IDENTIFICATION OF SYSTEM PARAMETERS USING TUNED VIBRATION ABSORBER FOR CAPTURING SYSTEM DYNAMICS

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ABSTRACT

For the systems, excited externally or internally the Tuned Vibration Absorber (TVA) are the passive device that absorb the vibration energy.Here we have focused on the effect of damping ratio and mass ratio of the TVA on the vibration characteristics of the combined system.The effect of alteration of attachment of the damping element of TVA on system dynamics and the effect of the primary system damping on combined system dynamics has been identified.The equation of motion are formed by using Lagrange's method and the dynamic response of the optimally tuned combined system are simulated in matlab®.

***Keywords**: Tuned Vibration Absorber, Damped TVA, Damped Primary system, Optimum Damping ratio, Mass ratio.*

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INTRODUCTION

The Tuned Vibration Absorber (TVA) is a widely used passive vibration control device. The vibration absorber is a mechanical device, consists mainly of a mass- spring-damper. The idea of the undamped TVA was pioneered by Watts [1] in 1883 and Frahm [2] in 1909. However an undamped TVA has a limited operating region and its performance deteriorates as the excitation frequency changes. Thus they find limited use, mainly for the constant frequency or for very narrow band. The damped TVA was presented by Ormondroyd and Den Hartog [3]. Den Hartog had found the optimum solution for the TVA parameters for classical primary system by using the "fixed-points" theory. Brock [4] derived an analytical solution for the optimum damping ratio for damped TVA. Ghosh and Basu [5]derived an approximate analytical solution for the optimum tuning parameter based on the assumption that "fixed-points" theory approximately holds, when a damped TVA attached to a lightly or moderately damped primary system. Liu and coppola [6] had given the optimal damping factor for the TVA whose damper connected to the ground. Here an attempt has been made to observe the effect of the optimal damping factor along with the system damping charecteristics on the dynamic response of the system.

DAMPED TVA AND DAMPED PRIMARY SYSTEM

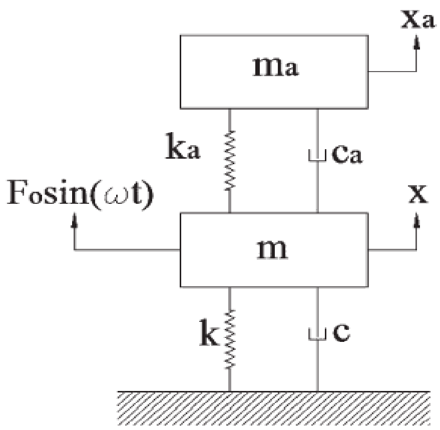

Figure 1: Damped primary system with damped vibration

The equations of motion are,

$$m\ddot{x} + (c - c_a)\dot{x} + (k - k_a)x = F_0 \sin \omega t \quad (1)$$

$$m_a\ddot{x}_a + c_a(\dot{x}_a - \dot{x}) + k_a(x_a - x) = 0 \quad (2)$$

For the classical primary system i.e. $\zeta p=0$ we can have optimum tuning parameter $\beta = 1/(1+\mu)$ and optimum damping of absorber [5] $\zeta a = \sqrt{\frac{3\mu}{8(1+\mu)}}$, both the parameters are in terms of frequency ratio μ.

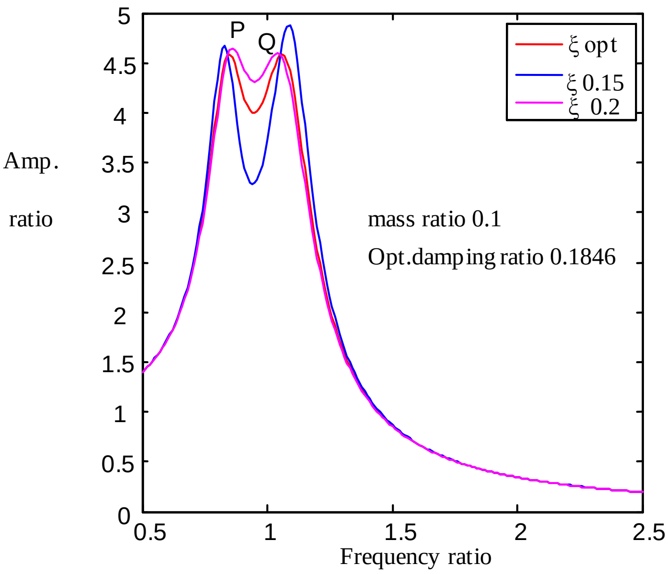

Figure 2: Response of combined system, mass ratio 0.1

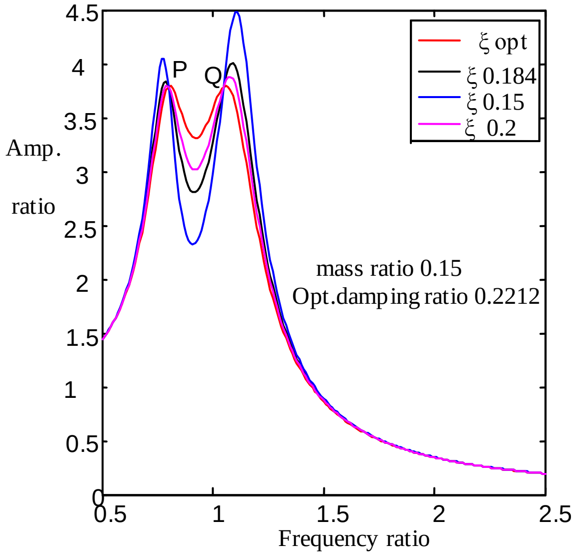

Figure 3: Response of combined system, mass ratio 0.15

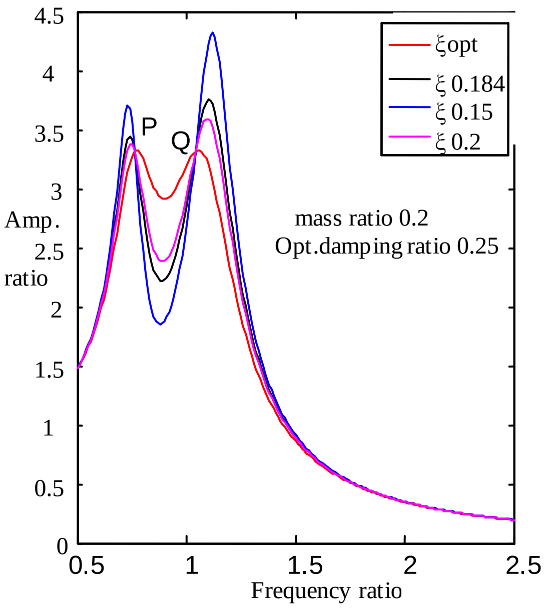

Figure 4: Response of combined system, mass ratio 0.2

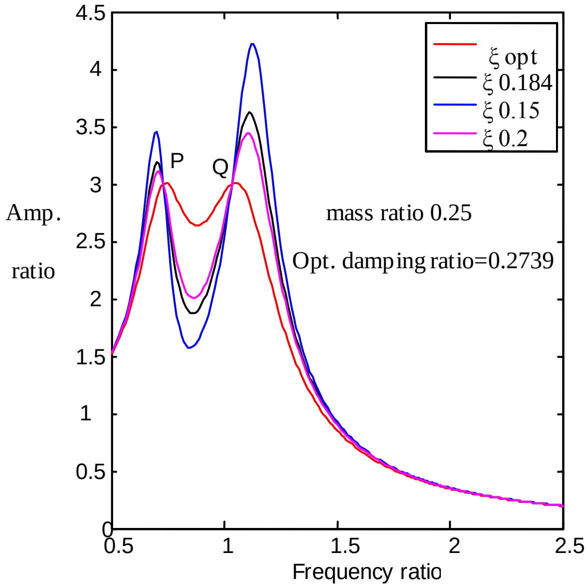

Figure 5: Response of combined system, mass ratio 0.25

DAMPED TVA AND DAMPED PRIMARY SYSTEM WITH ALTERATION OF DAMPING ELEMENT CONNECTION.

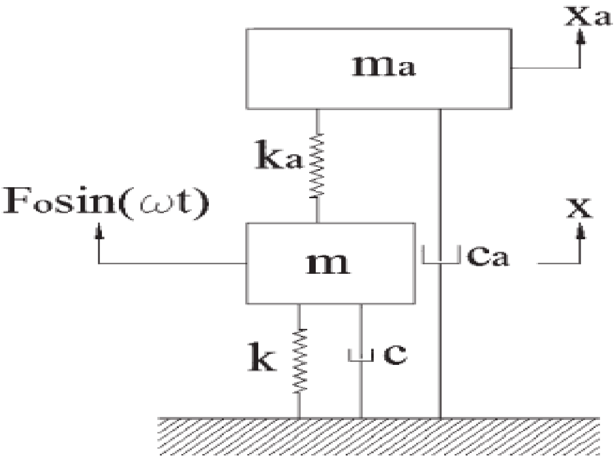

Figure 6: Damped primary system and damped TVA with damping element connected to ground.

The equation of motion is as follows

$$m\ddot{x} + c\dot{x} + (k - ka)x = F_0 \sin \omega t \qquad (3)$$

$$m_a\ddot{x}_a + c_a\dot{x}_a + k_a(x_a - x) = 0 \qquad (4)$$

by using this two equation the amplitude ratio equation is formulated. The optimum damping ratio[5] is $\zeta_a = 0.5\sqrt{\frac{3\mu}{(2-\mu)}}$ and The optimum tuning frequency ratio[5] is $\beta = \sqrt{\frac{1-4\zeta p^2}{1-\mu}}$.

The effect of mass ratio and also the effect of variation of the primary system damping factor are simulated as follows,

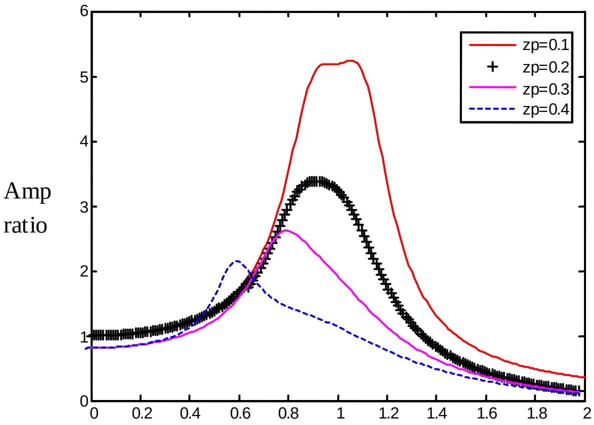

Figure 7: System response for mass ratio μ=0.05

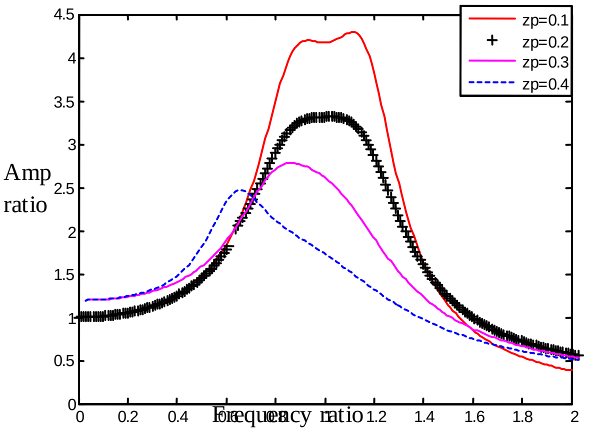

Figure 8: System response for mass ratio μ=0.1

z_p = Damping factor of primary system

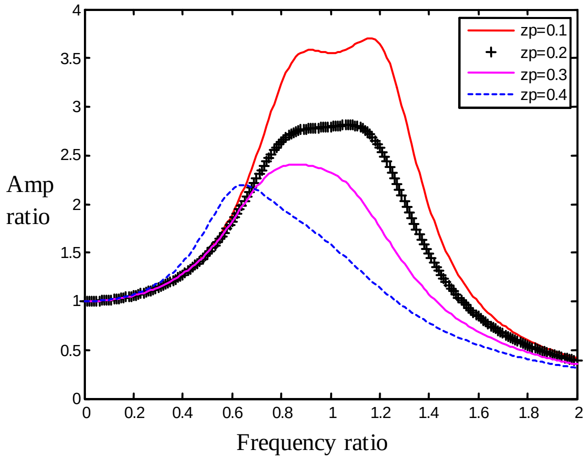

Figure 9: System response for mass ratio μ=0.15

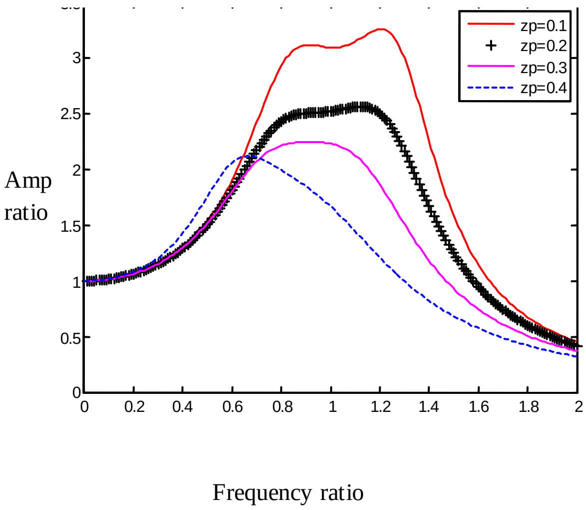

Figure10: System response for mass ratio μ=0.2

Here primary system damping (zp) has been changed from 0.1 to 0.4 and their effect on the primary system response has been seen we find appreciable amplitude reduction with the higher primary system damping. With the mass ratio 0.25 the following is the result.

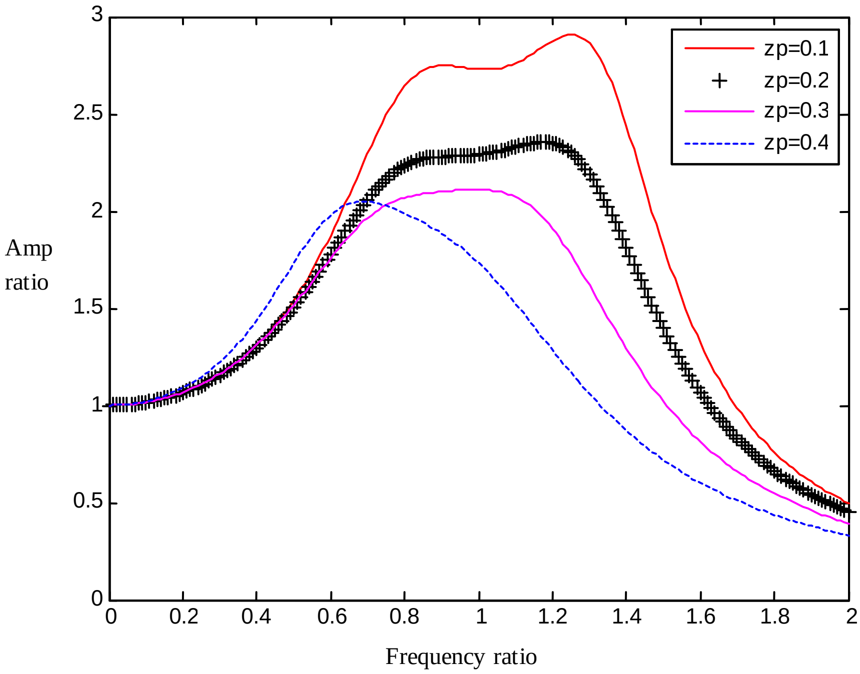

Figure 11: System response for mass ratio μ=0.25

CONCLUSION

For the model ‘A’ as shown in Figures 2 to 5, as the mass ratio increases amplitude of vibration decreases. As mass ratio increases the optimum damping also increases. One observation we can made from the FRF’s is the response curve becomes flatter as the mass ratio increases. Thus with the higher mass ratio system response becomes same as of single degree of freedom system response. For the model ‘B’ as shown in Figure 7 to 11, as the mass ratio increases the amplitude ratio decreases and as primary system damping factor increases the system resonance response moving towards the lower frequency ratio.

FUTURE SCOPE

The theoretical results obtained here are to be verified practically. The practical work are undergoing for the same.

REFRENCES

1. Watts, P., ‘‘On a method of reducing the rolling of ship at sea,’’ Transactions of the Institute of Naval Architects, Vol. 24, 165–190 (1883).
2. Frahm, H., ‘‘Device for damping vibrations of bodies,’’ U.S. Patent No. 989958, 1909.
3. Ormondroyd, J. and Den Hartog, J.P., ‘‘Theory of the dynamic vibration absorber,’’Transactions of the American Society of Mechanical Engineers, Vol. 50, 9–22(1928).

4. Brock, J. E., ‘‘A note on the damped vibration absorber,’’ Journal of Applied Mechanics, Vol. 68, A-284(1946).
5. Ghosh, A. and Basu, B., ‘‘A closed-form optimal tuning criterion for TMD in damped structures,’’ Structural Control and Health Monitoring, Vol. 14, 681–692(2007).
6. Liu,K., and Coppola,G., “Optimal design of damped dynamic vibration absorber for damped primary systems,” Canadian society for mechanical engineering, 34, 119-135 (2010).