Amirkabir Journal of Civil Engineering

Amirkabir J. Civil Eng., 51(5) (2019)303-306
DOI: 10.22060/ceej.2018.14219.5590

Semi-active fuzzy control of SDOF systems under loading of rotary machines by tuned mass dampers

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ABSTRACT: Dynamic vibrations of mechanical equipment might undesirably affect their performance and the structures on which they are installed. The gradual increase in angular velocity of such equipment and getting close to natural frequency of the structure leads to some phenomena such as resonance, pseudo resonance, beating and pseudo beating phenomenon. Therefore, dynamic responses of these structures should be reduced. Tuned mass dampers (TMDs) as one of the most reliable and simplest instruments to achieve this goal have been attracted by experts. The inertia force makes this kind of dampers vibrate in opposite direction and cause reduction in response of structure. Within seconds, by dedicating various parameters for TMD, their great performance can be augmented. There are lots of different strategies to assign these variable parameters. In this study, semi-active control approaches have been used to decrease the response of a single degree of freedom structure subjected to the above-mentioned probabilistic phenomena. In addition, some existing optimum functions have been applied to determine the TMD's frequency and damping parameters. These parameters of semi-active TMD are predicted utilizing two different strategies: the fuzzy logic system and ground-hook algorithm. The logic of making alteration to damping ratio is based on regaining the equilibrium of structure as it vibrates. Copping with different phenomena, results of this investigation indicate the advantages of using semi-active tuned mass damper to dramatically decrease the system displacement by 32 to 47 percent. Moreover, using fuzzy logic systems to set damping parameters of TMD, results in 1.5 to 6.2 percent displacement reduction in comparison with Ground-Hook algorithm. The conducted analysis for a wide range of optimal frequencies illustrate that fuzzy logic system is less sensitive to mistuning of TMD's optimal frequency.

Review History:

Received: 3/16/2018
Revised: 7/20/2018
Accepted: 8/9/2018
Available Online: 8/20/2018

Keywords:

Semi-active tuned mass damper
Fuzzy logic and Ground-Hook algorithm
Rotary machines
Resonance and Pseudo Resonance
Beating and Pseudo Beating

1. INTRODUCTION

Different kinds of machineries are used in industry that by paying attention to their special usages, they can apply the different kind of dynamic loads to the degrees of freedom that they are connected to them. Therefore, foundation of these structures in addition to static loads must be resistant to dynamic loads. By paying attention to unhealthy effects of vibrations on human, decreasing response amplitude of these structures seems vital too [1].

Tuned Mass Damper (TMD) is one of the most reliable and the simplest tools used for controlling the vibrations of new and old structures [2-4]. One of the important challenges of designing these tuned mass dampers is optimal tuning of their parameters such as stiffness and damping ratio. Considering the existence of uncertainty in loading, these parameters can keep away from optimal situation and cause reduction in damper's performance [5-7].

In this study, in order to decrease the sensitivity of tuning TMD parameters in passive situation, semi-active tuned mass damper is used with ground-hook and fuzzy controller algorithm for controlling the vibrations of a single degree of freedom system under phenomena such as resonance, pseudo resonance, beating and pseudo beating. In this paper, using previous research on vibration control of structures, the objective is to decrease vibration of industry machineries having rotary engines as the indicator of dynamic loads created by phenomena such as resonance, pseudo resonance, beating and pseudo beating. By clearing and applying these systems in smaller scales, such as washing machines in which rotary movement causes vibrations in body and driving system of bars, they can prevent undesirable and destructive vibration in large extent and intensify their beneficial life too.

2. RESPONSE OF SINGLE DEGREE OF FREEDOM SYSTEMS TO THE ROTARY MACHINE EXCITATIONS

Sinusoidal loading is one of the simplest periodic loading called simple harmonic loads. This kind of load can be created with existence of the mass that is eccentric in a rotary machine.

In Fig. 1, a simple beam is shown with a rotary machine in

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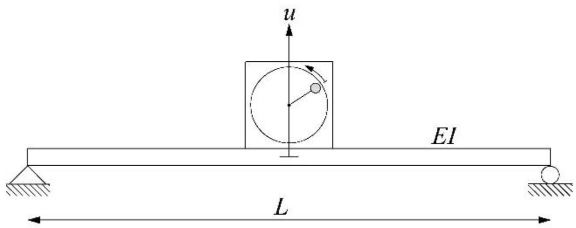

Fig. 1. Rotary machine in the middle of a simple beam

Table 1. Comparison of criteria for different control strategies (Note: less value indicates better performance)

Phenomena		J_1	J_2	J_3	J_4	J_5	J_6
Resonance	Passive	0.768	0.767	0.767	0.809	0.807	0.807
	On-Off	0.305	0.304	0.303	0.220	0.219	0.219
	FLC	0.289	0.287	0.288	0.194	0.194	0.194
Pseudo resonance	Passive	0.843	0.834	0.825	0.891	0.884	0.876
	On-Off	0.444	0.411	0.397	0.614	0.615	0.629
	FLC	0.429	0.393	0.381	0.590	0.591	0.606
Beating	Passive	0.747	0.746	0.745	0.799	0.800	0.801
	On-Off	0.463	0.472	0.480	0.739	0.742	0.745
	FLC	0.401	0.408	0.420	0.668	0.671	0.673
Pseudo beating	Passive	0.720	0.719	0.719	0.649	0.657	0.667
	On-Off	0.425	0.435	0.444	0.549	0.564	0.577
	FLC	0.380	0.389	0.402	0.534	0.547	0.560

the middle of its span. While there is a machine with mass m, in the middle of simple beam's span with length L, moment of inertia, I and modulus of elasticity E, by assuming eccentric mass m' and the gap from being out of the center is e, while the eccentric mass rotates with Ωt angle, the applied load is equal to:

$$P(t) = P_0 \sin(\Omega t), \quad P_0 = m' e_0 \Omega^2 \tag{1}$$

3. NUMERICAL STUDIES

In this study, single degree of freedom system with mass m, and stiffness k are used as the representative of rotary systems under external harmonic loading.

3.1. Resonance

When the loading frequency (Ω) of the single degree of freedom system under harmonic loading become equal with the system's natural frequency (ω_N) it causes the appearance of the resonance phenomenon.

3.2. Pseudo resonance

By starting these machines to work, angular velocity increases from zero to the maximum by paying attention to these device capacities. While the first mode structural frequency is less than the rotary machine's loading frequency, rotary machine frequency crosses from the first mode frequency of that machine. According to the pseudo resonance phenomenon, by reducing the gap between two frequencies, at first the responses will increase then they decrease after crossing that phase region.

3.3. Beating

While the frequency of harmonic loading is close to the first mode frequency of undamped system and the term of equation 2 occurs, the beating phenomenon will appear.

$$\Omega = (1+\varepsilon)\omega_N, \quad |\varepsilon| \ll 1 \tag{2}$$

By existence of that condition, system's response will increase through passing time and after getting to the maximum, with the same inverse slope, it will decrease until reaching zero.

3.4.Pseudo Beating

While the frequency of harmonic loading is close to the first mode frequency of the damped system, by passing time the system's response will increase and after arriving at the maximum it will decrease. These processes will continue until the response converges to the mean. This phenomenon is called pseudo beating.

In order to observe the controlling system's performance six criteria are used. J_1, J_2, and J_3 criteria are respectively the ratio of maximum responses in controlled situations to uncontrolled ones for displacement, velocity, and acceleration. J_4, J_5, and J_6 Criteria are the norms of responses in controlled situations to uncontrolled ones for displacement, velocity and acceleration respectively.

According to Table 1, the semi-active damper with fuzzy controller has better results than other controlling method. Decreasing maximum displacement and velocity responses will provide suitable situations for designing structure's elements and convenience of users. On the other hand, decrease in norm of responses will speed up damping in vibration of elements and cause's decrease in the possibility of fatigue appearance.

4. CONCLUSIONS

When rotary machine frequency is close to system frequency it may lead to some phenomena such as resonance, pseudo resonance, beating and pseudo beating in the system. In this study, we tried to reduce the undesirable response of single degree of freedom system according to probable phenomena using the passive and semi-active mass dampers. In order to do this, 2% mass ratio was used and frequency and damping ratio were offered by optimized relations as passive TMD parameters. To determine damping ratio in semi-active TMD, on-off ground- hook controller and fuzzy logic system were used. Fuzzy system needs suitable set of rules and membership function depending on structure in order to have good performance. Therefore, these rules are designed to return the structure to equilibrium condition. Semi-active TMD with fuzzy controller can have better performance than the on-off ground-hook algorithm, and also it is less sensitive to TMD frequency. So it can be used more confidently than other controllers. The results show that semi-active TMD with fuzzy controller as the best controller reduces the maximum displacement at resonance, pseudo resonance, beating, and pseudo beating 71, 57, 60, 62% respectively.

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HOW TO CITE THIS ARTICLE

M. Ramezani, A. Bathaei, S.M. Zahrai, Semi-active fuzzy control of SDOF systems under loading of rotary machines by tuned mass dampers, Amirkabir J. Civil Eng., 51(5) (2019) 303-306.

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