Active vibration suppression of a flexible structure using smart material and a modular control patch

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Abstract: This paper presents experimental results of vibration suppression of a flexible structure using smart materials and a miniaturized digital controller, called the modular control patch (MCP). The MCP employs a TI-C30 digital signal processor and was developed by TRW for the United States Air Force for future space vibration control. In this research, the MCP is used to implement different control algorithms for vibration suppression of a cantilevered aluminum beam. The beam is equipped with smart sensors and actuators, and both are made of piezoceramics. Positive position feedback (PPF) control, strain rate feedback (SRF) control and their combinations were implemented. Experiments found that PPF control is most effective for single-mode vibration suppression, and two PPF filters in parallel are most effective for multimode vibration suppression. Experiments also demonstrated the capacity of smart material being used as sensors and actuators for vibration suppression. The MCP was shown to be capable of implementing various real-time control laws.

Keywords: miniaturized digital controller, positive position feedback control, strain rate feedback control, vibration suppression, multimode vibration suppression, smart sensors and actuators

NOTATION

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>constant</td>
</tr>
<tr>
<td>$d_{31}$</td>
<td>lateral strain coefficient of the PZT</td>
</tr>
<tr>
<td>$E_b$</td>
<td>Young’s modulus of the beam</td>
</tr>
<tr>
<td>$E_p$</td>
<td>Young’s modulus of the PZT</td>
</tr>
<tr>
<td>$G$</td>
<td>feedback gain</td>
</tr>
<tr>
<td>$L$</td>
<td>beam length</td>
</tr>
<tr>
<td>$L_a$</td>
<td>length of PZT actuators</td>
</tr>
<tr>
<td>$L_s$</td>
<td>length of PZT sensor</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
</tr>
<tr>
<td>$t_b$</td>
<td>beam thickness</td>
</tr>
<tr>
<td>$t_p$</td>
<td>PZT actuator and sensor thickness</td>
</tr>
<tr>
<td>$w_b$</td>
<td>beam width</td>
</tr>
<tr>
<td>$w_p$</td>
<td>PZT actuator and sensor width</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>magnitude of the assumed single degree-of-freedom vibration of the beam</td>
</tr>
<tr>
<td>$\varepsilon_3^{\perp}$</td>
<td>absolute permittivity of the PZT</td>
</tr>
<tr>
<td>$\zeta$</td>
<td>damping ratio of the structure</td>
</tr>
<tr>
<td>$\zeta_c$</td>
<td>damping ratio of the compensator</td>
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<tr>
<td>$\eta$</td>
<td>coordinate of the compensator</td>
</tr>
<tr>
<td>$\xi$</td>
<td>modal coordinate describing the displacement of the structure</td>
</tr>
<tr>
<td>$\rho_b$</td>
<td>beam density</td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>PZT density</td>
</tr>
<tr>
<td>$\phi$</td>
<td>phase angle</td>
</tr>
<tr>
<td>$\omega$</td>
<td>natural frequency of the structure</td>
</tr>
<tr>
<td>$\omega_c$</td>
<td>natural frequency of the compensator</td>
</tr>
</tbody>
</table>

1 INTRODUCTION

The current trend of spacecraft design is to use large, complex and lightweight space structures to achieve increased functionality at a reduced launch cost. The combination of a large and lightweight design results in these space structures being extremely flexible and having low fundamental vibration modes. Active vibration control has been increasingly used as a solution for spacecraft structures to achieve the degree of vibration suppression required for precision pointing.
accuracy. This paper examines the effectiveness and suitability of the modular control patch (MCP), a miniaturized onboard digital controller, to implement various control algorithms to achieve active vibration control on flexible structures with embedded piezoceramic sensors and actuators.

The MCP is a miniaturized digital controller for future space applications in vibration suppression. The MCP was developed by TRW for the United States Air Force and uses a digital microprocessor to implement control algorithms. In this research, the MCP is used for vibration suppression of a cantilevered beam. The first two modes of the beam are found to be dominant. The cantilevered beam has piezoceramic sensors and piezoceramic actuators. Piezoceramics have several desirable characteristics for this type of application. These include high strain sensitivity, high stiffness, low noise, good linearity, temperature insensitivity, ease of implementation and low power consumption [1, 2].

Positive position feedback (PPF) control [3, 4] and strain rate feedback (SRF) control [5] were designed and implemented using piezoceramic sensors and actuators and the MCP. These control laws were used independently and in combination in order effectively to suppress vibrations of the first two modal frequencies of the cantilevered beam. The PPF was found to be most effective for single-mode vibration suppression. Two PPF filters in parallel provide the most effective multimode damping. Experiments demonstrated the capacity of piezoceramics to be used as both smart sensors and actuators for vibration suppression. The MCP was also shown to be capable of implementing various real-time control laws. The major novelties and contributions of this paper are:

(a) the use of the MCP, a miniaturized onboard digital controller, for digital data acquisition and real-time control;
(b) multimode vibration suppression using a combination of PPF and/or SRF controls.

2 EXPERIMENTAL SET-UP

The purposes of the experiment are to examine the effectiveness of the MCP for digital control and to implement various vibration suppression methods. A schematic of the equipment set-up for vibration control using the MCP is shown in Fig. 1. A cantilevered beam (its properties are shown in Table 1) is used as the object for vibration control. The beam has a piezoceramic sensor and three actuators on each side. Properties of the piezoceramics are shown in Table 2. The aluminium beam was clamped such that its length was parallel to the granite table below it. This allowed the bending to be strictly in the horizontal plane. The MCP is used to implement vibration suppression algorithms. The algorithm is first designed in a PC and then downloaded to the MCP. Using a TMS320C30 microprocessor, the MCP processes the data from the PZT sensor and generates a control signal according to the control algorithm. The control signal is then amplified and lastly sent to the PZT actuator(s) to suppress vibrations. A picture showing the aluminium beam, the MCP, the MCP analogue interface, the power supply for the MCP, the analogue interface low-voltage power supply and the analogue high-voltage power supply is presented in Fig. 2. The MCP power supply provides power to the MCP. The MCP analogue interface offers an interface between the MCP and the PZT sensors and actuators. The MCP analogue interface is connected to the PZT

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**Table 1** Cantilevered beam properties (aluminium beam type 7075 T-6)

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Description</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$t_b$</td>
<td>Beam thickness</td>
<td>mm</td>
<td>1.95</td>
</tr>
<tr>
<td>$w_b$</td>
<td>Beam width</td>
<td>mm</td>
<td>76.2</td>
</tr>
<tr>
<td>$L$</td>
<td>Beam length</td>
<td>m</td>
<td>1.176</td>
</tr>
<tr>
<td>$\rho_b$</td>
<td>Beam density</td>
<td>kg/m$^3$</td>
<td>2800</td>
</tr>
<tr>
<td>$E_b$</td>
<td>Young’s modulus</td>
<td>N/m$^2$</td>
<td>$7.1 \times 10^{10}$</td>
</tr>
</tbody>
</table>

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Fig. 1 Experimental set-up schematic
Table 2 Properties of the piezoceramic actuators and sensor [type PZT-5A (Navy type II)]

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Description</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d_{33}$</td>
<td>Lateral strain coefficient</td>
<td>C/N</td>
<td>$1.8 \times 10^{-10}$</td>
</tr>
<tr>
<td>$E_p$</td>
<td>Young’s modulus</td>
<td>N/m²</td>
<td>$6.3 \times 10^{10}$</td>
</tr>
<tr>
<td>$\varepsilon_3^1$</td>
<td>Absolute permittivity</td>
<td>F/m</td>
<td>$1.5 \times 10^{10}$</td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>PZT density</td>
<td>kg/m³</td>
<td>$7.7 \times 10^{11}$</td>
</tr>
<tr>
<td>$l_p$</td>
<td>PZT actuator and sensor thickness</td>
<td>mm</td>
<td>0.5</td>
</tr>
<tr>
<td>$w_p$</td>
<td>PZT actuator and sensor width</td>
<td>mm</td>
<td>38.1</td>
</tr>
<tr>
<td>$L_a$</td>
<td>Length of PZT actuators</td>
<td>mm</td>
<td>63.5</td>
</tr>
<tr>
<td>$L_s$</td>
<td>Length of PZT sensor</td>
<td>mm</td>
<td>25.4</td>
</tr>
</tbody>
</table>

sensor and sends beam vibration information to the MCP. The analogue interface is also connected to the PZT actuators and sends amplified control voltage to drive the PZT actuators. The high-voltage supply provides voltage amplification for the analogue interface to drive the PZT actuators. The low-voltage supply provides the regular operation of the analogue interface.

A dSPACE digital data acquisition system was used to record the experimental data. The dSPACE system incorporates a TMS320C40 digital signal processor. Using a DS2003 MUX/AD board, dSPACE can convert up to 32 analogue inputs to digital signals for processing. The real-time trace module of dSPACE, a windows-based graphical user interface, was used for data acquisition. The trace module permits saving of data in the Matlab.MAT format for post-processing and plotting. Matlab programs were written to identify the modal frequencies, calculate the modal energy drop in dB and plot the results.

Three inputs were provided to dSPACE for data recording. The first two were the PZT sensor output and the MCP output. The third one was the beam tip displacement. The displacement was detected by an NAIS ANL1651AC infrared laser analogue displacement sensor. The laser provides an output of 0.1 V/mm and has a dynamic range that is adjustable up to 1 kHz. It was set at 100 Hz for the beam experiments. This was more than sufficient since the first two modes of primary interest are below 10 Hz.

2.1 Experimental procedure

Both open- and closed-loop tests were performed. All tests were started by manually exciting the beam. This was a simple and effective method to excite the beam.

For single-mode vibration suppression, tests were run with either all three actuators operational or only the first actuator operational. For multimode suppression, only the first actuator was used since the locations of the second and the third actuators adversely impact damping for higher modes. For each test, data were obtained for a time interval of 15 s after beam excitation. This allowed ample time to measure damping effects. The experimental data were then processed to show the effectiveness of the tested control algorithm. A fast Fourier transform (FFT) was performed in Matlab to provide a power spectral density (PSD) plot of the beam response. The PSD gives a measure of signal energy level at different frequencies. A comparison of the ratios of the last-second modal energy level in dB to the initial one provides an indication of the damping effectiveness on this particular mode. Also, a direct comparison of the modal energy

![Diagram](image-url)
level drop with that of an open-loop response can indicate the effectiveness of the control algorithm.

Figure 3 shows the PSD plots for a multimode open-loop vibration. The solid line is for the first second of

the 15 s test and the dashed line for the last second. A Matlab program was written to identify the modes
excited and to compute the difference between the initial and final energy level in dB at the identified modal fre-
quencies. Table 3 shows energy level drops in dB for the first four modes in the 15 s free vibration. It is clear
that vibrations of the third and fourth modes quickly damp out. The first and second modes become the major concern for vibration suppression.

### Table 3: Natural damping of the aluminium beam

<table>
<thead>
<tr>
<th>Mode</th>
<th>First 1.33 Hz</th>
<th>Second 7.1 Hz</th>
<th>Third 19.0 Hz</th>
<th>Fourth 38.2 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>dB drop in 15 s</td>
<td>9.52</td>
<td>22.38</td>
<td>48.98</td>
<td>61.94</td>
</tr>
</tbody>
</table>

### 3 MODULAR CONTROL PATCH

The MCP was developed by TRW [6] and was aimed to develop a miniaturized multichannel digital controller
specifically for space-based vibration control and pointing systems. The one used at the Naval Postgraduate
School is a MCP-III controller. It has eight analogue inputs and six analogue outputs. In order to handle the
multiple analogue input and output channels, a time division multiplexing approach was adopted. This design
moves all the digitized data to and from the processor using the C30 expansion bus. Since the expansion
bus moves the data in parallel from the different inputs and outputs, the data from all of them can be moved
in a single processor cycle. Timing of the numerous devices is controlled by the ACTEL field programmable
gate array (FPGA). The Texas Instrument TMS320C30
(C30) incorporates a 32 bit floating point arithmetic, parallel instruction capability and on-chip random
access memory (RAM) [7].

The analogue board is specifically designed for piezo-ceramic sensors and actuators. Figure 4 shows a func-
tional overview of the MCP and its interface. The only

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**Fig. 3** PSD plots of free vibration of the beam

**Fig. 4** MCP overview
component not shown is the ACTEL FPGA which is used to control the timing of the numerous channels through the C30. The MCP digital board was intentionally designed to contain only the digital signal processing components and the A/D and D/A converter. All sensor and actuator electronics were omitted from the MCP. The intent was to make it possible to design analogue interface boards for each individual application. In this way, the MCP could be kept as a general-purpose device and the analogue interface could be made to utilize a number of different sensors and actuators. The analogue interface at the Naval Postgraduate School is designed to have all piezoelectric inputs and outputs.

The basic flow is to send an address through the FPGA to the input multiplexer (MUX), instructing it as to which analogue input to receive. A field effect transistor (FET) is used to select the desired input and transfer it to the MDAC. The output buffer amplifier of the MDAC then applies to the signal a gain from 0 to 9 before it is sent to a sample and hold device. The sample and hold device ensures that the input signal remains stable while it is being converted to a digital signal. The A/D device is the 12 bit Analog Devices 774, a CMOS device that operates at 80,000 samples/s. A resolution of 12 bit is currently the greatest resolution available in rad-hard devices. The analogue signal range of all the input devices is ±10 V. The A/D converter is operated with a 10 V reference so that 0 V signals get an output code in the middle of the range. The code corresponding to the digitized analogue signal is then transmitted to the processor across the expansion bus. The offset is subtracted in software before calculations are performed on the signal [6].

Data from all of the input channels are acquired before digital signal processing commences. Following calculations for each time step, digital command data are fed to all of the output channels for conversion to analogue. The offset is added back to the actuator command code before it is sent to the D/A converter, as the D/A converter also employs a 10 V offset signal [6].

The D/A converter signal is scaled by the output MDAC, which behaves identically to the input MDAC. The analogue output is then held by one of eight sample and hold devices. These sample and holds were built from a combination FET switch and hold circuit. The FPGA commands the FET switch to steer the voltage from the output MDAC to one of the eight hold circuit signals.

4 METHODS FOR VIBRATION SUPPRESSION

Two vibration suppression methods, namely PPF control and SRF control, are reviewed in this section. These two methods are implemented to suppress vibrations of the flexible beam using the MCP.

4.1 Positive position feedback control

For control of the flexible structures, the PPF control scheme shown in Fig. 5 is well suited for implementation utilizing the piezoelectric sensors and actuators. In PPF control, structural position information is fed to a compensator. The output of the compensator, magnified by a gain, is fed directly back to the structure. The equations describing PPF operation are given as

\[ \ddot{\xi} + 2\zeta\omega\dot{\xi} + \omega^2\xi = Go^2\eta \]

(1)

\[ \ddot{\eta} + 2\zeta\omega_\epsilon\dot{\eta} + \omega_\epsilon^2\eta = \omega^2\xi \]

(2)

where \( \xi \) is a modal coordinate describing displacement of the structure, \( \zeta \) is the damping ratio of the structure, \( \omega \) is the natural frequency of the structure, \( G \) is a feedback gain, \( \eta \) is the compensator coordinate, \( \zeta_\epsilon \) is the compensator damping ratio and \( \omega_\epsilon \) is the frequency of the compensator.

To illustrate the operation of a PPF controller, assume a single-degree-of-freedom vibration of the beam in the form

\[ \xi(t) = o\epsilon e^{int} \]

(3)

Then the output of the compensator at the steady state will be

\[ \eta(t) = \beta e^{i(o\epsilon - \phi)} \]

(4)

![Fig. 5 PPF block diagram](image_url)
In equation (4), the magnitude $\beta$ is given as

$$\beta = \frac{A(\omega / \omega_c)}{\sqrt{(1 - \omega^2 / \omega_c^2)^2 + (2\zeta \omega_c / \omega)^2}}$$

where $A = \alpha \omega_c / \omega$.

In equation (4), the phase angle $\phi$ is given as

$$\phi = \tan^{-1}\left(\frac{2\zeta \omega / \omega_c}{1 - \omega^2 / \omega_c^2}\right)$$

(5)

When the structure vibrates at a frequency much lower than the compensator natural frequency, the phase angle $\phi$ approaches zero according to equation (5). Substituting equation (4) with $\phi = 0$ into equation (1) results in

$$\ddot{\xi} + 2\zeta \omega \dot{\xi} + (\omega^2 - G\beta \omega^2)\xi = 0$$

(6)

It is clear from equation (6) that the PPF compensator in this case results in the stiffness term being decreased, which is called active flexibility. When the compensator and the structure have the same natural frequency, it can be derived from equation (6) that the phase angle $\phi$ approaches $\pi/2$. Substituting equation (4) with $\phi = \pi/2$ into equation (1), the structural equation becomes

$$\ddot{\xi} + (2\zeta \omega + G\beta \omega)\dot{\xi} + \omega^2 \xi = 0$$

(7)

Equation (7) shows that the PPF compensator in this case results in an increase in the damping term, which is called active damping. When the structure frequency is much greater than that of the compensator, the phase angle $\phi$ approaches $\pi$. Substituting equation (4) with $\phi = \pi$ into equation (1) results in

$$\ddot{\xi} + 2\zeta \omega \dot{\xi} + (\omega^2 + G\beta \omega^2)\xi = 0$$

(8)

It is clear from equation (8) that the PPF compensator in this case results in an increase in the stiffness term, which is called active stiffness. A plot of the phase angle $\phi$ versus frequency $\omega$ is shown in Fig. 6. As can be seen from the figure, to achieve maximum damping, $\omega_c$ should be closely matched to $\omega$. Also, any structural natural mode below $\omega_c$ will experience increased flexibility.

The effect of the damping ratio, $\zeta_c$, is as follows. Larger values of the damping ratio $\zeta_c$ will result in a less steep slope, thereby increasing the region of active damping. Figure 7 shows the Bode plot for $\zeta_c = 0.5$ and for $\zeta_c = 0.1$. The difference in the slopes of the phase angle can be easily seen. A larger value of $\zeta_c$ ensures a larger region of active damping and therefore will increase the robustness of the compensator with respect to uncertain modal frequencies. However, it is expected to result in slightly less effective damping and in increased flexibility at lower modes as a trade-off.

4.2 Strain rate feedback control

Strain rate feedback (SRF) control is achieved by feeding the structural velocity coordinate to the compensator. The compensator position coordinate is then fed back to the structure after a negative gain is applied. When using a PZT sensor and a PZT actuator, this is realized by feeding the derivative of the voltage from the sensor, which is proportional to the strain rate, to the input of the compensator and applying the negative compensator output voltage to the actuator. The equations of motion in modal coordinates are

$$\ddot{\xi} + 2\zeta \omega \dot{\xi} + \omega^2 \xi = -G\zeta \xi$$

(9)

$$\eta + 2\zeta \omega \dot{\eta} + \omega^2 \eta = \omega^2 \dot{\xi}$$

(10)

where the variables are the same as those defined for the case of PPF in the previous section. A block diagram illustrating this control scheme is shown in Fig. 8.
The phase plot for SRF is shown in Fig. 9. Again, assuming a single-degree-of-freedom vibration for the beam

\[ \ddot{\xi} = \alpha e^{j\omega t} \]  

(11)

the output of the compensator at steady state is

\[ \eta(t) = \beta e^{j(\omega t + \pi/2 - \phi)} \]  

(12)

In equation (12), the magnitude \( \beta \) is given by

\[ \beta = \frac{A(\omega/\omega_c)}{\sqrt{1 - \omega^2/\omega_c^2}^2 + (2\zeta/\omega_c)^2} \]

where \( A = \alpha\omega_c/\omega \). Also in equation (12), the phase angle \( \phi \) is given by

\[ \phi = \tan^{-1}\left( \frac{2\zeta/\omega_c}{1 - \omega^2/\omega_c^2} \right) \]  

(13)

When the structure vibrates at a frequency much lower than the compensator natural frequency, the phase angle \( \phi \) approaches zero according to equation (13). Substituting equation (12) with \( \phi = 0 \) into equation (9) results in

\[ \ddot{\xi} + 2\zeta\omega_c\dot{\xi} + \omega_c^2\xi = 0 \]  

(14)

It is clear from equation (14) that the SRF compensator in this case results in an increase in the damping ratio, which is called active damping. When the compensator and the structure have the same natural frequency, the phase angle \( \phi \) approaches \( \pi/2 \). In such a case, after substituting equation (12) with \( \phi = \pi/2 \) into equation (9), the structural equation becomes

\[ \ddot{\xi} + 2\zeta\omega_c\dot{\xi} + (\omega_c^2 + G\beta\omega^2)\xi = 0 \]  

(15)

Equation (15) shows that the SRF compensator in this case causes an increase in the stiffness term, which is called active stiffness. When the compensator frequency is much greater than that of the structure, the phase angle \( \phi \) approaches \( \pi \). Substituting equation (12) with...
\[ \phi = \pi \] into equation (9) results in a structural equation of the form

\[ \ddot{\xi} + (2\xi\omega - G\beta\omega)\dot{\xi} + \omega^2 \xi = 0 \]  \hspace{1cm} (16)

It is clear from equation (16) that the effect of the SRF compensator in this case is a decrease in the damping term, which is referred to as active negative damping. Thus, in implementing SRF, the compensator should be designed so that the targeted frequencies are below the compensator frequencies.

The SRF has a much wider active damping frequency region, which gives the designer some flexibility. Selecting a precise compensator frequency for SRF is not as critical as for PPF. As long as the compensator frequency is greater than the structural frequency, a certain amount of damping will be provided. A big limitation to SRF is that the magnitude of the transfer function in the active damping region becomes extremely small very quickly. Therefore, the amount of damping provided over a certain range is limited.

5 EXPERIMENTAL RESULTS OF SINGLE-MODE VIBRATION SUPPRESSION

5.1 PPF experimental results

Positive position feedback control was implemented using the MCP to suppress the vibration of the first and second modes respectively. Experimental results of the PPF implementation are shown in Table 4. The last column shows the percentage ratio of achieved modal energy drop in dB to that of the free vibration. Different gain values were tested. Table 4 indicates that higher gains achieved high vibration reduction. However, experiments revealed that larger gains were more likely to cause instability. The compensator damping ratio \( \zeta_c \) was set to 0.5. This was chosen as a compromise between damping effectiveness and robustness.

Figure 10 shows the result of PPF control, targeting the first mode. All three actuators were used. The energy level of the first mode dropped 72 dB during the 15 s PPF active control, compared with only 9.52 dB for free vibration. Figure 11 shows the result of PPF control, targeting the second mode. Only the first actuator was employed. A drop of 44 dB is observed for the second mode during the 15 s active control. By comparison with the 22.38 dB drop in the free vibration, PPF achieved 97 per cent more vibration reduction in terms of the modal energy drop. In this case, the energy level of the first mode also dropped, by 24 dB. This is attributed to the large value of the compensator damping ratio, \( \zeta_c = 0.5 \), which provides a wide frequency region for active damping. In all cases, there was no excitation in higher modes.

5.2 SRF experimental results

Strain rate feedback controls were implemented using the MCP on the aluminium beam. The compensator damping ratio \( \zeta_c \) for this experiment was set to 0.02. The compensator frequency was chosen so that the
targeted frequency fell in the active damping range with its magnitude as high as possible. This was to limit the active stiffness area and to maximize the active damping region with as much gain as possible.

Experimental results of SRF implementations are shown in Table 5. The last column shows the percentage ratio of achieved modal energy drop in dB to that of the free vibration of the beam. As can be seen from the table, SRF was not as effective in damping the targeted mode as PPF. It only reached 50 per cent of the damping achieved by PPF on the first mode. It achieved the same reduction when targeting the second mode.

Figure 12 shows a PSD plot of an SRF filter using three actuators and targeting the first mode. Only limited vibration reduction was achieved on the targeted mode. Negative damping was observed for higher frequencies.

Fig. 10  PSD plot for PPF using three actuators and targeting the first mode

Fig. 11  PSD plot for PPF using one actuator and targeting the second mode
Table 5  SRF results on single-mode damping

<table>
<thead>
<tr>
<th>Target mode</th>
<th>Parameters</th>
<th>Modal dB drop</th>
<th>Change (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$\omega_0 = 2.5$ Hz, $\zeta_c = 0.02$, gain = 1, 3 actuators</td>
<td>22.79</td>
<td>139</td>
</tr>
<tr>
<td>1</td>
<td>$\omega_0 = 2.5$ Hz, $\zeta_c = 0.02$, gain = 2, 3 actuators</td>
<td>32.34</td>
<td>240</td>
</tr>
<tr>
<td>1</td>
<td>$\omega_0 = 2$ Hz, $\zeta_c = 0.02$, gain = 2, 1 actuator</td>
<td>32.60</td>
<td>242</td>
</tr>
<tr>
<td>2</td>
<td>$\omega_0 = 10$ Hz, $\zeta_c = 0.02$, gain = 2, 1 actuator</td>
<td>44.10</td>
<td>97</td>
</tr>
</tbody>
</table>

The energy level of the third mode was higher than that of the case of free vibration. This observation reflects one disadvantage of the SRF control—active negative damping in the higher frequency region.

6  EXPERIMENTS ON MULTIMODE VIBRATION SUPPRESSION

The lack of success in damping the first two modes of the beam with a single control law led to the use of two control laws in parallel to increase the effectiveness of multimodal damping.

6.1  PPF–PPF control

Two PPF filters in parallel were tested. The first PPF filter targeted the first mode and the second PPF filter targeted the second mode. Since a PPF filter introduces active flexibility for frequencies lower than the targeted one, the second PPF filter may adversely affect the vibration suppression of the first mode. Therefore, the second PPF filter initially used a relative small gain of 2, while the first PPF filter used a gain of 6. To increase the robustness of both filters, a damping ratio of 0.5 was used for both. A Bode plot of this PPF–PPF controller is shown in Fig. 13. Note that the phase angle reaches a value of approximately 45° at the first mode. Positive
Table 6 Experimental results on multimode damping

<table>
<thead>
<tr>
<th>Controller type</th>
<th>Controller parameters</th>
<th>First mode dB drop (%)</th>
<th>Second mode dB drop (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PPF (first mode)</td>
<td>$\alpha = 1.3$ Hz, $\zeta = 0.5$, gain = 6</td>
<td>68.02</td>
<td>31.68</td>
</tr>
<tr>
<td>PPF (second mode)</td>
<td>$\alpha = 7.1$ Hz, $\zeta = 0.5$, gain = 2</td>
<td>(614)</td>
<td>(42)</td>
</tr>
<tr>
<td>PPF (first mode)</td>
<td>$\alpha = 1.3$ Hz, $\zeta = 0.5$, gain = 6</td>
<td>58.33</td>
<td>44.00</td>
</tr>
<tr>
<td>PPF (second mode)</td>
<td>$\alpha = 7.1$ Hz, $\zeta = 0.5$, gain = 6</td>
<td>(513)</td>
<td>(97)</td>
</tr>
<tr>
<td>PPF (first mode)</td>
<td>$\alpha = 1.3$ Hz, $\zeta = 0.5$, gain = 6</td>
<td>50.85</td>
<td>52.11</td>
</tr>
<tr>
<td>SRF (first mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.9</td>
<td>(434)</td>
<td>(133)</td>
</tr>
<tr>
<td>SRF (second mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.04</td>
<td>72.84</td>
<td>38.98</td>
</tr>
<tr>
<td>SRF (first mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.04</td>
<td>(665)</td>
<td>(74)</td>
</tr>
<tr>
<td>SRF (second mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.04</td>
<td>16.09</td>
<td>42.70</td>
</tr>
<tr>
<td>SRF (first mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.04</td>
<td>(69)</td>
<td>(91)</td>
</tr>
<tr>
<td>SRF (second mode)</td>
<td>$\alpha = 10$ Hz, $\zeta = 0.02$, gain = 0.04</td>
<td>30.04</td>
<td>34.04</td>
</tr>
</tbody>
</table>

damping on this mode was expected. Experimental data confirmed this expectation. The results are shown in the first row of Table 6: a 68.02 dB drop for first mode and a 31.68 dB drop for the second mode. The strong damping on the first mode suggests that the adverse effect of the second filter on the first one is very limited. Based on this observation, the gain for the second PPF filter was increased to 6 to increase damping on the second mode. Experimental data verified that increasing the gain of the second filter from 2 to 6 increased the damping on the second mode. The energy level on the second mode dropped further to 44.0 dB. Although this did lessen the damping on the first mode slightly, it doubled the percentage damping on the second mode, resulting in a more effective controller. Figure 14 is a PSD plot showing the effectiveness of the controller with a gain of both PPF filters of 6.

6.2 PPF–SRF control

The next pair tested was a PPF filter with an SRF filter.

Fig. 14 PSD plot for two PPF filters in parallel

Fig. 15 Bode plot for PPF combined with SRF
Since the SRF filter introduces active negative damping at frequencies higher than the targeted one, a PPF is chosen to target the first mode and an SRF to target the second. To increase the robustness of the PPF filter, a damping ratio of 0.5 was used. For the SRF filter, a gain of 0.9 was initially used. A Bode plot for this controller is shown in Fig. 15. The Bode plots indicated that both the first and second modes should have positive damping, but the gain does not drop quickly for higher frequencies. It was suspected that high modes might be excited by the high SRF gain through strong damping for the two lower modes. Experimental data shown in the third row of Table 6 confirmed this expectation. The first mode and the second mode had an energy drop of 50.85 dB and 52.11 dB respectively, but the fourth mode was excited. This can be seen on the graph of the PSD in Fig. 16. The gain on the SRF was then lowered to 0.04. No higher mode was excited and strong damping was still observed for both modes: a 72.84 dB energy drop for the first mode and a 38.98 dB drop for the second mode, as shown in the fourth row of Table 6.

6.3 SRF–SRF control

The next combination tested was a controller using two SRF filters. Since the SRF filter introduces active negative damping at frequencies higher than the targeted one, the first SRF filter will adversely affect the performance of the second. Therefore, SRF–SRF control is
not expected to be very effective. Table 6 shows two cases of SRF–SRF control. A Bode plot for SRF–SRF control is shown in Fig. 17. The gain was initially set at 0.04 for both filters. This produced effective damping on the second mode but very little on the first. For the second case, the gain was increased to 1 for the first filter to improve the damping on the first mode. The first mode energy drop was increased from 16.09 to 30.04 dB, but the increased gain also increased the active negative damping from the first filter and thus the second mode energy drop decreased from 42.7 to 34.04 dB. This can be seen on the PSD plot in Fig. 18. Compared with the PPF–PPF control cases, the SRF–SRF control is not effective in damping out multimode vibrations.

8 CONCLUSION

This research presents the experimental results of vibration suppression of a flexible structure using the modular control patch, a miniaturized electronic controller. The MCP employs a TI-C30 digital signal processor and is used to implement the control algorithms in this research. The flexible structure is a cantilevered beam with attached sensors and actuators. Positive position feedback and strain rate feedback controls were implemented independently for single-mode vibration suppression and in combinations for multimode vibration suppression. Experiments found that PPF control is most effective for single-mode vibration suppression and that parallel PPF–PPF control is most effective for the multimode case. Piezoceramics proved to be good material for both the sensor and actuator for vibration control. During the experiments, the MCP demonstrated the capability effectively to implement real-time control laws. The MCP has the potential to be used in space-based vibration controls.

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