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ACTIVE VIBRATION SUPPRESSION OF A SMART BEAM VIA PID CONTROLLER DESIGNED THROUGH WEIGHTED GEOMETRIC CENTER METHOD

Abdullah Turan[1]
Inonu University,
Malatya, Turkey

Cem Onat[2]
Firat University,
Elazig, Turkey

Melin Sahin[3]
Middle East Technical University
Ankara, Turkey

ABSTRACT

In this study, the effectiveness a PID controller designed through weighted geometric center (WGC) method is investigated in order to actively attenuate the free and the forced vibrations of a smart beam comprising piezoelectric sensor and actuator pairs. In practice, the smart beam is an aluminum beam with a Lead Zirconate-Titanate (PZT) patches attached to the surfaces in the bimorph configuration and fixed from a single end and the other end is free-standing. First the frequency response of the beam in the range covering the first resonance frequency is experimentally obtained. Then, a PID controller based on the weighted geometric center method is designed and applied to the obtained system model. The experimental results indicated that, the suppression of the vibration actively around the first resonance frequency of the smart beam is a viable one through the proposed controller.

INTRODUCTION

The concept of smart structures has started a new structural revolution. A smart structure typically consists of a host structure incorporated with sensors and actuators coordinated by a controller. The integrated structure system is called a smart structure because it has the ability to perform self-diagnosis and adapt to the environment change. The recent developments in sensor and actuator technologies allows the usage of smart structures in the active vibration control of light and flexible structures; especially in aeronautical structures. The piezoelectric materials can serve both as sensor and actuator. The PZT (Lead-Zirconate-Titanate) piezo ceramic smart materials are widely used both as embedded within the passive composites and/ or being surface bonded to the passive metals and/ or composites in order to form a smart structure [Onat, et al. 2017]. For active vibration control, the design of piezoelectric smart structures needs both the structural dynamics and control theories to be considered.

The research for the active vibration control of smart beams can be broadly classified into three main categories as classical PID, adaptive and robust controller [Ros, et al. 2015]. High performance requirements, uncertainties in the system model and different operating

[1] Post Graduate Researcher, in the Department of Mechanical Engineering, Email: abdullah.turan@inonu.edu.tr
[2] Assoc. Prof. Dr. in the Department of Airframe and Power-plant, Email: conat@firat.edu.tr
[3] Assoc. Prof. Dr. in the Department of Aerospace Engineering, Email: msahin@metu.edu.tr

conditions require more effective control algorithms such as adaptive ones [Fadil, et al. 2013; Onat, et al. 2011; Onat, et al. 2013; Saad, et al. 2012; Shouwei, et al. 2010 and Zoric, et al. 2014]. The adaptive control architecture on the other hand has its own requirements such as the application of an adaptation scheme which makes the controller a complicated one to design. If one would like to avoid the performance losses due to the system and the modeling uncertainties, then he inevitably resort to use robust control methodologies. Robust control can appropriately handle the parameter uncertainties and nonlinearities in the system model as well. Nestorovic and Oveisiby proposed a combined H_2/H_∞ approach, experimentally proved the efficacy of their design [Nestorovic and Oveisi, 2015]. The authors were obtained the system model through Finite Element Method (FEM) and successfully were suppressed the vibration level at the first resonance frequency of the beam by 20 dB by using a laser displacement sensor in the experiments. Sridevi and Madhavasarma designed an H_∞ controller and compared with an LQG (Linear Quadratic Gussian) controller [Sridevi and Madhavasarma, 2010]. In their experiment, they used PZT patches. They showed that the developed controller was efficient at high frequencies when the closed loop responses were considered. They also showed that an H_∞ controller gave better performance as compared to an LQG controller. Omidi and Mahmoodi designed and applied controller for the active vibration control of smart structures [Omidi and Mahmoodi, 2014]. In the study, H_∞ MPPF (H_∞ Modified Positive Position Feedback) and H_∞ MPVF (H_∞ Modified Positive Velocity Feedback) designs were implemented where PZT patches were used as actuators. It is also shown that an H_∞ MPVF controller performs superior to an H_∞ MPPF one. The impressive performance of this type of controller is due to the parallel compensator used in addition to the H_∞ controller. In another study, Onat et al. have designed and implemented gain scheduling H_∞ controller for a smart beam [Onat, et al. 2017]. The proposed control architecture depends on the programming parameter. This is a visionary study for systems where parameter change in the system is known or predicted continuously. However, the controllers designed in the mentioned studies either require very complex control and / or adaptation architectures or the synthesized controllers have a high degree.

On the other hand, PID controllers are low-order controllers which have been simply used for decades to control higher-order processes. Industries attracted to used PID controller because it has low cost, easy to maintain, simplicity in control structures and easy to understand. Fadil et al. implemented common PID controller and PID tuned by iterative learning algorithm (ILA) [Fadil, et al. 2013]. PID-ILA gave higher performance than common PID in term. Rahman and Alam applied the PID controller to reduce the vibration of different beams using PZT. They demonstrated the validity and efficiency of the PID controller with the experimental results obtained using the active vibration control system [Alam and Rahman, 2010]. Khot et al. also obtained similar results [Khot, et al. 2012]. Chhabra et al. performed simulation studies about active vibration control with piezoelectric actuator and sensor layers integrated to different parts of the beam. The structure was modeled using finite element method (FEM) and state-space techniques. It has shown that the numerical simulation by applying the PID controller can provide adequate vibration control by the proposed method [Chhabra, et al. 2012]. Kumar et al. used PID control as a medium to suppress the vibration of beam by finding the optimal placement of piezoelectric sensor/actuator on surface of a beam [Kumar et al. 2014].

In the Department of Aerospace Engineering at Middle East Technical University, there are various theoretical and experimental structural model characteristics and active vibration control studies [Sahin, et al. 2008]. It has been shown that PZT patches work effectively in active control of vibrations of the smart beam in controller applications where they are used as stimulators and sensors. Several controllers have also been designed for active vibration damping on the smart beam. These controllers are $H\infty$ [Yaman, et al. 2003], PID [Onat, et al. 2010], LQG [Onat, et al. 2011], CFE [Onat, et al. 2011], LPV [Onat, et al. 2011] and LQR [Akin and Sahin, 2015].

In the PID control applications mentioned so far, there is no PID design procedure developed specifically for smart structures, and their vibration suppression performances are worse than controllers in which they are compared. More efficient design procedures are needed to set PID controller parameters.

One of the current studies on setting PID control parameters has been proposed by [Ozyetkin, et al. 2018]. In the study, the weighted geometric center and stability boundary curve method has been recommended for PID controller design. For the first time, the weighted geometric center concept proposed by Onat for PI control of time-delayed systems is based on the time line of calculating the region of PI controller parameters that makes it stable [Onat, 2013]. The concept of predominantly geometric center has so far been used in successful applications in PI / PID design [Onat, et al. 2012; Onat, 2013; Onat, et al. 2017; Onat, 2018; Ozyetkin et al. 2018 and Ozyetkin et al. 2019].

In this study, an original PID controller design procedure for suppressing vibrations in the first the mode of a smart beam was presented. In the first stage of the proposed procedure, the PID controller was designed with the method introduced in [Ozyetkin, et al 2018], considering a very small time delay for the beam model whose anti-resonance dynamic was omitted. In the second step, the neglected anti-resonance dynamics is taken into account in the PID controller. In the last stage, the gain of the obtained controller is increased up to the upper limit output value of PZT patches used as actuator. The experimental results show that the PID controller designed by the proposed procedure exhibits very good vibration suppression performance.

Smart beam

The smart beam (Figure 1a) consists of an aluminum material with a size of 350x30x2 mm, equipped with symmetrically oriented BMP500 type PZT patches (Figure 1b), 25.37x25.37x0.50 mm in size and located as two on the front and two on the back surface configuration. These piezoelectric patches are entitled as A1, A2, D, and S according to their intended use (Figure 1a). A1 and A2 are used as control piezoelectric patches (i.e. actuators) and are connected in bimorph configuration S stands for the sensor and D represents the disturbance given to the piezoelectric patches. Figure 2 shows the experimental setup of the system.

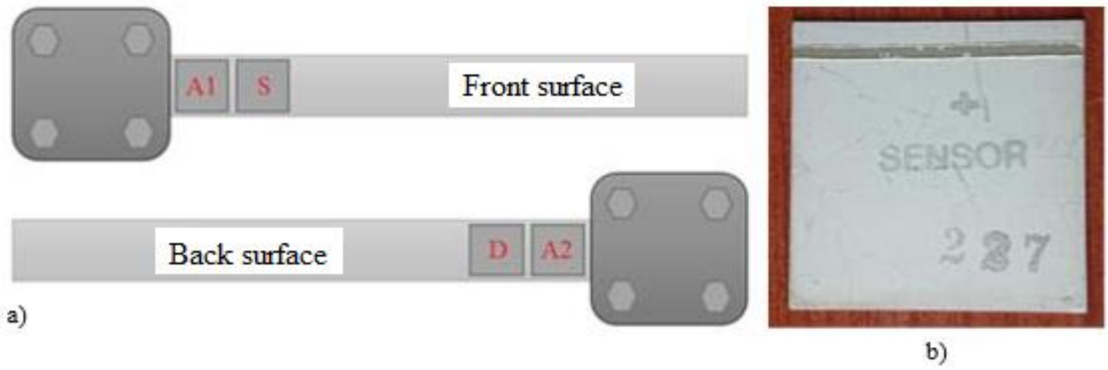

Figure 1: a) Labels of the PZT patches, b) PZT patch (BMP500) [Akın,2015]

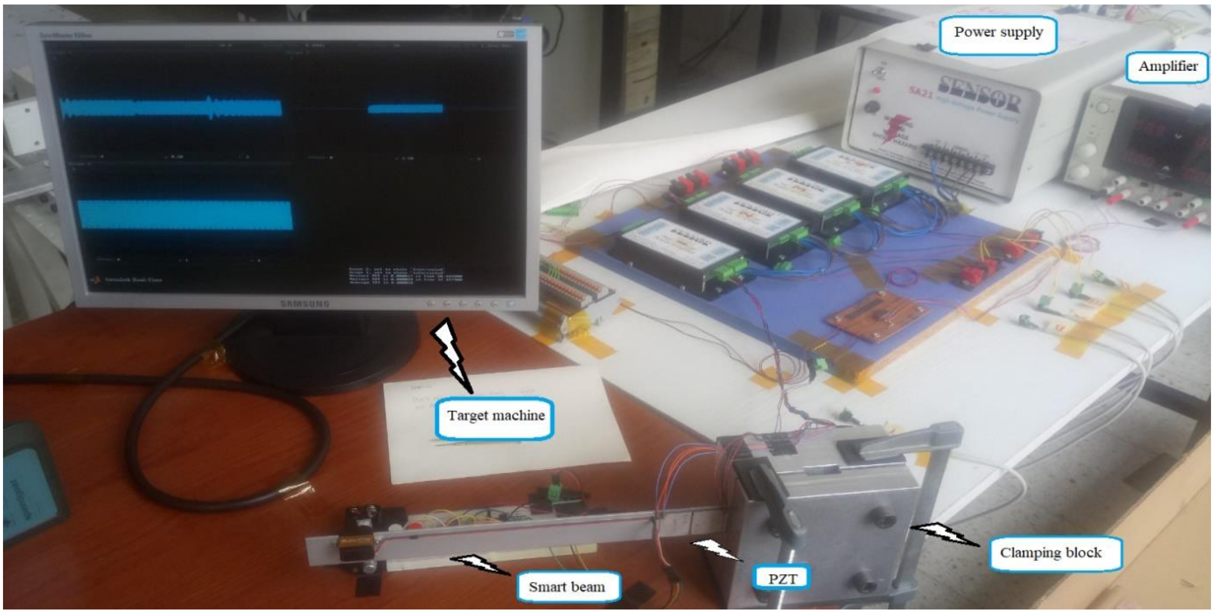

Figure 2: Experimental setup of the system

The piezoelectric patch D is stimulated by exponentially increasing sine waves in the frequency range of 5 Hz to 30 Hz and the response of the system is measured with the S piezoelectric patch which is used as a sensor. This input-output data pair is than converted to the frequency response function using the Fast Fourier Transform (FFT) using the MATLAB program. According to the experimental data, the second order transfer function is obtained as below in Eq. (1)

$$G_p(s) = \frac{N_P(s)}{D_P(s)} = \frac{0.0169s^2+0.4183s+283.0299}{s^2+1.6754s+8175.1} \tag{1}$$

where $G_p(s)$ represents the transfer function of the smart beam.

Figure 3 shows the experimental and adapted analytical frequency response function of the smart beam. Accordingly, it is seen that the adapted second order transfer function fits with great precision, especially in the resonance region.

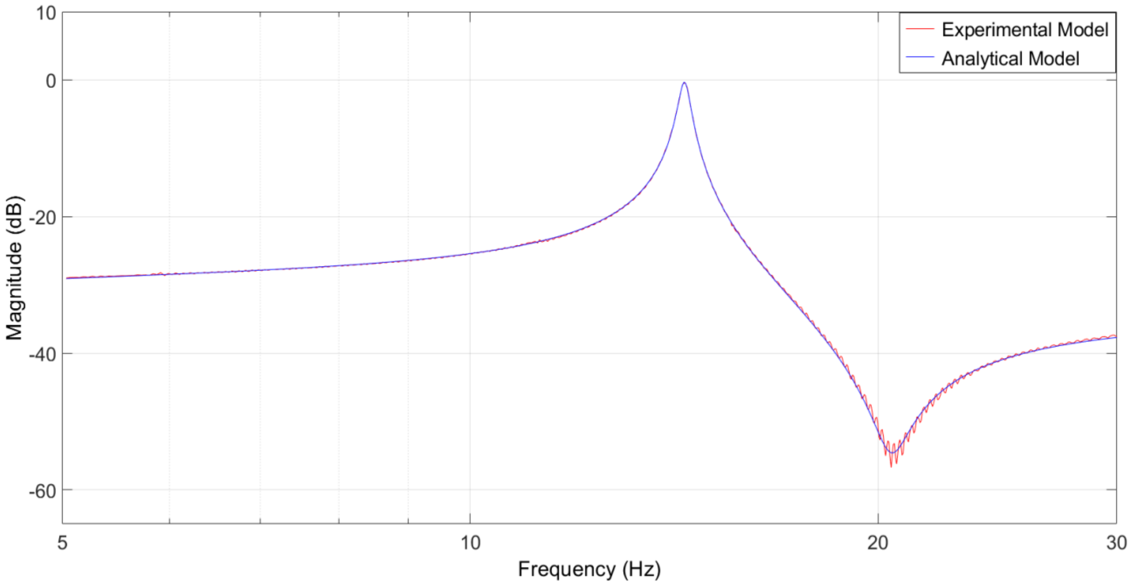

Figure 3: Frequency response of experimental and analytical models of smart beam

PID CONTROLLER DESIGN

The PID control system is shown in Fig. 4. Smart beam transfer function $G_p(s)$ ignored anti-resonance dynamics is defined as follows.

$$G_P(s) = \frac{N_P(s)}{D_P(s)} e^{-\tau s} = \frac{\mathbf{1}}{\mathbf{s^2+1.6754s+8175.1}} e^{-0.01s} \tag{2}$$

$C_{PID}(s)$ refers to the transfer functions of the PID controller. It is defined by Eq. (3).

$$C_{PID}(s) = \frac{N_{PID}}{D_{PID}} = \frac{k_p s + k_d s^2 + k_i}{s} \tag{3}$$

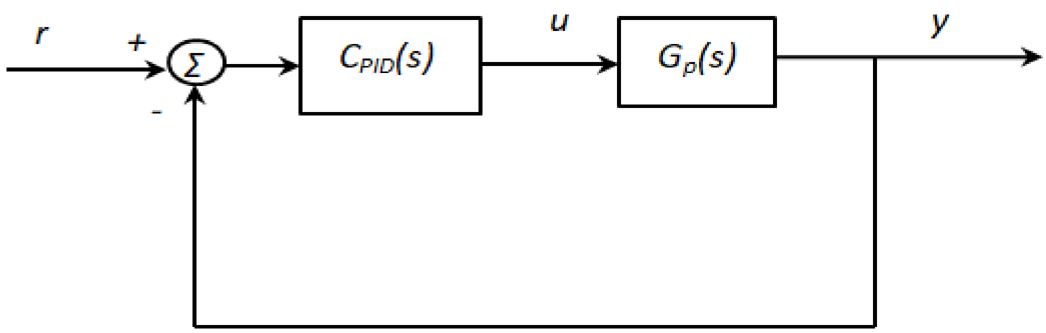

Figure 4: PID control system

Here, k_p, k_d, k_i and τ are proportional, derivative and integral gains of the PID controller and dead time of the system respectively. The closed loop characteristic equation (Δ) is given in Eq. (4).

$$\Delta(s) = D_P(s)D_{PID}(s) + N_P(s)N_{PID}(s)e^{-\tau s} = 0 \tag{4}$$

In the design, first, stabilizing parameters of C_{PID} are computed for the closed loop. In this way, the stability region is graphically determined in k_p–k_i plane according to derivative gain k_d. For this purpose, substituting Eq(5) in Eq. (4), one can obtain

$$s = j\omega, \;\; e^{-\tau s} = \cos(\tau\omega) - j\sin(\tau\omega) \tag{5}$$

$$\Delta(j\omega) = D_P(j\omega)D_{PID}(j\omega) + N_P(j\omega)N_{PID}(j\omega)(\cos(\tau\omega) - j\sin(\tau\omega)) = 0. \tag{6}$$

Decomposing Δ into its real and imaginary parts, one can write

$$\Delta = R_\Delta + jI_\Delta = 0 \tag{7}$$

Here, R_Δ and I_Δ are functions of k_p, k_i, k_d and ω. Equating real and imaginary parts of Δ to zero, two equations with three unknown parameters (k_p, k_i and k_d) are obtained. The equation system is given in Eq. (8). This equation system can be solved for derivative gain k_d.

$$\begin{aligned} R_\Delta(k_p, k_d, k_i, \omega) &= 0 \\ I_\Delta(k_p, k_d, k_i, \omega) &= 0 \end{aligned} \tag{8}$$

Solving the equation system depending on the frequency (ω) for the different values of k_d, and then plotting obtained k_p and k_i parameters in k_p–k_i plane, stability region is determined. The details can be found in [Onat, et al. 2012; Onat, 2013].

Remark

If more than one real value of ω which satisfies Eq. (8) exist, then frequency axis can be divided into a finite number of divisions and by testing each division the stability region can be computed.

Depending on the derivative gain (k_d), the k_{dC} value is selected which maximizes the expansion of the region of the controller parameters that makes it stable in the k_p-k_i plane. The details can be found in [Ozyetkin, et al. 2018]. By calculating the weighted geometric center of the maximized stability area, the k_p and k_i parameters are also determined. The coordinates (k_{pC} and k_{iC}) of the weighted geometric center of the maximized stability zone are calculated using Eq. (9) and (10).

Here, k_{pm} shows the coordinates of the points forming the boundary curve of the stability region with respect to the horizontal axis (k_p axis) and k_{im} shows the coordinates relative to the vertical axis (k_i axis).

$$k_{pC} = \frac{1}{n}\sum_{m=1}^{n} k_{pm} \tag{9}$$

$$k_{iC} = \frac{1}{2n}\sum_{m=1}^{n} k_{im} \tag{10}$$

Once the PID control parameters have been determined, the anti-resonance dynamics of the neglected smart beam in the first stage of the proposed procedure are taken into account as in Eq. (11). Here, k is the controller gain.

$$K(s) = k\frac{C_{PID}(s)}{A(s)} \tag{11}$$

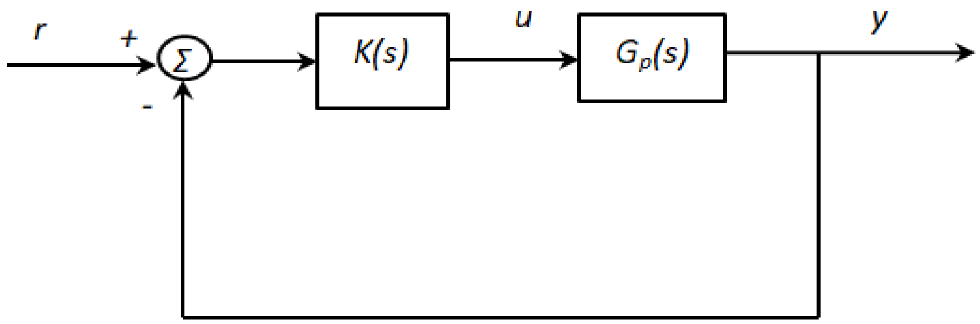

Figure 5: Real time closed loop system

If the proposed design procedure is run by considering the smart beam model given in Eq. (1) and the dead time of τ = 0.01 s, the region of the controller parameters which makes the stabilization dependent on derivative gain k_d is obtained as in Figure 6.

Ozyetkin, et al. have shown that the k_d value maximizing the stability region of the derivative gain along the horizontal axis (k_p axis) and the weighted geometric center parameters of the corresponding stability region provide better performance [Ozyetkin et al, 2018]. When the k_d = 60 value of the derivative gain is exceeded, it is seen that the stability region narrows with respect to the horizontal axis. For this, the value of the derivative gain was chosen as k_d = 60. The stability region for k_d = 60 and its weighted geometric center were given in Figure 7.

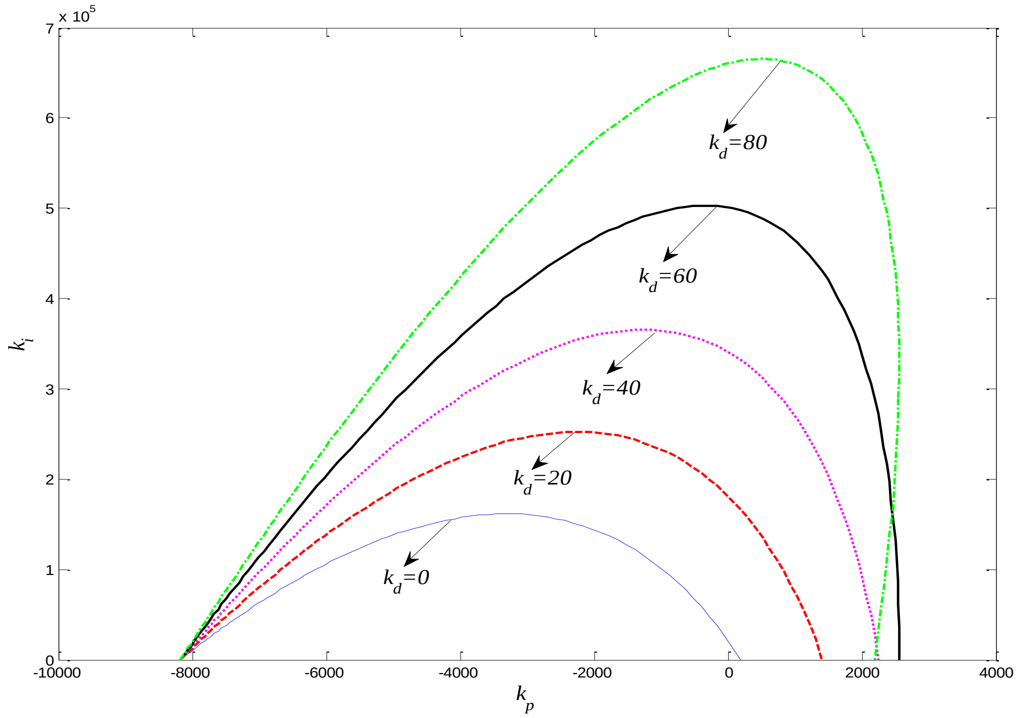

Figure 6: Regions of stable controller parameters according to derivative gain

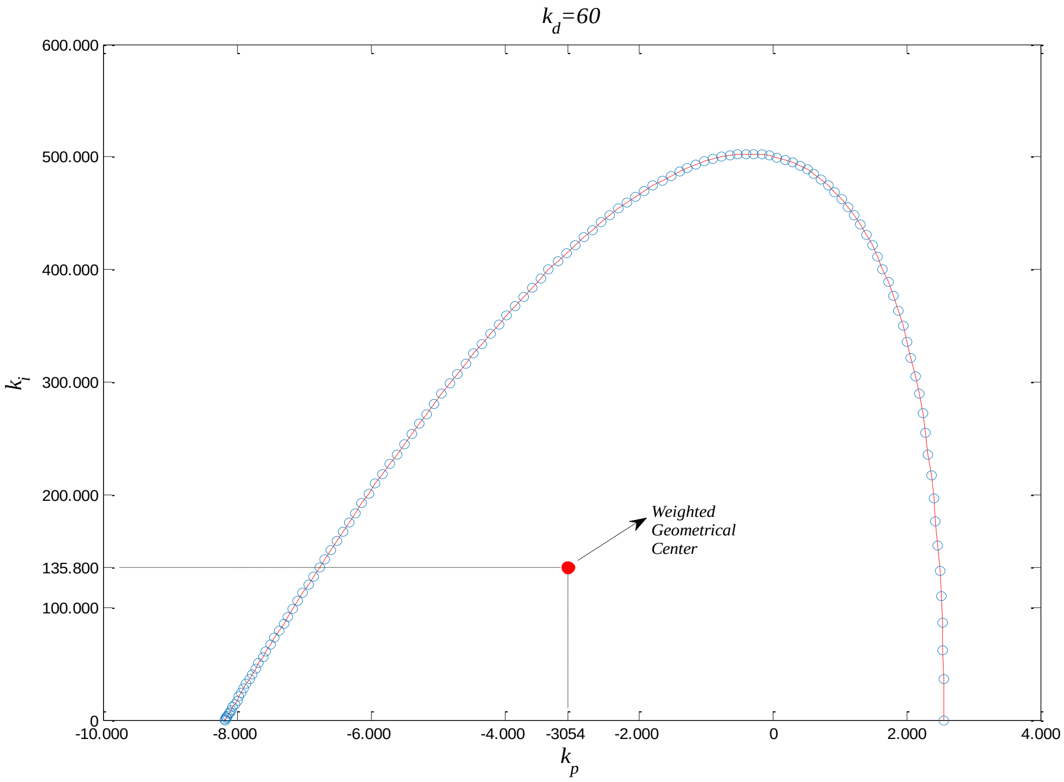

Figure 7: k_d = 60 which makes it stable for the k_p-k_i controller parameters region and its weighted geometric center

Accordingly, the parameters of the proposed controller were calculated as k_p = -3054, k_i = 135800 and k_d = 60. The transfer function of the designed controller was given in Eq. (12).

$$K(s) = \frac{k_p s + k_d s^2 + k_i}{s\,A(s)} = \frac{-3054s + 60s^2 + 135800}{s\,(0.0169\text{s}^2 + 0.4183\text{s} + 283.0299)} \quad (12)$$

The weighted geometrical center point of this region is computed by using coordinates of the boundary points. The superiority of the WGC method over other methods (such as; the Ziegler-Nichols tuning, the Astrom-Hagglund auto tuning) [Ho, et al. 1996; Ho, et al. 1997] is the numerical calculation of the control parameters without any optimization.

EXPERIMENTAL RESULTS

Figure 8 shows the experimental response of the smart beam in the case of free vibration. It can easily be observed from the figure that the settling time is around 6.5 seconds for the open loop system. On the other hand, the designed controller managed to suppress the free vibrations of the smart beam with a settling time of nearly 0.8 seconds.

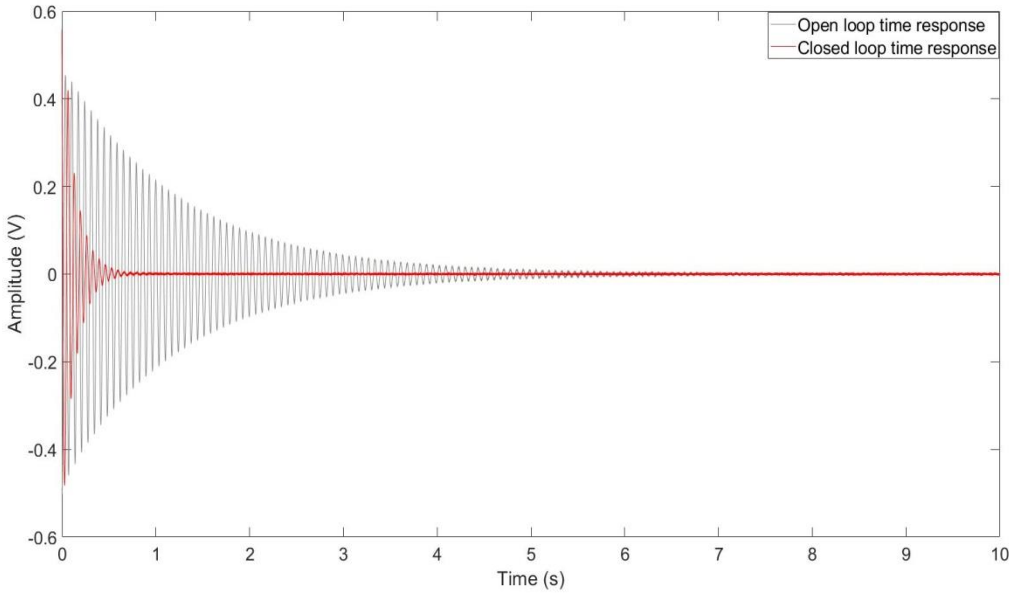

Figure 8: Experimental response of the smart beam: Free vibration case

Figure 9 shows the experimental response of the smart beam in the case of forced vibration at the first resonance frequency. The forced vibration suppression performance of the PID controller is defined in Eq. (11) as approximately 82 % as it can be seen in Figure 9. So, the designed controller proves to be very effective on suppressing the free and forced vibration of the smart beam.

$$\text{Suppression rate} = \frac{[\text{Open loop magnitude}]_{\text{max}} - [\text{Closed loop magnitude}]_{\text{max}}}{[\text{Open loop magnitude}]_{\text{max}}} * 100 \quad (11)$$

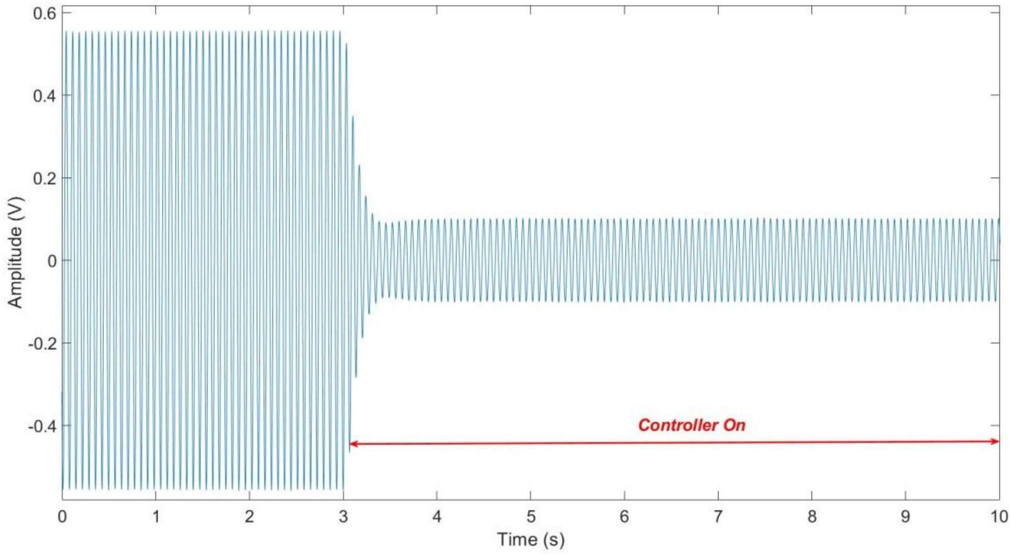

Figure 9: Experimental response of the smart beam: Forced vibration case at the first resonance frequency

CONCLUSION

As a result of experimental studies with the designed WGC method based PID controller, the proposed controller demonstrates a successful performance and shows experimental verification on the suppression of the smart beam both in the free and forced vibration at the first resonance frequency cases.

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