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# Optimum Design of Damped Dynamic Vibration Absorber – A Simulation Approach

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**Abstract**: Optimum design of Damped Dynamic Vibration Absorber (DDVA) presentsconflicting requirements. Mass ratio (μ), damping ratio (ξ) and tuning ratio (f) need to be optimally selected to realize satisfactory performance of modified system. Simulation provides valuable insights in dynamical behaviour and gives quantitative understanding of the dimensionless parameters mentioned aforesaid. Taguchi method parametrically optimizes to reach concrete design decisions.
**Keywords:** Dynamic Vibration Absorber, Optimum Design, Simulation

## I. INTRODUCTION

The *vibration absorber*[1], also called *dynamic vibration absorber (DVA),* is a mechanical device used to reduce or eliminate unwanted vibration. It consists of auxiliary system attached to the main system that needs to be protected from excessive vibration. Thus the main system and the attached absorber mass constitute a two-degree-of-freedom system; hence the modified system will have two natural frequencies.

The vibration absorber is commonly used in machinery that operates at constant speed, because it is tuned to one particular frequency and is effective only over a narrow band of frequencies. Common applications of the vibration absorber include reciprocating tools, such as sanders, saws, and compactors, large reciprocating internal combustion engines and pumps which run at constant speed (for minimum fuel consumption). In these systems, the vibration absorber helps balance the reciprocating forces. Without a vibration absorber, the unbalanced reciprocating forces may make the device impossible to hold or control.

A spring mass DVA [2] operates over narrow frequency range and suffers from performance deterioration with variation in excitation frequency. Performance robustness can be improved by introducing damping. A Damped Dynamic Vibration Absorber (DDVA) has expectedly much lower response than undamped DVA with broadened operational frequency range.

Design of DDVA possess formidable manufacturing difficulties. Making a DDVA suitable for wide operating range requires elaborate procedure.

## II. MATHEMATICALMODELING AND ANALYSIS OF DDVA

Addition of auxiliary system to main system renders it potentially a two degree of freedom system. Parametric model of the DDVA is presented inFig. 1 [2].

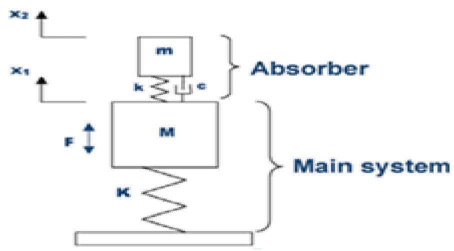


Fig. 1 Two DOF damped DVA model for Simulation

In the Fig. 1, an auxiliary system comprising of a small mass ‘m’ is attached to main system of mass ‘M’ so that vibrations of primary mass gets controlled.

**M, m** = masses of the primary and auxiliary systems in kg respectively.
**K, k** = spring stiffnessof the primary and auxiliary systems in N/m respectively.
**F** = excitation force in N
**$x_1$, $x_2$**= displacement of primary and auxiliary systems in meter.
**c** = damping in N-s/m

The equationsof motion for above system are as given below [3],

$$M\ddot{x}_1 + Kx_1 + k(x_1 - x_2) + c(\dot{x}_1 - \dot{x}_2) = F_0 \sin \omega t \quad (01)$$

$$m\ddot{x}_2 + k(x_2 - x_1) + c(\dot{x}_2 - \dot{x}_1) = 0 \quad (02)$$

For simplification we convert these equations to dimensionless form by using following symbols:

$\mu = m/M$ = mass ratio = absorber mass / main mass
$\omega_a^2 = k/m$ = natural frequency of absorber
$\omega_m^2 = K/M$ = natural frequency of main system
$f = \frac{\omega_a}{\omega_m}$= tuning ratio (natural frequencies)
$g = \frac{\omega}{\omega_m}$ = forced frequency frequency ratio
$x_{st} = \frac{P_0}{K}$ = static deflection of system
$c_c = 2m\omega_a$ = ‘critical’ damping

$\xi = \frac{c}{c_c}$ = damping ratio (03)

The main system response $x_1$ needs to be controlled as given by the Eq. (04),

$$\frac{x_1}{x_{st}} = \sqrt{\frac{\left(2\frac{c}{c_c}g\right)^2 + (g^2 - f^2)^2}{\left(2\frac{c}{c_c}g\right)^2 (g^2 - 1 + \mu g^2)^2 + [\mu f^2 g^2 - (g^2 - 1)(g^2 - f^2)]^2}} \quad (04)$$

The equation clearly indicates the parameters to be controlled viz. mass ratio (μ), damping ratio (ξ) and tuning ratio (f).

## III. SIMULATION BASED DESIGN OF DDVA

Whether we are interested in the behavior of an automotiveclutch system, the flutter of an airplane wing or the effect of the monetary supply on the economy, simulation helps to understand wide variety of real world phenomena. Simulation of DDVA is shown in Fig. 2 [4][5].

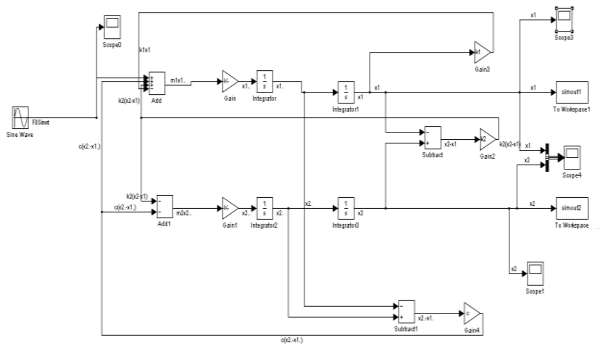

Fig. 2 Simulation of two DOF damped DVA model using Simulink

Simulation results are obtained for mass ratio (μ) varying from 0.05 to 0.2,damping ratio (ξ) from 0.0 to 0.5 and tuning ratio (f) from 0.7 to 1.0 (Refer Table 1).

TABLE 1

RESPONSE OF PRIMARY AND ABSORBER SYSTEM

| Sr. No. | μ (m/M) | f (□a/□m) | □=0.0 x1 | □=0.125 x1 | □=0.25 x1 | □=0.5 x1 |
|---|---|---|---|---|---|---|
| 1 | 0.05 | 0.7 | 28.5 | 30 | 30 | 31.5 |
| 2 | 0.1 | 0.7 | 27 | 27 | 28 | 32 |
| 3 | 0.15 | 0.7 | 26 | 28 | 25 | 30 |
| 4 | 0.2 | 0.7 | 23 | 21.5 | 24 | 30 |
| 5 | 0.05 | 0.8 | 39 | 40 | 40 | 45 |
| 6 | 0.1 | 0.8 | 32 | 35 | 37 | 45 |
| 7 | 0.15 | 0.8 | 27.5 | 30 | 33 | 43 |
| 8 | 0.2 | 0.8 | 24 | 27 | 30 | 41 |
| 9 | 0.05 | 0.9 | 46 | 50 | 67 | 87 |
| 10 | 0.1 | 0.9 | 34 | 40 | 52 | 80 |
| 11 | 0.15 | 0.9 | 28 | 32 | 43 | 73 |
| 12 | 0.2 | 0.9 | 25 | 28 | 33 | 61 |
| 13 | 0.05 | 1 | 45 | 61 | 95 | 160 |
| 14 | 0.1 | 1 | 32 | 38 | 50 | 80 |
| 15 | 0.15 | 1 | 26 | 30.5 | 39.5 | 53 |
| 16 | 0.2 | 1 | 22.5 | 26 | 30.5 | 45 |

## IV. PARAMETRIC OPTIMIZATION USING TAGUCHI METHOD

Taguchi method [5]is used for parametric optimization ofmass ratio (μ), damping ratio (ξ) and tuning ratio (f).

TABLE 2

MATRIX FOR PARAMETRIC OPTIMIZATION USING TAGUCHI METHOD

| Parameters | Values | | | |
|---|---|---|---|---|
| Frequency ratio f | 0.7 | 0.8 | 0.9 | 1.0 |
| Mass ratio μ | 0.05 | 0.10 | 0.15 | 0.20 |
| Damping ratio ξ | 0.0 | 0.125 | 0.25 | 0.5 |

After parametric optimization we get below optimized parameters: f=0.7 to 0.8, μ=0.2 and ξ=0.125.

## V. RESULTS AND DISCUSSIONS

Fig. 3 shows response of main system without auxiliary system. Resonance conditions at tuning frequency are clearly evident from it.

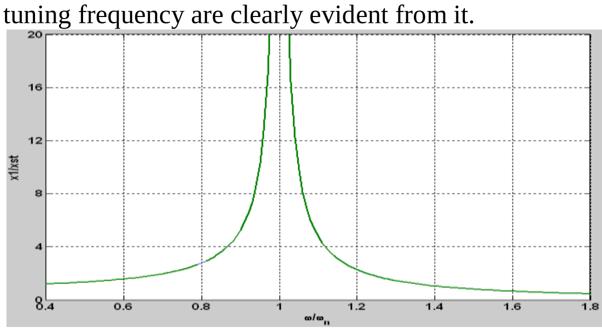


Fig. 3 Response of main system without auxiliary system

Response after addition of auxiliary system without damping for mass ratio (μ) = 0.2 and tuning ratio (f) = 0.7 to 0.8 is shown in Fig. 4.The modified system has two natural frequencies one above and one below the original natural frequency. The spread is controlled by the mass ratio (μ).

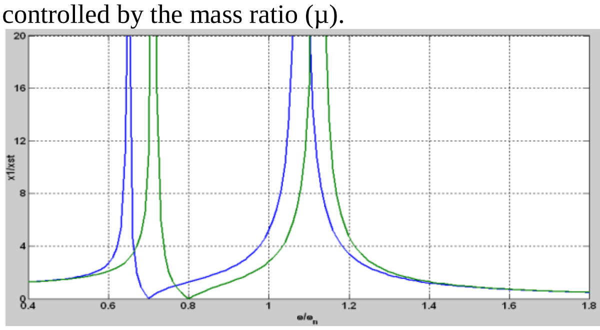


Fig. 4 Response of main system after addition of auxiliary mass

The larger absorber mass separates the two natural frequencies wider resulting in safe operating range. But larger absorber mass is highly impractical especially for large machinery. In such cases addition of damping provides better attenuation in a wide frequency range. With increment in the damping, system response further alleviates. (Refer Fig. 5).

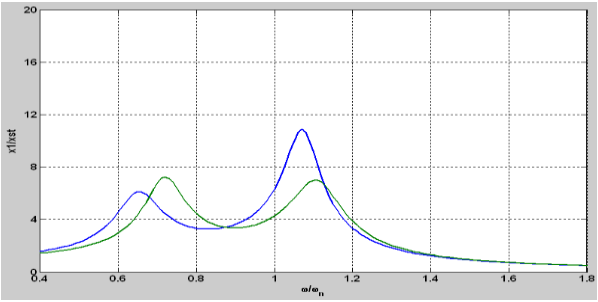


Fig. 5 Response of main system with damping

Fig. 6 shows the system response afterretaining mass ratio (μ) = 0.2, damping ratio (ξ) = 0.25 and tuning ratio (f) = 0.7 to 0.8.

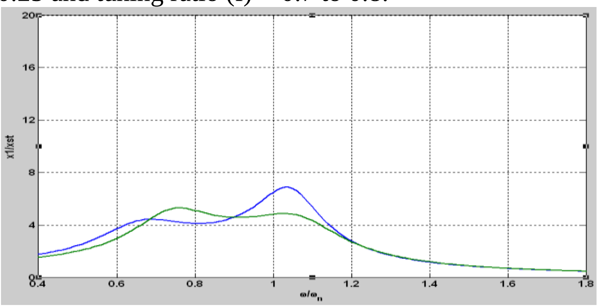


Fig. 6 Response of primary system for μ = 0.2, ξ = 0.25, f = 0.7 to 0.8

In Fig. 7, system response is shown considering mass ratio (μ) = 0.2, damping ratio (ξ) = 0.125 and tuning ratio (f) = 0.9 to 1.0.

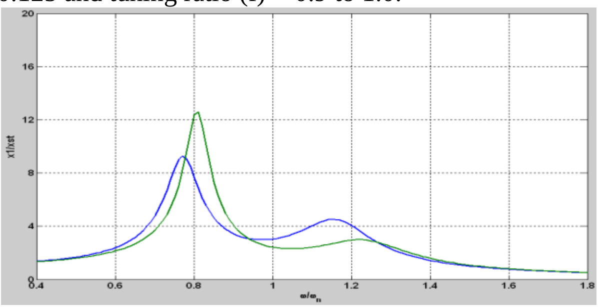


Fig. 7 Response of primary system for μ = 0.2, ξ = 0.125, f = 0.9 to 1.0

Comparison of results from Fig. 5 andFig. 7 helps in selecting tuning ratio for obtaining optimum results. Merely increase in the damping may reduce the response but it also narrows operating range.

## VI. CONCLUSION

For a dynamic damped vibration absorber, an increase in mass ratio results in diminishing response of main system. On the contrary, response of the absorber system increases with increase in damping. Similarly, increase in tuning ratio results in increased system response. So choice of design parameters needs optimization. By and large, low damping ratio and low tuning ratio can be preferred in the early part of design. However, tuning ratio should be high for high operating speeds. It is obvious that mass ratio is governed by practicalities.

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