Vibration Control of Clamped Affected by Dynamic Load With Electromagnetic Actuator

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Abstract

The dynamic behavior of fixed-fixed beam, resting on a nonlinear vibration isolator at middle point and subjected to dynamic load, is investigated. The vibration isolator consists of nonlinear spring and nonlinear damper. The beam is modeled as Euler beam and in order to find the nonlinear equations of motion, the method of assumed-mode is used. A moving force with different linear velocities is subjected along the beam length. In addition, the magnitude and direction of the force are varied. The effect of the linear velocities and frequencies of dynamic load on the isolation of the beam are discussed. Furthermore, Proportional-Integral-Derivative (PID) controller and electro-magnetic actuator are integrated to the system to achieve an optimal elimination of the beam deflection. The optimal output feedback PID gains are obtained using the pole placement method, and then further tuned by gradient descent optimization. The proposed controller showed a significant elimination of the beam deflection compared to non-controlled system.

Keywords— harmonic moving load, nonlinear vibration isolator, electromagnetic actuator, and PID controller.

1. Introduction

During operation of the machines, the mechanical parts suffer from changeable dynamic forces which vary in their magnitude and direction. These forces produce the vibration in these machines. Therefore, engineering designers always that the dynamic response of the beam was depended on. In addition, the researcher presented a variety of references that describe various problems of moving load which solved by different techniques. A number of investigations into the beam vibration were carried out by Mardani [12]. The amplitude of the steady state vibration as well as the angular frequency was studied. In addition, the case of beam subjected to a concentrate moving load was considered. Moreover, the bending moment, stress and deflection for the beam were obtained.

Over the past century there has been a dramatic increase in studying of moving load problem. A considerable amount of literature has been published on this topic for example references [4],[5],[7],[16]. Many types of stationary load problems related with structural dynamics were explained in excellent monograph by [5].Recently, because of developments in the fields of high-speed transportation, flight, robotics, machining with high prediction and the imperative necessity to the rapid lifting and supporting systems in the factories as well as in the shipyards, the study and investigation of structures that excited by moving and fluctuated loads are very important. Stefan and Andreas [17] presented a method to analyze robot arm issue that affected by moving loads and flexible bodies. The arm was considered as a multiple beam system and the robot was simplified as controlled multibody encourage reducing the vibration as low as possible to avoid catastrophic machine failure and to increase the product quality as well. The study of dynamic responses for mechanical parts is very important because it gives a good visualization about the mechanical behavior of these parts according to different dynamic forces excitation. Moreover, the physical limitations for the machines operation are specified, such as angular rotation which may cause resonance and linear velocity which translate through mechanical parts.

In the dynamic structures, it is very difficult to ignore the difficulty of the finding dynamic responses when the vibration arises according to the excitation of moving load. Such types of load can be found in guide ways, cranes, cableways, rails, pipelines, roadways, runways and bridges. The dynamics of structures are affected by significant parameters of the moving object which are inertia, velocity and acceleration.

The dynamic performance of a beam when loaded with accelerate mass was discussed in [8]. For the purpose of considering the effects of inertia alongside gravity force for the mass, a moving finite element was suggested to represent the acceleration of movable mass that was translated on the beam. It was confirmed that the acceleration of the mass and the mass ratio (mass of the load / the mass of the beam) were the main parameters at the middle of the beam. Also, an optimal PID controller is proposed to minimize the effects of the forces transmitted.

Nomenclature

A=cross section of area(m2)	w_c = is the outer coil width
c1= linear parameter of damper (Ns/m)	h_m = is the permanent magnet length
c2= nonlinear parameter of damper (Ns3/m3)	t_c = is the outer coil thickness
E= modules of elasticity (Pa)	z = is the coordinate relating the upper surface of the permanent magnet at mid
Iy= moment of inertia about y-axis (m4)	z_2, z_3 = is the coordinates describing the upper and lower ends of the core of the electromagnet, respectively
Fo= force amplitude (N)	z_{e1}, z_{e2} = is the coordinates relating the upper and lower ends of the coil of the electromagnet, respectively
k1= linear parameter of spring (N/m)	z_h = is the distance between the two magnets
k2= nonlinear parameter of spring (N/m3)	B = is magnetic flux densities
kij= stiffness element (N/m)	I= input current
L= length of panel (m)	μ_0 = is the permeability of magnetic in the air
mij= mass element (kg)	J = is the coil electric current per unit cross-sectional area
Qi=generalized force	N= is the turns number for the coil
qi= generalized coordinate	h = is the height of the coil
t= time (sec)	T =is the stress tensor in the magnetic field
V= velocity of moving load (m/sec)	F = is the total force vector for the magnetic field
xo= position of damper and spring (m)	
xp= position of moving load (m)	
w = beam displacement in z-direction (m)	
$\beta_i L$=weighted frequency	
ρ= density (kg/m3)	
ω_1=forced frequency (rad/sec)	
ψ_i= coordinate	

systems. The model showed how the response of flexible arm changes due to forces of the moving load affect. The possibility of forces connection was also presented.

In addition, not only the analysing and modelling of the mechanical structures dynamic is important issue but also the controlling of vibration for these structures is a very interesting issue. Recent developments in the fields of structural design and its flexibility have led to a renewed interest in vibration control of structure subjected to non-stationary load [3],[6],[21].

There is a large volume of published studies describing the role of vibration control of structure. References [2],[15], [18],[20] discussed different methods of vibration control including the main types such as (passive, semi-active, and active) as well as hybrid types. Researchers showed a big attention in active control method for structures because of its efficient and effective results than the others [1],[10]. The structure was formulated as a beam which is the commonly approach utilized in such study. This is because of numerous researchers can modeled as beams subjected to translating load in the design of machining processes. In this paper, a beam which is clamped at each end subjected to harmonic moving load is studied. In addition, the deflection of the beam is calculated when integrating nonlinear spring, nonlinear damper and electromagnetic actuator

$$PE = \frac{1}{2}EI\int_0^L \left(\frac{\partial^2 w}{\partial x^2}\right)^2 dx + \frac{1}{2}k_1 w(x_{\text{mid}})^2 + \frac{1}{4}k_2 w(x_{\text{mid}})^4$$
(2)

$$DE = \frac{1}{2}c_1 w(x_{\text{mid}})^2 + \frac{1}{4}c_2 w(x_{\text{mid}})^4$$
(3)

For the general solution of any beam problem, the displacements can be assumed to take the below form

$$w(x,t) = \psi_1(x)q_1(t) + \psi_2(x)q_2(t) + \cdots + \psi_i(x)q_i(t)$$
(4)

where q_i is generalized coordinates, and $\psi_i(x)$ is the admissible beam functions which satisfy fixed-fixed beam boundary conditions and is defined as follows [13]:

$$\psi_i(x) = \cosh(\beta_i \frac{x}{L}) - \cos(\beta_i \frac{x}{L}) - \nu[\sinh(\beta_i \frac{x}{L}) - \cosh(\beta_i \frac{x}{L})]$$
(5)

where the first mode

$$\nu = \frac{\sinh(\beta_i L) + \sin(\beta_i L)}{\cosh(\beta_i L) - \cos(\beta_i L)}$$

is taken for displacements of beam i.e.

$$w(x,t) = \psi_1(x)q_1(t) \qquad (6)$$

Table 1. System parameters

A	16e-4 m^2	***L***	10 m
E	200e9 N/m^2	$\beta_1 L$	3.027
I_y	2.133e-7 m^4	$\beta_2 L$	6.085
F_o	1000 N	ρ	8000 kg/m^3

function a = is the permanent magnet width	n = is the normal vector $\gamma 1,2,3$=is constants

2. Mathematical models

2.1 Model of supported beam

The beam of fixed-fixed ends which is integrated with a nonlinear vibration isolator at the middle and subjected to a harmonic moving load, is shown in Fig. 1. The vibration isolator consists of cubic nonlinear spring and damper. The equation of motion of the beam is derived base on the energy methods and Lagrange concept along with the applying assumed–mode method as follows:

The kinetic energy KE of the beam can be represented as [13],

$$KE = \frac{1}{2}\rho A\int_0^L \left(\frac{\partial w}{\partial t}\right)^2 dx \qquad (1)$$

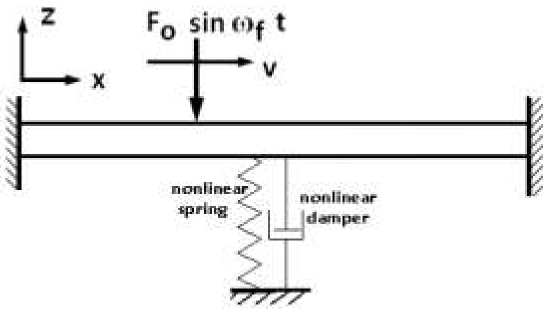

Fig 1. A schematic diagram of fixed-fixedbeam when a harmonic moving load is applied.

The potential energy *PE* of the system and the damping energy DE can be written as [13], damper elements are not appear. After deep searching, the actuator which was adopted from reference **[9]** is considered as the better choice for this work. It is operated in force control mode and controlled by the actuator current which is proportional to the actuator force. The main dimensions for the electromagnetic actuator as well as the electromagnet and a permanent magnet coordinates are illustrated in Fig. 3. According to references [9],[14] the representation of total densities for magnetic flux at any random point (x, y, z) that induced because of the effects of an input current which flow through a rectangular coil can be written as

$$B_x(x,y,z,I) = B_{fx}(x,y,z,I) - B_{mx}(x,y,z)$$
(8)

$$B_y(x,y,z,I) = B_{fy}(x,y,z,I) - B_{my}(x,y,z)$$
(9)

$$B_z(x,y,z,I) = B_{fz}(x,y,z,I) - B_{mz}(x,y,z)$$
(10)

Also, the main assumption is that both of thickness and width of rectangular coil are the same length in the x and y directions. The overall magnetic force vector in the magnetic field is formulated by adopting Maxwell's stress tensors [14].

$$T = \begin{bmatrix} T_{xx} & T_{xy} & T_{xz} \\ T_{yx} & T_{yy} & T_{yz} \\ T_{zx} & T_{zy} & T_{zz} \end{bmatrix} = \frac{1}{\mu_0}\begin{bmatrix} \frac{B_x^2-B_y^2-B_z^2}{2} & B_xB_y & B_xB_z \\ B_yB_x & \frac{-B_x^2+B_y^2-B_z^2}{2} & B_yB_z \\ B_zB_x & B_zB_y & \frac{-B_x^2-B_y^2+B_z^2}{2} \end{bmatrix}$$
(11)

Substituting equation (6) into equations (1, 2 and 3) and applying Lagrange's equation, the equation of motion for the beam is formulated as,

$$m\ddot{q}_1 + ac_1\dot{q}_1 + bc_2\dot{q}_1^3 + (k+dk_1)q_1 + ek_2q_1^3 = Q[F_0\sin(\omega_f t)] \quad (7)$$

Where

$$a = d = \psi_1\left(\frac{L}{2}\right)^2, \quad b = e = \psi_1\left(\frac{L}{2}\right)^4$$

$$k = EI_y\int_0^L \left\{\frac{\partial^2\psi}{\partial x^2}\right\}^2 dx, \quad m = \rho A\int_0^L m\psi_1^2 dx$$

$$Q = \cosh\left(\beta_i\frac{Vt}{L}\right) - \cos\left(\beta_i\frac{Vt}{L}\right) - \nu\left[\sinh\left(\beta_i\frac{Vt}{L}\right) - \cosh\left(\beta_i\frac{Vt}{L}\right)\right]$$

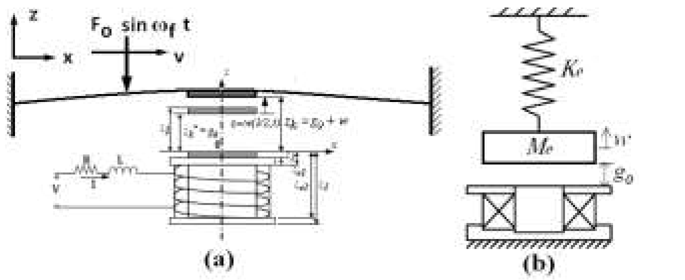

Fig 2. Graphical diagram of the system. (a) The system excited by an electromagnetic actuator at a midpoint. (b) The system in equivalent shape.

The numerical values of parameters employed in the calculation are listed in Table 1.

2.2 Model of actuator

The electromagnetic actuator, EMA, is integrated into the system, and positioned at the midpoint of the beam as shown in Fig. 2 where the spring position of the air gab between these permanent magnets is given by $z_k^* = g_0$ which represents the equilibrium position. The value of input current $I = 0$ represents the static equilibrium. From equation (14), the consequent actuating force is given by

$$F_z(z_k^*) = F_s(z_k^*)$$
(15)

When the input current is non-zero, the control force is defined as

$$F_c(z_k, I) = F_z(z_k, I) - F_s(z_k^*)$$

$$= \alpha_2(z_k)I^2 + \alpha_1(z_k)I + [F_s(z_k) - F_s(z_k^*)]$$
(16)

The air gaps between the two permanent magnets and input currents are the main parameters that the forces produced by the actuator are dependent on. According to the results obtained from calibration of the magnetic force, it is establish that the term $\alpha_2(z_k)I^2$ is very small comparing to the other terms appears in Equation (16), in particular when the value of current is very small so it can be ignored.

$$F_c(z_k, I) \approx \alpha_1(z_k)I + [F_s(z_k) - F_s(z_k^*)]$$

$$\approx \gamma_1 z_k + \gamma_2 I + \gamma_3 w I$$

$$\approx \gamma_1 g_0 + \gamma_1 w + (\gamma_2 + \gamma_3 g_0)I + \gamma_3 w I$$
(17)

According to reference [9], the numerical values of parameters

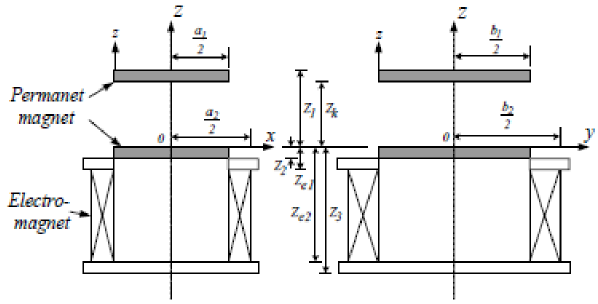

Fig 3. The coordinates of adopted EMA [9].

The electromagnetic actuator interactive force is expressed as [9],

$$F_z(z_k, I) = \iint_a (T_{zx}\hat{n}_x + T_{zy}\hat{n}_y + T_{zz}\hat{n}_z)da$$
(12)

The components of the normal vector are $n_x = n_y = 0$ because of the magnetic field is symmetrical about the z-axis. Replacing these into Equation (12) gives

$$F_z(z_k, I) = \frac{1}{2\mu_0}\int_{-\frac{a_1}{2}}^{\frac{a_1}{2}}\int_{-\frac{b_1}{2}}^{\frac{b_1}{2}}[B_z^2(x, y, z_k, I) - B_x^2(x, y, z_k, I) - B_y^2(x, y, z_k, I)]dxdy$$
(13)

Substituting Equations (8, 9 and10) into Equation (14), the actuating force has the form;

$$F_z(z_k, I) = \alpha_2(z_k)I^2 + \alpha_1(z_k)I + F_s(z_k)$$
(14)

As shown in Fig. 2 (a), the two permanent magnets are distributed into two positions, one of them is linked to the electromagnet while the other one is linked at the mid of the beam. The calculates the reaction according to the rate at which the error has been changing. The signal of the controller (h), which represents the output signal, will sent to the plant and the new output will obtained. This new output will be feed back to the sensor again to sense the new error signal (e). The PID controller computes the derivative and integral of the new error signal again.

There are many approaches for regulating a controller of type PID. The most useful approach normally involves the improve-ment of some form of process model, and then selecting the constants terms, P, I, and D, according to the parameters of the dynamic model. To get the desired response, the following procedure can be adopted as a common rule: 1) to get better rise time, a proportional element term is added, 2) to enhance the overshoot, a derivative element term is added, 3) to reduce steady-state error, an integral element term is added. In this study, the PID controller was tuned by implementing the pole placement approach and the optimization capabilities of MATLAB which makes use of the gradient descent method, for more details see [11].

3. Simulation

The Matlab/Simulink environment was used to perform the simulations in this study. Fig. 4 demonstrates the Simulink represent-ation of the control scheme strategy for the controlled constant are substituted in equation (17) which becomes as equation (18) and it used for calculating actuator force in the current work;

$$F_c(z_k, I) = 6.40 \times 10^{-3} z_k + 1.6 \times 10^{-2} I + 0.546 z_k I \quad (18)$$

where:

$z_k = g_0 + w(l/2, t)$,
$g_0 = 3.2 \times 10^{-2}$ m

2.3 PID Controller model

The PID controller is the controller that used to minimize the differences between the desired set points and the measured variable by correcting the error [19]. This type of controller is adopted in the current study due to its easy practical usage as well as the simplicity of structure in the control loops. The PID controller is represented by:

$$h(t) = K_p e(t) + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt}, \quad (19)$$

Where the constants $K_p, K_i, \text{and} K_d$ represent the proportional term, integral term and derivative term respectively and e is the position error. Each of these constant has specific role, the proportional gain calculates the reaction to the current error, the integral term calculates the reaction according to the sum of up to date errors, and the derivative term the dynamic performance of the beam system. The parameters, given in Table I, were used during the simulation. The PID controller was used as for the controlled system and it was compared and evaluated against an open loop system.

The simulation of the uncontrolled and controlled schemes is performed with no change in all the conditions and the significant parameters are used to satisfy the fair and realistic comparison.

system. While the scheme of uncontrolled system is the same as in Fig. 4 but not included the electromagnetic actuator and the controller as well. The simulation consists of fixed–fixed beam dynamic system, electromagnetic actuator dynamic system, PID controller, and optimization block included the constraints. It also has one output which is the beam deflection, x_{mid}, before and after controlling which act as the major parameter of responses or concern to be analyzed and rated for

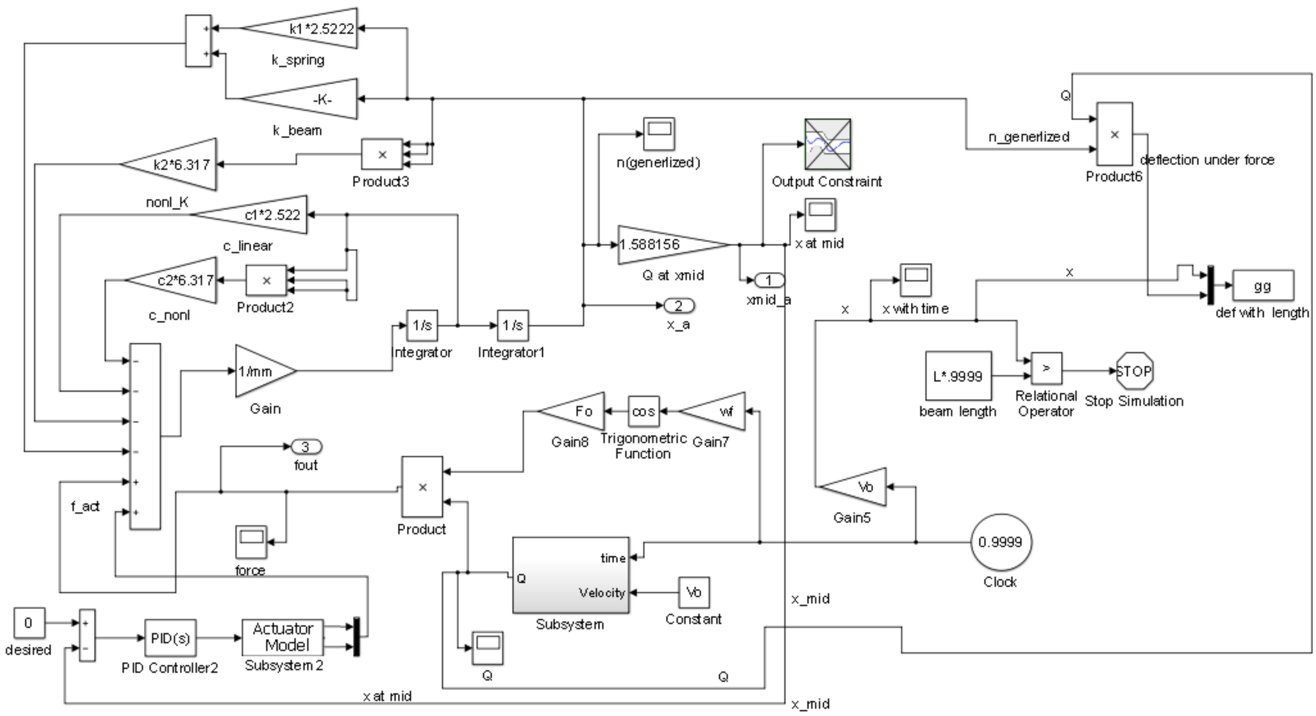

Fig 4. PID Controller schemes MATLAB simulink.

to increase the moving load velocity. This is because when performing that, the frequencies of external forces will approximately have no effect (can be ignored).

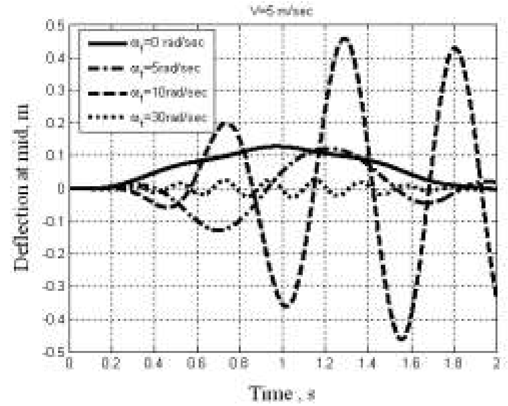

Fig 6. The time vibration response of uncontrolled system with different external force frequencies and moving load =5 m/s.

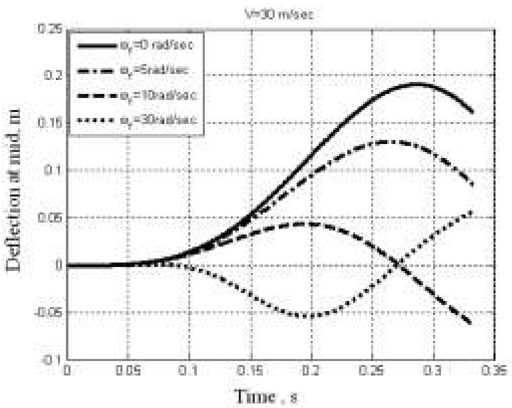

Fig 7. The time vibration response of uncontrolled system with different external force frequencies and moving load =30 m/s.

4. Outputs and discussions

4.1 The uncontrolled system perform-ance analysis

In this section, the effect of velocity and forced frequency is discussed. The time responses are shown in Figs. 5-7. Fig. 5 shows a typical case when considering the load as being fixed at mid distance of beam and its magnitude is changing as sine wave function. It is clear that when the value of forced frequency is close to the natural frequency of the beam which has value of 12.9 rad/sec, the displacement response is increased. Moreover, when the linear velocity of moving load increased the amplitude at resonance will be suppressed as shown in Figs. 6 and 7.

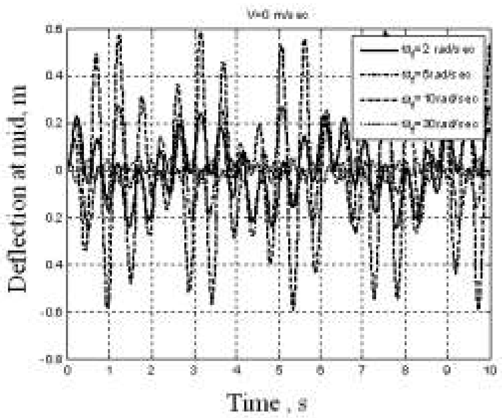

Fig 5. The time vibration response of uncontrolled system with different external force frequencies and no moving load.

The results of this study indicate that in some applications of beams and when it is the necessary to apply moving load which may be near from the first mode, it could be applicable work. It is clear that the peak response is also efficientlysuppressed. According to the maximum peak values, the controlled vibration deflection performance showed an enhancement of about 90.1%. The outputs noticeably demonstrate the effort of the proposed controlling system.

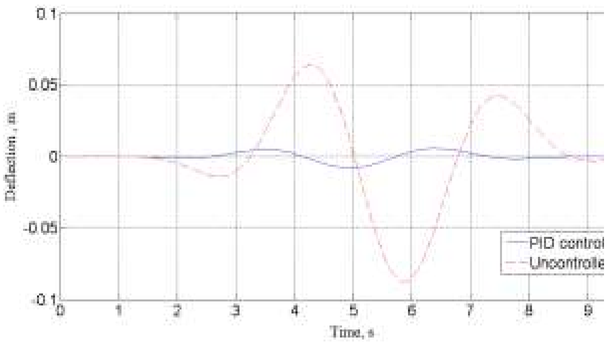

Fig 8. Vibration response of uncontrolled system vs controlled system in time domain.

Fig. 9 shows the performance of both uncontrolled and PID controlled in the frequency domain. It is obvious that the value of the first mode frequency for the beam is around 12.9 rad/sec. From Fig. 6, not only the PID controller is very significant to reduce the vibration of beams but also it indicated that the control force in the PID controller is somewhat higher than that in the external forces subjected on the beam (see Fig. 10). It can be concluded also that a new value of the peak is around 132.8 rad/sec (see Fig. 9) which produced by the PID controller. In some

4.2 The controlled system performance analysis

The PID controller of the system model is tuned first and after that it is optimized to obtain very adequate performance over the corresponding operating region. The pole placement approach is used to obtain the optimal output feedback PID gains. The gradient descent optimization method is used for further tuning. The proposed objective function, F_{obj} , can be written as follows,

$$\min(F_{obj} = \sum |\dot{q}_i|), \quad (20)$$

Equation (20) is derived under the application of a constraint represented by the beam deflection, x_{mid}, which is less than 0.001 m. Therefore, the optimal gains of PID controller are given in Table 2.

Table 2. Optimal gainsof PID controller

P $\times 10^3$	I $\times 10^3$	D $\times 10^3$
52.6086	-50.2914	1.563
where the gains evaluated when velocity of moving load, V, is 10 m/s while forced frequency, w_f, is 20 rad/s.		

Fig. 8 illustrates the simulated vibration displacement response in the time domainfor both the PID controlled system and the open loop system. It can be seen that PID controller can eliminate the vibration efficiently, and the PID controller performing a significant role in this applications, the structure under control may be detrimental. However, the uncontrolled system does not have this drawback.

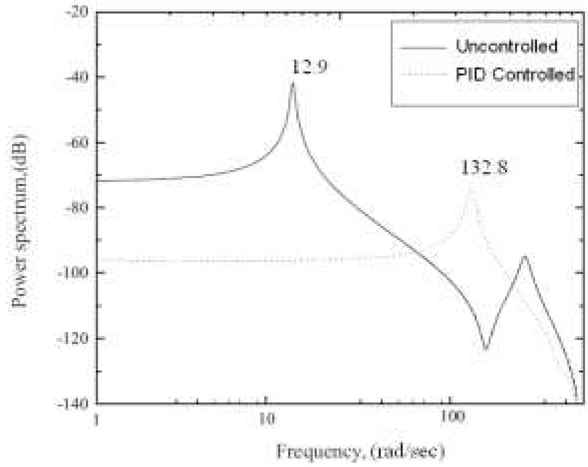

Fig 9. Vibration response of uncontrolled system Vs controlled system in frequency domain.

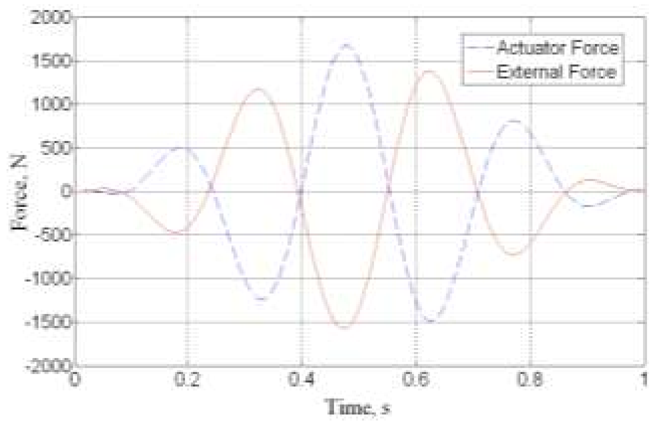

Fig 10. Actuator force provided by electro-magnetic actuator when the beam subjected to external force.

5. Conclusion

This research studied the vibration control of beams affected by non-stationary load when the two ends of the beams were clamped. It is a very important issue in the field of vibration control that related to flexible structures. The velocity effect of the dynamic load and its frequency on the displacement response of fixed-fixed beam was investigated. Mechatronic actuator was integrated at the mid of the beam in addition to nonlinear spring damper system existence. General method was used for tuning the controller gains and an optimization method was combined to solve this problem efficiently. The benefits and drawbacks of the controller were extensively investigated in this study. It can be concluded that the PID controller has significant effect in reducing the vibration of the beam when affected by dynamic load. In addition, the controller can reduce the peaks of deflection efficiently when the dynamic mass is applied on the beam. Moreover, the PID controller may provide little bit higher frequency vibration which can be considered as being not useful to the structure in some applications. In future investigations it might be possible to use different schemes of controller for improving some disadvantages of current scheme. An experimental work also suggested.

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السيطرة على اهتزازات عتبة مثبتة الاطراف متاثرة بحمل ديناميكي بواسطة مشغل كهرومغناطيسي

م.د. وجدي صادق عبود

قسم هندسة الاطراف والمساند الصناعية

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الخلاصة

في هذا البحث، تم فحص السلوك الديناميكي لعتبة مثبتة من طرفيها ومستقرة على مخمد صدمات لاخطي في منصفها ومعرضة الى حمل ديناميكي.

يتكون مخمد الاهتزازات من نابض ومخمد لاخطيين. تم نمذجة العتبة كعتبة اويلرولايجاد المعادلات اللاخطية للحركة، استخدامت طريقة فرض النسق. تم تعريض العتبة على طول امتادها الى القوة متحركة بمختلف السرع الخطية ، بالاضافة الى ان قيمة هذه القوة تتغير في القيمة والاتجاه. تم مناقشة تأثير هذه السرع الخطية وتردد القوة الديناميكية على عزل العتبة .

علاوة على ذلك ،تم دمج مسيطر من نوع التناسبي التفاضلية التكاملي(PID) مع مشغل كهرومغناطيسي مع العتبة لتقليل الانحرافات الحاصة في العتبة .

تم الحصول على الثوابت للمسيطرنوع (PID) ذو التغذية العكسية عن طرق استخدام طريقة تثبيت الاقطاب ومن ثم اجريت عملية التوليف بعد ذلك باستخدام طريقة التدرج النسبي للحصول على الثوابت المثلى. اظهر المسيطر المقترحة تخفيض مميز لانحناء العارضة عند مقارنتها مع نتائج انحناء العارضة بدون مسيطر