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# Resetting Velocity Feedback: Reset Control for Improved Transient Damping

M.A. Mohan<sup>1 2</sup>, M.B. Kaczmarek<sup>2</sup> and S.H. HosseinNia<sup>2</sup>

**Abstract**—Active vibration control (AVC) is crucial for the structural integrity, precision, and speed of industrial machines. Despite advancements in nonlinear control techniques, most AVC techniques predominantly employ linear feedback control due to their simplicity and ability to be designed in the frequency domain. In this paper, we introduce a reset-based nonlinear bandpass filter that uses velocity feedback to improve transient damping of vibrating structures. The approach is motivated from an energy-based mechanistic analysis, which incentivizes the use of reset. A novel feature of our approach is that it works for non-ideal, naturally damped systems, and enables control design in the frequency domain, inline with industrial practice. We demonstrate the effectiveness of this new filter by numerical simulations and experimental validation on a single degree-of-freedom flexure stage.

## I. INTRODUCTION

As the age of digitalization evolves rapidly, there is an ever-increasing demand for improving precision and decreasing production times for industrial automation in general, and semiconductor manufacturing in particular. As these complex machines incorporate flexural elements to overcome friction and backlash, structural vibrations pose a new challenge. The structural resonance modes result in vibrations that reduce the precision, and considerably increase settling times, and thus decreasing the productivity of such devices. Hence, the need for controlling and quickly damping these vibrations is paramount.

Active vibration control is a well-studied problem [20]. In most industries, linear control methods are the most dominant. These techniques have the advantages of being easy to analyze and tune, owing to their analysis and design in the frequency-domain. This is particularly advantageous in an industrial setting where loop-shaping methods are preferred. However, nonlinear control techniques hold better promise compared to linear ones, as they are not restricted by Bode's gain-phase relationship, and hence offer better flexibility and trade-offs [11].

The majority of existing nonlinear active vibration control techniques focus on steady-state damping performance. In [7], a nonlinear technique called QMPF was used to damp forced vibrations. In [15], another nonlinear technique called delayed resonant feedback was used, based on the principle

of an electrical realization of a delayed vibration absorber, to absorb the steady-state resonant oscillations. Apart from not considering transient damping performance, frequency-domain analysis tools are rendered ineffective for the latter. Therefore, we require a nonlinear control technique that can be designed in the frequency-domain, thereby increasing its relevance for industries.

Reset control satisfy both these requirements as it is a nonlinear hybrid control technique that uses jumps in state trajectories to improve transient performance [11][17][18], and allows design in the frequency-domain using Describing Functions (DF). To this end reset was used in the insightful work in [4] which introduced Resetting Virtual Absorbers (RVAs) to achieve finite-time vibration attenuation for plants without damping. This analysis was motivated by energy principles and damping injection through reset, thereby providing a clear incentive for the use of reset. However, the effectiveness of this technique reduces with non-zero plant damping. Moreover, the absence of frequency-domain techniques to design RVAs limits their adoption by industries. Hybrid Integrator Gain System (HIGS) is another nonlinear hybrid control technique initially introduced in [9]. In [12], a HIGS bandpass filter was used to improve transient response for active vibration isolation compared to linear techniques. Although the HIGS bandpass filter was designed using describing functions in the frequency-domain, the underlying mechanism by which it provides better transient response, and its relation to the frequency-domain attributes were not fully explored. In [16], reset control was used to inject damping by employing an optimal port-Hamiltonian approach. However, this also did not employ frequency-domain techniques to design reset controllers for active damping.

This begs the question: How can reset control be used to guarantee better transient damping performance for damped systems compared to linear control? Furthermore, how can they be systematically designed in the frequency-domain, to ease tuning, implementation, and adoption by industries?

In this paper, we introduce a novel reset-based bandpass filter that employs velocity feedback to achieve finite-time vibration suppression for damped systems and compare its performance to a linear bandpass filter. We start with the work of [3] and adopt a velocity feedback framework to extend its effectiveness to damped plants. Since the original analysis in [3] stems from an energy-based mechanistic approach, our technique also provides an understanding of the underlying mechanism for the improved transient response, as opposed to HIGS. Furthermore, owing to the more

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<sup>1</sup>M.A. Mohan is currently with the Institute for Dynamic Systems and Control, ETH Zurich, Switzerland and was with the Department of Precision and Microsystems Engineering, Delft University of Technology, The Netherlands while conducting this study mmohan@ethz.ch

<sup>2</sup>M.B. Kaczmarek and S.H. HosseinNia are with the Department of Precision and Microsystems Engineering, Delft University of Technology, The Netherlands S.H.HosseinNiaKani@tudelft.nl

aggressive nature of reset compared to HIGS, improvement in transient response is possible with lower control gains which makes our method more energy efficient.

We also develop tuning rules based on describing functions to design this filter in the frequency-domain, thereby increasing its relevance for industries. Staying true to this ethos, the stability of the novel bandpass filter and the closed-loop system is proven using passivity arguments, which only require the base linear transfer function of the reset element and the transfer function of the plant.

We also experimentally demonstrate the effectiveness of the Resetting Velocity Feedback (RVF) framework for transient damping using a single degree-of-freedom flexure stage. While preliminary, our experimental results show great promise and agree with numerical simulations.

This paper is organized as follows: In Section II we introduce the preliminaries on Reset Control, Negative Derivative Feedback, and Resetting Virtual Absorbers independently. Combining ideas from these approaches, we develop the Resetting Velocity Feedback framework in Section III, and analyze its stability properties. Numerical and experimental results of this novel technique follow in Section IV. Finally, Section V summarizes the study and suggests recommendations for future work.

## II. PRELIMINARIES

### A. Reset Control

Reset control systems are a class of hybrid dynamical systems. In this study we are concerned with zero-crossing reset systems with the general dynamics given by:

$$R: \begin{cases} \dot{x}(t) = Ax(t) + Bu(t) & u(t) \neq 0 \\ x(t^+) = A_r x(t) & u(t) = 0, \\ y(t) = Cx(t) + Du(t) \end{cases} \quad (1)$$

where the matrices  $A, B, C$ , and  $D$  describe the state-space matrices of the reset element,  $u(t)$  is the error input,  $x(t)$  are the states, and  $y(t)$  is the controller output. The linear dynamics given by the first and third equations of Equation (1) are referred to as the base linear system. The controller states propagate according to the base linear system if the input  $u(t) \neq 0$ . Whenever the reset conditions are met, i.e.,  $u(t) = 0$ , specified controller states are reset according to the reset matrix  $A_r$ . This work focuses on reset elements based on zero-crossings of the input signal  $u$  (velocity), with full reset ( $A_r = 0$ ), as they are the most widely studied, applied, and tested [18].

### B. Describing Functions

Describing Functions (DF) are a quasi-linearization of a nonlinear element subject to certain excitation input used to approximately analyze the behaviour of nonlinear systems. DF is a powerful tool for investigating behaviours of elements with hard nonlinearities including dead zone, backlash, and hysteresis, and has been applied in limit cycle predictions and control design [19]. Furthermore, DFs allow us to analyze reset systems in frequency domain and apply

loop-shaping like techniques for control design, which is the de-facto standard in industries [17][21].

The Sinusoidal Input DF (SIDF), which uses sinusoidal inputs as excitation signals, is the most widely used describing function technique to analyze reset systems [10][17]. The SIDF of a reset element can be represented as  $G(j\omega)$  as introduced in [10]

$$\begin{aligned} G(\omega) &= C(j\omega I - A)^{-1} (I + j\Theta_D(\omega)) B + D \\ \text{with } \Theta_D(\omega) &= -\frac{2\omega^2}{\pi} \Delta(\omega) [\Gamma_r(\omega) - \Lambda^{-1}(\omega)] \\ \Lambda(\omega) &= \omega^2 I + A^2 \\ \Delta(\omega) &= I + e^{\frac{\pi}{\omega} A} \\ \Delta_r(\omega) &= I + A_r e^{\frac{\pi}{\omega} A} \\ \Gamma_r(\omega) &= \Delta_r^{-1}(\omega) A_r \Delta(\omega) \Lambda^{-1}(\omega) \end{aligned} \quad (2)$$

where  $A, B, C, D$  are the state-space matrices of the reset element. Although reset systems are nonlinear, DFs only depend on the frequency of the input signal and not on the magnitude, unlike certain other nonlinear systems. This makes them an ideal candidate for frequency-domain analysis.

### C. Negative Derivative Feedback

Direct Velocity Feedback (DVF) is a well understood active damping technique in which structural (modal) velocity is negatively feedback to impart damping [1]. In [6], this technique was extended by feeding back velocity through a bandpass filter tuned to the eigenfrequency of the mode to eliminate low- and high-frequency spillover. This is termed as Negative Derivative Feedback (NDF).

For such a system, a physical analogy can be drawn as pointed out by [14]. The closed-loop negative feedback interconnection of a plant and a linear bandpass filter (controller) represents an oscillator-vibration absorber setup, where the controller performs the role of an emulated dynamic vibration absorber. This mechanical analogy will be exploited later on in Section III to develop RVF and prove stability.

### D. Resetting Virtual Absorbers

The concept of a resetting virtual absorber is based on the insightful analysis found in [3][4]. This shows how reset can inject damping into an undamped system consisting of a single degree-of-freedom oscillator and an emulated vibration absorber, for finite-time vibration suppression.

Consider a single degree-of-freedom undamped oscillator with mass  $M$  and stiffness  $K$ , in series with an emulated dynamic vibration absorber with mass  $m$  and stiffness  $k$ , whose states (position and velocity), can be instantaneously reset to zero. This is shown in Fig. 1. According to the physical analogy of vibration controllers introduced earlier in Section II-C, this emulated vibration absorber can be considered an active damping controller, which uses the position of mass  $M$  as its input. The state-space matrices of such a controller are shown in Equation (3). This is the state

space representation of a mass spring system with velocity and position as its two states.

$$A = \begin{bmatrix} 0 & -\frac{k}{m} \\ 1 & 0 \end{bmatrix}, \quad B = \begin{bmatrix} 1 \\ 0 \end{bmatrix}, \quad C = \begin{bmatrix} 0 & -\frac{k^2}{m} \end{bmatrix}, \quad D = k \quad (3)$$

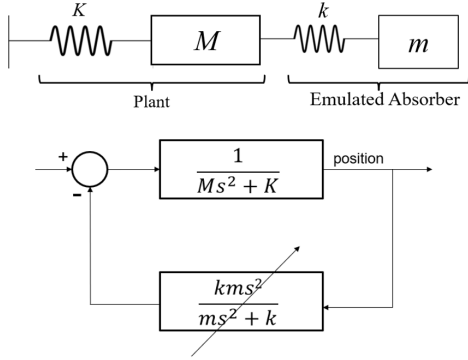


Fig. 1. The model of a Resetting Virtual Absorber and its block diagram representation. The diagonal arrow denotes a reset element

For an impulse excitation to mass  $M$ , for  $m = 1.33M$  and  $k = 1.33K$  [3], the response of the system is shown in Fig. 2. When the position of mass  $M$  reaches zero for the first time, the velocities of masses  $M$  and  $m$  are also simultaneously zero, as indicated in Fig. 2. Therefore, the energy contained in the system at this instant is uniquely due to the compression of spring  $k$ , i.e., the non-zero position of mass  $m$ . In other words, all the energy is contained within the emulated absorber.

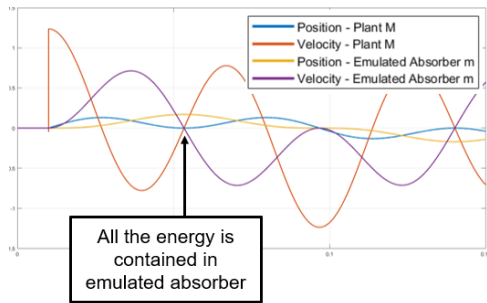


Fig. 2. Response of the system in Fig. 1 to an impulse on mass  $M$

If the states (virtual position and velocity) of the emulated absorber (controller) are reset to zero at this exact instant, the total energy in the system is instantaneously dissipated before it redistributes it to the states of mass  $M$ . Hence, damping is injected into the system by resetting the states of the controller. This results in finite-time suppression of the oscillations of mass  $M$ , as shown in Fig. 3. Reset is the only source for damping as this system has no natural damping whatsoever. The control effort is the force provided by the compression of spring  $k$  and can be seen to follow a smooth

trajectory until the point where the position of mass  $M$  hits zero. It is then instantaneously reset to zero as shown in Fig. 3.

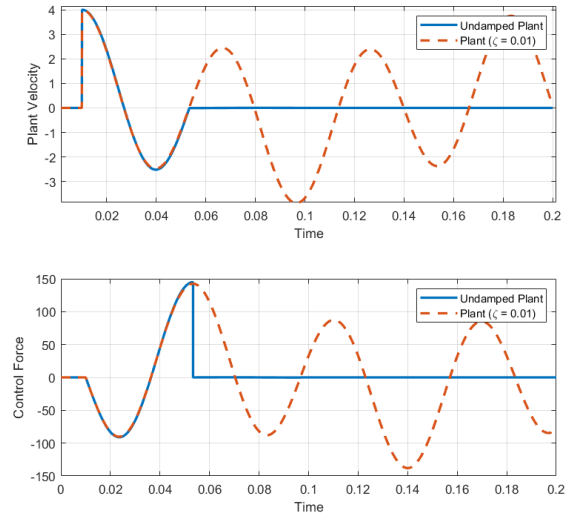


Fig. 3. Finite-time vibration suppression using RVA for undamped plants (in blue). The RVA framework is rendered ineffective even for very low levels of inherent plant damping ( $\zeta = 0.01$ ), as shown in dashed-red, resulting in prolonged oscillations. This motivates velocity feedback

### III. FRAMEWORK

In this section, the three independent concepts introduced in Section II are combined to develop the Resetting Velocity Feedback (RVF) framework. The framework is summarized schematically in Fig. 4.

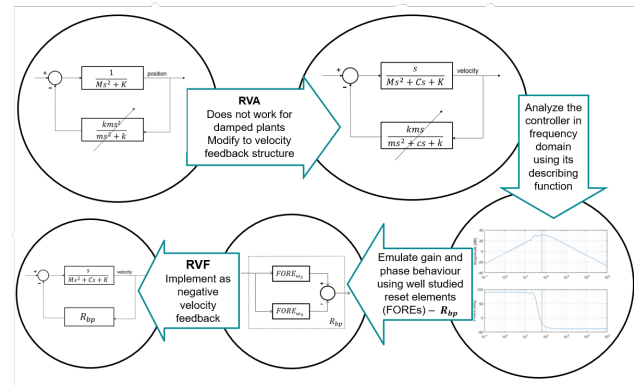


Fig. 4. Development of the Resetting Velocity Feedback framework at a glance

#### A. From RVA to RVF

As seen earlier, RVAs are based on position feedback of undamped plants, designed to reset when plant position hits zero. However, therein lies the problem: for damped plants, the plant position no longer hits zero at the desired point, but remains ever so slightly positive, as shown in Fig. 5. Even though this time instant corresponds to a point of minimum

plant and maximum controller energy, a reset law based on zero crossings of position law no longer holds, and does not result in finite-time vibration suppression, as shown in Fig. 3. Fortunately, the plant velocity does cross zero at this point (of minimum plant energy) and thus motivates the use of a reset law based on velocity feedback. This ensures the reset of controller states when the controller has maximum energy, thereby taking away most of the energy in the system and injecting damping.

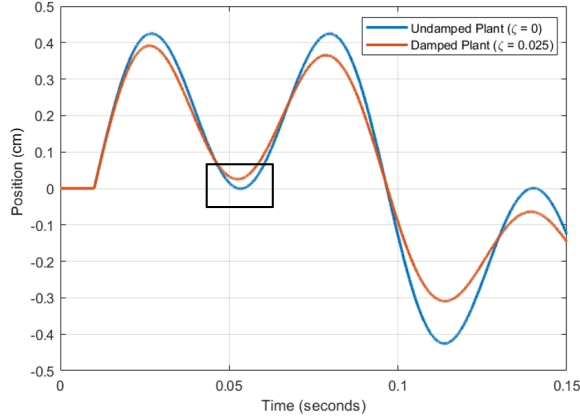


Fig. 5. Position trajectories for damped and undamped plants showing the ineffectiveness of zero-crossing position feedback. The point-of-interest is highlighted with a rectangular box

Our feedback framework needs to be slightly modified to account for velocity feedback compared to Fig. 1. This framework resembles the NDF system presented earlier, with the linear bandpass controller being replaced by a reset alternative. A damping term,  $c$ , is also added to the controller, with the same damping ratio of the plant ( $c = \frac{\sqrt{kmC}}{\sqrt{KM}}$ ), to match the resonance frequency of the emulated absorber to that of the plant.

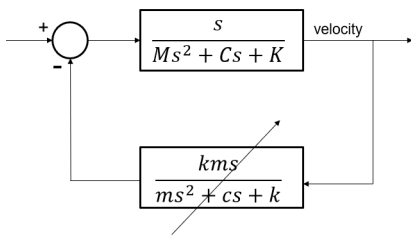


Fig. 6. Modified block diagram to account for velocity feedback which also includes a damping term in the controller structure

As the main aim of our work is to systematically design reset controllers in frequency domain, we analyze this “new” reset filter in frequency domain using its describing function. This is shown in Fig. 7. Two aspects are worth noting:

- At the resonance frequency of the controller  $\omega_c = 65 \text{ rad/s}$ , the describing function has a gain of 31.9 dB. This value depends on  $k$  and  $m$ , which are in turn related to  $K$  and  $M$  as mentioned in the previous section. For systematic design and tuning, it is essential to introduce a non-dimensional parameter which

can be used to design controllers for any given plant parameters. We define the non-dimensional parameter  $\zeta_{active} = \frac{km}{2\sqrt{KM}}$  for this purpose. We see that a gain of 31.9 dB corresponds to a dimensionless active damping ratio  $\zeta_{active} = 0.55$ .

- At  $\omega_c$ , phase is approximately  $-20^\circ$  (different from a linear bandpass filter which has a phase of  $0^\circ$  at its resonance frequency). The phase becomes slightly less negative with increased damping,  $c$ . As long as the  $\frac{k}{m}$  ratio is maintained, changing  $m$  and  $k$  does not affect the phase characteristics, but only adds a constant gain.

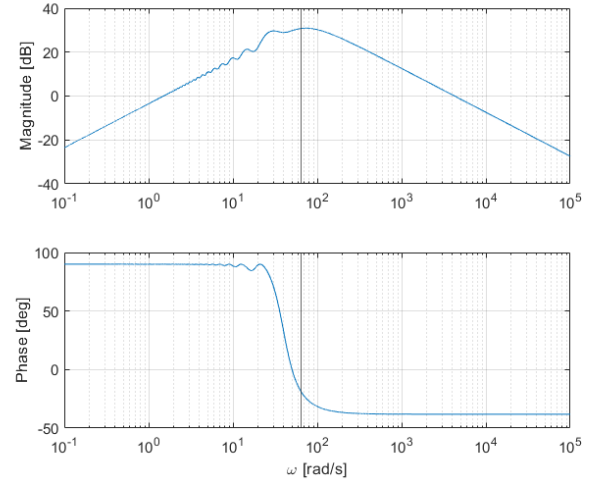


Fig. 7. Describing function of the reset element in Fig. 6. The vertical line highlights the target frequency

Although this reset filter appears promising from describing function analysis, reset systems with second-order bandpass structures are not well-studied. Reset systems are also special in the sense that, for the same base linear system, different state-space realizations result in different closed-loop behaviour. Hence, our aim is to “emulate” the  $\zeta_{active}$  and phase of this describing function at  $\omega_c$ , with a reset bandpass filter built from commonly-used and well-understood reset elements. This is obtained by subtracting a First Order Reset Element (FORE) [13] with corner frequency  $\omega_1$  from a FORE with corner frequency  $\omega_2$ , with  $\omega_2 > \omega_1$ . Since FOREs have a unique state-space realization, this approach eliminates ambiguities on which state-space realization to implement. The well-studied nature of FOREs in the reset control community also makes the choice of using them straight-forward. These are tuned to the appropriate gain and phase value at  $\omega_c$ . The resulting FORE-based bandpass filter for active damping can be represented as

$$R_{bp} = g(\text{FORE}_{\omega_2} - \text{FORE}_{\omega_1}), \quad (4)$$

where  $g$  is the gain required to ensure  $\zeta_{active} = 0.55$ . The corner frequencies of the FOREs are obtained by solving an optimization problem involving the phase of the describing function of  $R_{bp}$ , formulated as:

$$\operatorname{argmin}_{\omega_1, \omega_2} |\angle R_{bp} - 20^\circ|.$$

The describing function of this novel FORE-based bandpass filter is shown in Fig. 8 and compared to the second-order bandpass filter introduced earlier. Clearly, the magnitude and phase at the plant eigenfrequency  $\sqrt{K/M} = 65 \text{ rad/s}$ , are equal.

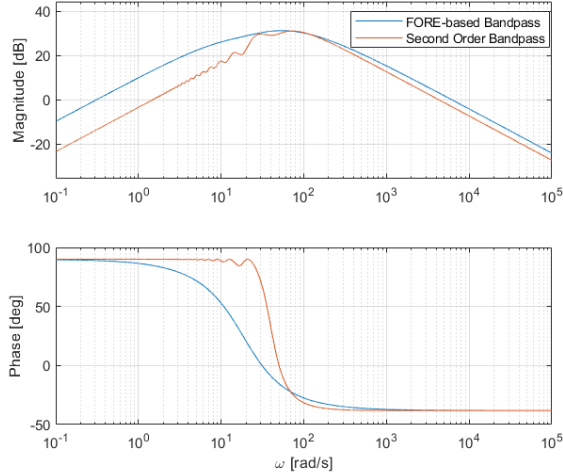


Fig. 8. Describing function of a FORE-based bandpass filter ( $R_{bp}$ ) compared to the second-order bandpass filter in Fig. 6. Both provide the same magnitude and phase values at the target frequency of 65 rad/s

For easy implementation, the following heuristics are nearly optimal for plants with 1%-2% natural damping:

- Choose  $\omega_1 = 0.4\sqrt{K/M}$  and  $\omega_2 = 0.75\sqrt{K/M}$ . This ensures a phase of about  $-20^\circ$  at the target frequency. This value need not be exact, and small deviations of about  $1^\circ$ , can be easily compensated by tuning the gain. This is further illustrated by performing sensitivity analysis in Section IV. This approximate nature of the phase and gain probably stems from the fact that describing functions themselves are an approximation of the actual reset system. This is also advantageous if the plant's eigenfrequency is not known exactly, as fine-tuning the gain can result in optimal performance.
- Choose gain  $g$ , such that the magnitude of the describing function of  $R_{bp}$ ,  $|R_{bp}|_{\omega=\sqrt{K/M}} = 2\zeta_{active}\sqrt{KM}$ , where  $\zeta_{active} = 0.55$ , provided the plant transfer function is of the form as shown in Fig. 6. This ensures a dimensionless active damping ratio of  $\zeta_{active} = 0.55$  at the plant's eigenfrequency, which is required for optimal performance.

Once the parameters have been calculated, the gain value can be fine-tuned to achieve optimal transient damping performance. The closed-loop system with such a FORE-based bandpass filter interconnected with a damped single degree-of-freedom plant constitutes the Resetting Velocity Feedback (RVF) framework.

## B. Stability Analysis

We now address the stability properties of (1) the reset bandpass filter  $R_{bp}$  and, (2) the closed-loop system of RVF. The following three theorems on passivity will serve as a baseline for this analysis:

**Theorem 1:** [5] For an LTI system with transfer function  $H(s) = C(sI - A)^{-1}B + D$ , with  $A$  Hurwitz and the pair  $(A, B)$  controllable, it holds that: The system is passive if and only if  $\operatorname{Re}(H(j\omega)) \geq 0$  for all  $\omega$ . The system is Output Strictly Passive (OSP) if and only if there is an  $\epsilon$  such that  $\operatorname{Re}[H(j\omega)] \geq \epsilon|H(j\omega)|^2$  for all  $\omega$ .

**Theorem 2:** [5] A full reset compensator  $R$  is passive, Input Strictly Passive (ISP), OSP, or Very Strictly Passive (VSP) if the base compensator is passive, ISP, OSP, or VSP, respectively.

In our case, the reset bandpass filter  $R_{bp}$  is formed by two FOREs in parallel with corner frequencies  $\omega_1$  and  $\omega_2$  with  $\omega_2 > \omega_1 > 0$ , resulting in the following base linear transfer function:

$$R_{bp}(j\omega) = \frac{(\omega_2 - \omega_1)j\omega}{(j\omega + \omega_1)(j\omega + \omega_2)} \quad (5)$$

$$\operatorname{Re}[R_{bp}(j\omega)] = \frac{\omega^2(\omega_2 - \omega_1)(\omega_2 + \omega_1)}{(\omega_1\omega_2 - \omega^2)^2 + (\omega_1 + \omega_2)^2\omega^2} \quad (6)$$

$$|R_{bp}(j\omega)|^2 = \frac{\omega^2(\omega_2 - \omega_1)(\omega_2 + \omega_1)}{(\omega_1\omega_2 - \omega^2)^2 + (\omega_1 + \omega_2)^2\omega^2}. \quad (7)$$

Clearly,  $\operatorname{Re}[R_{bp}(j\omega)] \geq 0$  for all  $\omega$ , and  $\operatorname{Re}[R_{bp}(j\omega)] \geq \epsilon|R_{bp}(j\omega)|^2$  for  $\epsilon = 1$ . Hence, according to Theorem 1 and Theorem 2, the reset-based bandpass filter is OSP.

Similarly, it can be shown that the plant, with the transfer function from force to velocity, is also OSP, as its transfer function also has a similar structure.

**Theorem 3:** [5] The negative feedback interconnection between an LTI plant  $P$  and a full reset compensator  $R$ , with base linear compensator  $R_{bl}$ , is finite-gain stable if one of the following conditions are satisfied:

- $R_{bl}$  is ISP, and  $P$  is ISP
- $R_{bl}$  is OSP, and  $P$  is OSP
- $R_{bl}$  is VSP, and  $P$  is passive
- $R_{bl}$  is passive, and  $P$  is VSP

We can assert that the negative feedback interconnection of the reset-based bandpass filter and the plant with velocity output is finite-gain stable, as they are both OSP. Hence the Resetting Velocity Feedback framework is finite-gain stable.

## IV. RESULTS

While describing functions simplify the analysis of reset systems, they are still only approximations of the actual system. Empirical evidence is needed to determine whether the framework developed earlier holds true in practice. In this section we report evidence on the previously developed Resetting Velocity Feedback structure, through numerical simulations and experimental testing.



To focus on controller validation, the physical system is chosen to be a simple single degree-of-freedom flexure stage. A Lorentz actuator (ETEL 025C) is used to provide both, the disturbance signal and the control force. A Polytec OFV-505 Laser Doppler Vibrometer (LDV) is used as a velocity sensor, which provides voltage signals proportional to the vibration velocity. National Instruments compactRIO FPGA is used to acquire the signals, and compute and deliver the control signal to the current-source power amplifier. LabVIEW 2020 is used to interface the host computer with compactRIO. The experimental setup is shown in Fig. 9.

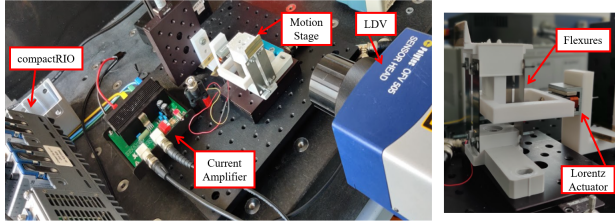


Fig. 9. Experimental setup (left) and motion stage (right)

System identification is performed by exciting the system with a chirp (swept sine) signal. Using MATLAB's System Identification Toolbox, the Frequency Response Function (FRF) of the system is obtained, as shown in Fig. 10. As expected, this corresponds to the transfer function from force to velocity, and is a reasonable estimate of a single degree-of-freedom lumped-mass model, with an eigenfrequency at 13.2 Hz. A transfer function is fitted to the experimental data using MATLAB's `tfestimate` function, and is given by

$$P(s) = \frac{555.2s}{s^2 + 2.882s + 6972}. \quad (8)$$

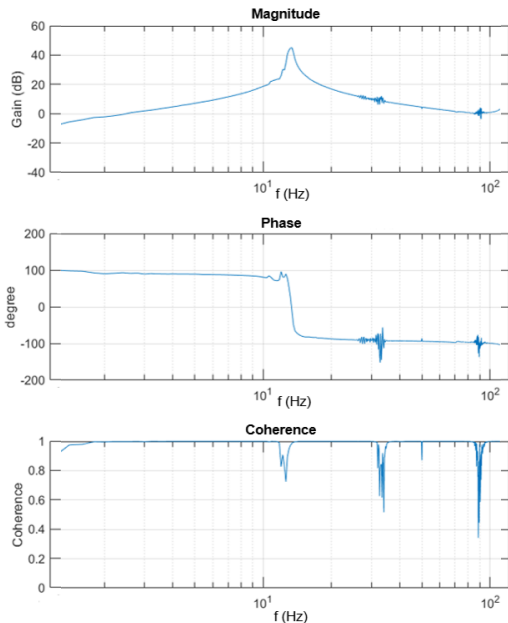


Fig. 10. FRF of the experimentally identified system

With the plant parameters identified experimentally, the method described in Section III is followed to design the reset bandpass filter,  $R_{bp}$ . For the plant  $P(s)$ , the  $R_{bp}$  parameters are:  $\omega_1 = 33.4$  rad/s,  $\omega_2 = 62.62$  rad/s and  $g = 0.875$ .

For experiments, the sampling frequency is chosen to be 10 kHz. This is significantly higher than the eigenfrequency to avoid aliasing, but well within the limits of the FPGA's computational capabilities. Before deploying the discrete controller on the experimental setup, it is numerically simulated using Simulink. A pulse disturbance is imparted to the system revealing its transient damping performance. For a comparative study, a linear bandpass filter  $L_{bp}$  is designed with its transfer function given by

$$L_{bp}(s) = \frac{36s}{(s + 167)(s + 41.75)}. \quad (9)$$

This linear bandpass filter is designed to have the same gain as  $R_{bp}$ , and  $0^\circ$  of phase, at the plant's eigenfrequency of 13.2 Hz. The equal gain value of both filters ensure a similar peak control force and is a practical measure to compare control performance. The describing function of  $R_{bp}$  and the FRF of  $L_{bp}$  is shown in Fig. 11.

#### A. Numerical Results

Fig. 12 shows the simulation results for the pulse disturbance. The reset-based bandpass filter (RVF) provides a 120.3% improvement in settling time compared to the linear bandpass filter (NDF). In Fig. 13, the plant, controller, and total energies are shown, for the pulse disturbance. The total energy monotonically decreases, and decreases sharply at the reset instants. These instants also corresponds to the time of minimum plant energy.

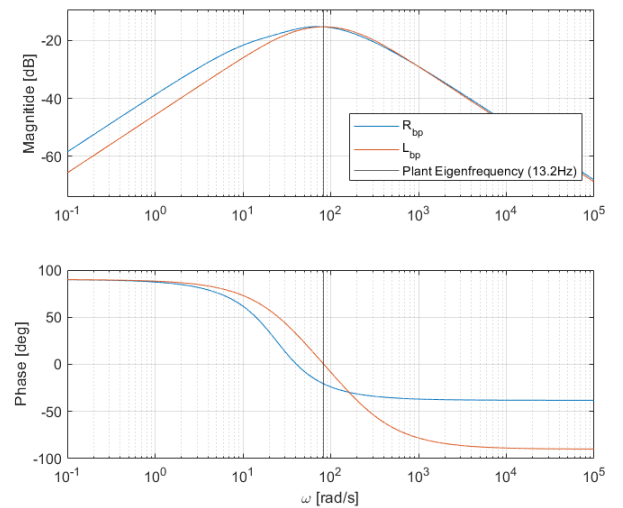


Fig. 11. Describing function of  $R_{bp}$  compared to the FRF of  $L_{bp}$ . Both are tuned to provide the same gain at 13.2 Hz



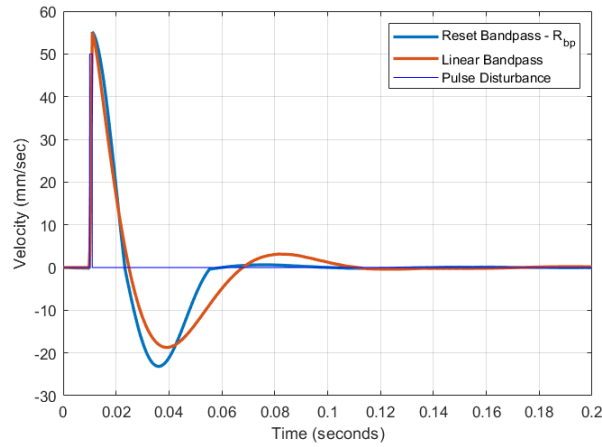


Fig. 12. Comparison of system responses to a pulse disturbance, showing superior transient damping performance provided by the reset bandpass filter (simulation)

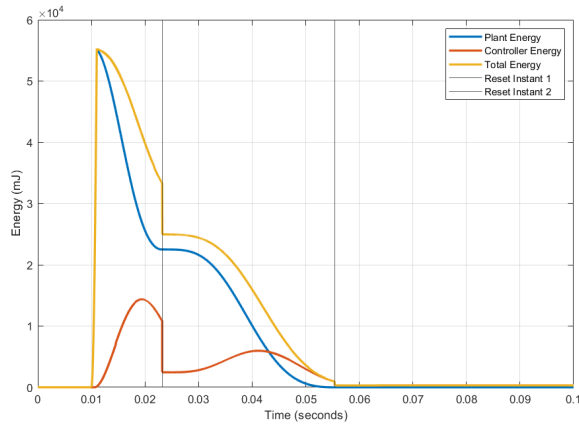


Fig. 13. Plant, controller, and total energies. The total energy decreases monotonically, and decreases considerably at the two reset instants. This validates the claim that reset injects damping into the system. This is similar to the study shown in [2], which considered undamped plants. By employing  $R_{bp}$ , this is easily extended to non-ideal damped plants

### B. Experimental Results

For the experimental study, a finite-time sinusoidal excitation at 13.2 Hz is applied for 2708 ms (36 complete cycles). This is done as a finite-time sinusoidal disturbance reveals both steady-state and transient damping performance. These results are shown in Fig. 14, with the emphasis placed on transient response. The experimental results are representative of the numerical simulations. RVF provides a 173.1% reduction in settling time compared to NDF, for the same control gain. The steady-state damping performance of RVF is marginally worse compared to NDF. This is to be expected as the focus of RVF design was to ensure better transient performance.

### C. Sensitivity Analysis

It is important to quantify its sensitivity to tuning parameters (control gain,  $g$ ) experimentally. We see that for a 10%

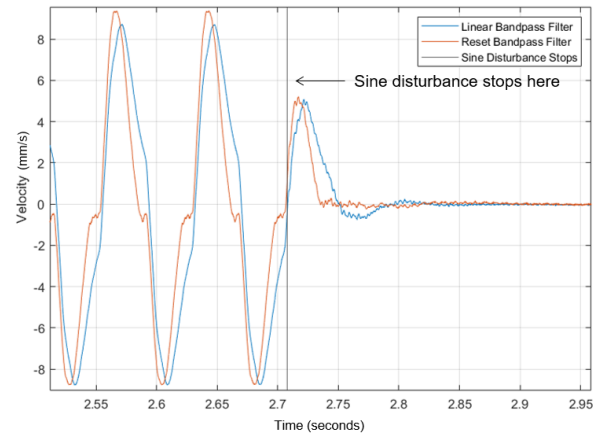


Fig. 14. Comparison of system responses to a finite sinusoidal excitation (experiment)

variation in control gain, the transient performance is still superior to the nominal linear bandpass filter. The results are summarized in Table I. These results illustrate that even though the reset bandpass filter is not explicitly designed to be robust, it still shows relatively low sensitivity to tuning parameters.

TABLE I  
SENSITIVITY OF SETTLING TIMES TO CONTROL GAIN

Controller	Settling Time	Deterioration w.r.t nominal $R_{bp}$
$R_{bp}$ (gain = 110% of nominal gain)	41 ms	29.9%
$R_{bp}$ (gain = 90% of nominal gain)	53 ms	69.8%
Nominal $L_{bp}$	82 ms	173.1%

### D. Delay Compensation

As far as (known) system delays are concerned, the reset bandpass filter can be tuned to provide a certain phase at the plant eigenfrequency such that the net phase including the delay amounts to about  $-20^\circ$ . This further emphasizes the flexibility offered by using two independent FOREs, as it is possible to manipulate the phase characteristics without considerably changing the magnitude characteristics. This would be not possible for a linear bandpass filter. For a system delay of 2 milliseconds (amounting to  $10^\circ$  of phase lag at the target frequency), the simulation results for a pulse disturbance, of a re-tuned reset bandpass filter with delay compensation (phase of  $-10^\circ$  instead of  $-20^\circ$ ) is compared to a reset bandpass without delay compensation in Fig. 15. The control gain is kept constant as the nominal case without system delays. The bandpass filter with delay compensation performs similar to a system without delays. This illustrates that reset can also be used to actively compensate for known time-delays.

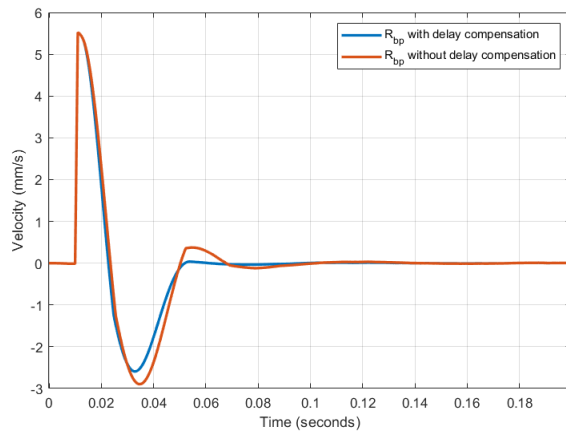


Fig. 15. RVF performance for systems with known time-delays

## V. CONCLUSIONS

This paper presented our methods at using reset control to achieve improved transient damping compared to linear control methods. The approach is motivated by an energy-based mechanistic analysis of the base linear systems, which provides clear insight for the use of reset in such problems. Translating these insights into the frequency-domain using describing functions and tuning heuristics for systematic control design, greatly simplifies the application of reset control for vibration control problems and makes it relevant for industrial applications. The numerical and experimental results validate the approach and demonstrate its superiority compared linear to methods such as NDF, by providing a 173.1% reduction in settling time for the same control gain. We also empirically showed that this novel reset-based bandpass filter is robust with respect to tuning parameters and can also compensate for known system delays.

As future work, the study can be extended to more complex distributed parameter systems. Multiple (and non-collocated) modes may render the plant non-passive and affect stability. Presently, the controller is not explicitly designed to be robust to plant variations. An adaptive scheme can be incorporated into this framework to improve its performance in uncertain environments. The design of a robust version of the controller could also be explored. Although the analysis is motivated from energy principles and describing functions, a rigorous mathematical treatment on how the reset bandpass filter provides better transient response is yet to be performed. Finally, a theoretical treatment on the effect of delays on system performance is also left out. Further theoretical questions remain unanswered in these directions and could pave the way for interesting future research.

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