한국기계가공학회지, 제18권, 제1호, pp.1～8(2019.01) ISSN 1598-6721(Print)
Journal of the Korean Society of Manufacturing Process Engineers, Vol. 18, No. 1, pp.1~8(2019.01) ISSN 2288-0771(Online)

https://doi.org/10.14775/ksmpe.2019.18.1.001

Design Parameter Analysis of a Dynamic Absorber for the Control of Machine Body Vibration

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기계 진동의 수동적 제어를 위한 동흡진기 설계인자 해석

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(Received 17 December 2018; received in revised form 27 December 2018; accepted 7 January 2019)

ABSTRACT

The optimal design parameters of a dynamic absorber (DA) in a machine body (that is considered as a rigid body) are discussed in this paper. The bounce and rotation motions of the rigid body have been controlled passively by a DA, which consists of a mass and a spring. The rigid body is subjected to a harmonically excited force and supported by linear springs at both ends. To define the motion of a rigid body with a DA, the equation of motion was expressed in the third-order matrix form. To define the optimal design conditions of a DA, the reduction of dynamic characteristics, represented by the amplitudes of bounce and rotation, and the transmitted powers, were evaluated and discussed. The level of reduction was found to be highly dependent on the location and spring stiffness of the DA.

Keywords : Optimal Design Parameters(최적 설계 인자), Dynamic Absorber(동흡진기), Bounce(상하운동), Rotation(회전운동), Transmitted Power(전달 동력)

1. Introduction

In the industrial fields, lots of machines and structures had experienced the troubles caused by vibration and unbalance phenomena. Numerous studies have been done to control the vibrations of the machines in operation in the past decades.

Currently it is known well that the fast and the accurate motions in the machining duration becomes one of the common trends for the optimal design. In the high speed operation, the vibration which may occurs at each parts of machine should be controlled actively or passively for the stability of the machine structure.

The vibration problems of machines and structures had been studied by lots of researchers. The vibration energy flow and the dynamic response of

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a beam, plate or some compound system have been analyzed. By using a model of an elastic beam, the vibrational intensity and control skills had been presented[1-4]. The structure analysis and the control of the vibrations of the machine tools had been introduced [5-10].

It is known that a body or part of machinery or robots in the industrial fields experience a few types of dynamic response being induced by a bouncing movement, a rotation and/or the constant reciprocal motions. So to control the above mentioned responses, the dynamic absorber is simply applied to the machine. The Dynamic Absorber (DA)[11] is known to be effective over narrow band of frequencies and is designed to make the natural frequencies of the main system be away from the forcing frequency. Hence in this study a machine which is subjected to a harmonically excited force is modelled theoretically as a rigid body in a two dimension and employed to be controlled passively by a Dynamic Absorber which consists of a mass and a spring. Two supports of a body are considered as linear springs having only the elastic property. Based on the equation of motion, two motions (bounce and rotation about the center of gravity (CG)) of a rigid body will be discussed.

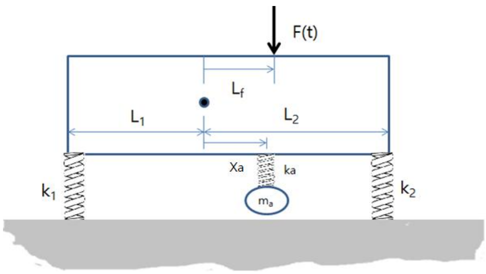


Fig. 1: A rigid body with two supports F(t) (the external force), Dynamic Absorber (DA), Distances based on the center of gravity (CG) [L1(to the left support), L2(to the right support), Xa(to the a DA), Lf(to the external force)] Mass items [m(a rigid body), ma(DA)] Stiffness items [k1(a left side), k2(a right side), ka(DA)]

To define the optimal design parameters of a DA, the cost function which is the ratio of the dynamic characteristics of a body without DA to that of a body with DA will be evaluated and discussed. Fig. 1 shows the theoretical model of a rigid body which is subjected to a harmonically excited force F(t) and has a Dynamic Absorber (DA) which is located at La being distanced from the center of gravity (CG).

2. The Governing Equations

The governing equations for the motions of a rigid body which has a dynamic absorber(DA), can be written in matrix form as:

$$\begin{pmatrix} m & 0 & 0 \\ 0 & m_a & 0 \\ 0 & 0 & J_G \end{pmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{\theta} \end{Bmatrix} + \begin{pmatrix} k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{31} & k_{32} & k_{33} \end{pmatrix} \begin{Bmatrix} x \\ y \\ \theta \end{Bmatrix} = \begin{Bmatrix} F(t) \\ 0 \\ F(t) L_f \end{Bmatrix} \qquad (1)$$

where x and Θ mean the bouncing and rotating displacements of a rigid body, respectively and y is the displacement of a Dynamic Absorber(DA), J_G is a mass moment of inertia of a rigid body. The elements of a matrix are given as

$$\begin{gathered} k_{11} = k_1 + k_2 + k_a,\ \ k_{12} = -k_a = k_{21}, \\ k_{13} = k_2 L_2 - k_1 L_1 = k_{31},\ \ k_{22} = k_a, \\ k_{23} = \mp k_a X_a = k_{32}, \\ k_{33} = k_1 L_1^2 + k_2 L_2^2 + k_a X_a^2 \end{gathered} \qquad (2)$$

In the above equations, the displacements of a rigid body and a DA are expressed into the amplitude of a spatial function and the time harmonic function. Then by suppressing the time term in Eq. (1), it can be rearranged in terms of amplitudes as,

$$\begin{pmatrix} z_{11} & z_{12} & z_{13} \\ z_{21} & z_{22} & z_{23} \\ z_{31} & z_{32} & z_{33} \end{pmatrix} \begin{Bmatrix} X \\ Y \\ \Theta \end{Bmatrix} = \begin{Bmatrix} f \\ 0 \\ f L_f \end{Bmatrix} \qquad (3)$$

where $z_{11} = k_1 + k_2 + k_a - \omega^2,\ z_{12} = -k_a = z_{21},$
$z_{13} = k_2 L_2 - k_1 L_1 = z_{31},\quad z_{22} = k_a - m_a \omega^2,$
$z_{23} = \mp k_a X_a = z_{32},$
$z_{33} = k_1 L_1^2 + k_2 L_2^2 + k_a X_a^2 - J\omega^2$

Here the amplitudes can be obtained by multiplying both sides of Eq.(3) by $[Z]^{-1}$ which is the inverse of a third-order matrix [Z] in Eq.(3). Finally the amplitudes become as follows,

$$X = \frac{f(z_{22}z_{33} - z_{23}z_{32}) + fL_f(z_{12}z_{23} - z_{13}z_{22})}{\triangle}$$
$$X_a = \frac{f(z_{23}z_{31} - z_{21}z_{33}) + fL_f(z_{13}z_{21} - z_{11}z_{23})}{\triangle} \qquad (4)$$
$$\Theta = \frac{f(z_{21}z_{32} - z_{22}z_{31}) + fL_f(z_{11}z_{22} - z_{12}z_{21})}{\triangle}$$

where 'Δ' represents the determinant of a third-order matrix [Z] and is expressed as

$$\triangle = z_{11}z_{22}z_{33} - z_{12}z_{21}z_{33} + z_{13}z_{21}z_{32} - z_{11}z_{23}z_{32} + z_{12}z_{23}z_{31} - z_{13}z_{22}z_{31}$$

The determinant will be used for determining the resonant frequencies to control the vibration of a rigid body. All values obtained in this study have been expressed into the dimensionless forms which are based on a certain quantity. The dimensionless mass (m_a) of a DA which is divided by a mass of rigid body (m) is given as 0.1 ~ 0.3 and the mass moment of inertia (J_G) is kept a constant as 0.1 based on the quantity of mL^2.
The center of a gravity (CG) of a lot machines are observed to deviate away from a geometric center of a body. So the dimensionless lengths (L_1, L_2) of both supports from CG are given as 0.4 and 0.6, respectively. In Table 1, the dimensionless properties and the base quantity are introduced as

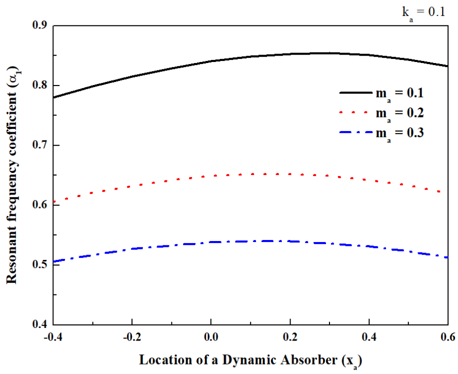
Table 1 Dimensionless properties

Items	Expression forms	Base quantity
masses	m, m_a,(0.1~0.3)	m
Inertia	J_G (= 0.1)	mL^2
force	f (= 1)	kL
Lengths	X, Y, L_1, L_2, Xa, L_f	$L=L_1+L_2=1$
stiffness	k_1, k_2, k_a	$k=k_1+k_2=1$
frequency	α	$\omega_n=(k/m)^{1/2}$

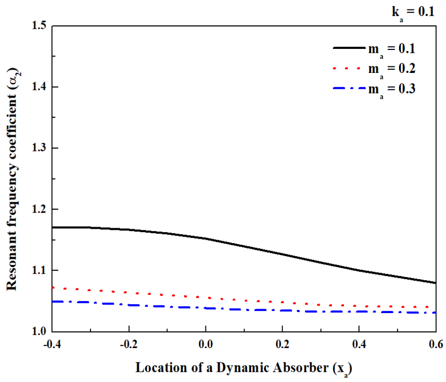
3. Resonant frequency coefficients (α)

In Eq.(4), the denominator part vanishes for certain specific values of the frequency ratio(α). The roots of denominator which are expressed in α are called the resonance frequency coefficient for the rigid body with a DA. For three mass ratios (m_a = 0.1, 0.2 and 0.3) of a DA stiffness ratio (k_a=0.1), the variations of three resonance frequency coefficients (α1, α2, α3) versus location of a DA (Xa) are plotted in Fig. 2 (a), (b) and (c), respectively. Based on the CG, the left direction is marked negative sign (0 ~ -0.4) and the positive sign (0 ~ 0.6) is given for the right direction. As the mass is increasing, the values of the coefficients are getting higher. But the differences of the value is getting smaller as be higher the order of the coefficient. It is confirmed that as the total mass is increasing, is getting lower the value of the resonant frequencies.

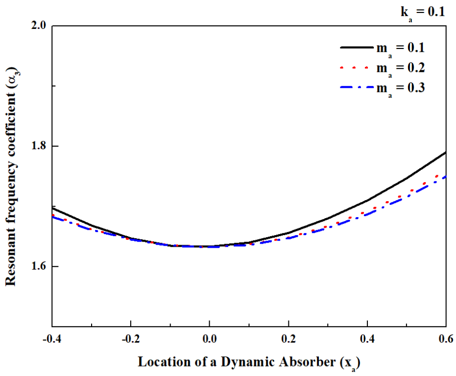
In Fig. 2 (a), (b) and (c), the variations of the three frequency coefficients are plotted versus the location of a DA for three stiffness ratios (ka = 0.1, 0.3 and 0.5) of a mass ratio (ma=0.2), respectively. It is noted that the higher stiffness produces the higher coefficient. The variation of each coefficient along the length of a body is found to show the similar trend for the same order of coefficient.



(a) α1

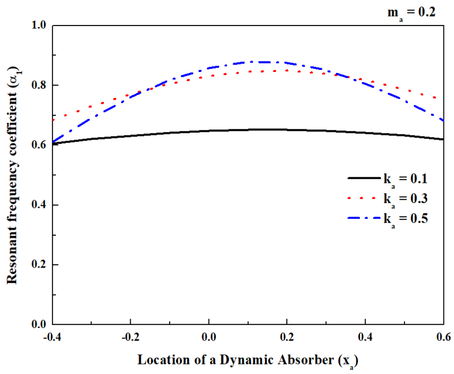


(b) α2

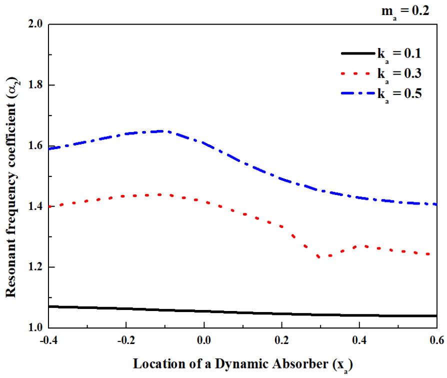


(c) α3

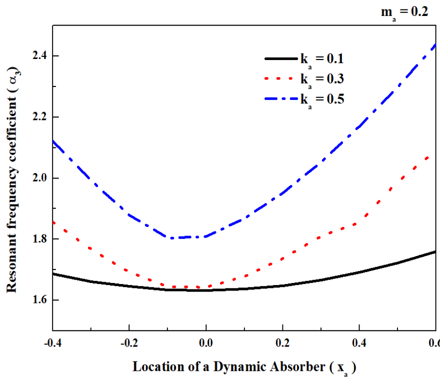
Fig. 1 Resonant frequency coefficients s Xa ka=0.1; L1=0.4; L2=0.6; JG=0.1; k1=0.5; k2=0.5 solid [ma=0.1], dot [ma=0.2], dash-dot [ma=0.3]



(a) α1



(b) α2



(c) α3

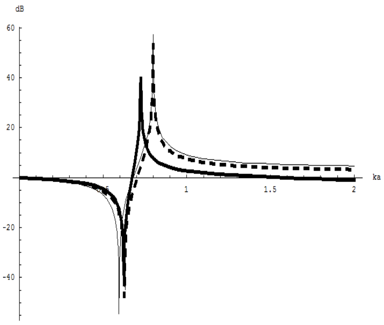
Fig. 2 Resonant frequency coefficients vs Xa ma=0.2; L1=0.4; L2=0.6; J=0.1; k1=0.5; k2=0.5 solid [ka=0.1], dot [ka=0.3], dash-dot [ka=0.5]

4. Control of Dynamic Characteristics

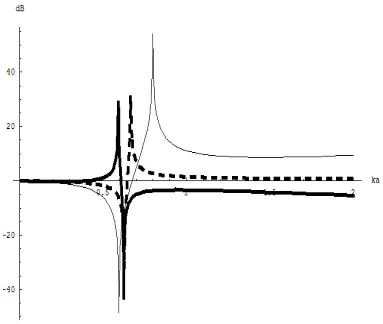
4.1 Reduction of amplitudes

In order to control passively the dynamic responses of a rigid body, In Eq. (4), two amplitudes for the bounce (X) and the rotation (Θ) are employed for the responses of a rigid body. The reduction level of the dynamic response (RD) is evaluated in comparison with two cases - a response without DA and a response with DA. The given external force ratio (f) is set to be a unity and the forcing frequency (α_f) is randomly entered. The ratio of a response without DA to one with DA is expressed in terms of decibel [dB] as follows;

$$RD[X] = 20Log_{10}(|X_{noDA}|/|X_{DA}|)$$
$$RD[\Theta] = 20Log_{10}(|\Theta_{noDA}|/|\Theta_{DA}|) \qquad (5)$$



(a) bouncing amplitude

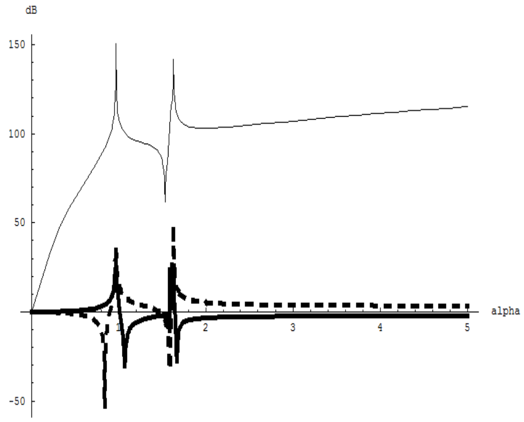


(b) rotating amplitude

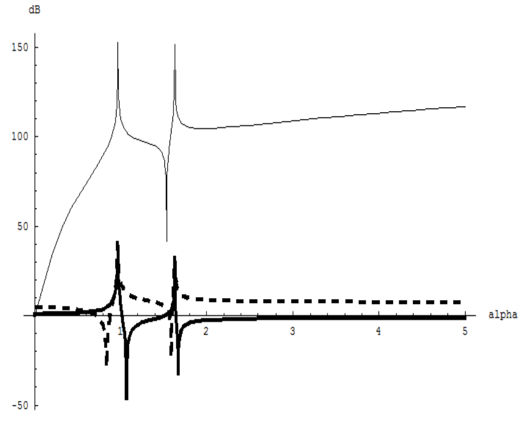
Fig. 3 Reduction of amplitude (RD) vs ka
ma=0.2; α_f = 2; Lf=0.1; J=0.1; thick solid[Xa=-0.1], thin solid[Xa=0.1], dot[Xa= 0.0]

In Fig.3, the reduction of two amplitudes (bounce and rotation) for ma = 0.2 and α_f = 2, are plotted along the DA stiffness ratio (ka) for three locations of a DA - Xa = -0.1 (the left side from CG), Xa = 0.0 (positioned at CG) and Xa = 1.0 (positioned at the location of the external force (Lf)).

In case of bouncing motion, the huge reduction is observed near or at the value of ka = 0.8 for three locations and otherwise the reduction is found to be small or worse.



(a) bouncing amplitude



(b) rotating amplitude

Fig. 4 Reduction of amplitude (RD) vs α
ma= 0.2; Xa = Lf = 0.1; J=0.1; thick solid [ka= 0.5k optimal], thin solid[ka= koptimal], dot[ka= 2koptimal]

The large reduction of rotating case occurs at the value of ka = 0.8 only for the case of Xa = 0.1 which is the same place of the external force. Here the value of ka = 0.8 matches the optimal stiffness of a DA being reported previously[11] such that the natural frequency of DA should be set equal to the forcing frequency of a main system. So the optimal value of the stiffness ratio (ka) can be given as;

$$k_a|_{optimal} \cong m_a \times \alpha_f^2 \qquad (6)$$

In Fig.4, the value of ka being estimated by Eq. (6) gives the large reduction level compared to those of other values of ka. It is proved that the optimal value of ka in Eq. (6) is still effective to control two motions of a rigid body (bounce and rotation).

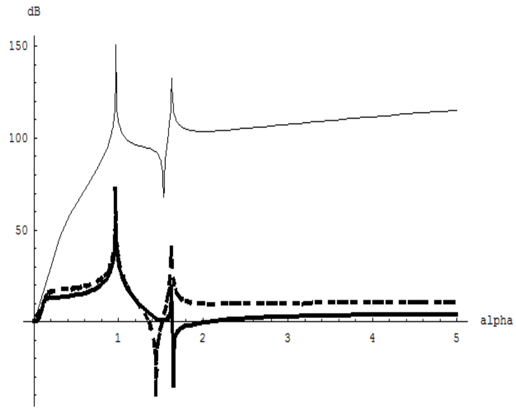
4.2 Reduction of transmitted power

The Vibrational energy which is induced by the harmonic motion of the external force is known to transmit to the main and the neighbour systems. In practice, a lot machines and frames are known to be troubled by excessive energy which is transmitted to the machine body and to the surrounding parts. Hence power which is the energy rated by time is evaluated and is expressed as follows;

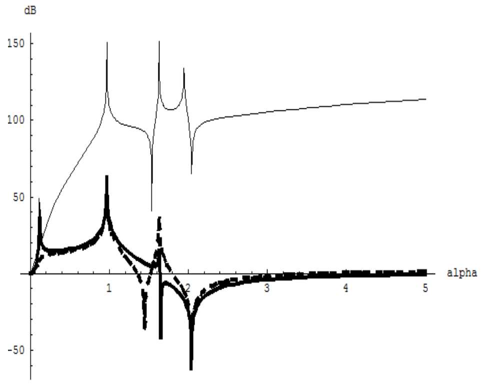
$$\begin{aligned} P[m] &= -\alpha^3 X^2 \\ P[L] &= \alpha * k_1 (X - L_1 \Theta)^2 \\ P[R] &= \alpha * k_2 (X + L_2 \Theta)^2 \end{aligned} \qquad (7)$$

where P[m], P[L] and P[R] represent the power be transmitted to a rigid body, to the left support and to the right support, respectively. The reduction level of power (RP) is expressed in terms of decibel [dB] as;

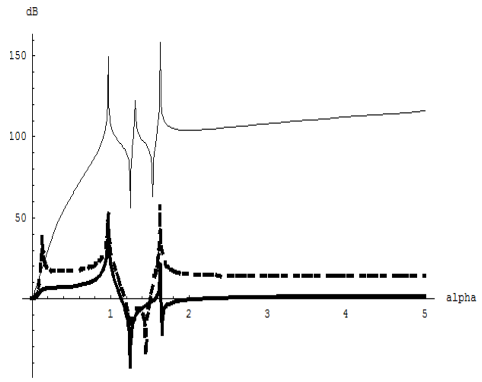
$$RP = 10 * Log_{10}\left(\frac{|P[\,]_{noDA}|}{|P[\,]_{DA}|}\right) \qquad (8)$$



(a) a rigid body



(b) a left support



(c) a right support

Fig. 5 Reduction of power (RP) vs α ma= 0.2; Lf=0.1; J=0.1; ka = Eq. (6) thick solid[Xa=-0.1], thin solid[Xa=0.1], dot[Xa= 0.2]

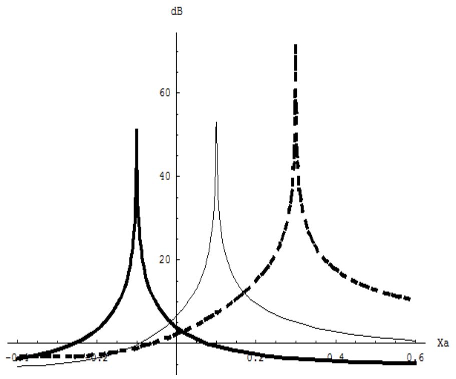


Fig. 6 Reduction of power to a body vs La
ma= 0.2; α_f = 2; J=0.1; ka = Eq.(6)
thick solid[Lf=-0.1], thin solid[Lf=0.1], dot[Lf= 0.3]

For three parts of a system (a rigid body, a left and right supports), Fig.5 shows the reductions of the transmitted power (RP) of three DA locations (Xa= -0.1, 0.1 and 0.2) versus the forcing frequency coefficient (α_f) for the optimal value of a DA stiffness ratio (ka=Eq. (6)). As shown in Fig.5, reduction level of the location of Xa = 0.1 is exclusively high compared to other locations. In Fig.6, the reductions of the transmitted power of a rigid body for three cases of a force location (Lf= -0.1, 0.1 and 0.3) are plotted against the location of a DA (Xa). It is observed that the powers be reduced largely near the same location of a DA and a external force. It is surely noted that the optimal location of a DA be near or at the place of an external force.

5. Conclusion

In this paper, the passive control of a vibrating machine with a Dynamic Absorber is studied to determine the optimal design parameters of a DA. On the bases of the analyses in this study, the conclusions are obtained as follows,

1) A Dynamic Absorber is observed to have the satisfactory results for the reduction of the dynamic characteristics, which are the dynamic responses and the power be transmitted to a rigid body and two supports.

2) The location of a DA should be positioned near or at the place of a external force for the control of two motions of a rigid body - bounce and rotation.

3) The variation of a mass of a DA might be a helpful tool to avoid the resonant phenomena by changing the range of the natural frequencies.

4) It is proved that the optimal stiffness of a DA which is determined by Eq. (6) is still conservative as the design parameter of a DA.

Acknowledgment

This paper was supported by Research Fund, Kumoh National Institute of Technology.

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