MECHANICAL VIBRATION CONTROL USING VARIABLE STIFFNESS ACTUATOR

Bahaa Ibraheem Kazem and Yarub Omer Naji
Mechatronics Engineering Dept.,Mechanical Engineering Dept.
University of Baghdad,Baghdad,Iraq

ABSTRACT

Active control strategy had been used for controlling the vibration response, with considering the simplification in the design and keep away from complexity and high cost in order to extend the actual application of the active control systems. The control program is responsible for identifying the acquired data and make comparisons and decisions required for increase or decrease the actuator stiffness, by instructing the control signal to open the inlet or the exhaust valve respectively. The PC sound card has been used as a Data Acquisition Card. The proposed control strategy had been simulated and tested to control the vibration response of a single degree of freedom spring-mass-damper system excited harmonically at a fixed frequency and with variations of different system parameters (mass, damping constant, and excitation force). Two testing cases are used to investigate the ability of using the proposed actuator and the control strategy in the vibration control field.

الخلاصة

استراتيجية السيطرة الفعالة تم استخدامها للسيطرة على استجابة الاهتزاز، باعتماد البساطة في التصميم و البقاء بعيدا عن التعقيد و الكلفة العالية من اجل توسعة التطبيق الحقيقي لأنظمة السيطرة الفعالة,حيث ان استجابة الأهتزازيتم قياسها بولسطة (accelerometer), ثم تترجم الأشارة المقاسة الىاشارة سيطرة التي هي مسؤولة عن السيطرة على استجابة الأهتزاز. المسيطر متمثل ببرنامج الحاسب بال(MATLAB version 6.5) و هذا البرنامج مسؤول عن تعريف البيانات المكتسبة و عمل المقارنات و القرارات ألللازمة, حيث يتم مقارنة البيانات ألمقاسة مع البيانات ألمرغوبة والتي تمثل الأ ستجابة الترددية لنظام نموذجي(او ألمعيار) الذي يتم التعامل معه, ونتيجة هذه المقارنة تقرر زيادة او نقصان جسائة المشغل, ويتم ذلك بايعازاشارة السيطرة لفتح صمام الأدخال أو الأخراج. كما و اقترح أيضا، كفكرة جديدة و التي هي استخدام بطاقة الصوت الخاص بالحاسب الشخصي كبطاقة اكتساب البيانات. تم محاكاة استراتيجية السيطرة المقترحة و فحصها للسيطرة على استجابة الاهتزاز لنظام (كتلة نابض ذو درجة حرية حركة واحدة) يتم تحفيزه توافقيا وبتردد ثابت و للتغير في مختلف متغيرات النظام (الكتلة ، ثابت الاخماد، القوة المؤثرة) و النتائج بينت مطابقة جيدة مع المعيار المقترح مع أقصى نسبة خطا(٢٠±%).

KEYWORDS:
artificial muscle, mechanical vibrations, active vibration control

INTRODUCTION

The increasing use of high strength materials in mechanical parts, machines and structures design has made these parts to become relatively light and flexible, with a low level of internal damping .The vibration of these parts may therefore exceed safety criteria more often when compared to older parts, resulting in discomfort of occupant, malfunction of equipment and possibility of fatigue failure. Therefore great efforts had been spent and yet to reduce or mitigate vibrations.

The active control devices introduce mechanical energy into the controlled vibrated system. The demand for active vibration control devices and systems stems from one of the following needs: (i) increasing performances in precision systems, (ii) improving the riding and acoustic comfort of vehicles, (iii) allowing the construction of taller civil engineering structures and longer bridges.

(Shiyu Zhou and Jianjun Shi 2001) show that the vibration suppression of rotating machinery is an important engineering problem. A review of the research work performed in real-time active balancing and active vibration control for rotating machinery, as well as the research work on dynamic modeling and analysis techniques of rotor systems is presented. The basic methodology and a brief assessment of major difficulties and future research needs are also provided.

(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.

(Alder and 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 et. al.2002) show that the new generation of electrorheological fluids (ERFs) offera 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 and 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. A model which predicts the stability behavior of this system is presented. The model is then extended to include an active control system. The active control system is a feedback loop which consists of the following components. A transducer senses the velocity of the mass. The transducer output is passed to filter, then to a phase shifter which applies an adjustable phase shift, to a variable-gain amplifier, and finally to a shaker which is attached to the mass. The shaker exerts a control force in the tangential direction, and this force superimposes on the friction force. The stability behavior of the controlled system is analyzed in two ways by calculation of the complex eigenfrequencies and by considerations based on a balance equation for the oscillatory energy. The differences between this form of active control and conventional anti-vibration methods are discussed in this paper. Further studies, both on the theoretical and experimental front, are under way to extend the application of this kind of active control to curve squeal from trains.

(Ramutis et. al. 2002) analyzed the smart system for the automatic suppression of structural element oscillations by controlled dissipative forces. The system consists of composite coating of structural element and includes the piezoelectric layer of electrorheological fluid (ERF) contacting with it. The electric charge occurs in every section of the electrodes of the piezoelectric layer due to the direct piezo effect when oscillations in the structural element are exited. The ERF viscosity changes in its area of contact with electrodes in accordance with the voltage on them. Magnitostrictive materials in combination with the magnitorhelogical fluids (MRF) may be used for such system design as well. The method can be generalized for wider applications including the systems with several degrees of freedom.

(Genda and Derek 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. A change of 1% in natural frequency of a TMD experiencing a peak displacement of 0.25 in. can be achieved with the displacement proportional control when 750 volts are applied on the piezoelectric actuator. Similarly an increase of 33% in damping ratio can be achieved with the velocity proportional control and an equivalent damping ratio of 0.054 is obtained with the Bang-Bang control for a TMD of 0.024 original damping ratios.

BRAIDED PNEUMATIC ACTUATOR (BPA)

A BPA is a pneumatic device developed in the 1950's as an orthotic appliance for polio patients by J. L. Mckibben (Nickel et al., 1963). It consists of a rubber bladder encompassed by a tubular braided mesh. When the bladder is inflated, the actuator expands radially and undergoes a lengthwise contraction. **Fig.1**. is a picture of an inflated (bottom) and uninflated (top) actuator.

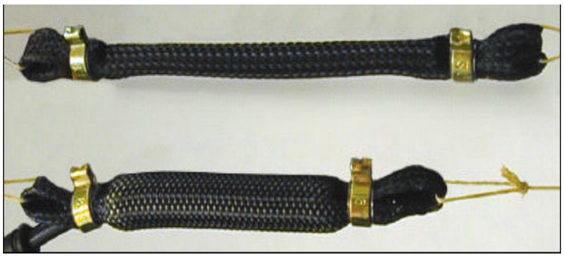


Fig.1. Pneumatic muscle

THE +MA ACTUATOR

BPA muscle can be stretched only and can not be compressed. It is possible therefore to deal with it as an extension spring. But in vibration systems, the spring component must be of the helical type that can be stretched and compressed . This is the reason for putting what is named +MA (Plus Muscles Actuator).

The +MA, shown in **Fig.2** is a subsystem that consists of two artificial muscles work antagonistically, and together behaves as a variable stiffness spring of the helical type. The two muscles placed into a rhombus frame of four movable links connected through joints.+MA connects in series with the TMD as shown in **Fig.2**.

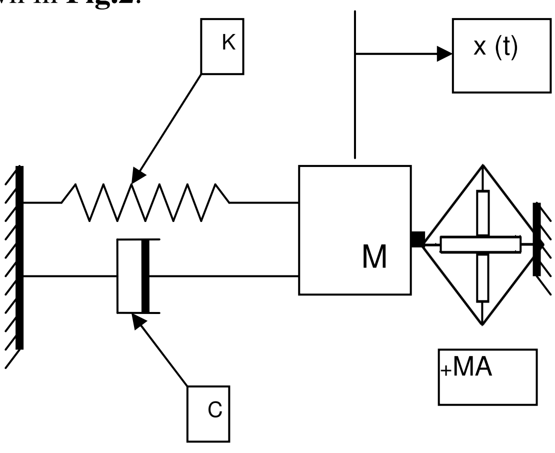


Fig.2. A schematic of TMD with +MA

One muscle has stiffness (k) that depends on its length (L) at a constant pressure (Pg), and that means (k) is proportional to the length of the muscle. When dealing with dynamic motion it will give a non-linear relationship.

The stiffness of the +MA is:

$$k_A = \frac{3P_g\, x(t)}{\pi n^2} \tag{1}$$

Eq.(1) indicate that the stiffness of the +MA changes with the change of the system response x (t) and the pressure (Pg). Therefore the variation in the stiffness of the actuator comes from the variation in these two parameters. The forces generated at each muscle and at +MA due to the responce x (t) at any time can be represented as:

$$F_{m1} = k_{m1}x(t) = \frac{3P_g}{2\pi n^2}(Lo - x(t))x(t) = \frac{3P_g}{2\pi n^2}\left[Lox(t) - x^2(t)\right] \tag{2}$$

$$F_{m2} = k_{m2}x(t) = \frac{3P_g}{2\pi n^2}(Lo + x(t))x(t) = \frac{3P_g}{2\pi n^2}\left[Lox(t) + x^2(t)\right] \tag{3}$$

$$F_A = F_{m2} - F_{m1} = \frac{3P_g}{\pi n^2}x^2(t) \tag{4}$$

From **equ.(4)**, FA is a nonlinear dynamic function of the vibration response x(t).
These three forces, as a test, at x(t) = Xo sinωt at a given constants, also can be represented as shown in

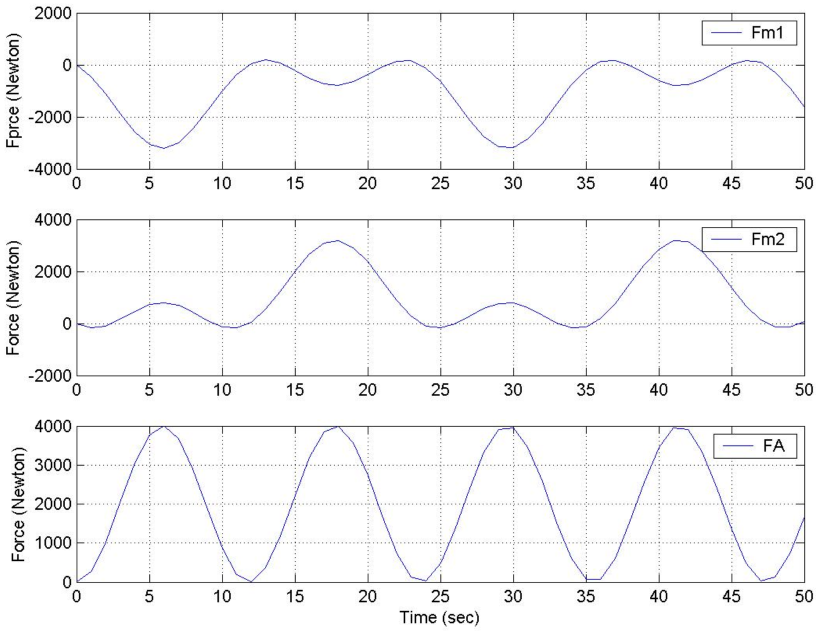


Fig.3.: Forces response to each muscle and to +MA at x(t) =Xo sinωt

The change of +MA stiffness due to the responce x(t) is not adjustable but it is possible to adjust the air pressure (Pg) inside +MA and therefore adjust its stiffness. So air pressure (Pg) will be used in controlling the system response x (t).

CONTROL SYSTEM OVERVIEW

The Plus Muscle Actuator (+MA) is built in conjugation with a passive Tuned Mass Damper (TMD) system for testing the actuator control theory. A more important part of the design is the actuator (i.e. +MA), where it is designed to control the overall system response.

An accelerometer based acceleration transducer is mounted on the mass of the system. The sensor signal is sent to the computer through a Data Acquisition Card (DAQ). An effective program used to make comparisons and decisions required. The control program generates, therefore an output signal and sends it on the same DAQ. This signal is adjusted to open or close solenoid valves through two separated lines that connected to two relays, and so the air is moved in and out of the actuator. **Fig .4**. shows the hardware overview of the system.

SIMULATION OF THE CONTROL STRATEGY

The purpose of the simulation is to create a general dynamic model of the passive tuned spring-mass-damper system working in conjugation with the +MA and the solenoid valves. The input to the model is simply valve state (i.e. open to supply, open to atmosphere, or chamber sealed), and the output is the system peak response Xo. **Fig.5.** is a schematic of the simulation flow.

In **Fig.5.,** the controller which represents the used computer program, identifies the system response through acquiring the measured data, compares these data with the criterion, and then make decision and modification required.

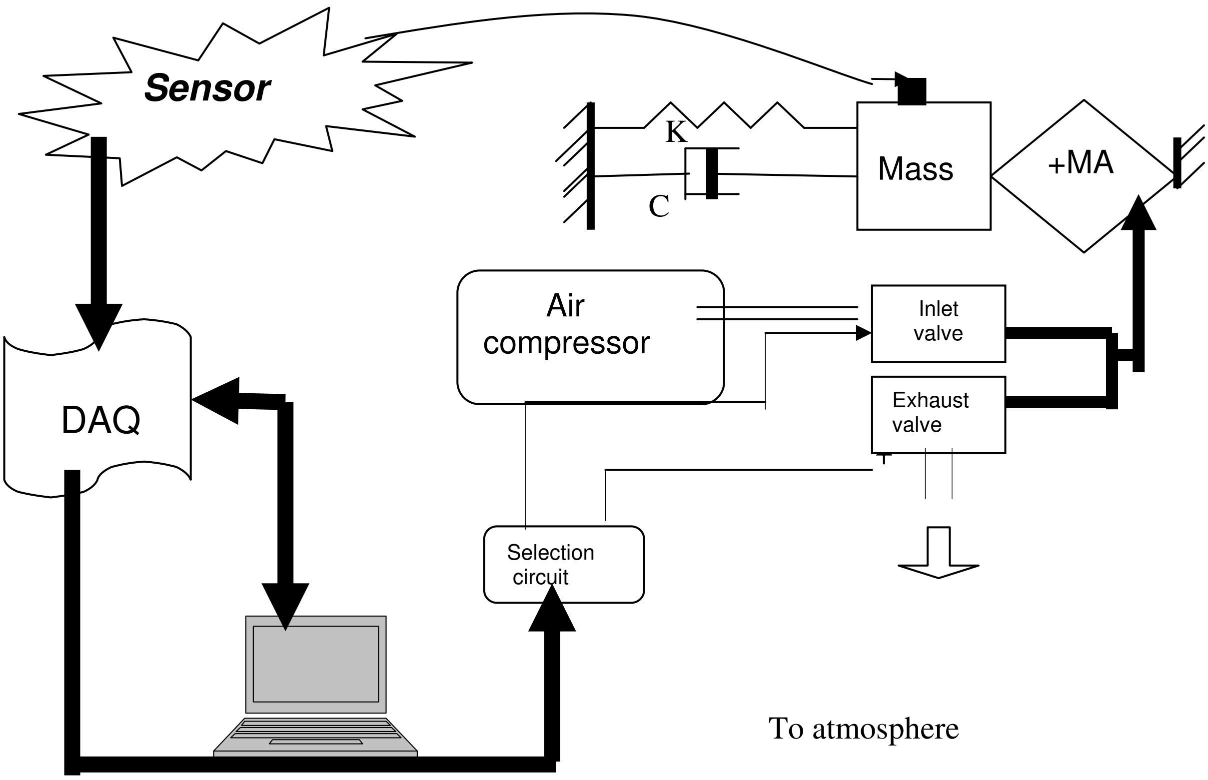


Fig..4. Hardware Overview

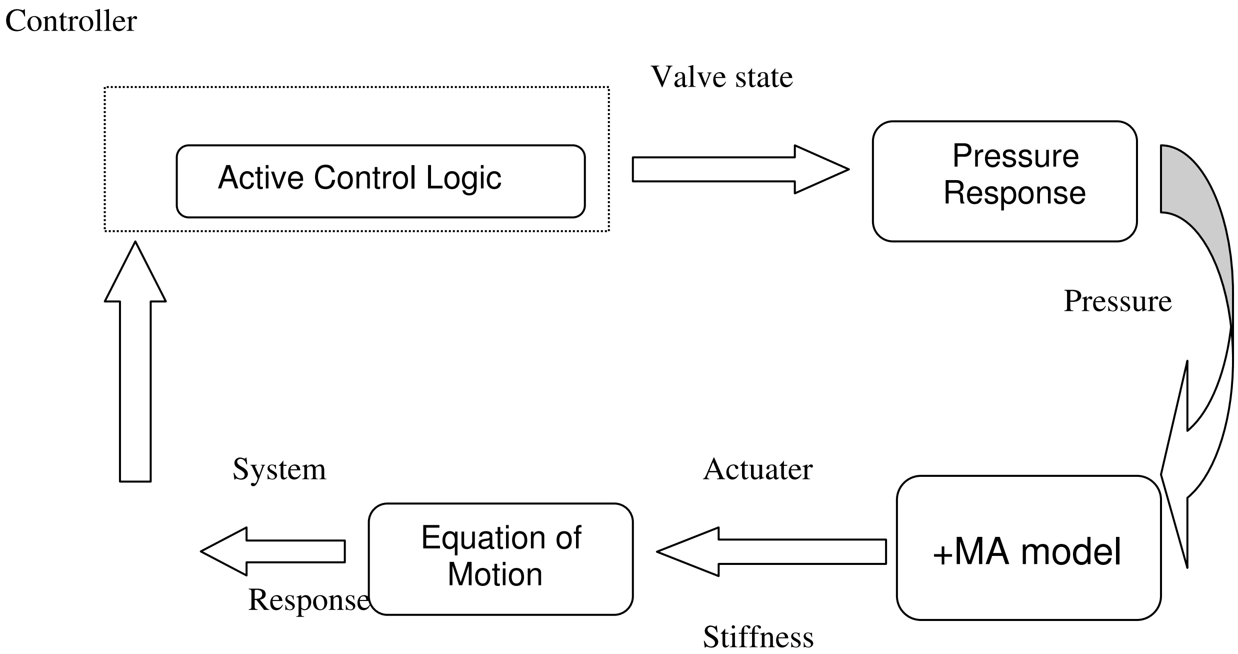
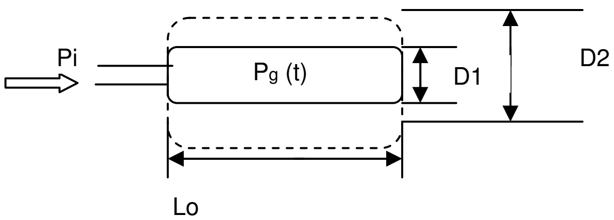


Fig.5.: Simulation Flow

In order to make the dynamic simulation of the control system, it is needed at first to make the following assumptions are used:

1. For +MA, at a constant pressure, the increase in the volume of one muscle due to x (t) is opposed by the decrease in the volume of the other muscle, therefore the volume of +MA is dynamically constant, and changes only with the change in pressure (Pg), and this change is radial for each muscle as shown in **Fig.6.**



Pi = Supplied pressure
D1 = Initial diameter
D2 = Increased diameter
Lo = constant initial length

Fig.6.: The volume change of one muscle of +MA due to pressure change Pg (t)

2. From **Fig.(7),** (shown later), the supplied pressure (Pi) is 689 kPa.

3. The typical system is consisting of the following data:

m = 20 kg
C = 60 N.sec/m
k = 3000 N/m
Fo = 10 N

These data is chosen arbitrarily, but in case at which the vibration peak does not exceed the maximum limit (i.e. 0.0175 m).The equation of the frequency response of the typical system is now:

$$Xo = \frac{10}{\sqrt{(3000 - 20\omega^2)^2 + (60\omega)^2}} \tag{5}$$

And it is represented by **Fig. 7**.

The required general dynamic model:

$$Xo = \frac{Fo}{\sqrt{\left[\left(k1 + 0.001P_g(t) - m\omega^2\right)\right]^2 + (C\omega)^2}} \tag{6}$$

4. Recall **equ.(1)**, it is easy to assume that n=3 from looking to **Fig.(1).**

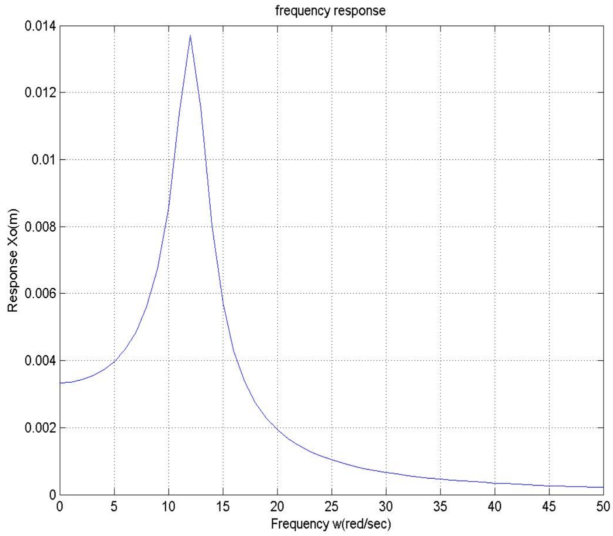


Fig.7. Frequency response curve of the typical system

As shown from this equation, the stiffness of +MA is changing with the vibration response x(t), but fortunately this change is small. Let Xo = S, and so the (equ.(1)) will be:

$$k_A = \frac{3P_g S}{\pi n^2}\sin\omega t = K_A \sin\omega t \qquad (7)$$

$$\therefore K_A = \frac{3P_g S}{\pi n^2} \qquad (8)$$

Now taking three different values to the S of (equ.(8)), in order to make test to the stiffness changing with the vibration response x(t). These different values are:

S = 0.1 cm (minimum value)
S = 0.925 cm (middle value)
S = 1.75 cm (the maximum vibration peak value)

Since K_A of (equ.(8)) is independent on the frequency (ω), therefore take the vibration at any frequency value, and let it to be ω = 25 rad/sec.

For these values of (S) and (ω) , the stiffness changing and its effect on the vibration response x(t) at a different pressure values (Pg) are shown in **Figures (8), (9),** respectively.

From Figures(9) [(a), (b), and (c)], it is easy to see that the change of the vibration response x(t) with the change in +MA stiffness (kA) due to (S) is small and therefore taking any value of the mentioned values of (S) then substitute it in the (equ.(8)). So, take the middle value of the (S), with a small error ratio as shown follow:

From Figure(9), (c), the maximum error is:

$$\max imum error = \pm\frac{Xo(S=0.925)-Xo(S=1.75)}{Xo(S=1.75)}$$
$$= \pm\frac{0.003-0.0025}{0.0025} = \pm 0.2 \qquad (9)$$

Or Maximum error ratio = ±20%.

and so the (equ.(8)) will be now:

$$K_A = \frac{3P_g\,(0.00925)}{\pi n^2} \approx 0.001 P_g \quad (10)$$

Or

$$K_A \approx 0.001 P_g \quad (11)$$

And let the initial Pg = 80 kPa which will make the initial stiffness KA =80 N/m.

CONTROL PROGRAM ALGORITHM

The control program written in MATLAB v 6.5, consist of three parts. Part 1 is for getting the data from the accelerometer through the sound card to the MATLAB engine], and analyze these data, compare with the criterion, and depending on the result of the comparison decide to translate to others two parts (2 and 3).

If the result of the comparison (which is the difference between the desired response and the actual one at the vibration frequency) is negative, then going to part 2. Part 2 is for generating an output analog signal at the first line which will represents the control signal that is responsible of activate the inlet valve to the given charging duration time. But if the result of the comparison is positive, then going to part 3. Also part 3 is for generating the same output control signal but at the second line in order to activate the exhaust valve to the given discharging duration time.

At the same time of which the program deals with part 2 or 3 there is no getting new data, since at each part the program will be in a wait state till it complete control signal generation. Then after generating each of the control signals, the program translates to part 1 to record the new modified response data, then making the new comparison and decision and so on.

Now if the comparison result is zero, the program will go back to part 1 to start again without modifications.

RESULTS AND DISCUSSIONS

<u>Case study1</u>

If for any reasons, the value of the damping constant C of the considered system is reduced to C=50 N.sec/m. The frequency response will be changed. And since the control algorithm is online, the controller will record directly the new change which is the increased response, and thus activate the inlet valve in this case to the given duration time (the duration time t= 0.1 sec for both charging and discharging) . The criterion and data changing can be represented by the **Table 1**.

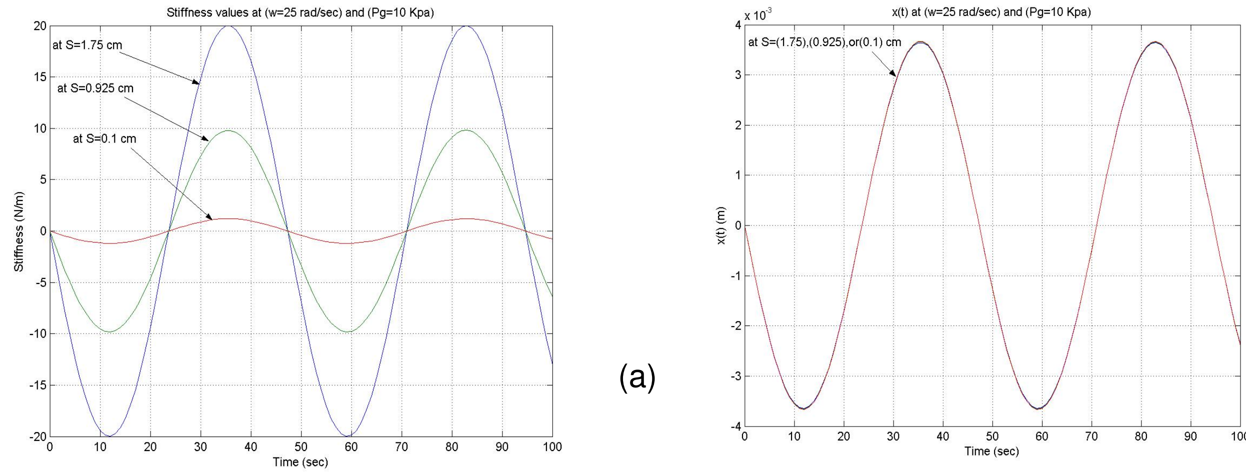
At frequency ω = 11 rad/sec, the response is changed (increased) and the control algorithm will take three iteration times charging and two times discharging to be agreed as shown in **Table 2.**

At frequencies (ω = 13, 14 rad/sec), which are higher than the natural frequency (ωn = 12.24 red/sec), the response is increased, but here the charging iteration times will never make the response agree the criteria because there will be a saturation case (i.e. the pressure inside +MA will be at its maximum range 689000 Pa),

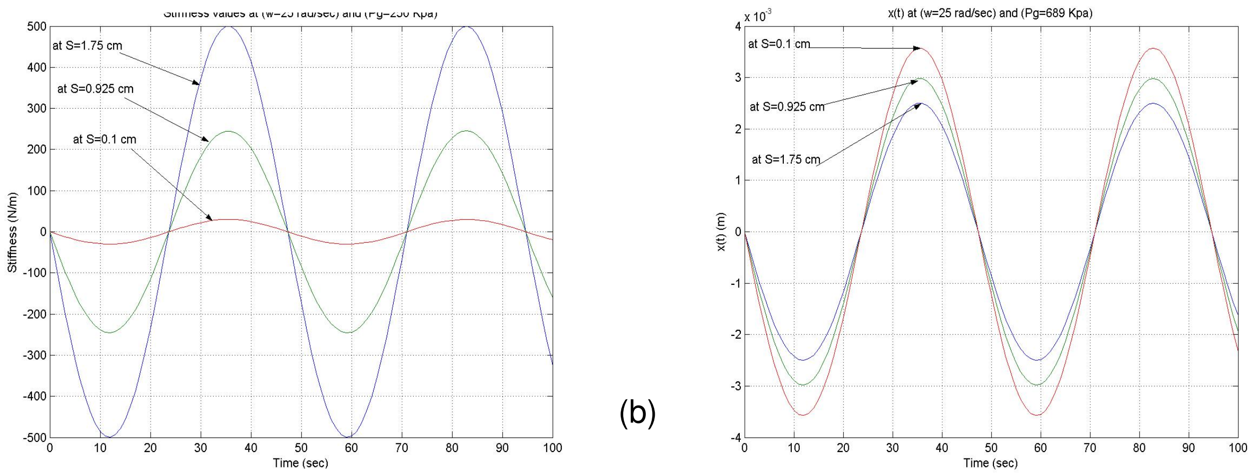
Case study2 The damping constant reduced to C = 50 N.sec/m, and at the same time the mass decreased to m = 15 kg, and Fo = 12 N. The new response changed also and the previous iteration process will occur same as the above states to reach the optimal performance. The criteria and the change are represented in **Table 3.**

At ω = 10, and 11 rad/sec, the response is decreased, and so there will be no change on the system response, exactly as in case study 2, when mass increased at frequency ω = 13 rad/sec.

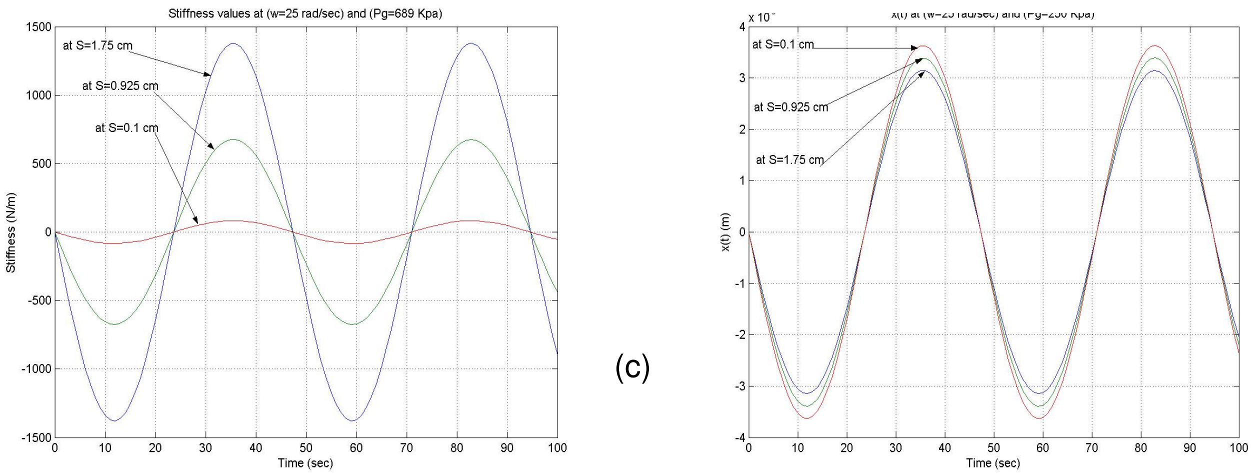
At ω = 13 rad/sec, eleven charging iteration times required to reach criteria as shown in **Table 4.**



(a)



(b)



(c)

From simulation results, it can be seen that within this simplified control system it is possible to control the vibration response and make the controlled cases despite of saturations agree the putted criterion with probability of a maximum error ratio (±20%), but that may occur after many extreme charging and/or discharging iteration times because of the selected duration time for both states.

Also it can be shown that the plus muscles actuator (+MA) is a good idea for use in vibrations control, such as controlling the vibration of sensitive equipments of the airplane, optimizing the effectiveness of the TMD used in the structures vibration damping, and more others. Where its high strength to weight ratio and variable linear stiffness properties make it more reliable. Also with this variable stiffness actuator it is possible to prevent the overdamping case that may occur when dealing with variable damping coefficient, where the increasing in damping coefficient may exceed the critical damping coefficient ($2\sqrt{mk}$) and the damping ratio therefore will be higher than unity and hence overdamping occur. With variation in stiffness the process is like a shifting case to the frequency response rather than direct damping.

Table 1: The response criterion and response changing when the damping constant C changed from 60 to 50 N.sec/m

ω (rad/sec)	**Criteria Xo (m) (at C=60 N.sec/m)**	**Change Xo (m) (at C=50 N.sec/m)**
1	0.0034	0.0034
2	0.0034	0.0034
3	0.0035	0.0035
4	0.0037	0.0037
5	0.0040	0.0040
6	0.0043	0.0043
7	0.0048	0.0049
8	0.0056	0.0057
9	0.0067	0.0069
10	0.0086	0.0089
11	**0.0114**	**0.0125**
12	0.0137	0.0163
13	0.0115	0.0133
14	0.0080	0.0087
15	0.0057	0.0060
16	0.0043	0.0044
17	0.0034	0.0034
18	0.0027	0.0028
19	0.0023	0.0023
20	0.0019	0.0020

Table 2 : Number of charging and discharging times to the system at ω = 11 rad/sec to reach criteria (**Char** = charging **Dis** = discharging)

Table 3. Response criteria and response changing due to changing in C to 50 N.sec/m, and m to 15 kg, and Fo to 12 N.

ω rad/sec	**Criteria Xo (m) (at C=60 N.sec/m)**	**Change Xo (m) (at C=50 N.sec/m)**	**Char 1**	**Char 2**	**Char 3**	**Dis 1**	**Dis 2**
10	0.0086	0.0089	0.0086	0.0083	0.0081	0.0082	0.0083
11	**0.0114**	**0.0125**	**0.0120**	**0.0115**	**0.0111**	**0.0113**	**0.0114**
12	0.0137	0.0163	0.0161	0.0157	0.0154	0.0155	0.0157
13	0.0115	0.0133	0.0137	0.0141	0.0144	0.0142	0.0141
14	0.0080	0.0087	0.0089	0.0092	0.0095	0.0094	0.0093
15	0.0057	0.0060	0.0061	0.0063	0.0064	0.0063	0.0063

ω (rad/sec)	Criteria Xo (m) (at C=60 N.sec/m, and m=20 kg, and Fo=10 N)	Change Xo (m) (at C=50 N.sec/m, and m=15 kg, and Fo=15 N)
1	0.0034	0.0040
2	0.0034	0.0041
3	0.0035	0.0042
4	0.0037	0.0043
5	0.0040	0.0046
6	0.0043	0.0048
7	0.0048	0.0052
8	0.0056	0.0058
9	0.0067	0.0065
10	0.0086	0.0076
11	**0.0114**	**0.0092**
12	0.0137	0.0116
13	0.0115	0.0150
14	0.0080	0.0171
15	0.0057	0.0143

Table 4. Charging times at ω = 13 rad/sec, to agree the Criteria when C = 50 N.sec/m, and m=15 kg, and Fo = 12 N

Ω rad/sec	Criteria Xo (m)	Change Xo (m)	Char1	Char2	Char3	Char4	Char5	Char6	Char7
13	**0.0115**	**0.0150**	**0.0145**	**0.0140**	**0.0136**	**0.0132**	**0.0129**	**0.0126**	**0.0123**

Char8	Char9	Char10	Char11
0.0121	**0.0118**	**0.0116**	**0.0115**

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