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Active Control of Residual Vibrations of a Cantilever Smart Beam

Zeki KIRAL, Levent MALGACA, Murat AKDAĞ
Dokuz Eylül University, Faculty of Engineering, İzmir-TURKEY
e-mail: zeki.kiral@deu.edu.tr

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Abstract

The aim of this study is to control the residual vibrations of a clamped-free beam subjected to a moving load. The dynamic response of the beam is calculated by using the finite element method in order to design a suitable control technique and the numerical results are verified by vibration measurements. All the numerical calculations are performed by a commercial finite element package. Two laser displacement sensors are used to measure the dynamic response of the beam. The moving load is obtained by pressured air directed to the beam via a nozzle, and the movement of the load is achieved by an industrial robot manipulator having 6 degrees of freedom. In this study, the suppression of the residual vibrations that occur after the moving load has left the beam is considered as the main subject. Piezoelectric actuators are used for active vibration control study and displacement feedback is employed. The numerical results agree well with the experimental results. The results show that the finite element method can be used effectively for designing a suitable vibration control strategy.

Key words: Active vibration control, Finite element method, Moving load, Piezoelectricity

Introduction

Engineering structures work frequently under dynamic excitations. The type of the dynamic excitation may vary but the results of these excitations are shown generally in the form of vibration. Vibration can be attributed as an unwanted output for many engineering structures due to waste of energy, precision losses, noise etc. and should be kept under control especially for lightweight structures. Researchers still pay great attention to vibration suppression due to its industrial importance and there are numerous studies related to active vibration control.

Lim et al. (1997, 1999) investigated vibration controllability of beams with piezoelectric sensors and actuators using finite element analysis in both frequency and time domain. They showed the suppression of vibration amplitudes with both constant displacement and velocity feedback. The sensor response is examined when a unit voltage is applied to the actuator. Celentano and Setola (1999) developed a simplified model of a beam-like structure with a bonded piezoelectric plate by integrating the usual electrical with the finite element method and mechanical models with a RLC circuit.

Manning et al. (2000) presented a smart structure vibration control scheme using system identification and pole placement. System identification is carried out in 3 phases: data collection, model characterization, and parameter estimation. Input-output data are collected by stimulating the piezoelectric actuators with a square wave signal and monitoring the strain gage response. Negative velocity feedback is used as the controller to reduce vibration amplitudes. Gaudenzi et al. (2000) investigated this problem both experimentally and numerically with position and velocity control approaches. The numerical simulation is developed with the finite element method based on an Euler-Bernoulli model.

Bruant et al. (2001) described the modeling of beam structures that contained piezoelectric devices with a simple finite composite element. A simple cantilever beam structure is studied. Six mechanical degrees of freedom and 4 electric degrees of freedom are used in the model. They developed a methodology for the determination of the optimal geometries of piezoelectric devices. Halim and Moheimani (2002) aimed to develop a feedback controller that suppresses vibration of flexible structures. The controller (H_∞) is applied to a simple-supported PZT laminate beam and it is validated experimentally. Yaman et al. (2002) described the role of smart structures in aerospace engineering applications.

Kusculuoglu et al. (2004) developed a new FE model for a beam with a piezoceramic patch actuator. Each layer is treated as a Timoshenko beam. Two experimental studies are performed to validate the theoretical developments. They observed that the use of the introduced model became more important when the piezoceramic and base layer thickness were large and shear and related rotational inertia became more important.

Karagülle et al. (2004) simulated the active vibration control of a cantilever beam having piezoelectric patches by ANSYS finite element package. They reported that modeling the smart structures, locating the actuators and sensors at the most suitable positions, and determining the feedback gain is possible by commercial computer programs such as ANSYS even for complicated structures. Xu et al. (2004) studied the effect of the sensor/actuator location on the vibration control performance for a clamped-free beam using ANSYS. Malgaca (2007) studied the active control of free and forced vibrations of beam structures by piezoelectric actuators both numerically and experimentally. Kıral et al. (2007) studied the moving load problem both experimentally and numerically and presented the dynamic response of a cantilever beam for different magnitudes and velocities of the moving load.

In the present study, active control of residual vibrations of a clamped-free beam subjected to a moving load is considered. The moving load is a type of dynamic excitation in which the position of the load varies in time. The vibration control problem is studied both experimentally and numerically. The residual vibrations of the beam that occur after the load has left the beam are suppressed effectively using the piezoelectric actuators via displacement feedback with zero reference input. The dynamic displacements during the moving load can be reduced by using suitable gain values for the proportional controller. The selection of the controller gain can be achieved by programmable finite element programs in which the active elements can be employed. The moving load mechanism used in this study has not been encountered in the literature and is the original contribution of this study.

Beam Model and Experimental Set-Up

In this study, a clamped-free beam having a rectangular cross section with dimensions $1.5 \times 20 \times 1000$ (mm) is considered. The material of the beam is aluminum with the Young modules E = 69 GPa and the density $\rho = 2676$ kg/m^3. The schematic view of the experimental set-up is shown in Figure 1.

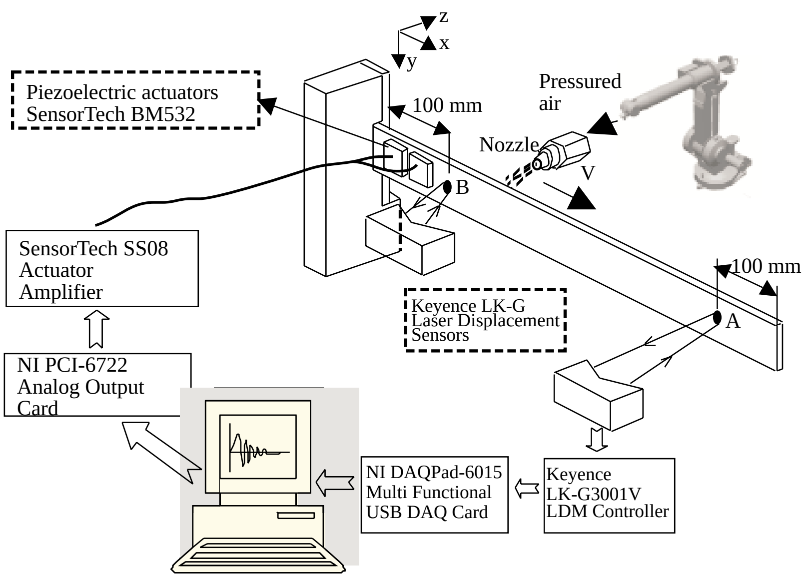


Figure 1. Schematic view of the experimental set-up.

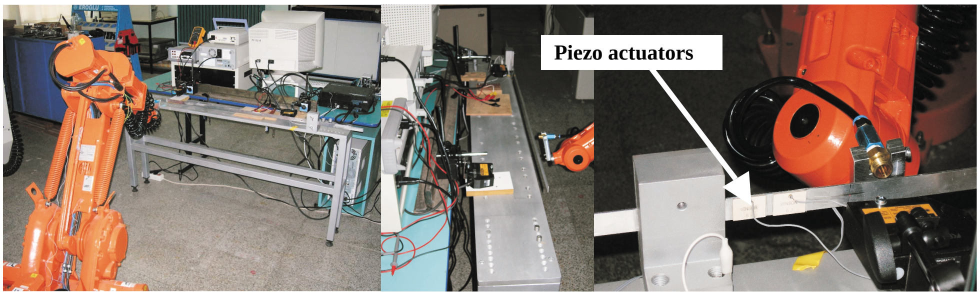


Figure 2. View of the experimental set-up, smart beam and piezoelectric actuators.

The force acting on the smart beam is obtained by pressured air via a nozzle and the movement of the force is established by a robot manipulator ABB IRB 1400 shown in Figure 2. The robot is programmed to move the nozzle with constant speed. The displacement response of the beam is measured via 2 laser displacement sensors located at the positions A and B shown in Figure 1. Two piezo patches located 10 mm away from the clamped edge, Sensortech BM532, are glued on the aluminum beam and used as the smart portion of the beam in monomorph configuration as shown in Figure 2. The rectangular cross section of a piezo patch has the dimensions 1 × 20 × 25 (mm).

The measurement ranges of the laser sensors used for displacement measurement for point A and point B are 150 ± 40 mm and 30 ± 5 mm, respectively. The sensitivities of the laser sensors A and B are 0.5 μ and 0.05 μ, respectively. Dynamic displacement response recorded for point A is used as the feedback for the vibration control study. The feedback signal is acquired via a LabVIEW (2003) program and then processed to form the control signal.

Experimental Results

Firstly, the natural frequencies of the smart beam are determined by free vibration analysis. Then 30 mm initial displacement is given to the free end of the beam and the free vibrations of the smart beam are measured in terms of displacement via laser displacement sensors. The displacement signal measured by the laser sensor B contains 3 fundamental frequencies of the beam as shown in Figure 3. The fundamental frequencies are determined by fast Fourier analysis.

The load velocity is chosen as 0.25 m/s and the air pressure is regulated as 2.25 atm. Load velocity of 0.25 m/s is not a critical velocity and is chosen in order to observe the moving load response of the beam for a sufficiently long time period. The moving load velocity can be changed in a broad velocity range from 0.005 m/s to 4 m/s in the experimental set-up. Figure 4 shows the step response of the beam measured at point A to a 2.25 atm step load applied 0.98 m away from the clamped edge. The distance between the nozzle and the beam surface is 25 mm, similar to the moving load case. The mean value of the steady-state displacement response is calculated as –35.89 mm as shown in the figure and the load value is calculated as 0.0427 N by the static deflection expression

$$F = \frac{6\,E\,I_y\,u_{zs}}{x^2\,(3\,a - x)} \tag{1}$$

where I_y denotes the inertia moments of area about the y axis, u_{zs} denotes the steady-state or static displacement at point A, x denotes the coordinate of point A for which the static displacement is calculated, and a denotes the position of the load. The oscillations around the steady-state response can be attributed to the small changes in the air pressure and the solid/fluid interactions on the beam surface.

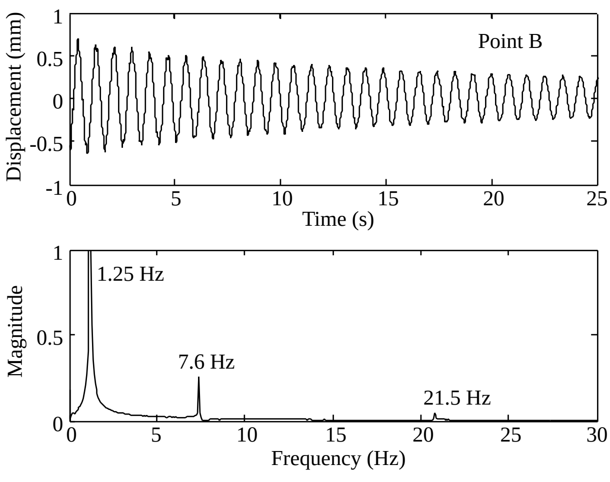


Figure 3. Free vibration response of the smart beam for point B and its frequency content.

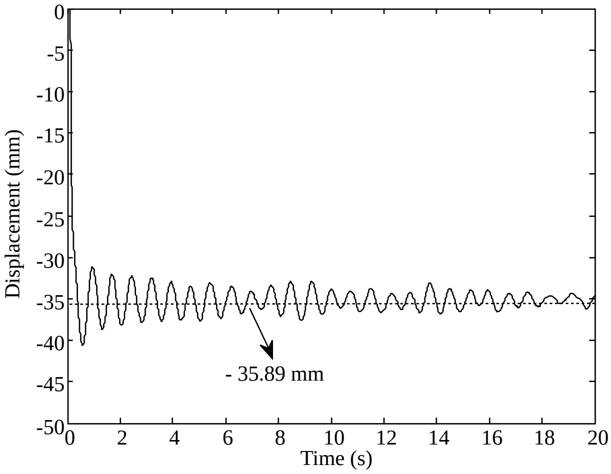


Figure 4. Step response of beam measured at point A for 2.25 atm.

The air nozzle is moved from the clamped end to the free end of the smart beam by robot manipulator and the displacement responses during the moving load acting on the beam and after the moving load has left the beam are recorded for control off and control on cases. The closed loop block diagram of the active vibration control employed in this study is shown in Figure 5.

As shown in Figure 5, the dynamic displacement response of the beam u_z is used as a feedback signal and compared with the reference signal, which is 0 V, representing the case in which the beam is at rest. The error signal is processed by a proportional controller and the actuation signal is sent to the piezoelectric actuators after amplification. The amplification constant of the piezo amplifier is 30. The piezoelectric actuators drive the beam in order to reduce the dynamic displacements. The upper limit of the voltage value applied to the piezo actuators is ±270 V. Figure 6 shows the experimental dynamic responses of the smart beam for control off and control on cases at 0.25 m/s moving load velocity.

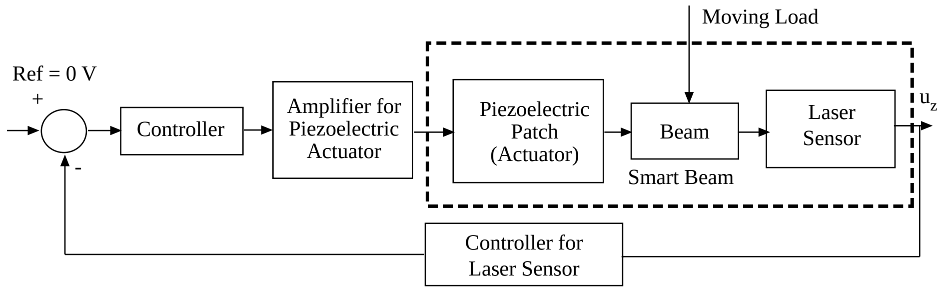


Figure 5. Block diagram of the closed loop control system.

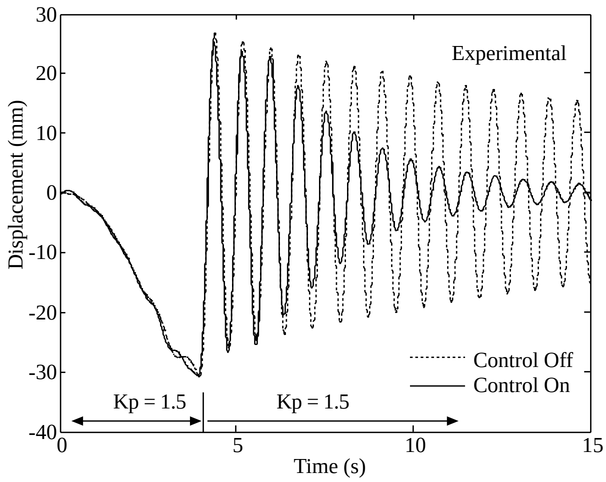


Figure 6. Dynamic responses of the smart beam for point A, 2.25 atm, 0.25 m/s.

The moving load completes its travel on the beam in 4 s. The controller gain Kp is chosen as 1.5 for both cases: moving load acting on the beam and the residual vibrations. It is shown from Figure 6 that the closed loop control with proportional controller and piezoactuators does not have a considerable effect on the moving load response but vibration control is very effective in suppressing the residual vibrations for the selected controller gain value.

Simulation Results with ANSYS

The finite element method has been used for many years to examine the dynamics of structures. In this study, a commercial finite element package ANSYS 10.0 (2006) is used to simulate the vibration control of the smart beam. The active portion (piezoelectric patches) of the beam is modeled by Solid5 elements and the passive portion (aluminum beam) is modeled by Solid45 elements as shown in Figure 7.

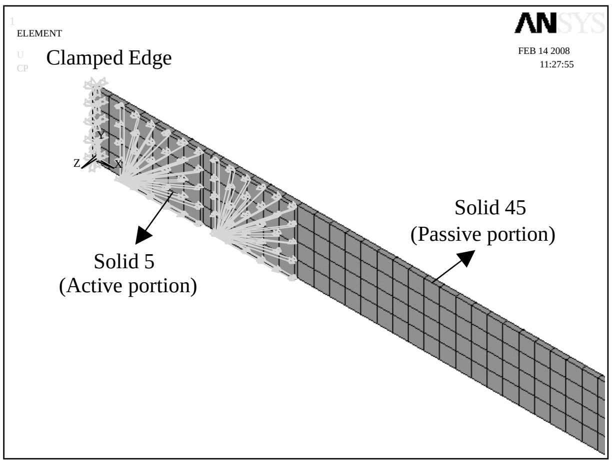
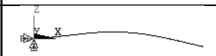
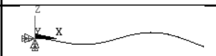


Figure 7. The finite element model of the smart beam.

An APDL (ANSYS Parametric Design Language) code is developed to create the finite element model of the smart beam. Modal and transient analyses are performed via this code. The APDL code enables us to perform a closed loop control for vibration suppression. The finite element model of the smart cantilever beam consists of 456 elements for the passive portion and 40 elements for the active portion. Having obtained the finite element model, the natural frequencies of the smart beam are calculated as given in the Table.

As shown in the Table, the experimental and numerical values of natural frequencies are in good agreement. The third step in the simulation is the dynamic response analysis with closed loop control. All these steps are carried out by the APDL code, a small part of which is given in the appendix. Rayleigh damping, $[C] = \alpha\,[M] + \beta\,[K]$, with coefficients $\alpha = 0.00462$ and $\beta = 0.00231$, is used in the dynamic analyses. The values of the damping coefficients are determined by a trial and error procedure. These damping coefficients give the same experimental and numerical decrement values calculated from the successive peaks in the free vibration response. The movement of the load in the simulation is established in the APDL code by altering the node at which the force is applied for each time step. For every substep in the transient analysis the variation of the load is assumed to be linear. Figure 8 shows the simulated dynamic responses of the beam for control off and control on cases.

Table. First 3 natural frequencies of the smart beam.

		Natural Frequencies (Hz)	
		Experimental	ANSYS
1^{st} mode		1.25	1.26
2^{nd} mode		7.60	7.82
3^{rd} mode		21.50	21.68

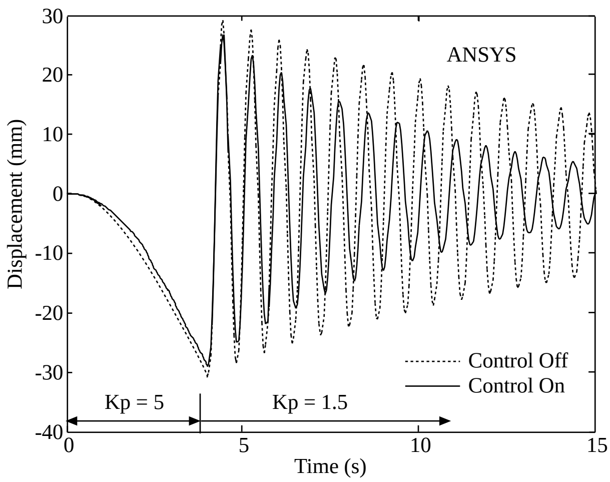


Figure 8. Simulated dynamic responses of the smart beam for point A, 2.25 atm, 0.25 m/s.

In the simulation, the controller gain Kp is chosen as 5 during the moving load acting on the beam and Kp is chosen as 1.5 after the load has left the beam. Different gain values could not be used in the experiments due to instrumental restrictions. The simulated actuation voltages applied to the piezoelectric actuators are shown in Figure 9. The simulation results given in Figure 8 show that the dynamic displacements during the moving load can be reduced slightly, but the residual vibrations are suppressed efficiently with active control. The finite element analysis package ANSYS is used successfully to design a suitable vibration control strategy. The simulation results agree well with the experimental results.

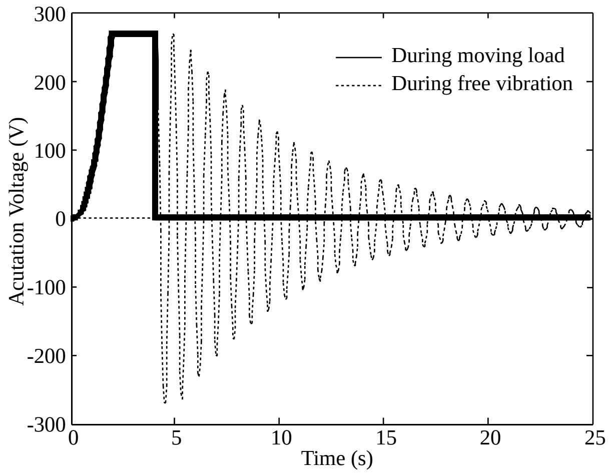


Figure 9. Simulated actuation voltage.

Conclusions

Active vibration control of a cantilever smart beam is considered both experimentally and numerically in this study. A contactless moving load mechanism and contactless vibration measurement system are used in the experimental study. The moving load mechanism used in this study has not been encountered in the literature. The simulation of the closed loop vibration control with displacement feedback is achieved by using a commercial finite element package. The piezoelectric elements that actuate the beam to suppress the dynamic response are used in the experimental system and are modeled by a commercial finite element package for the numerical analyses. Residual vibrations of the smart beam are suppressed successfully by proportional control. The simulated and experimental results are in very good agreement. The design of active vibration control of more complex structures can be achieved with the programmable finite element packages, which enable us to use active elements.

Nomenclature

a	position of the load in x direction
F	load magnitude
I_y	inertia moments of area about y axis
u_z	dynamic displacement in z direction
u_{zs}	static displacement in z direction
[C]	damping matrix
[K]	stiffness matrix
[M]	mass matrix
α	mass proportional damping coefficient
β	stiffness proportional damping coefficient

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Appendix

A1. A sample part of the APDL code for active vibration control of smart beam.

```
/prep7
/title, Cantilever smart beam model with discrete multiple
!actuators for moving load analysis
!Parameters for Smart Structure
npzt=1       ! Number of PZT : (0.No PZT, 1.A)
l1=1000e-3   ! Length of Metal Beam
b1=20e-3     !  Width
h1=1.5e-3    ! Height
l2p=25e-3    ! Length of PZT
b2p=b1       ! Width of PZT
h2=1e-3      ! Height of PZT
l1s=10e-3    ! Length of space to locate PZT#1
l2s=5e-3     ! Length of space to locate PZT#2
nebx=100     ! Number of elements in X direction for Metal
nepx=5       ! Number of elements in X direction for PZT
nepy=4       ! Number of elements in y direction for PZT
nebsx=2      ! Number of elements in X direction for
nebsy=0      ! Number of elements in y direction for Space
!-------------------------------------------------
!Material Properties
!-------------------------------------------------
et,1,solid45
et,2,solid5,3
r,1,0,0,mz
```

```
/solu                  ! Modal Analysis
d,nv0,volt,0
d,nv,volt,0
alphad,alphar
betad,betar
antype,modal,new
modopt,lanb,6
solve
*get,f1,mode,1,freq
*get,f3,mode,3,freq
Finish
!-----------------------------------------------------------------
/solu            ! Transient analysis for moving load problem
antype,trans,new
outres,all,all
kbc,0
!-----------------------------------------------------------------
*if,contsel,eq,1,then
*do,t,to+2*dt,ts,dt      ! Active control after moving load
*get,uza,node,ncen,u,z
   err=ref-ks1*uza
   va=kpf*kv*err
 *if,va,ge,vmax,then
  va=vmax
```