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Tuning Condition Modification of Damped and Un-damped Adaptive Vibration Absorber

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Abstract:

In this paper a tuned vibration absorber (TVA) is realized using different tuning conditions. The main objective of adaptive tuned vibration absorber is to maximize the vibration attenuation of main system vibration. The tuning condition is used to track the excitation frequency. By making some modification on tuning condition the frequency response of primary system (FRC) will have better performance. This paper shows that the FRC can be reduced by 10%. This enhancement is dependent on system parameters in addition to excitation frequency.

Keywords — **Adaptive tuned vibration absorber, control of undesirable vibrations, tuning condition**.

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I. INTRODUCTION

The most common type of tuned vibration absorbers TVAs are: (1) Passive; (2) Semi-active and (3) Active, are widely used in industries to suppress undesired vibrations of some machines excited by harmonic forces. However, ,the TVA is only effective over a considerably narrow frequency range.

Semi-active or adaptive tuned vibration absorbers ATVA are used in case of exciting frequency and system characteristics uncertainty. In which the absorber's frequency tuned on-line control. As reactive forces are used, semi-active absorbers devices consume less power than active absorbers. Moreover, ATVA can behave as passive devices in case of loss of power.

ATVA have two important topics to discuss: (1) How the ATVA varys its natural frequency which mean the adaptive absorber has adapted mechanism to change parameters (stiffness, damping…etc.) (2) The tuning conditions and tuning strategies are used to adapt the frequency of ATVA which maximize attenuation of primary system vibration.

Franchek (1995) designed an adaptive-passive vibration absorber using a variable spring as the adaptive component. The stiffness was controlled using a spring inserted through a sliding plate, which could then be moved to alter the effective number of coils in the spring [4]. Nagaya (1999) discussed a method of vibration control of a structure by using a variable stiffness vibration absorber [3]. Brennan (2005) investigated different strategies for tunable stiffness [5].

Jalili (2002) proposed a novel ATVA which vary the mass distribution by mechanical mechanisms [2]. Mirsanei (2012) developed a new design for adaptive tuned dynamic vibration absorber (ATDVA) based on smart slider-crank mechanism to control of undesirable vibrations. The smart tuned mass damper (STMD) is used also in the building field. Yalla (2001) and Nagarajaiah (2005) developed a STMD capable of varying its stiffness and retuning its frequency due to real time control continuously [13,14].

Many researchers used non-traditional ATVA with smart materials. Rustighi (2005) developed a continuously tunable device by using shape memory alloy (SMA) elements [8]. The elastic modulus of SMA changed with adjusted temperature. Gong et. al. (2012),used an electro-rheological fluid (ERF) as an adaptive material which changes its material properties quickly and reversibly in relation to an electric current [9]. This behaviour was discovered by Winslow in 1947 [1]. Eroglu and Neil (2014) presented the development of an active-damping-compensated magnetorheological elastomer (MRE) adaptive tuned vibration absorber (ATVA) [15]. Nicklas (2014) investigated an adaptive multi-body absorber prototype filled with Electro-rheological fluid (ERF) experimentally and a numerical model was validated using measurements [16].

Adaptive-passive absorber must be tuned by matching its natural frequency ω_a to the excitation frequencyω. Hollkamp (1994) developed a global search algorithm. This approach looks for the absorber frequency that produces the maximum suppression of the main system vibration. This algorithm will automatically tune a vibration absorber to a time invariant excitation frequency but will not working well when the excitation frequency continuously changes [17]. .Franchek

(1995) modified the global tuning strategy which is based on minimizing the voltage amplitude from an accelerometer. It is a robust and insensitive tuning strategy [4]. Mianzo (1992) used another type of vibration control. That is the open loop control of an adaptive absorber in which the natural frequency of the ATVA and the excitation frequency must be known precisely at all times [10].

In this paper, the tuning condition of ATVA will be discussed and developed to minimize the amplitude of the primary system as possible. The most common tuning condition can be achieved by matching the ATVA natural frequency ω_a with the excitation frequency ω for harmonic excitation [5-8]. It is not the optimum tuning condition. Some modifications should be made to have better performance of the ATVA. Then the tuning condition after modification will be applied on the pendulum like ATVA. It is supposed to be an adaptive device that can enhance the vibration attenuation capacity by adapting the pendulum natural frequency. Thus, the natural frequency of the pendulum-like ATVA can be controlled by tuning the position of the sliders to trace the external excitation frequency. The novelty is in the simple configuration of the device.

II. FORMULATION OF MOTION EQUATIONS

2-1 Mathematical modeling of linear absorber

The mathematical model for the vibrating system appended with a dynamic vibration absorber is shown in fig. 1. It shown as a two degree of freedom lumped parameter model. The main system mass, stiffness and damping are denoted by m_a, k_p and c_p the absorber mass, stiffness and damping are denoted by m_a, k_a and c_a. The main system excitation is a harmonic forcing of amplitude F and frequency ω. The governing differential equations for the systems are

$$\begin{bmatrix} m_p & 0 \\ 0 & m_a \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} c_a + c_p & -c_a \\ -c_a & c_a \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} k_p + k_a & -k_a \\ -k_a & k_a \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} Fe^{j\omega t} \\ 0 \end{bmatrix} \quad (1)$$

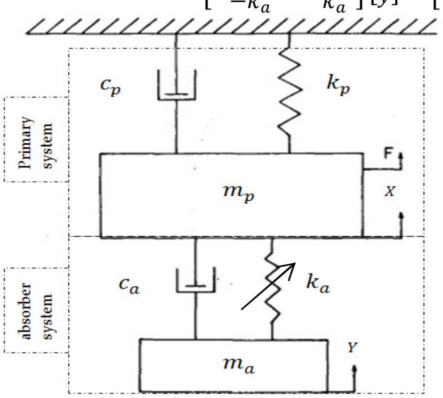

Fig.1 Model A of linear vibration absorber.

The steady state response of the two masses after the transients have vanished is given by the equations:

$$\begin{bmatrix} \alpha \\ \gamma \end{bmatrix} = \begin{bmatrix} 1 + \frac{\mu}{T^2} - \Omega^2 + 2\left(\zeta_p + \frac{\zeta_a}{T}\mu\right)\Omega j & -\frac{\mu}{T^2} - \frac{2\zeta_a}{T}\mu\Omega j \\ -\frac{\mu}{T^2} - \frac{2\zeta_a}{T}\mu\Omega j & \frac{\mu}{T^2} - \mu\Omega^2 + \frac{2\zeta_a}{T}\mu\Omega j \end{bmatrix}^{-1} \begin{bmatrix} 1 \\ 0 \end{bmatrix} \quad (2)$$

Where

$$\alpha = \frac{X}{F/k_p}, \quad \zeta_p = \frac{c_p}{2m_p w_p}, \quad w_p = \sqrt{\frac{k_p}{m_p}},$$

$$\gamma = \frac{Y}{F/K_p}, \quad \zeta_a = \frac{c_a}{2m_a w_a}, \quad w_a = \sqrt{\frac{k_a}{m_a}}, \quad (3)$$

$$\Omega = \frac{\omega}{\omega_p}, \quad \mu = \frac{m_a}{m_p}, \quad T = \frac{\omega_p}{\omega_a},$$

This parameter called normalized parameters where α is the normalized vibration amplitude of the primary mass and γ is the normalized vibration amplitude of the absorber mass.

2-2 Mathematical modeling of pendulum absorber

The mathematical model for SDOF primary system with the proposed pendulum-like ATVA is described in Fig. 2. The pendulum-like ATVA can be considered as one dynamic mass swaying around pendulum axis. For simplification, the mass of pendulum arm is massless with respect to dynamic mass. The governing differential equations for the systems are

$$\begin{bmatrix} m_a + m_p & m_a l \\ m_a l & m_a l^2 \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{\theta} \end{bmatrix} + \begin{bmatrix} c_p & 0 \\ 0 & c_a \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} k_p & 0 \\ 0 & m_a g l \end{bmatrix} \begin{bmatrix} x \\ \theta \end{bmatrix} = \begin{bmatrix} Fe^{j\omega t} \\ 0 \end{bmatrix} \quad (4)$$

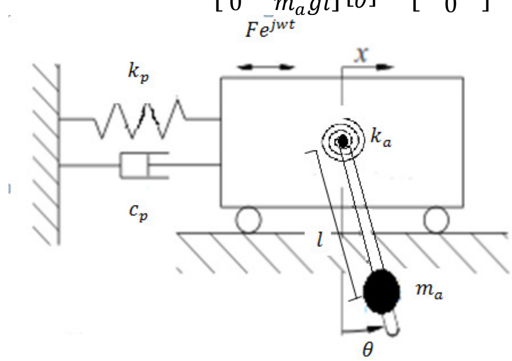

Fig. 2 Model B pendulum vibration absorber.

The steady state solution of Eq. 4 is:

$$x = Xe^{j\omega t} \quad (5)$$

$$\theta = \Theta e^{j\omega t} \quad (6)$$

Taking the following non-dimensional variables:

$$\alpha = \frac{X}{F/k_p}, \quad \zeta_p = \frac{c_p}{2m_p w_p}, \quad w_p = \sqrt{\frac{k_p}{m_p}},$$
$$\gamma = \frac{\theta l}{F/K_p}, \quad \zeta_a = \frac{c_a}{2m_a l^2 w_a}, \quad w_a = \sqrt{\left(\frac{m_a g l + k_a}{m_a l^2}\right)}, \quad (7)$$
$$\Omega = \frac{\omega}{\omega_p}, \quad \mu = \frac{m_a}{m_p}, \quad T = \frac{\omega_p}{\omega_a},$$

from Eqs. (5), (6) and (7), we substitute into Eq. (4) yields

$$\begin{bmatrix}\alpha\\ \gamma\end{bmatrix} = \begin{bmatrix} -(1+\mu)\Omega^2 + 2\zeta_p j\Omega + 1 & -\mu\Omega^2 \\ -T^2\Omega^2 & -T^2\Omega^2 + 2\zeta_a Tj\Omega + 1 \end{bmatrix}^{-1} \begin{bmatrix}1\\0\end{bmatrix} \quad (8)$$

III. TUNING CONDITION OF DAMPED AND UN-DAMPED MODELS

3.1 Tuning of un-damped ATVA

The FRC of the main system at μ=0.1 and ζ_p=0.1 (standard system) with un-damped absorber is illustrated in Figure 2. Both models A and B have the same FRC. It is show the best tuning condition for maximum vibration attenuation is:

$$T * \Omega = 1 \quad (9)$$

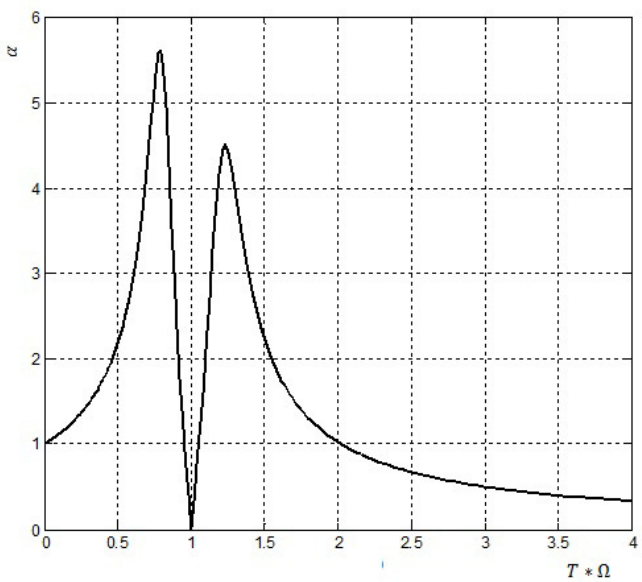

Fig.3 FRC of the primary system for different tuning condition

Unfortunately, this performance is realized at one frequency only and is extremely sensitive to proper tuning due to the narrow frequency band between the resonant peaks. Consequently, the performance of a vibration absorber is sensitive to variations in both normalized frequencies $\Omega and T$ around tuning condition Eq.9.

Design example:

Consider a system with the following characteristics: μ=0.1, ζ_p=0.1 and the normalized excitation frequency $\Omega = 1.2$, then search for the best choice of T to tune the ATVA at maximum main system attenuation. Fig. 4 shows the FRC of the main system for different T. the Fig. 3 have only one minimum point at T=0.833 which is the best tuning point when $T * \Omega \cong 1$. Also there is great sensitivity of α around tuning condition at T=0.833.

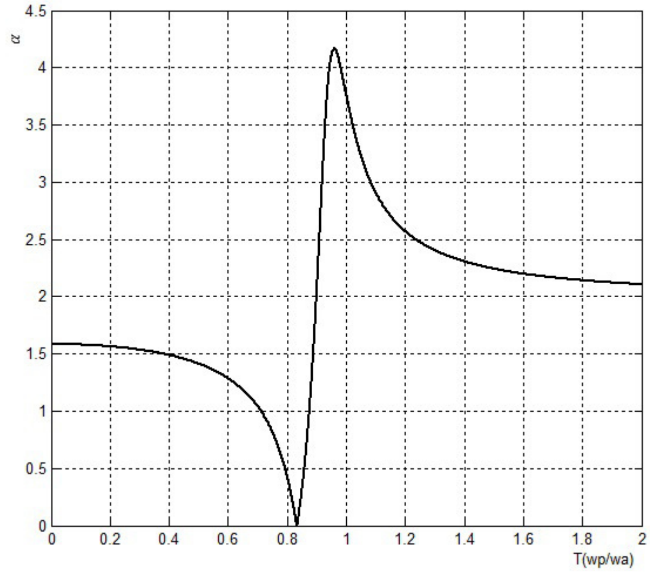

Fig.4 $\alpha vs T$ relationship of un-damped ATVA at $\Omega = 1.2$

3.2 Tuning of damped ATVA

When the natural frequency of the absorber matches the excitation frequency, the displacement amplitude of the main system is completely attenuated but when the absorber is un-damped. Absorber damping ζ_a effects on the tuning condition Eqs (9). Consider a system with the following characteristics: μ=0.1, ζ_p=0.1 and the normalized excitation frequency $\Omega = 1.2$. Fig. 5 shows that $T * \Omega \neq 1$ for different ζ_a. In other words, we can say $T * \Omega = f$ where tuning factor $f = 1$ for un-damped ATVA and $f \neq 1$ for damped ATVA. The modified tuning condition is:

$$T * \Omega = f \quad (10)$$

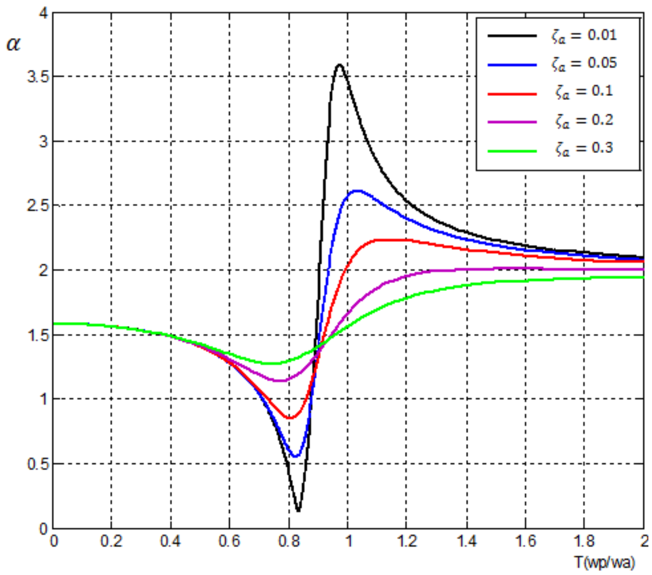

Fig.5 $\alpha vs T$ relationship of damped PDVA at $\Omega = 1.2$

Also the FRC of the ATVA increase dramaticallyaround exitation frequency $\omega = 1.2$ during the ATVA damping decreased. Fig. 6 show that.

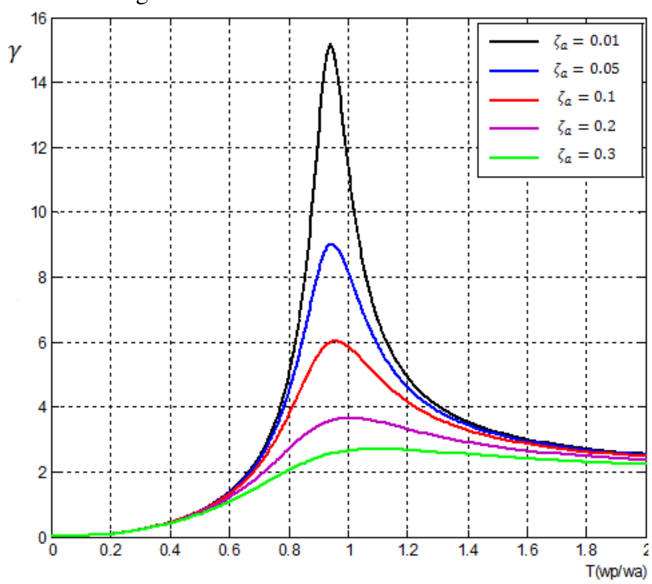

Fig.6 $\gamma vs T$ relationship of damped PDVA at $\Omega = 1.2$

Tuning factor f

The tuning factor f doesn't have unit value as is customary. In general, it is dependent on four parameters: excitation frequency Ω, ATVA damping ζ_a, primary system damping ζ_p and mass ratio μ. For standard primary system f should be greater than one for $\Omega < 1$ and less than one for $\Omega > 1$ as shown in the Fig. 7. Moreover f has direct proportional deviation from unity with ATVA damping.

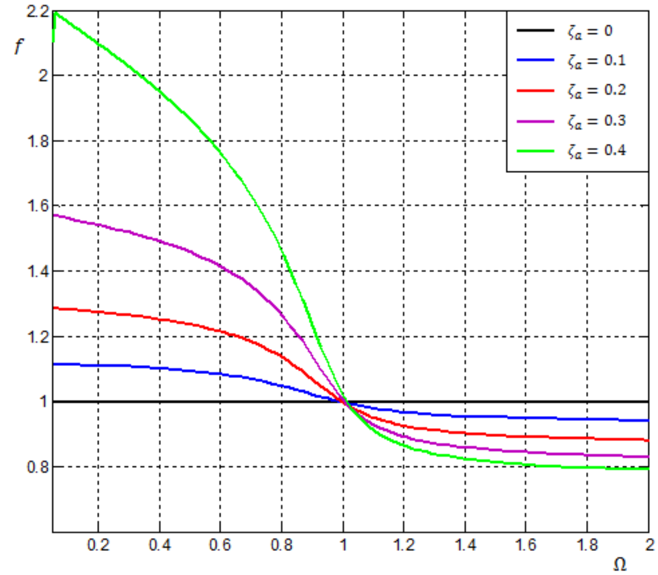

Fig.7 Tuning factor $f vs \Omega$ for primary system at μ=0.1 and ζ_p=0.1

Applying tuning factor to ATVA must be safe more attenuation for the main system vibration. Fig. 7 shows the possible minimum α with tuned ATVA for excitation frequencies from 0 to 2 at different ATVA damping. It is necessary to look at the frequency response of the absorber Fig. 8 shows the FRC of tuned ATVA corresponds to tuning factor in Fig. 6.

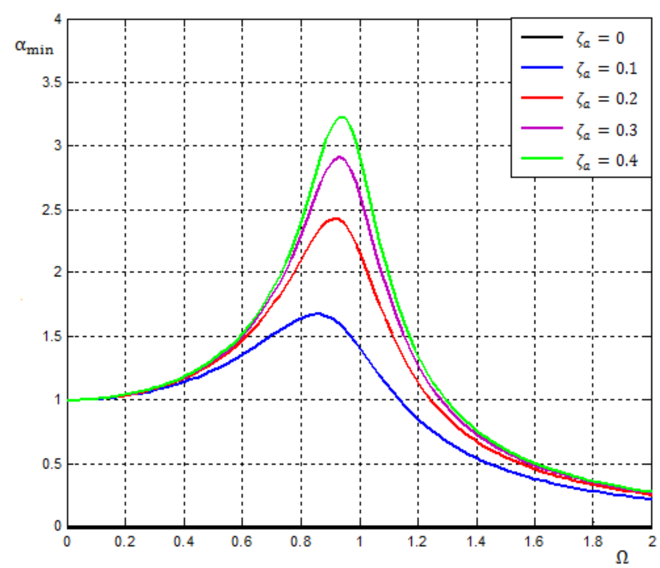

Fig. 8 FRC of primary system with tuned PDVA at μ=0.1 and ζ_p=0.1

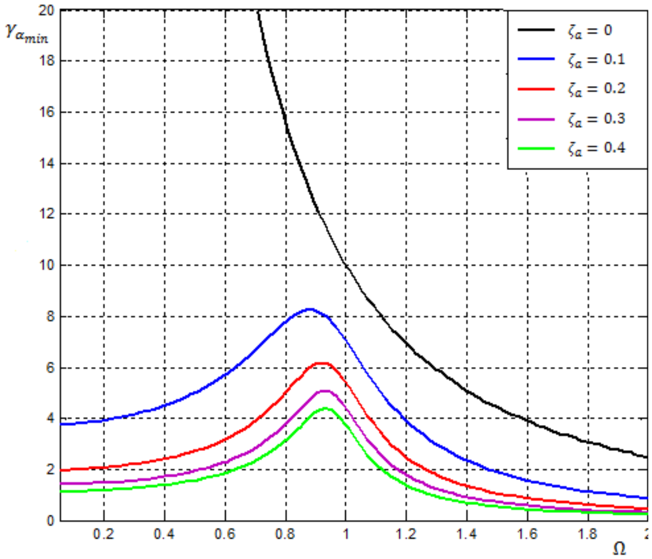

Fig.9 FRC of ATVA at μ=0.1 and ζ_p=0.1

IV. ADAPTIVE PENDULUM VIBRATION ABSORBER

The mass distribution can be changed by adjusting the position of the sliding blocks at the pendulum axis.
Thus, the natural frequency of the pendulum-like ATVA can be controlled by tuning the pendulum arm to trace the external excitation frequency. When the tuned pendulum-like ATVA matches the excitation frequency, the vibration can be attenuated significantly. This point will be theoretically addressed in the following sections.

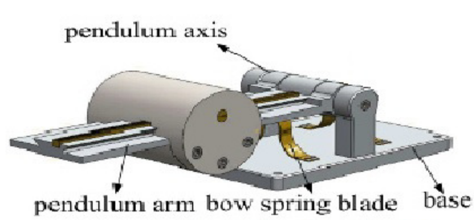

Fig.10 Model B of single pendulum [9].

By taking the previous non-dimensional variables:
Apply this tuning condition on un-damped model B the length tuning condition. From Eqs.7 and 9, the tuned length is given by:

$$l_{tun} = \frac{g + \sqrt{\left(g^2 + \frac{4w^2k_a}{m_a}\right)}}{2\omega^2} \tag{11}$$

In general, the length of the pendulum$l$ in damped model B can be adjusted to achieve the previous tuning condition.

$$l_{tun} = \frac{gf^2 + \sqrt{\left(g^2f^4 + \frac{4w^2f^2k_a}{m_a}\right)}}{2\omega^2} \tag{12}$$

V. CONCLUSIONS

The ATVA performance of vibration attenuation reduces as long as the absorber damping increased. For example, $\alpha_{min}$is 0.1338 at $\zeta_a = 0.01$ and become 1.28 at $\zeta_a = 0.3$ as shown in Table 1. The goal is to maximize the attenuation performance that means we should reduce absorber damping as possible but we must consider the sharp boost of γ as long as absorber damping decreases as shown in Fig. 6.

Table 1 Damped ATVA performance for μ=0.1 and ζ_p=0.1 at Ω = 1.2

ζ_a	α_{min}	$\gamma_{\alpha_{min}}$
0.01	0.1338	6.6795
0.05	0.5476	5.2467
0.1	0.8513	3.9052
0.2	1.1431	2.4640
0.3	1.2786	1.7742

Fig. 11 shows the effect of using the modified tuning condition (Eq. 10). The minimum possible amplitude of primary system is obtained by using the tuning condition.

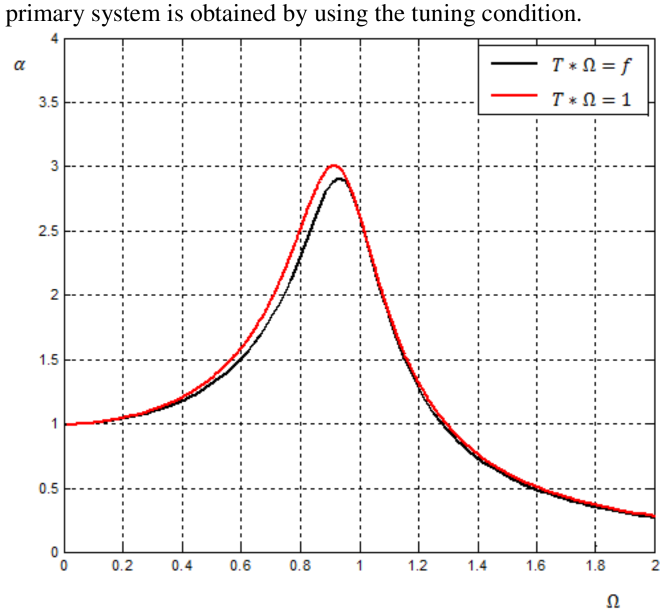

Fig.11 The FRC of primary system at μ=0.1, $\boldsymbol{\zeta_p}$=0.1 and $\boldsymbol{\zeta_a = 0.3}$ for two types of tuning condition.

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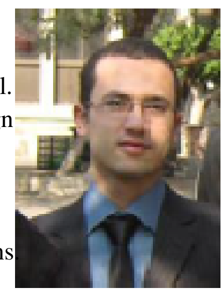
- Reviewer of a number of international journals.



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