Proportional-Derivative PD Vibration Control with Adaptive Approximation Compensator for a Nonlinear Smart Thin Beam Interacting with Fluid

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***Abstract*—This work is concerned with the vibration attenuation of a smart beam interacting with fluid using proportional-derivative PD control and adaptive approximation compensator AAC. The role of the AAC is to improve the PD performance by compensating for unmodelled dynamics** using **the concept of function approximation technique FAT. The key idea is to represent the unknown parameters using the weighting coefficient and basis function matrices/vectors. The weighting coefficient vector is updated using Lyapunov theory. This controller is applied to a flexible beam provided with surface bonded piezo-patches while the vibrating beam system is submerged in a fluid. Two main effects are considered: 1) axial stretching of the vibrating beam that leads to the appearance of cubic stiffness term in beam modelling, and 2) fluid effect. Fluid forces are decomposed into two components: hydrodynamic forces due to the beam oscillations and external (disturbance) hydrodynamic loads independent of beam oscillations. Simulation experiments are implemented using MATLAB/SIMULINK to verify the correctness of the proposed controller. Two piezo-patches are bonded on the beam while an impulse force with multi-pulse is applied to excite the beam vibration. The results show the strength of the proposed control structure.**

***Index Terms*—fluid-structure interaction, smart beam, PD control, adaptive approximation control, hydrodynamic forces**

I. Introduction

Flexible structures interacting with fluid play an important role in applications of many fields such as offshore extraction, underwater robotic vehicles, flexible aircraft structures, and resonant cantilevers for probing surface properties or measuring liquid density and viscosity [1-3]. These structures undergo instability and even damage if they are subjected to unwanted loads. Therefore, active vibration control is a potential solution for motion regulation and stability recover. Piezoelectric materials can be integrated with flexible structures for vibration suppression. They behave as actuators and/or sensors making the flexible structure adaptable to external disturbances [4]. However, the vibrating flexible structure still requires a specific control system to stabilize the motion. As a result, this work is focused on active vibration control of a flexible nonlinear beam immersed in a fluid. Several collocated piezo-patches are bonded on the surface of the vibrating beam working as actuators/sensors in the control system. The dynamics of the vibrating beam is required to design a suitable control structure. The governing partial differential equation PDF of the target beam is derived considering three important issues: 1) axial stretching resulted from large deflection, 2) fluid loadings and 3) piezoelectric transducers. The axial stretching results in a nonlinear restoring force with cubic stiffness term that complicates the control task. On the other hand, the piezo-actuator moments are considered as input controls to regulate the beam vibration. What important here is the modelling of the fluid loadings. In fact, two concepts are available to determine the fluid effect [5]:

1. If the flexible beam is constrained from oscillations while there are some incident waves, then the fluid forces can be decomposed into Froude-Krylov and diffraction forces/moment.
2. If there is no incident wave while the flexible beam is enforced to vibrate with fluid oscillations, then the fluid force is decomposed into inertial, damping and restoring forces. The restoring forces are neglected in this work. The inertia force is proportional to the acceleration of the vibrating beam while the damping force is proportional to the velocity of the beam. The added fluid mass makes the resonance frequency decrease while the increase of damping makes the Q-factor decrease [1]. For modelling of the coupled beam-fluid system, the reader is referred to [1-3, 6-13].

Due to the presence of nonlinear restoring force related to axial stretching, the coupled smart beam-fluid system is no longer linear and advanced nonlinear vibration control is required. Three well-known control strategies are used for nonlinear control purposes with uncertain modelling: 1) adaptive feedback linearization control, 2) adaptive backstepping control, and 3) virtual velocity error-based adaptive control VVEC with (or so-called passivity based control in the community of

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robotics control) [14-16]. In general, the adaptive scheme can be classified into two categories: regressor-based adaptive control and adaptive approximation control. The regressor algorithm is based on system physics while the adaptive approximation control is a model-free control algorithm. The latter is a promising tool to deal with unknown parameters. It includes a representation of uncertainty in terms of weighting coefficient and basis function matrices. Then the weighting coefficient matrices are updated based on Lyapunov theory. For more details, see [16-21]. To get a simple control structure, the key idea of this work is to enhance the PD controller using adaptive approximation compensator AAC with ensured stability. This control structure is equivalent to VVEC mentioned above with lumped uncertainty being approximated using orthogonal basis functions.

In view of the above, this paper proposes proportional-derivative PD control with adaptive approximation compensator AAC to stabilize the coupled smart beam-fluid system. The effect of axial stretching is considered that complicates the control problem due to the appearance of a coupled cubic stiffness term. In addition, the fluid loading can be decomposed into two components: a hydrodynamic force due to the beam vibration $f_m(x,t)$ and an external exciting hydrodynamic force $f_e(x,t)$ independent of the beam motion. On the other hand, $f_m(x,t)$ can be further modelled as inertia and damping forces [1,2]. The PDE for the coupled system is derived and is transformed into multi-modal ODEs using the Galerkin approach. Simulation experiments are performed on a simply supported beam with two piezo-patches immersed in a fluid. An impulse force with multi pulses is used for exciting the coupled beam-fluid system. The results show the strength of the proposed control architecture to attenuate the produced vibration. The contribution of this paper can be summarized as follows:

1. Design of a PD controller integrated with an AAC to compensate for nonlinear terms.
2. Considering the axial stretching of the beam vibration complicates the modelling and control problems.

The rest of the paper is organized as follows. Section 2 introduces the modelling of a coupled smart beam-fluid system in detail. Section 3 presents the control structure while simulation results and discussions are described in Section 4. Section 5 concludes.

II. Dynamics of Coupled Smart Beam-fluid System

This section describes in some detail the modelling of a flexible beam with surface bonded piezo-patches interacting with fluid, see Fig. 1. To determine the effect of hydrodynamics pressure, we should recognize two sub-problems for analyzing the fluid effects [5, 22]

1. If the flexible structure is constrained from oscillations, then the hydrodynamic forces are composed of Froude-Krylov and diffraction forces/moments.
2. If the flexible structure is enforced to vibrate with the fluid excitation frequency, then the hydrodynamic forces are determined by the concept of added mass, damping and restoring terms. The current work is concerned with this category of problem.

In view of the above, the following assumptions are imposed [4, 22]:

1. The beam is forced to oscillate with fluid excitation frequency in any mode shape.
2. No incident waves occur and hence the fluid loading is composed of inertial and damping forces.
3. The axial stretching on the beam is considered.
4. The modal amplitudes are measurable via a sufficient number of piezo-sensors.

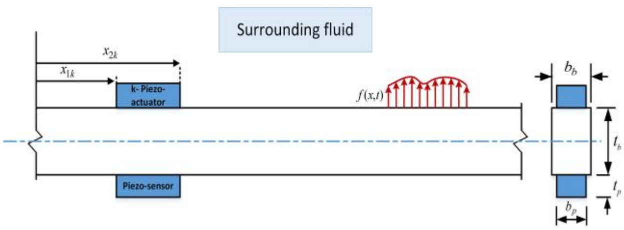

Figure 1. A flexible beam with piezo-patches interacting with fluid

The governing PDE for the transverse deflection $w(x,t)$ of a smart beam-fluid system can be expressed as follows [4, 23]

$$E_b I_b \frac{\partial^4 w}{\partial x^4} - \frac{E_b A_b}{2 l_b} \int_0^{l_b} \left(\frac{\partial w}{\partial x}\right)^2 dx \left(\frac{\partial^2 w}{\partial x^2}\right) + \rho_b A_b \frac{\partial^2 w}{\partial t^2} = f(x,t) - \frac{\partial^2 M_p}{\partial x^2} \tag{1}$$

$$M_p = D v_a(t) \frac{\partial^2}{\partial x^2}\left[H(x-x_1) - H(x-x_2)\right] \tag{2}$$

$$f(x,t) = f_m(x,t) + f_e(x,t) \tag{3a}$$

where E_b , I_b , A_b , and ρ_b are Young's modulus, moment of inertia, cross-sectional area, and density of the beam respectively. M_p is the moment per unit length exerted by the piezo-actuator to regulate system motion, D is a constant depending on properties of the regular beam and smart materials [23]. $H(.)$ is a Heaviside step function of the beam displacement. On the other hand, $f(x,t)$ denotes to the hydrodynamic forces that composed of two terms: $f_m(x,t)$ denoting to the hydrodynamic force per unit length due to the beam motion and $f_e(x,t)$ referring to an external hydrodynamic force independent of beam motion, e.g. turbulent forces. The second term on the left-hand side of Eq. (1) comes from the axial loading affected on the vibrating beam.

Remark 1. The hydrodynamics forces $f_m(x,t)$ consists of two terms: inertia and damping forces and hence they can be expressed as [1]

$$f_m(x,t) = -m_a \frac{\partial^2 w}{\partial t^2} - c_a \frac{\partial w}{\partial t} \tag{3b}$$

where m_a and c_a are the added mass and damping coefficient respectively. On the other hand, $f_e(x,t)$ is a disturbance hydrodynamic force found experimentally. In this work, it will be imposed as an impulse force with multi-pulse for excitation purposes.

The next step is to transform the PDF of Eq.(1) into multi-modal ODEs using the Galerkin approach, hence the deflection can be approximated as

$$w(x,t) = \sum_{i=1}^{N} \phi_i(x) q_i(t) \tag{4}$$

where $\phi_i(x)$ is the mode shape and $q_i(t)$ is the modal amplitude.

Substituting Eq. (4) into Eq. (1), multiplying by an arbitrary $\phi_j(x)$ and integrating along the beam length exploiting the orthogonal conditions for a simply supported beam, we can get

$$\begin{aligned}&\left(\frac{\rho_b A_b l_b}{2}\right)\ddot{q}_j + \left(\frac{E_b I_b \lambda_j^4}{2}\right) q_j + \left(\frac{E_b A_b \pi^4 j^2}{8 l_b^3}\right)\sum_{k=1}^{N} k^2 q_k^2 q_j + \\ &g_{mj}(x,t) + g_{ej}(x,t) = -\mu_j v_a(t)\end{aligned} \tag{5}$$

where

$$\begin{aligned}\mu_j &= D(\varphi_j'(x_2) - \varphi_j'(x_1)),\\ g_{mj}(x,t) &= -\int_0^{l_b} f_m(x,t)\varphi_j(x)dx,\\ g_{ej}(x,t) &= -\int_0^{l_b} f_e(x,t)\varphi_j(x)dx,\\ \omega_{n_j} &= \lambda_j^2 \sqrt{\frac{E_b I_b}{\rho_b A_b}},\\ \lambda_j &= \frac{j\pi}{l_b}.\end{aligned}$$

Equation (5) includes using one piezo-actuator. For N_a -piezo-atuators, the equation can be reformulated as

$$\begin{aligned}&\left(\frac{\rho_b A_b l_b}{2}\right)\ddot{q}_j + \left(\frac{E_b I_b \lambda_j^4}{2}\right) q_j + \left(\frac{E_b A_b \pi^4 j^2}{8 l_b^3}\right)\sum_{k=1}^{N} k^2 q_k^2 q_j + \\ &g_{mj}(x,t) + g_{ej}(x,t) = -\sum_{i=1}^{N_a} \mu_{ji} v_{ai}(t)\end{aligned} \tag{6}$$

With

$$\mu_{ji} = D(\varphi_j'(x_{2i}) - \varphi_j'(x_{1i}))$$

It is suitable to integrate the structural damping at this stage, however, a viscous damping coefficient is added. Then, Eq. (6) becomes

$$\begin{aligned}&m_j \ddot{q}_j + b_j \dot{q}_j + k_j q_j + \eta_j - g_{mj}(x,t) \\ &- g_{ej}(x,t) = u_j,\ j = 1,2,3,\dots,N\end{aligned} \tag{7}$$

With

$$m_j = \left(\frac{\rho_b A_b l_b}{2}\right), b_j = (2\zeta_j w_j) m_j, -1 \le \zeta_j \le 1, k_j = \frac{E_b I_b \lambda_j^4}{2}, w_j^2 = k_j / m_j,$$
$$\eta_j = \left(\frac{E_b A_b \pi^4 j^2}{8 l_b^3}\right)\sum_{k=1}^{N} k^2 q_k^2 q_j, u_j = -\sum_{i=1}^{N_a} \mu_{ji} v_{ai}(t)$$

In matrix representation, Equation (7) becomes

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{B}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} + \boldsymbol{\eta} + \mathbf{g}_m + \mathbf{g}_e = \mathbf{u} \tag{8}$$

$$\mathbf{q} = \begin{bmatrix} q_1 \\ \vdots \\ q_N \end{bmatrix} \in R^N, \mathbf{M} = \begin{bmatrix} m_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & m_N \end{bmatrix} \in R^{N\times N},$$

$$\mathbf{B} = \begin{bmatrix} b_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & b_N \end{bmatrix}, \mathbf{K} = \begin{bmatrix} k_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & k_N \end{bmatrix} \in R^{N\times N},$$

$$\boldsymbol{\eta} = \begin{bmatrix} \eta_1 \\ \vdots \\ \eta_N \end{bmatrix} \in R^N, \mathbf{g}_m = \begin{bmatrix} -g_{m1} \\ \vdots \\ -g_{mN} \end{bmatrix} \in R^N, \mathbf{g}_e = \begin{bmatrix} -g_{e1} \\ \vdots \\ -g_{eN} \end{bmatrix} \in R^N,$$

$$\mathbf{u} = \begin{bmatrix} u_1 \\ \vdots \\ u_N \end{bmatrix} = -\begin{bmatrix} \mu_{11} & \cdots & \mu_{1N_a} \\ \vdots & \ddots & \vdots \\ \mu_{N1} & \cdots & \mu_{NN_a} \end{bmatrix}\begin{bmatrix} v_{a1} \\ \vdots \\ v_{aNa} \end{bmatrix} \in R^N$$

III. CONTROL STRUCTURE

The vibrating beam is equipped with distributed piezo-patches. The task of piezo-patches is to actuate the input control and/or to measure the state variables of the dynamic system. In fact, the measurement of modal amplitudes occurs via installing piezo-sensors such that a mathematical relation is established between the modal amplitudes and sensor voltage readings. The proposed control system described here consists of two terms: PD control term and an AAC term for compensating unmodelled dynamics. Therefore, the intuitive control law is selected as

$$\mathbf{u} = \hat{\boldsymbol{\varsigma}} - \mathbf{K}_p \mathbf{e} - \mathbf{K}_d \dot{\mathbf{e}} - \boldsymbol{\gamma}\,\mathrm{sgn}(\dot{\mathbf{e}}) \tag{9a}$$

$$\begin{aligned}\mathbf{e} &= \mathbf{q} - \mathbf{q}_d,\\ \hat{\boldsymbol{\varsigma}} &= \hat{\mathbf{M}}\ddot{\mathbf{q}}_d + \hat{\mathbf{B}}\dot{\mathbf{q}}_d + \hat{\mathbf{K}}\mathbf{q}_d + \hat{\boldsymbol{\eta}} + \hat{\mathbf{g}}_m + \hat{\mathbf{g}}_e,\\ \hat{\boldsymbol{\varsigma}} &= \hat{\mathbf{W}}^T \boldsymbol{\theta}\end{aligned} \tag{9b}$$

where $(\hat{\cdot})$ refers to estimation, $\mathbf{K}_p \in R^{N\times N}$ and $\mathbf{K}_d \in R^{N\times N}$ are the proportional and derivative gains respectively, $\boldsymbol{\gamma} \in R^{N\times N}$ is a positive definite diagonal matrix, $\mathbf{q}_d \in R^N$ is the desired reference vector that should tracked by the controller (it is a null vector in the case of vibration suppression), $\hat{\mathbf{W}} \in R^{N\beta\times N}$ and $\boldsymbol{\theta} \in R^{N\beta}$ are the weighting coefficient and the basis function matrix/vector, and β is the number of basis function used for the function approximation technique FAT.

Equating Eq. (9a) to Eq. (8) to obtain the following closed loop dynamics

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{B}\dot{\mathbf{q}} + \underbrace{\mathbf{K}\mathbf{q} + \boldsymbol{\eta} + \mathbf{g}_m + \mathbf{g}_e}_{\lambda} = \hat{\mathbf{M}}\ddot{\mathbf{q}}_d + \hat{\mathbf{B}}\dot{\mathbf{q}}_d + \underbrace{\hat{\mathbf{K}}\mathbf{q}_d + \hat{\boldsymbol{\eta}} + \hat{\mathbf{g}}_m + \hat{\mathbf{g}}_e}_{\hat{\lambda}} - \mathbf{K}_p\mathbf{e} - \mathbf{K}_d\dot{\mathbf{e}} - \boldsymbol{\gamma}\,\mathrm{sgn}(\dot{\mathbf{e}}) \quad (10)$$

By adding $(-\mathbf{M}\ddot{\mathbf{q}}_d - \mathbf{C}\dot{\mathbf{q}}_d - \boldsymbol{\lambda})$ to Eq. (10) leads to

$$\mathbf{M}\ddot{\mathbf{e}} + (\mathbf{B}+\mathbf{K}_d)\dot{\mathbf{e}} + \mathbf{K}_p\mathbf{e} + \boldsymbol{\gamma}\,\mathrm{sgn}(\dot{\mathbf{e}}) = -\tilde{\boldsymbol{\varsigma}} + \boldsymbol{\varepsilon} \quad (11)$$

where $\boldsymbol{\varepsilon} \in R^N$ is the approximation modelling error. Now, it is time to prove the stability of control law of Eqs. (9a) and (11) while deriving the suitable update law for the weighting coefficient vector $\hat{\varsigma}$. Let us consider the non-negative function along Eq. (11) to get

$$V = \frac{1}{2}\dot{\mathbf{e}}^T\mathbf{M}\dot{\mathbf{e}} + \frac{1}{2}\mathbf{e}^T\mathbf{K}_p\mathbf{e} + \frac{1}{2}tr(\tilde{\mathbf{W}}^T\mathbf{G}^{-1}\tilde{\mathbf{W}}) \quad (12)$$

Taking time derivative of the above equation to obtain

$$\dot{V} = \dot{\mathbf{e}}^T(-(\mathbf{B}+\mathbf{K}_d)\dot{\mathbf{e}} - \mathbf{K}_p\mathbf{e} - \boldsymbol{\gamma}\,\mathrm{sgn}(\dot{\mathbf{e}}) - \tilde{\varsigma} + \boldsymbol{\varepsilon}) + \dot{\mathbf{e}}^T\mathbf{K}_p\mathbf{e} - tr(\tilde{\mathbf{W}}^T\mathbf{G}^{-1}\dot{\hat{\mathbf{W}}}), \dot{\mathbf{M}} = \mathbf{0} \quad (13)$$

Equation (13) can be manipulated to get

$$\dot{V} = -\dot{\mathbf{e}}^T(\mathbf{B}+\mathbf{K}_d)\dot{\mathbf{e}} + \dot{\mathbf{e}}^T\boldsymbol{\varepsilon} - \dot{\mathbf{e}}^T\boldsymbol{\gamma}\,\mathrm{sgn}(\dot{\mathbf{e}}) - tr(\tilde{\mathbf{W}}^T(\boldsymbol{\theta}\dot{\mathbf{e}}^T + \mathbf{G}^{-1}\dot{\hat{\mathbf{W}}}) \quad (14)$$

From Eq. (14), the suitable update law for the weighting matrix to get a stabilized system is

$$\dot{\hat{\mathbf{W}}} = -\mathbf{G}^{-1}\boldsymbol{\theta}\dot{\mathbf{e}}^T \quad (15)$$

Equation (14) is reduced to

$$\dot{V} = -\dot{\mathbf{e}}^T(\mathbf{B}+\mathbf{K}_d)\dot{\mathbf{e}} + \dot{\mathbf{e}}^T\boldsymbol{\varepsilon} - \sum_i \gamma_i\left|\dot{e}_i\right| \quad (16)$$

Selecting
$\gamma_i \geq \dot{e}_i + \delta_i$, where δ_i is a positive constant can stabilize the closed-loop dynamics leading to

$$\dot{V} = -\dot{\mathbf{e}}^T(\mathbf{B}+\mathbf{K}_d)\dot{\mathbf{e}} - \sum_i \gamma_i\left|\dot{e}_i\right| \quad (17)$$

Equation (17) is stable according to Lyapunov theory.

IV. Simulation Results and Discussions

This section attempts to verify the validity of the proposed control architecture by making a simulation experiments on a smart simply supported beam interacting with fluid, see Fig. 1. The piezo-patches are placed on $0.3l_b$ from both ends of the beam. The external excitation hydrodynamic force is assumed a multi-pulse impulse force having 1 N pulse peak, 2% pulse width and 2 s pulse period. Besides, it is applied at the middle of the beam. See Table I for more details on properties of beam, piezo-materials and fluid used in simulation experiments.

TABLE I. Properties of Beam, Piezo-materials, and Fluid

Beam	$\rho_b = 8000kg/m^3, l_b = 0.4m, E_b = 190\times10^3 MPa,$ $A_b = 0.04\times0.001m, b_1 = 007N.s/m,$ $b_2 = 0.03N.s/m$
Piezo-material	$l_p = 0.08m, A_p = 0.035\times0.0004m,$ $E_p = 70\times10^3 MPa.$
Fluid (FC-72)	$c_m = 2.1, c_v/\mu = 2280, \mu = 0.4\times10^{-6}m^2s,$ $\rho_f = 1.68\times10^3 kg/m^3.$

The proposed control law associated with the update law described in Eqs. (9a) and (15) are applied on the vibrating beam using MATLAB/SIMULINK package. The feedback and adaptation gain matrices used for simulation are: $K_p = 300I_2, K_d = 100I_2, W = 20I_{22}, \beta = 11, N = 2$. In the tuning process, the gains are used from zero to a value at which the noise or instability occurs then the value is halved. Chebyshev polynomials are used as approximator for adaptive technique. The results show that the controller is safely and precisely attenuate the resulted vibrations. Figs. 2 and 3 show the modal response and input control. In fact, it is assumed that the actuators are strong enough to produce any response; however, an algorithm is required to limit the output of the piezo-actuators. On the other hand, here we used 2 piezo-patches with two-mode shapes and hence determining the input control for piezo-voltages is easy, however, if the number of piezo-actuators is not equal to the number of the mode shapes, then pseudo-inverse matrix should be used alternatively to compute the input voltages.

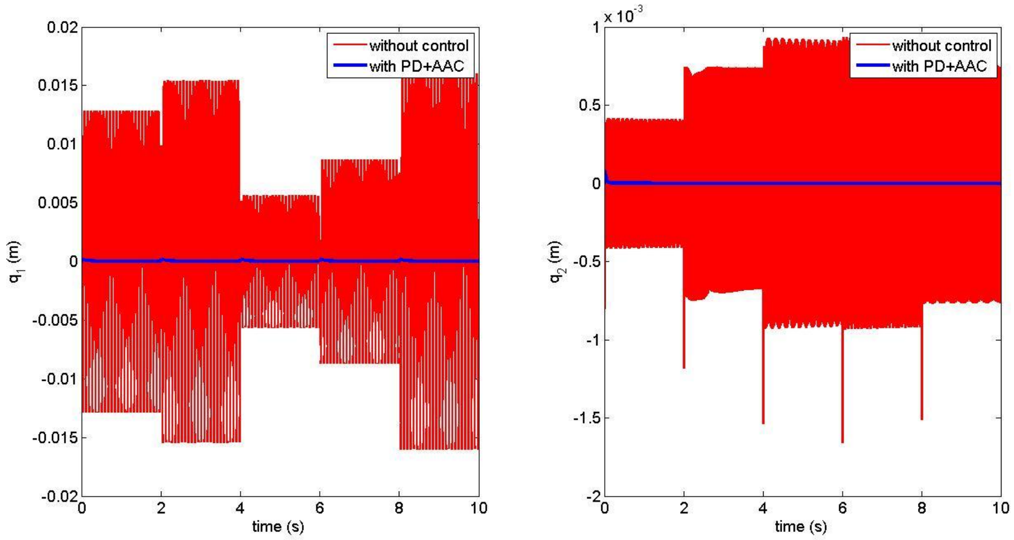

Figure 2. Modal amplitude response. As we see, there are sudden pulses at times: 2s, 4s, 6s, 8s, and 10s. Despite of the hard abrupt impulses, the controller smoothly attenuates the vibration motion of the beam.

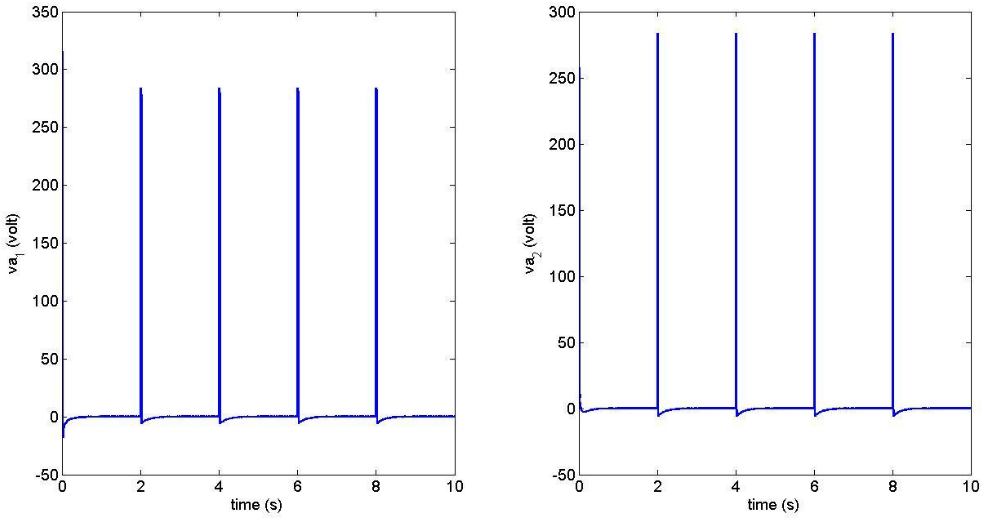

Figure 3. Input control voltages. As we see, there are sudden pulses at times: 2s, 4s, 6s, 8s, and 10s.

V. CONCLUSIONS

This work proposes a PD control with AAC for vibration suppression of smart beam interacting with fluid. The axial stretching is considered in dynamic modelling of the beam that complicates the control problem due to the appearance of nonlinear cubic stiffness term. In addition, the hydrodynamic loads are assumed equivalently to consist of two terms: inertial and damping forces using (i.e., added mass and damping). Standard multi-modal ODEs are derived and regulated using the proposed controller. In effect, our algorithm is sufficient to deal with any complex systems, however, future work is required to deal with the following points:

1. The effect of fluid loads using incident waves.
2. Generalizing the algorithm to deal with aeroelastic plates and shells.
3. Compensating for unmodelled mode shapes.

CONFLICT OF INTEREST

The authors declare no conflict of interest.

AUTHOR CONTRIBUTIONS

Conceptualization, methodology, resources and investigation were performed by Abdnoor Jameel Shaheed Al-hamadani and H.F.N. Al-Shuka. Paper writing was conducted by Kareem Jawad Kadhim

REFERENCES

[1] T. Naik, E. K. Longmire, S. C. Mantell, "Dynamic response of a cantilever in liquid near a solid wall," *Sensors and Actuators A: Physical*, vol. 103, no. 2, pp. 240-254, 2003.

[2] W. Lin, N. Qiao, "Vibration and stability of an axially moving beam immersed in fluid," *International Journal of Solids and Structures*, vol. 45, no. 5, pp. 1445-1457, 2008.

[3] M. Fadaee, M. Talebitooti, "Active vibration control of carbon nanotube-reinforced composite beam submerged in fluid using magnetostrictive layers," *Mechanics Based Design of Structures and Machines*, 2020.

[4] D. Wagg, S. Neild, *Nonlinear Vibration with Control,* SMIA, Springer Verlag, 2010.

[5] O. M. Faltinsen, *Sea Loads on Ships and Offshore Structures*, Cambridge University Press, 1990.

[6] J. T. Xing, W. G. Price, M. J. Pomfret, and L. H. Yam, "Natural vibration of a beam-water interaction system," *Journal of Sound and Vibration*, vol. 199, no. 3, pp. 491–512, 1997.

[7] J. E. Sader, "Frequency response of cantilever beams immersed in viscous fluids with applications to the atomic force microscope," *Journal of Applied Physics*, vol. 84, no. 1, pp. 64–76, 1998.

[8] S. Zhao, J. T. Xing, and W. G. Price, "Natural vibration of a flexible beam–water coupled system with a concentrated mass attached at the free end of the beam," in *Proc. the Institution of Mechanical Engineers, Part M: Journal of Engineering for the Maritime Environment*, vol. 216, no. 2, pp. 145–54, 2002.

[9] W. Lin and N. Qiao, "Vibration and stability of an axially moving beam immersed in fluid," *International Journal of Solids and Structures*, vol. 45, no. 5, pp.1445–57, 2008.

[10] R. Shabani, H. Hatami, F. G. Golzar, S. Tariverdilo, and G. Rezazadeh, "Coupled vibration of a cantilever micro-beam submerged in a bounded incompressible fluid domain," *Acta Mechanica*, vol. 224, no. 4, pp. 841–50, 2013.

[11] Q. Ni, M. Li, M. Tang, and L. Wang, "Free vibration and stability of a cantilever beam attached to an axially moving base immersed in fluid," *Journal of Sound and Vibration*, vol. 333, no. 9, pp. 2543–55, 2014.

[12] S. Eftekhari and A. A. Jafari, "A mixed modal-differential quadrature method for free and forced vibration of beams in contact with fluid," *Meccanica*, vol. 49, no. 3, pp. 535–64, 2014.

[13] M. Rezaiee-Pajand, M. S. Kazemiyan, and S. A. Aftabi, "Solving coupled beam-fluid interaction by DTM," *Ocean Engineering*, vol. 167, pp. 380–96, 2018.

[14] J. A. Farrell, M. M. Polycarpou, "Adaptive approximation based control: Unifying neural, fuzzy and traditional adaptive approximation approaches," John Wiley, 2006.

[15] Petros Ioannou and Barış Fidan, "Adaptive control tutorial," *SIAM*, 2006.

[16] A. C. Huang, M. C. Chien, "Adaptive control of robot manipulators: a unified regressor-free approach," World Scientific Publishing Co. Pte. Ltd. Singapore, 2010.

[17] H. F. N. Al-Shuka, B. Corves, W. H. Zhu, "Function approximation technique-based adaptive virtual decomposition control for a serial-chain manipulator," vol. 32, no. 3, pp. 375-399, 2013.

[18] H. F. N. Al-Shuka, "On local approximation-based adaptive control with applications to robotic manipulators and biped robots," *Int. J. Dynam. Control*, vol. 6, pp. 339–353, 2018.

[19] H. F. N. Al-Shuka, R. Song, "Hybrid regressor and approximation-based adaptive control of piezoelectric flexible beams," in *Proc. 2018 2nd IEEE Advanced Information Management, Communicates, Electronic and Automation Control Conference (IMCEC)*, Xi'an, 330-334, 2018.

[20] H. F. N. Al-Shuka, R. Song, "Hybrid regressor and approximation-based adaptive control of robotic manipulators with contact-free motion," in *Proc. 2018 2nd IEEE Advanced Information Management, Communicates, Electronic and Automation Control Conference (IMCEC)*, Xi'an, 2018, 325-329, 2018.

[21] H. F. N. Al-Shuka, R. Song, "Decentralized adaptive partitioned approximation control of high degrees-of-freedom robotic manipulators considering three actuator control modes," *International Journal of Dynamics and Control*, vol. 7, pp. 744-757, 2019.

[22] E. H. Dowell, *Aeroelastic Plates and Shells*, Noordhoff International Publishing, 1975.

[23] S. Le, "Active vibration control of a flexible beam," Master thesis, San Jose State University, USA, 2009.

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