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Semi-Active Damping of Mechanical Vibrating Systems Using Variable Stiffness Actuator

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Abstract

In this research, a variable stiffness actuator is proposed to enhance the damping of the mechanical vibrating system. The frequency response analysis of the vibrating system is dependant in order to analyze and synthesis this semi-active damping, where the suggested process is using active filter to estimate the present frequency of the vibration system, and this will limit the value of the stiffness of the vibrated system. Two active filter s are needed, low-pass-filter (LPF) to choose the higher stiffness of the actuator at small frequencies as well as more damping and high-pass-filter (HPF) to choose the lower stiffness of the actuator at high frequencies as well as more damping, and so the result will be good damping as a wholre.

These smart systems and others will increase the importance of the mechatronics systems. This job has a case study to explain the semi-active damping system proposed.

Keywords: Semi-active system, vibrations damping, active filters, variable stiffness actuator

1. Introduction

The mechatronics system really was developed to make the mechanical systems more intelligent [1].

During the last few years, a great efforts had been spent and yet to damp-out the vibrations of the mechanical systems because of the strength limitations of most parts of these systems [2], but the matter is just mitigating them. One of these efforts is the semi-active damping that who differ from the passive damping throughout having a modification property and differ from active damping throughout making the required damping without needing an external energy to make it. Semi-active systems are dependent systems now days [3].

Here, in this research, the ability of variable stiffness spring is exploited to develop an adaptive damping to the vibrated systems. This spring is added to a spring-mass-damper, and depending on the frequency response characteristics of this system, then decide the overall stiffness of the system depending on a predefined response criterion.

A. Abu Hanieh et al. 2002 show that as future astronomic missions will require more and more stringent resolution requirements, the high demand for an environment clean of vibrations and disturbance appears. This also leads to the need for high precision steering devices for fine pointing of sensitive optics with the highest possible accuracy. Several methods exist to reduce vibration levels: the first consists in isolating the sensitive system from the perturbation and the second in damping the structure vibration modes. Therefore, two Stewart platforms have been designed, manufactured and tested. The first is a soft hexapod that provides 6 degree-of-freedom (DOF) active isolation and the second is a stiff hexapod that provides active damping to whatever flexible payload attached/mounted to it. In addition, both hexapods have steering capabilities.

Lawrence J. Alder and Stephen M. Rock 1993 explain the goal to develop control techniques that provide precise high-bandwidth end-point control of flexible-link manipulators, while simultaneously damping any internal oscillations of the payload.

Ronald H. W. Hoppe et. al. 2002 show that the new generation of electrorheological fluids (ERFs) offers a wide range of applicability in fluid mechatronics with automotive ERF devices such as ERF shock absorbers mentioned at first place. The optimal design of such tools requires the proper modeling and simulation both of the operational behavior of the device itself as well as its impact on the dynamics of the complete vehicle, this research addresses these issues featuring an extended Bingham fluid model and its numerical solution as well as substitutive models of viscoelastic-plastic system behavior. Also control issues for optimal active suspension of vehicles with controllable ERF shock absorbers are discussed.

Maria A. Heckl and I. D. Abrahams 1996 introduce an active control technique that combats oscillations driven by dry friction forces. Dry friction can act as the excitation mechanism for some high amplitude oscillations; curve squeal from trains is a well known and notorious example. Another, more elementary, example is the oscillation of a mass spring system sliding on a moving Belt. A model which predicts the stability behavior of this system is presented. The model is then extended to include an active control system.

Genda Chen, and Derek Smith 2002 show that a Piezoelectric Wedge Actuator (PWA) is proposed to improve the seismic effectiveness of a passive tuned mass damper (TMD). A PWA is composed of a thin aluminum plate and two piezoelectric sheets bounded on the aluminum plate. The actuator is connected in series with a TMD and it is used as a variable stiffness device or a damping unit regulated with applied voltage. Several control algorithms are considered. Numerical results indicate that a PWA is effective for a light mass damper and rapidly ineffective as the weight of the damper increases.

2. Hardware Description

2.1. The Mechanical system

F(t)= force excitation (N)
m= mass (kg)
k=original stiffness (N/m)
ka= actuator stiffness (N/m)
C= damping coefficient (N-sec./m)
X(t)= Vibration response (m)

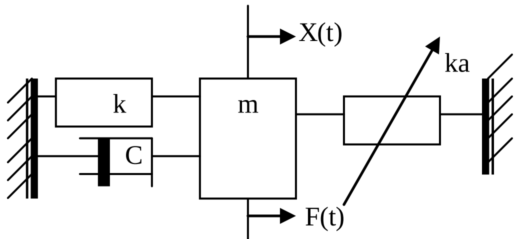
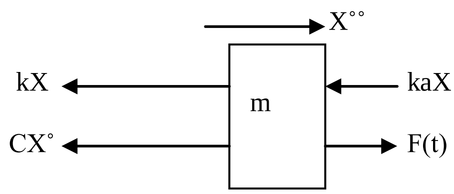


Fig. 1. Mechanical System.

The free-Body-Diagram to **Figure(1)**, is: -



$$\sum F = m\ddot{X}$$

$$m\ddot{X} + c\dot{X} + (k + ka)X = F(t)........................(1)$$

Assume that the system is excited harmonically.

where F(t)=Fo sin wt……………….…….. (2)

Hence the vibration response will be

$$X(t) = Xo\sin(\omega t - \phi)..................................(3)$$

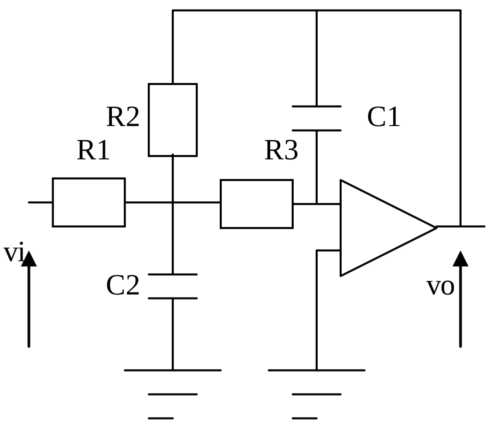
where:-

$$Xo = \frac{Fo}{\sqrt{[(k + ka) - m\omega^2]^2 + (c\omega)^2}}................(4)$$

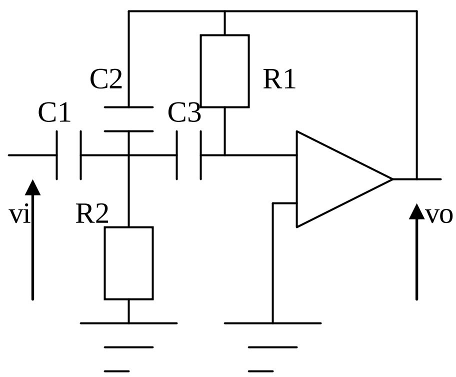
Equ. (4) is the required mathematical model to analyze the frequency response to the mechanical vibrating system.

2.2 Electrical Filters

Two types of active filters are used in this research they are low-pass-filter(LPF) with cutt-off frequency ωc and high-pass-filter (HPF) with the same cut-off frequency ωc .The LPF and HPF are shown in **Figure(2)**, (a) and (b) respectively [9].



(a) LPF



(b) HPF
Fig. 2. Active Filters.

The transfer function of the LPF is [9]:-

$$\frac{Vo}{Vi}=\frac{-1}{\left[1+\frac{R_1}{R_3}-R_1R_2C_1C_2\omega^2\right]+j\left[R_2C_1+2R1C1+\frac{R_1R_2}{R_3}C_2\right]\omega} \quad ..(5)$$

The transfer function of the HPF is [9]:-

$$\frac{Vo}{Vi}=\frac{R_1R_2C_1C_2\omega^2}{\left[1-(R_1R_2C_1C_2+R_1R_2C_2C_3)\omega^2\right]+j\left[R_2C_1+R_2C_2+R_2C_1+R_2C_3\right]\omega} \quad ...(6)$$

The ideal responses of LPF and HPF are shown in **Figure(3)**,(a) and (b) respectively

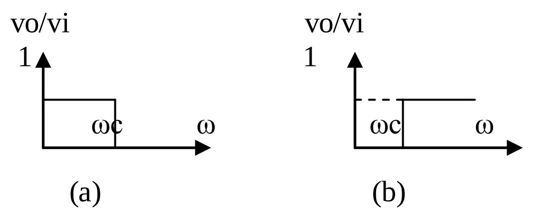


Fig. 3. ideal responses of LPF and HPF.

3. Semi-active Damping Strategy

The mechanism of changing the spring stiffness in order to enhance vibration damping is as follow:-

Taking two springs of the extension type as show in figure (4)a, by changing the length of this type of spring, it is possible to change its stiffness [10], and taking two springs in order to get the action of the helical spring, since the extension spring can extend only and cannot be compressed, at the same time, the type of the spring required in the vibration systems must be of the helical type [1].
Changing the length of these types of springs must
be occur without changing the setting location of the mass of the vibration system, and that may be happened by connecting the two mentioned springs as shown in figure (4) b.

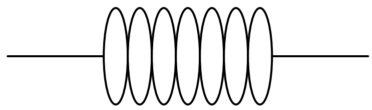


Fig. 4. (a). A schematic of an extension spring.

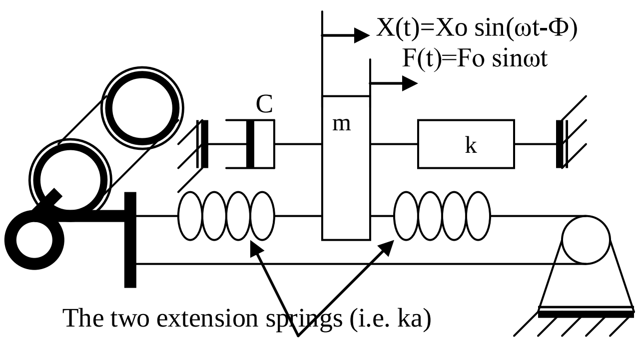


Fig. 4. (b). Spring-mass-damper system showing the two additional extension springs.

From **Figure (4)** (b), the purpose of the servo-motor is to change the length of the two extension springs (i.e., the developed helical spring), and thus change the spring stiffness as it is required to get dam ping required.

Note here, that each of extension springs must be initially stretched to have an initial length (that mean initial stiffness) to prevent the spring looseness cases as show in **Figure (5).**

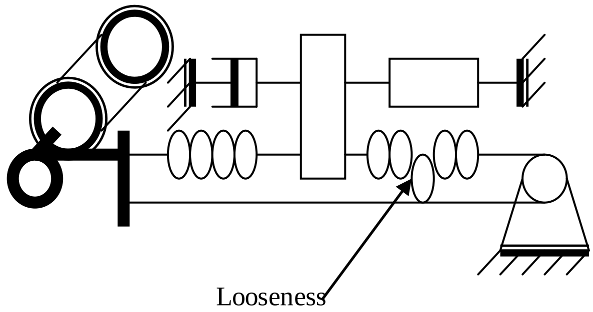


Fig. 5. Schematic of the system showing the looseness.

Also there must have mechanical stops to prevent the vibration responses from moving or rotate the servo-motor.

Now the overall semi-active damping system can be explained in the simulation flow diagram shown in **Figure(6)**:-

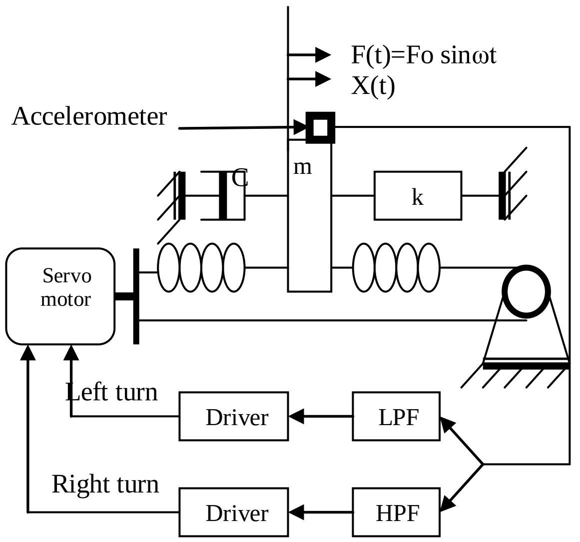


Fig. 6. Simulation flow diagram of semi-active damping system.

From **Figure(6),** the accelerometer senses the harmonic signal of the vibration, and this signal which is assumed as sine wave signal with its frequency will be input signal to the two active filters, and depending on the cut-off frequency of each filters, the signal will passed through either LPF or HPF, where if the frequency is low respectively, the signal will pass through the LPF and through the driver shown that will turn the motor left to increase the length of the extension spring (i.e. increasing its stiffness). And if the frequency of the signal is high respectively, it will pass through the HPF and through other driver that will turn the motor right to decrease the length of the extension spring (i.e. decrease its stiffness).

Here in this research since there are two constant motion to the motor (either full right or full left), then the control strategy that established here is by using Bang-Bang algorithm [11], which is meaning two states only to the control signal.

4. Case Study

Assume a spring-mass-damper system shown in **Figure(6)** with the following selected components:-

m= 60kg

c=50 (Ns/m)

k=3800 (N/m)

$$\text{ka} = \begin{cases} 100 \ \ (\text{N/m}) & \text{initial value} \\ 1000 \ \ (\text{N/m}) & \text{modified} \end{cases}$$

F(t)= Fo sinωt

Fo=10 N

The mathematical model:-

$$Xo = \frac{10}{\sqrt{\left[(3800 + ka) - 60\omega^2\right]^2 + (50\omega)^2}}$$

The frequency responses when ka=100N/m and 1000N/m are shown clearly in **Figure(7).**

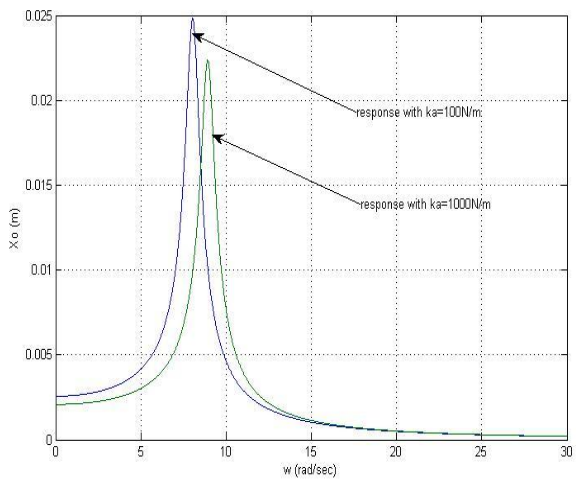


Fig. 7. Frequency responses of the proposed system when ka=100 and 1000N/m.

The process is to select the better response at the given frequencies. From **Figure(7)**, it is clearly shown that the response with ka=1000N/m is better than the response with ka=100N/m if the frequencies are ≤ 8.5 rad/sec (or 53.4 Hz), whereas the response with ka=100N/m is better than the response with ka=1000N/m if the frequencies are ≥ 8.5 rad/sec. So, since the proposed active LPF and HPF filters are responsible of the selection, therefore the cut-off frequency for each one must be ωc= 8.5 rad/s (or. 53.4 Hz). Also assume an ideal LPF and HPF.

Now when the frequencies of the vibration signal that sensed by the accelerometer is ≤ 8.5 rad/sec, the LPF will be activated and hence the response with ka=1000N/m will be selected, and when the frequencies exceed the 8.5 rad/sec, the HPF will be activated and hence the response with ka=100N/m will be selected, and so there will have been the modified response which is the better one.

5. Results and Discussion

The results of the previously assumed semi-active damping system are clearly shown in the **Table(1).**

Table 1
The vibration response when ka=100 and 1000N/m and the modified response.

Frequency ω (rad/sec)	Response Xo(m) when ka=100N/m	Response Xo(m) when ka=1000N/m	Modified response Xo(m)
0	0.0026	0.0021	0.0021
2	0.0027	0.0022	0.0022
5	0.0041	0.0030	0.0030
7	0.0098	0.0053	0.0053
8	0.0247	0.0096	0.0096
8.5	0.0160	0.0160	0.0160
9	0.0094	0.0220	0.0094
10	0.0046	0.0077	0.0046
11	0.0029	0.0040	0.0029
15	0.0010	0.0011	0.0010

It is clearly from **Table(1)**, that the modified response which is the response of the proposed semi-active damping system is very good response where it have a more damping than the two other responses which are without modification. The process is seems to be as a selection process to the better response values from the other two responses. The modified response is shown in **Figure (8)**.

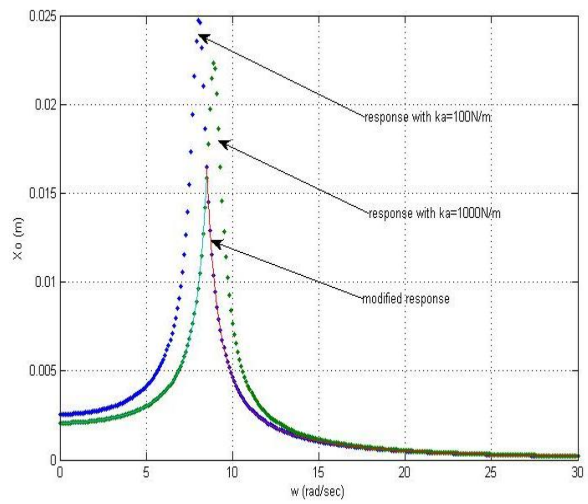


Fig. 8. Frequency response showing the modified response.

As shown from **Fig.8.** it is clearly that with the semi-active damping technique, it is possible to have more and more damping depending on the ability of the variation in spring stiffness.

6. Conclusion

The implementation of the semi-active damping system in hardware has been proven theoretically to be beneficial for intelligent systems requiring fast control response, and the simplicity of the proposed control system in this research had improved the speed of response of the damping to the vibrated system at steady-state. That is due to the hardware (not software) design.

Also the new idea of using the active filters approved its ability to pick-up the signal of the required frequency, and this idea may be develop to enhance the overall system performance.

The flexibility of the system here is limited due to the hardware design rather than software, but at the same time the system is more durable.

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الإخماد الشبه فعال لأنظمة الإهتزازات الميكانيكية بإستخدام جهاز متغير الجسائة

يعرب عمر ناجي العزاوي
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الخلاصة

في هذا البجث، جهاز متغير الجسائة تم اقتراحه لتحسين الأخماد لنظام الأهتزاز الميكانيكي .تحليل الأستجابة الترددية لنظام الأهتزاز تم اعتماده من اجل تحليل و تركيب نظام الأخماد الشبه فعال، حيث ان العملية التي تم اقتراحها هي باستعمال ال active filters لتخمين التردد الحالي لنظام الأهتزاز، وهذا سيحدد قيمة الجسائة لنظام الأهتزاز. اثنان من ال active filters طلوبة هما ال (LPF) لأختيار الجسائة الأعلى للجهاز في الترددات الصغيرة بشكل جيد كما هو الأخماد الأكثر ، و (HPF) لأختيار الجسائة الأقل للجهاز في الترددات العالية جيدا كاخماد أكثر. وهكذا النتيجة هي تخميد جيد. تلك الأنظمة الذكية و غيرها ستزيد من انظمة ال Mechatronics. كما و يتضمن العمل دراسة حالة لتوضيح نظام الأخماد الشبه فعال المقترح