared to the traditional model under harmonic loading. This performance ranged from 37.19% for mu = 0.05, passed by 9.78% for mu = 0.15 and reached 4.86% for mu = 0.25.

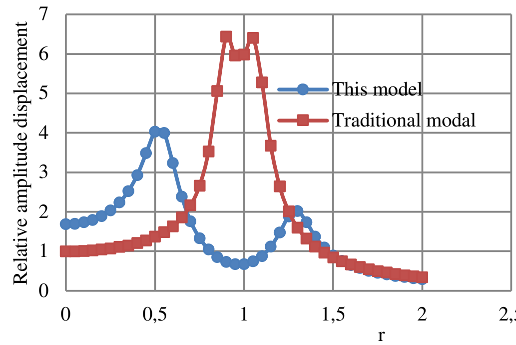

Fig. 7. Relative displacement amplitude versus frequency ratio for μ=0.05

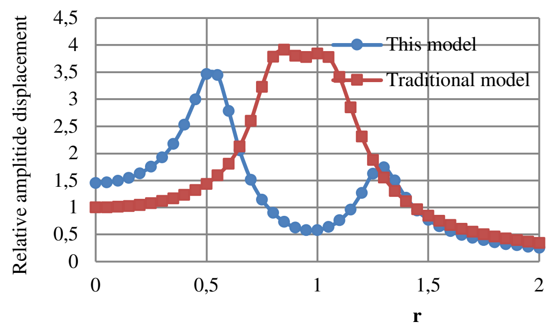

Fig. 8. Relative displacement amplitude versus frequency ratio for μ=0.15

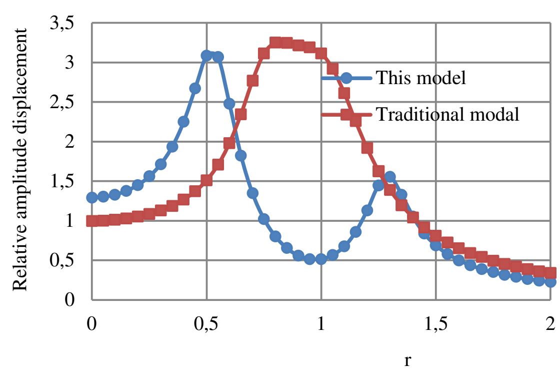

Fig.9. Relative displacement amplitude versus frequency ratio for μ=0.25

Adding, obtained results proved that the proposed TMD has a larger mitigation of resonant vibration amplitude of the main structure than the traditional TMD.

TABLE I: FORMULAS OF H $_\infty$ OPTIMAL PARAMETERS OF TMDS.

TMD MODEL	f^{opt}	ξ^{opt}	λ^{opt}
TRADITIONAL TMD	$\frac{1}{1+\mu}$	$\sqrt{\frac{3\mu}{8(1+\mu)}}$	-
THIS MODEL	$\sqrt{(1-\lambda\mu)}$	$\frac{1}{2}\sqrt{\frac{3\lambda^2\mu}{(2-\lambda^2\mu)}}$	$\frac{1}{\sqrt{2\mu}}$

5. CONCLUSION

A dynamic vibration absorber equipped with negative stiffness spring is investigated. The investigation is focused on the optimization of DVA parameters by using the two-point approach and H $_\infty$ optimization criterion. Thus, the optimum damping ratio, tuning frequency and pre-tensioned stiffness factor are formulated. The conclusions can be drawn are as follow:

1- The increase of mass ratio decreases the optimum pre-tensioned stiffness parameter. This diminution decreases from 3.162 to 1.416, it's about 223.6%.
2- The tuning frequency parameter decreases with increase of mass ratio and it tapers as soon as pre-tensioned stiffness parameter becomes important.
3- The optimum damping coefficient versus mass ratio is influenced by the pre-tensioned stiffening parameter. The pace of variation is dispersed in a unified way for $\lambda \leq 0.2$ but becomes a nonlinear relationship for $\lambda > 0.2$.
4- Obtained results show that DVA equipped with negative stiffness spring can provide a better control performance than traditional DVA.
5- Normalized amplitude displacement versus frequency ratio curves show a well improvement of the proposed DVA, a larger mitigation of resonant vibration amplitude of the main structure and a better control performance are well observed. Also, according to the mass ratio, the performance ranged from 37.19% ($\mu = 0.05$) to 4.86% ($\mu = 0.25$).

Finally, results of this study provide some technical information to design more effective vibration control device in engineering practices.

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OPTIMIZAREA INFINITĂȚII H A ABSORBILOR DIN VIBRARE DINAMICĂ CU REZIDENȚĂ NEGATIVĂ

Rezumat*: În prezenta lucrare, este dezvoltat un absorbant dinamic de vibrații cu rigiditate negativă. Acest studiu își propune să optimizeze componentele absorbantului utilizând abordarea în două puncte fixe și criteriul de maximizare H. Valorile optime ale rigidității împământate și ale componentelor reglate ale amortizorului de masă sunt derivate pentru a minimiza amplitudinea de rezonanță a unui sistem neamortizat sub excitație armonică. Teoria cu două puncte fixe este utilizată pentru a minimiza frecvența de reglare, raportul de amortizare și parametrul de rigiditate negativă. Parametrii optimi ai absorbantului sunt apoi formulați și utilizați pentru a cuantifica influența fiecărui factor asupra răspunsului sistemului primar. În cele din urmă, atenuarea amplitudinii oscilației de rezonanță a structurii primare este comparată cu cele ale celei tradiționale. Astfel, se arată că rigiditatea pre-tensionată asigură o atenuare bună la intervalul de vibrații rezonante. În plus, acest dispozitiv poate extinde, de asemenea, gama eficientă de frecvență a vibrațiilor.*

Cuvinte cheie*: absorbant dinamic de vibrații, rigiditate negativă pretensionată, optimizare H-infinit, atenuarea amplitudinii vibrațiilor rezonante, performanță de control.*

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