Approaches for Minimizing Tracking and Vibratory Errors in High-Bandwidth Beam Steering

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Abstract

Parallel advancements in the field of controls engineering have recently been commercialized which have application to the fields of active optics and high-bandwidth beam steering:

- Cost effective, industrial-class implementations of Momentum Compensation (also known as Frahm Damping) provide low-order cancellation of inertial inputs to supporting structures and is of particular applicability to structures with low natural resonance frequencies;
- Input Shaping®, a patented controls technique developed at the Massachusetts Institute of Technology, provides effective cancellation of structural resonances in arbitrary actuation;
- Input Preshaping™, a technique realized in both a priori and self-learning implementations, substantially eliminates following errors in repetitive actuation.

The author reviews applications of each of these, alone and together, in a comprehensive overview of the state of the art of high-bandwidth active optic positioning techniques.

Keywords: Frahm damping, momentum compensation, active optics, Input Shaping®, vibration cancellation, beam steering, descintillation, atmospherics, structural ringing, high-bandwidth positioning

1. Background

"Active optics" can mean different things to different people, ranging from real-time deformable optical surfaces in astronomy, to rapidly-steered deflecting optics in microlithography and biomedical microassays, to high-bandwidth path- or cavity-length compensation in atmospheric telescopy and laser stabilization and tuning, to continuous alignment of micro-photonic devices. The common thread is rapid (up to ~several kHz) physical motion of optical materials.

Rapid motions pose significant challenges: amplifier, interface, sensor and processing bandwidths pose obvious limitations, and the physical constraints posed by the lowest resonant frequency of the actuation mechanics provides a fundamental limit to how rapidly the optic may be repositioned. All of these are addressable in design choices (to a point), including: stiffer materials; optimization of moments of inertia along strategic vectors; maximization of the overall stiffness-to-weight ratio; improved kinematics; mass reduction; integration of higher-bandwidth actuation technologies such as piezoelectric elements; integral damping provisions; and so on. Such improvements allow the optic to be actuated more and more rapidly.

But too often, projections of system throughput do not adequately consider the consequences of the rapid actuation on the structure, including the optic, its fixturing, the supporting framework and adjacent componentry. Any actuation (except a pure sinusoid without beginning or end) embodies many Fourier components which drive resonances throughout the assemblage. The consequences of recoil-driven resonances worsen as the motions grow faster and more impulse-like (which increases the energy content in the high-frequency Fourier components of the impulse) and as stiffer structural components are incorporated (stiff materials tend to possess less inherent damping).

The generic physics of recoil-driven resonant reaction is straightforward: after excitation by a force such as the actuation or recoil resulting from it, the amplitude of the resonant ringing of each element in a structure scales as $e^{-t/\tau}$, where τ is the time constant for each element's resonant characteristics. For structures with damping characteristics typical of precision motion subassemblies (see Table 1), $\tau \sim (\omega_n\zeta)^{-1}$ where ω_n is the resonant angular frequency and ζ is the damping ratio for the resonance. ζ is commonly defined as the ratio of the damping for the resonance versus critical damping ($\zeta = C/C_c$) and varies from 0 (no damping) to 1 (critical damping) (see Fig. 1).

Table 1: Time Constant τ for Various Damping Coefficients, ζ, and Resonant Frequencies, ω_n

F_{res} (Hz)	**ω_n (rad/sec)**	ξ	τ
75	471.24	0.0005	4.244
		0.001	2.122
		0.005	0.424
		0.01	0.212
		0.05	0.042
		0.1	0.021
150	942.48	0.0005	2.122
		0.001	1.061
		0.005	0.212
		0.01	0.106
		0.05	0.021
		0.1	0.011

Clearly, this dictates worsening process throughputs as tolerances tighten and throughput needs escalate.

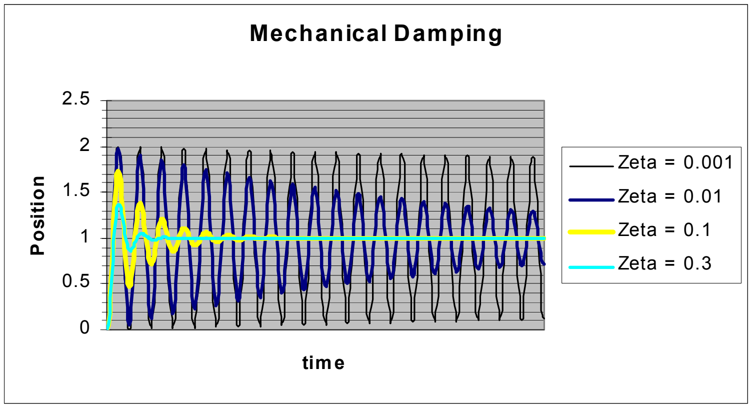

Fig. 1: Generic mechanical damping behavior after a rapid motion, shown for various material damping ratios, ζ. The amplitude of the ringing diminishes with time as $e^{-t/\tau}$, where $\tau \sim (\omega_n\zeta)^{-1}$.

Unfortunately, classical mechanical damping is of limited value in these situations. As mentioned above, the stiffer, lower-mass materials utilized to improve stiffness-to-weight ratios typically have low damping coefficients, and adding elastomeric or other materials to increase damping almost always lowers resonant frequency and travel range. And the bottom line is that increasingly in today's advanced applications, there is no spare time available for dwells in which vibrations can be turned into heat. "Active" vibration cancellation or servo techniques that form a servo loop around the actuated element are of minimal value (and high cost!) for the same reason: vibrations are addressed only after they are underway (i.e., the cows are already out of the barn), and any such technique is limited only to observable excitations within the bandwidth of the servo. What's needed are techniques which eliminate the vibrations in the first place, wherever they may occur in the assemblage—observable by on-board metrology or not.

2. Frahm Damping/Momentum Compensation

A classical approach to eliminating the recoil forces imparted to the underlying structure is the technique known variously as "Frahm Damping" or "momentum compensation." This involves the integration of an equal-and-oppositely-actuated mass which opposes the motion of the load, providing net cancellation of the impulse applied to the underlying structure. For example, the load might be actuated in the positive direction by *n* degrees in *t* milliseconds, and a dummy load would be actuated in the negative direction by the same amount and using the same motion profile. This can work well as long as the real and dummy loads and actuators are reasonably identical, have identical controls, and are configured to actuate coaxially so as to eliminate couple moments.

Figure 2 shows a simple example of such an assembly. In this case a single-axis coarse-fine angular positioner was required for the application. Very high bandwidth was needed for the fine angular positioning. However, recoil from the fast actuation of the fine positioner could dislodge the coarse rotary stage from its resting position. Closing a servo loop around the coarse angular positioner's platen to compensate for this would render the bandwidth of the whole assembly only as good as the servo bandwidth for that loop. Thus a method was needed to decouple the reaction forces from the ~10 kHz-bandwidth fine positioner from the coarse positioner.

An identical fine positioner with dummy load was configured coaxially with the main fine positioner and the coarse positioner. Actuated counter to the main fine positioner, the result is substantial nullification of the two positioners' impulses as perceived at the coarse positioner's rotary platen.

Fig. 2: Coarse/fine assembly composed of a rotary stage and an extremely high-bandwidth angular piezo mount (shown with simulated load which approximates the moment of inertia of the intended load). The rapid actuation of the piezo mount could dislodge the coarse rotary positioner from its resting position, so a coaxial, counter-actuated positioner was configured to substantially eliminate the reaction forces transmitted through the mounting structure to the rotation stage.

Drawbacks to this approach include the obvious disadvantage of cost and complexity, plus packaging challenges. In addition, the approach does nothing to address vibrations excited in the mount or in the actuated load (e.g., the optic and its moving fixture). Also, a small amount of residual impulse is usually inevitable (for example due to the torsional bending of the post assembly which connects the two fine-positioning mounts in Fig. 2). Still, this approach is proven, intuitive and highly effective.

In particular, momentum compensation is ideal when the underlying mounting structure has a very low resonance frequency or can be dislodged from a set-position by recoil forces from the positioner. Another, complementary technology, the Input Shaping™ controls algorithm (discussed below), yields zero residual resonant motion after approximately one cycle of the lowest resonant frequency in the system; in applications where this would be unacceptable due to the low resonance frequency of the underlying structure, momentum compensation is an effective (if usually costly) cure. In the case of professional telescopes, it has provided excellent results for the rapid modulation of secondary mirror position—for example for atmospheric de-scintillation—despite the inherently low resonant frequency of the massive telescope tube frame. Figure 3 shows such an assembly, composed of a six-degree-of-freedom hexapod micropositioner which compensates for telescope frame sag plus a high-bandwidth tip/tilt mirror mount (used for de-scintillation of starlight) with integral momentum compensation.

Fig. 3: Custom telescope secondary mirror package incorporates sophisticated internalized momentum compensation design. A high-bandwidth, piezo-actuated tip/tilt mount (for atmospheric Descintillation) is fixtured on a six-degree-of-freedom hexapod micro-positioner (for compensation of frame sag as the telescope is pointed around the sky). Momentum compensation prevents the high-bandwidth piezo tip/tilt positioner from imparting low-frequency structural vibrations in the telescope tube.

3. Input Shaping™

A patented, real-time feedforward technology called Input Shaping™ (Figs. 4 and 5) was developed based on research at the Massachusetts Institute of Technology and commercialized by Convolve, Inc., (New York, NY; http://www.convolve.com). It has been implemented in OEM NanoAutomation™ products by PI. The technique scales and times transitions in the command in real-time so motion-driven vibrations are canceled throughout the system. It uses a priori knowledge of resonances as quantified during installation and does not require feedback. Unlike notch filtering, it is insensitive to variations in frequency over a range >±15%. It is effective against multiple resonances, resonances occurring outside the servo loop, and resonances exceeding system bandwidths. Its robustness makes the technique attractive for OEM usage. In particular:

- It requires no changes to application setup, software or servo parameters.
- It cannot degrade servo stability.
- It is does not require specific configuration for specific motions.
- It is robust with changes in operating dynamics, such as unit-to-unit variations, or moderate changes in loading.

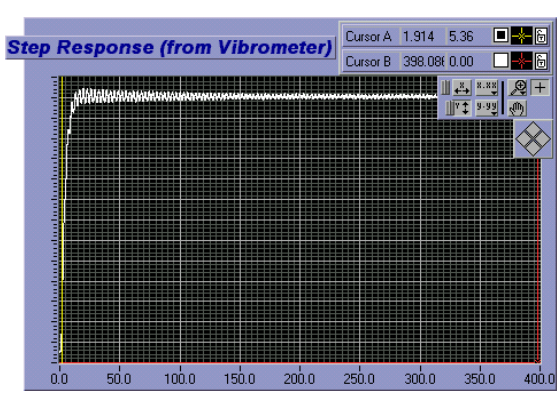

Fig. 4: Piezo devices are capable of millisecond-scale step-and settle. However, elements outside the servo loop ring (load, neighboring componentry…). External resonances are visualized here by a Polytec laser vibrometer measuring position vs. time.

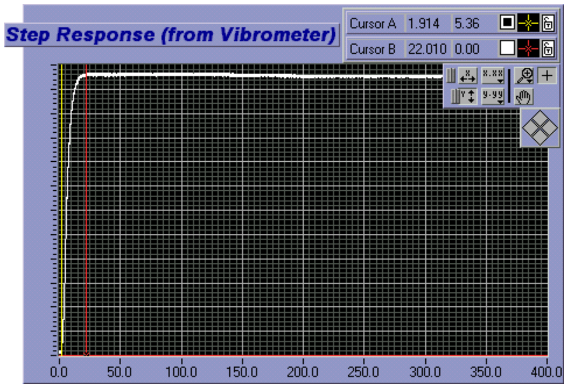

Fig. 5: Input Shaping™ eliminates motion-driven ringing of components outside the servo loop. Settling after risetime completes by $t \sim F_{res}^{-1}$.

As a rule of thumb, Input Shaping settles a system in about F_{res}^{-1} after the piezo risetime. Input Shaping also eliminates ringing during scanning, such as in scanned-microscopy applications, where one axis scans while the other steps. Structural ringing in such applications causes periodic image artifacts. This limits scan velocity and resolution and often necessitates over-scanning.

4. Combined Benefits

The simple coarse/fine angular positioner shown in Fig. 2 is an illustrative test-bed for demonstrating the complementary nature of Momentum Compensation and Input Shaping. First, measurement with a Polytec laser Doppler vibrometer (as described below) revealed the following resonances in the structure:

Frequency (Hz)	Zeta
665	0.2
805	0.001
4975	0.3

(To implement Input Shaping, it is not necessary to know precisely what is ringing in an assembly. Still, it's often interesting: in this case the resonance at 4975 Hz seemed to be the loaded mounts ringing, whereas the other resonances appeared to be in the mechanics and structure.)

Actuating the assembly in rapid sawtooth and square-wave motions revealed some interesting behavior (see Figs. 6-9), as imaged by the vibrometer:

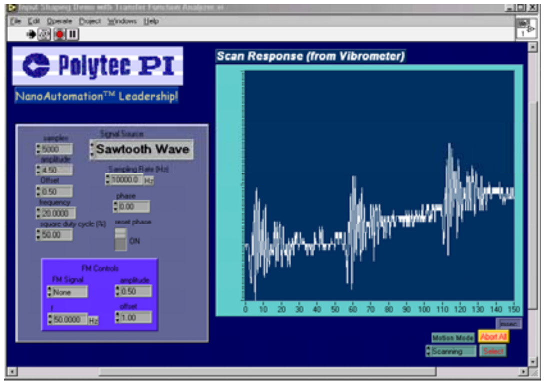

Fig. 6:. Vibrometer measuring mount base position vs. time. Actuation of one mount in a sawtooth wave without counter-actuation reveals significant vibration plus apparent angular "creep" of the mount base.

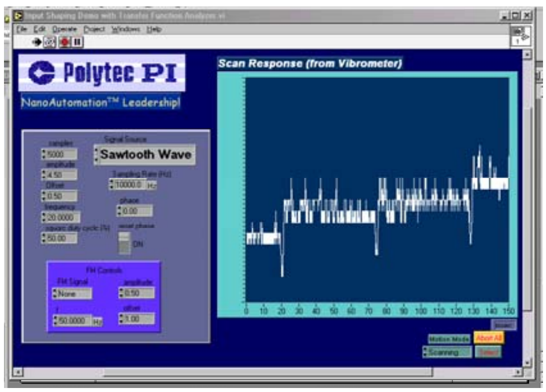

Fig. 7: Same as Fig. 6, but using Input Shaping™ to eliminate the structural ringing. The rotational creep of the mount base in sawtooth actuation is more clearly visible.

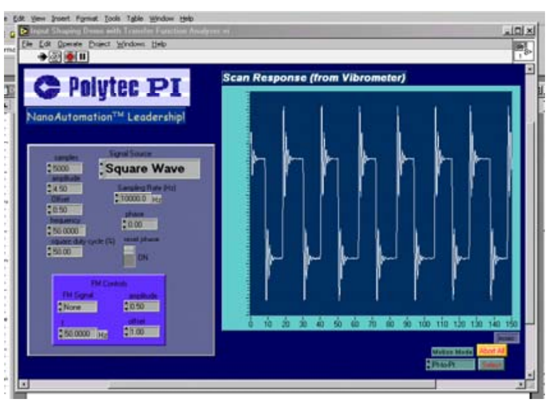

Fig. 8: Vibrometer measuring moving element of mount. Counter-actuation without Input Shaping. No creep of the coarse stage is observed, demonstrating the ability of Momentum Compensation to substantially eliminate impulses delivered to the supporting framework.

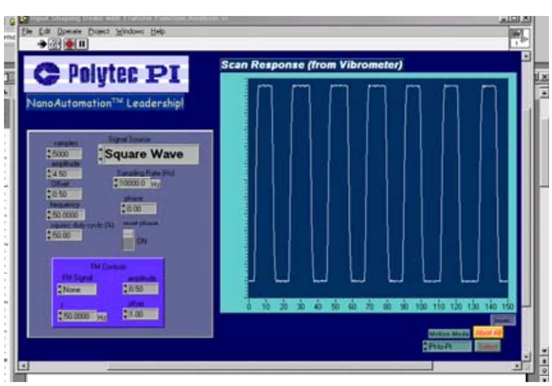

Fig. 9: Same as Fig. 8, but using Input Shaping to nullify resonances as well as Momentum Compensation to eliminate recoil deflections to the supporting structure.

5. Bandwidth Improvement: Input Preshaping

For applications with continuous, repetitive periodic inputs, a new pre-shaping technique can reduce the rolloff, phase error and hysteresis of the servo, improving the effective bandwidth and allowing more accurate tracking. It is implemented (1) either in object code based on an analytical approach where the complex transfer function of the system is calculated then mathematically transformed and applied in a feedforward manner, or (2) within digital controls as a self-taught setup capability that is transparent to the user. It is available as an integrated option in PI's latest piezo servocontrols.

Input Preshaping significantly reduces the tracking error and improves the effective bandwidth by a factor of 10. Accuracy and tracking error improved considerably versus what can be achieved by conventional P-I-D controls (Fig. 10). In the example shown below, overall following error is ~2% in this repetitively-actuated application; this includes contributions from servo-system rolloff as well as nonlinearities such as PZT hysteresis.

Input Preshaping is more effective than classical phase-shifting approaches in reducing tracking error in multifrequency applications. It can also be combined with Input Shaping™ to address resonances outside the servo/feedback loop.

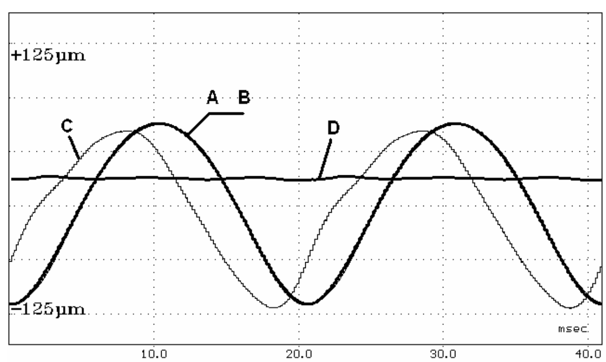

Fig. 10: Following error with pre-shaping. A: Commanded displacement. B: Actual displacement (virtually indistinguishable from commanded). C: Preshaped input signal. D: Tracking error (2%).

6. Conclusion

Active-optic applications are becoming both more numerous and more demanding. Mechanisms and controls exist which provide adequate bandwidth, range, degrees-of-freedom and accuracy, but systems are still vulnerable to recoil-generated vibrational errors occurring throughout the structure and to linearity and following-error issues which limit on-the-fly accuracy. These can be addressed with complementary technologies—used alone or together in various combinations—per Table 2.

Table 2: Complementary Technologies

Technology: **Addresses:**	**Momentum Compensation (Frahm Damping)**	**Input Shaping™**	**Input Pre-Shaping**
Resonances in Supporting Structure	Yes	Yes	
Resonances in Load		Yes	
Low-Frequency Structural Recoil Issues	Ideal	Settling is complete in $t \sim F_{res}^{-1}$	
Following Errors and Bandwidth Limitations		Often gain can be boosted (under-damped servo is just another Hookes oscillator), reducing following errors. Use in any actuation.	Yes— addresses servo rolloff, virtually eliminating following errors. Use in repetitive actuation.