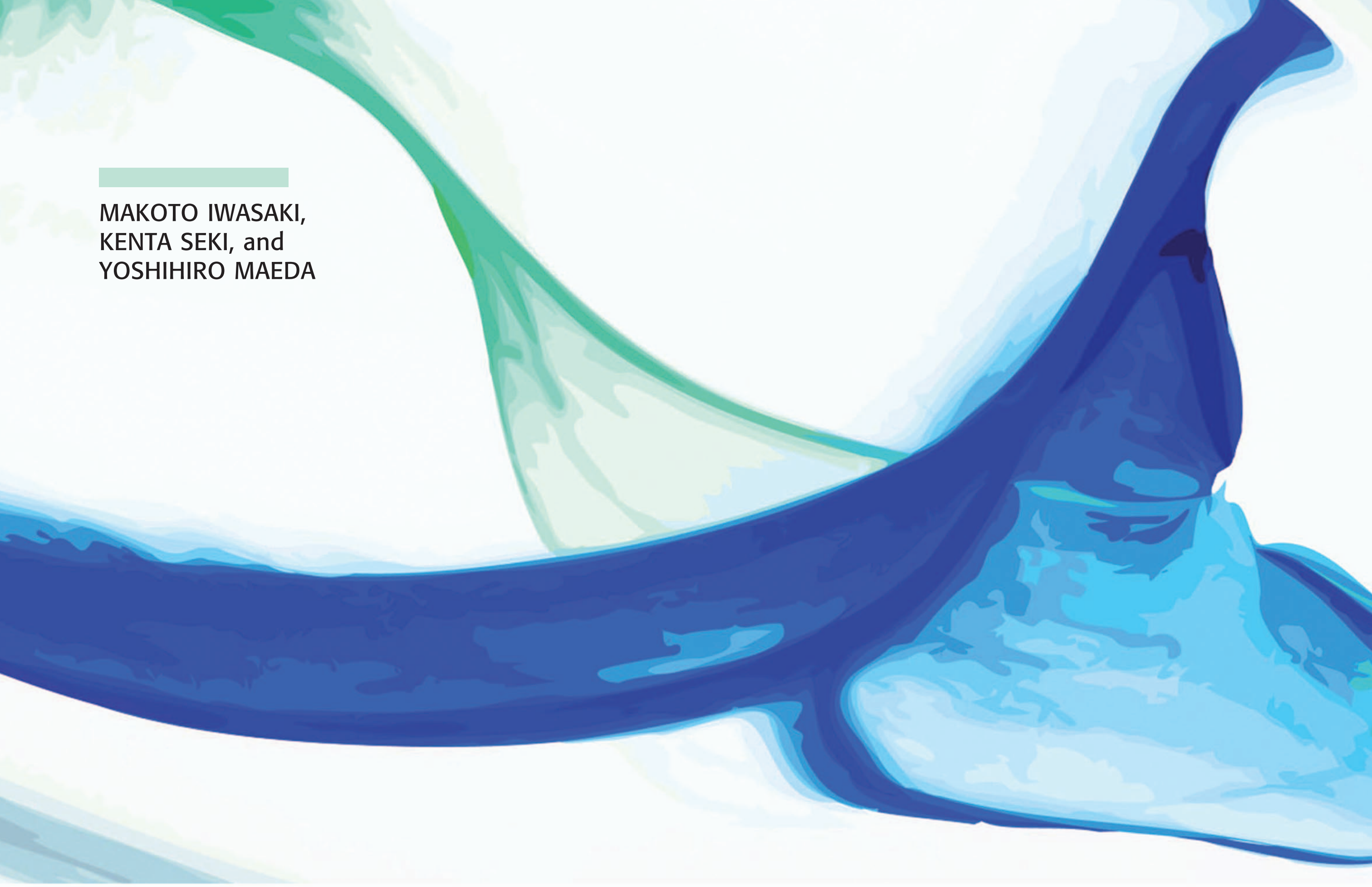

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High-Precision Motion Control Techniques

A Promising Approach to Improving Motion Performance

Fast response with high-precision motion control (on the levels of force, torque, velocity, and/or position) is an indispensable technique. This technique is used in a wide variety of high-performance mechatronic systems including micro- and/or nanoscale motion (such as data storage devices, machine tools, manufacturing tools for electronics components, and industrial robots) from the standpoints of high productivity, high quality of products, and total cost reduction. In these applications, the required specifications in motion performance, e.g., response/settling time and trajectory/settling accuracy, should be sufficiently achieved. In addition, robustness against disturbances and/or uncertainties, mechanical vibration suppression, and adaptation capability against variations in mechanisms should be the essential properties provided for performance.

In this article, the state of the art on high-precision motion control techniques is surveyed by referring to recent publications, mainly in the transactions and conferences of the IEEE

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Industrial Electronics Society (IES), where a 2-degree-of-freedom (DoF) control framework is considered a practical and promising approach to improving motion performance. The actual issues and relevant solutions for each component in the 2-DoF control structure are clarified. Next, one of the examples, a 2-DoF controller design for robust vibration suppression positioning, is presented as an application to industrial high-precision positioning devices.

High-Precision Motion Control Based on 2-DoF Control Framework

It is well known that an essential approach for achieving fast and precise motion performance is to expand the control bandwidth of a feedback control system; a higher bandwidth allows the system to be more robust against disturbances and/or uncertainties. However, mechanical vibration modes, dead-time/delay components, and varieties of nonlinearities generally prevent the control bandwidth from being wider than the viewpoints of system stability. The 2-DoF control framework with feedback and feedforward compensators should be a practical technique for achieving desired performance under the limitation of expansion in the control bandwidth. To ensure the required performance by the feedforward compensation, the precise modeling of the plant system should be indispensable; a more precise plant model provides higher compensation in the feedforward manner as well as a more progressive design in the feedback controller. Nonlinear components as disturbances, such as nonlinear friction and/or elastic elements in the microdisplacement region, are carefully analyzed and compensated because they behave in a different manner than in macroregions [1].

The motion performance, on the other hand, is deteriorated by a variety of factors in actual operating environments, e.g., aged deteriorations, temperature variations, and dispersion among products, while fault-tolerant capabilities are also required.

Precise modeling and identification for the target mechatronic systems should be indispensable from the standpoint of more accurate model-based feedforward compensation and/or more progressive design of feedback controllers.

Optimization, autotuning, and adaptive techniques should therefore be considered from the viewpoints of cost reduction, labor saving, and autonomous function in the control design, to provide a higher value-added production.

To give an overview of the typical issues faced in the design of a motion control system, let us survey the actual primary factors that deteriorate the motion performance of galvano scanners as an example of the fast and precise positioning device for industrial applications.

Figure 1 shows an outline of a laser drilling mechanism with galvano scanners for the via hole manufacturing of printed circuit boards. The configuration of the galvano scanner as a laboratory experimental setup is shown in Figure 2, where the galvano mirror position is controlled in a semiclosed manner using the motor sensor angle. The system requires inherent specifications in motion performance: the settling time and accuracy for the given point-to-point positioning strokes to satisfy the manufacturing productivity and quality. On the other hand, various factors can deteriorate the performance:

1) finite stiffness in the mechanism causes residual vibration modes, which lead to vibratory responses that affect the stability in the feedback control system
2) structured/unstructured uncertainties in electrical and mechanical components deteriorate the motion performance, i.e., parameter variations (especially, torque constant of actuator and vibration mode frequencies) due to temperature fluctuation and/or aged deteriorations, system delay components (dead time and servo lag), and unknown disturbances from environments
3) nonlinear components, such as friction and stiffness, should be compensated for by nonlinear control strategies.

In addition, several constraints, e.g., the saturation in control inputs, should be considered in the controller design.

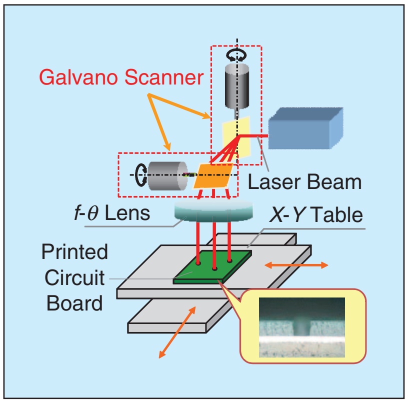

FIGURE 1 – Laser-drilling mechanism with galvano scanners for via hole manufacturing of printed circuit board: the laser beam is reflected by a pair of galvano mirrors and makes via holes on the printed board through f-θ lens, over 2,000 holes/s.

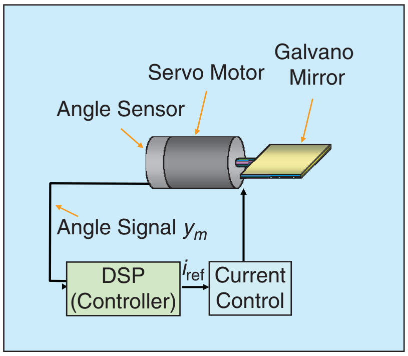

FIGURE 2 – Configuration of galvano scanner; the galvano mirror is controlled in a semiclosed control manner by detected motor sensor angle y_m while the servo motor is driven by a current-controlled amplifier with current reference i_{ref} generated by position controller [digital signal processor (DSP)].

Because the residual vibrations cannot be inherently attenuated more than the constraint by feedforward design, the feedback compensator should be redesigned as a coordinate design with a feedforward compensator.

Based on the background listed above, Figure 3 indicates a conceptual 2-DoF controller structure for high-accuracy motion control. The relevant solutions are briefly presented in the following by dividing the control system into four parts: 1) the feedback control system, 2) the feedforward control system and command shaping, 3) system modeling and identification, and 4) optimization and autotuning of the motion control system.

Feedback Control System

In precision motion control applications, the inherent properties that the feedback system should possess are a robust performance on system stability and/or controllability, disturbance suppression, adaptation capability against system variations, and issues due to the feedback control structure.

Novel approaches to achieve robust stability have been proposed by applying simple filters such as resonant [2] and phase lead/lag filters [3]. It is well known that a disturbance observer is one of the powerful tools used to enhance disturbance suppression performance [4], [5]: the analytical studies on parameter variations in the disturbance observer are discussed in [6]. Adaptive robust control framework [7] has also been applied to provide adaptation capabilities in motion control systems. For example, an adaptive transient performance against parametric uncertainties [8] and a multirate perfect tracking control for a plant with parametric uncertainty [9] could be achieved. Mode switching control is a common technique, especially in the hard disk drive (HDD) servo system, where the undesirable transient effects generated by switching should be compensated as an initial value compensation (IVC). A variety of approaches on IVC have been reported according to the application specifics [10]–[13].

Other approaches, including computational intelligence, have been introduced to improve feedback control performance: repetitive learning control provides a fast response and precise positioning [14]–[16], fuzzy control gives the maximum sensitivity for servo systems [17], [18], neuron-based proportional integral (PI) fuzzy controller suppresses mechanical vibration [19], and an optimized robust fuzzy controller compensates for the friction in robot joints [20].

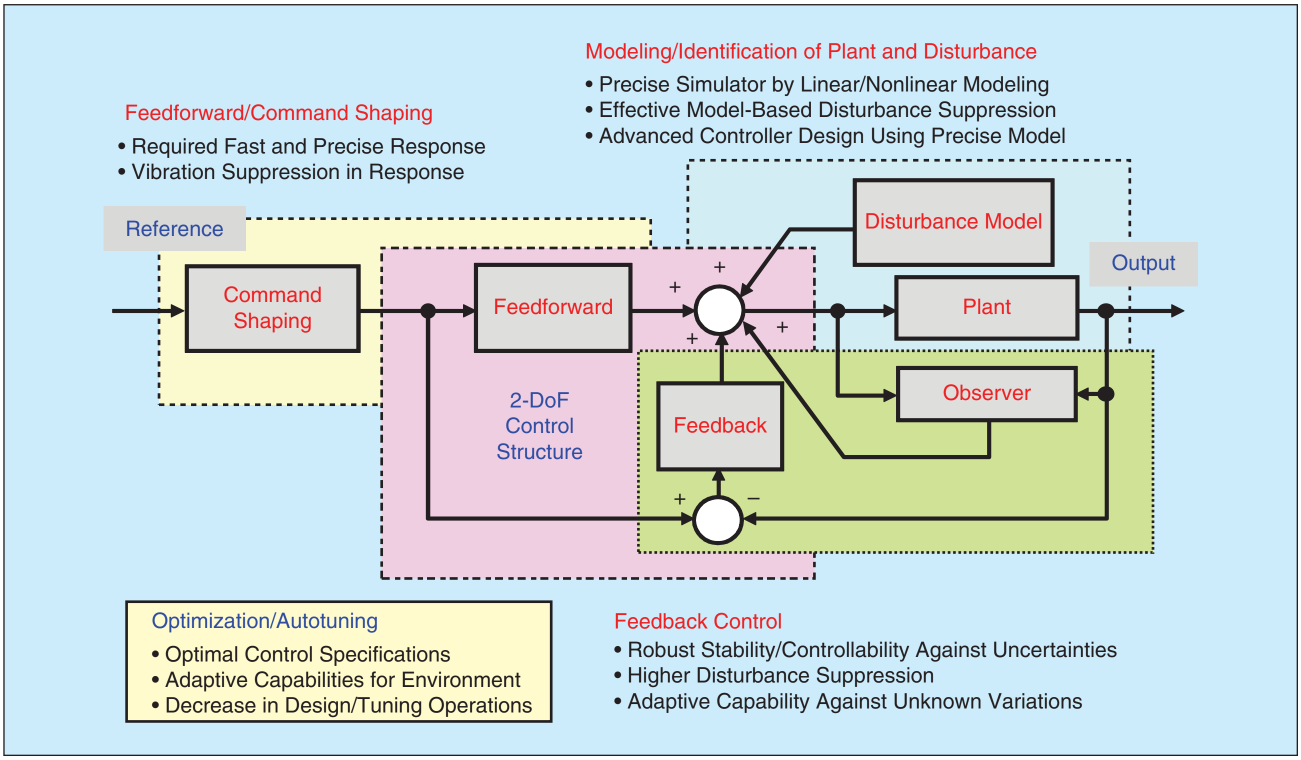

FIGURE 3 – Concept of 2-DoF control for a high-precision motion control system in industrial applications: control structure and issues to be considered.

Feedforward Control System and Command Shaping

Feedforward compensation can improve the motion performance in the 2-DoF control manner, e.g., fast response, high accuracy, and vibration suppression, independent of characteristics of the feedback system (system stability and disturbance suppression). The feedforward compensator can be systematically designed by, e.g., a coprime factorization [21], with consideration of mechanical vibration suppression. Perfect tracking performance, in addition, can be achieved by a multirate feedforward control approach [22].

Command shaping, on the other hand, should be an additional control freedom for feedforward compensation [23]: vibration suppression performance can be especially improved while settling speed and accuracy are ensured. Final state control (FSC) [24] is a systematic approach for optimizing motion reference under the constraints of a desired settling time and specified mechanical vibration attenuation. Minimum jerk control-based command shaping has also been applied to various positioning applications such as mass storage devices and table drive systems [25], [26]. Besides methodological proposals on command shaping, additional issues have been discussed. For example, a robust input shaping against parametric uncertainty could be achieved in [27], and a novel shaping scheme was given considering the target position correction during motion [28].

System Modeling and Identification

Precise modeling and identification for the target mechatronic systems should be indispensable from the standpoint of more accurate model-based feedforward compensation and/or more progressive design of feedback controllers. Among various methodologies and experiments on mathematical modeling and/or identification, a novel trial has been done; from the application specifics viewpoint, linear identification and control of plant dynamics above the system

The 2-DoF control framework with feedback and feedforward compensators should be a practical technique for achieving desired performance under the limitation of expansion in the control bandwidth.

Nyquist frequency have been discussed for HDDs [29], [30].

Nonlinear elements, on the other hand, should be carefully identified by analyzing their complicated behaviors and effects on the motion control system. In cases of nanoscale positioning, such as ultraprecision stage control in semiconductor and/or liquid crystal display manufacturing, nonlinear components directly deteriorate motion performance. Nonlinear friction in a mechanism interferes with motion performance as a disturbance; the friction may cause the tracking error for the trajectory reference generated by the feedforward compensator in the 2-DoF control system, deteriorating motion performances such as accuracy and vibration suppression. A wide variety of research has been reported on friction modeling and compensation, for example, a compensation for the stick–slip phenomena [31], a disturbance observer-based compensation [32], and a mathematical friction model-based compensation [33]. In addition to the friction compensation, system delay components (e.g., dead time and servo lag) and hysteresis/backlash properties have also been handled to improve performance [34]–[36].

Optimization and Autotuning of Motion Control System

From a practical viewpoint, it is important to optimize the whole motion control system (from the system modeling to compensator design and tuning) and to flexibly adapt to variations in various motion environments. Soft computing techniques and/or computational intelligence can be an attractive alternative to conventional approaches to possess autonomous functions in the controller design.

Genetic algorithms have been applied to controller design as an optimization approach [37], [38], where multiple objectives for the target system can be handled, e.g., motion speed and accuracy, vibration suppression, and energy minimization. Adaptation capability, on the other hand, has been introduced; an adaptive neural network compensation

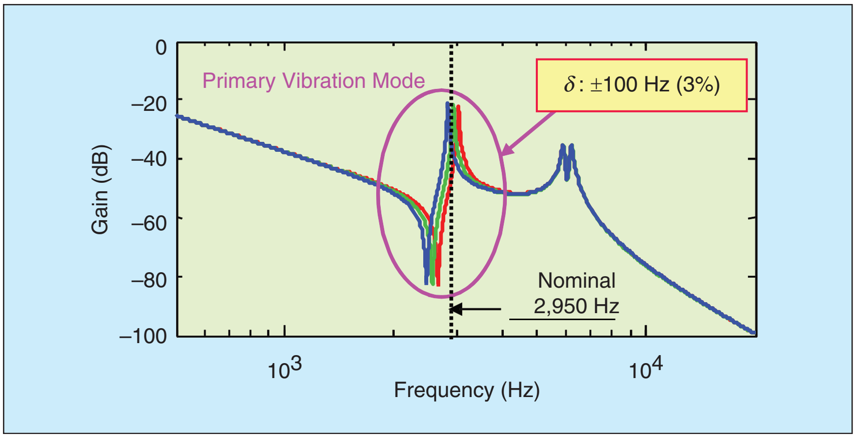

FIGURE 4 – Gain diagrams of a plant system with frequency variations $\delta = \pm 100$ Hz in primary vibration mode.

$$
\begin{cases} y_m[N] = X_r, \ \dfrac{dy_m}{dt}[N] = 0 & \text{for settling performance,} \\ J = \displaystyle\sum_{k=0}^{N-1} u_d{}^2[k] & \text{for smoothness in input.} \end{cases} \tag{1}
$$

for nonlinearity in HDDs [39], and fault-tolerant schemes against system failures [40]–[42].

Two-DoF Controller Design for Robust Vibration Suppression Positioning

As one of the practical applications of 2-DoF controller to fast and precise positioning devices for industrial applications, a linear matrix inequality (LMI)-based robust feedforward compensator design scheme is presented to provide robust properties in positioning performance against frequency perturbation of the vibration mode in the plant mechanism. Notice here that the proposed controller design scheme strongly relates to the above techniques on feedback, feedforward and command shaping, and optimization of the control system. In the following, a novel concept of "coordinate design between feedforward and feedback controllers" is also suggested to optimally design compensators and to provide a higher motion performance, although the feedforward and feedback compensators in the 2-DoF control structure are independently designed in general, as mentioned in the section "Feedforward Control System and Command Shaping."

Positioning Device: Galvano Scanner System

In the following, the galvano scanner system shown in Figures 1 and 2 is an example of a precision positioning device. Details of the actual system can be found in [43] and [44]. The scanner mechanism can be modeled by a multimass body system, whose gain characteristic of motor angle for motor current is shown in Figure 4. From the figure, the mechanism mainly includes the primary (2.95 kHz), second (5.95 kHz), and third (6.25 kHz) vibration modes up to 10 kHz, deteriorating the positioning performance as well as affecting the system stability. Frequency variations ($\delta = \pm 100$ Hz) in the primary mode are also indicated in the gain diagrams, as the frequency perturbation of the target plant mechanism due to aged and/or temperature variations and dispersion among products in actual industrial fields. One of the typically required control specifications is given as a point-to-point positioning with the settling time of 0.78 ms ($= NT_s$, $T_s = 20$ μs: position control sampling time, N: corresponding desired settling sample) and an accuracy of ± 3.5 μm for a typical position reference amplitude of $X_r = 1.5$ mm (in equivalent displacement on the board).

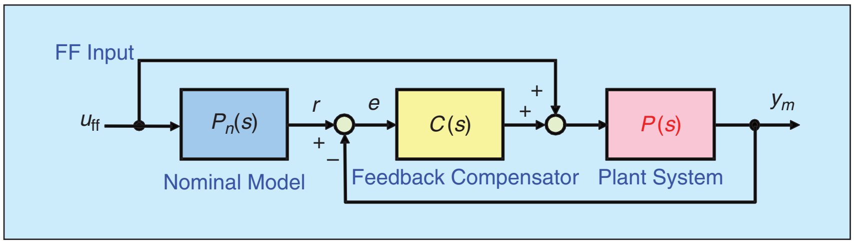

FIGURE 5 – Block diagram of an FSC-based 2-DoF positioning system for perturbed plant ($P(s) = P_n(s) + \Delta$): an FSC generates optimal feedforward motor input u_{ff} while optimal position reference r is given by $P_n(s)u_{ff}$.

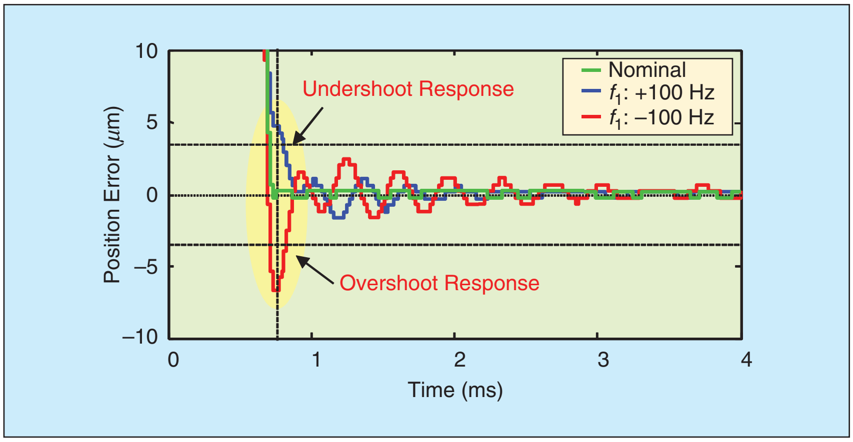

FIGURE 6 – Waveforms of position errors ($X_r - y_m$) for nominal and perturbed plant systems in step positioning with an amplitude of $X_r = 1.5$ mm: frequency perturbation leads to overshoot or undershoot responses in position errors that cannot satisfy a required settling accuracy of ± 3.5 μm indicated by horizontal broken lines.

Two-DoF Positioning Systems

Figure 5 shows the block diagram of an FSC-based 2-DoF positioning system, where $P(s)$ is a plant system with the frequency perturbation ($P(s) = P_n(s) + \Delta$, Δ: corresponding perturbation due to the frequency variations δ), $P_n(s)$ is a nominal plant model, $C(s)$ is a feedback controller, u_{ff} is a feedforward control input that is optimally generated by an FSC framework [24], [45], and r is an equivalent position reference given by the feedforward input as $r = P_n(s)u_{ff}$.

Here, the feedforward input can be designed by FSC for the nominal plant $P_n(s)$, to ensure the following final state condition for motor position/velocity and minimize the following index J for input [see (1) at the top of the page].

In (1), N is a specified sampling for the settling time ($= NT_s$) and u_d is the derivative of u_{ff}.

Figure 6 shows the experimental waveforms for the typical point-to-point

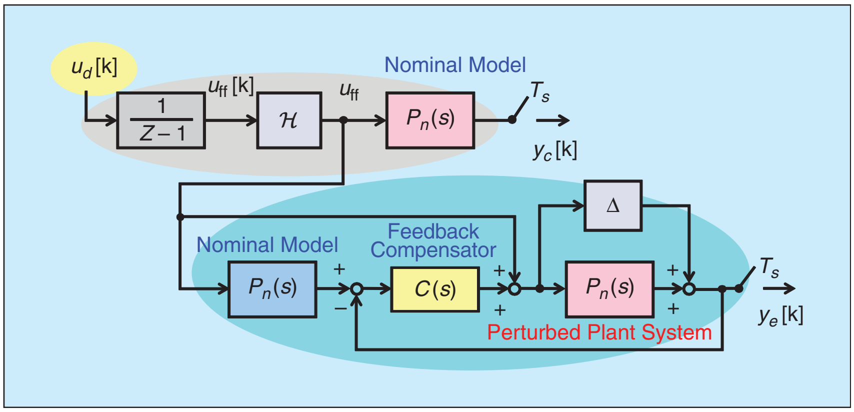

FIGURE 7 – Block diagram of an augmented system with a feedback compensator for perturbed plant system: constraints for control states ($y_c[k]$: motor position for a nominal plant model, $y_e[k]$: motor position for a perturbed plant, and $u_{ff}[k]$: motor torque) are formulated as LMI to design FSC input.

positioning with a step amplitude of $X_r = 1.5$ mm, where motor position errors ($X_r - y_m$: in equivalent displacement on the board) are shown for nominal and perturbed plant systems. From the figure, the residual vibration-free response that satisfies the required specifications (settling time by vertical broken line and accuracy by horizontal broken lines) can be achieved in a nominal condition. The frequency perturbation, on the other hand, leads to overshoot or undershoot responses that cannot satisfy the required performance since the feedforward input has been designed for the nominal plant with no consideration of perturbation.

Robust Feedforward Compensator Design Against Frequency Variation

To improve the motion performance under perturbation, a robust FSC-based feedforward compensator design against frequency variation is presented by additional state constraints formulated by LMI [24], [46]. In the proposed design, an alternative plant system shown in Figure 7, represented by both the nominal plant model (P_n, output: y_c) and the perturbed plant model ($P_n(s) + \Delta$, output: y_e) with feedback control loop, is introduced to design the FSC input with the following constraints as LMI formulations [see (2) at the bottom of the page].

In (2), u_m is an input saturation value, y_{em} is an output constraint for perturbed model, and N_m is the desired sample for settling of residual errors. Figure 8 shows a conceptual waveform of the constraints.

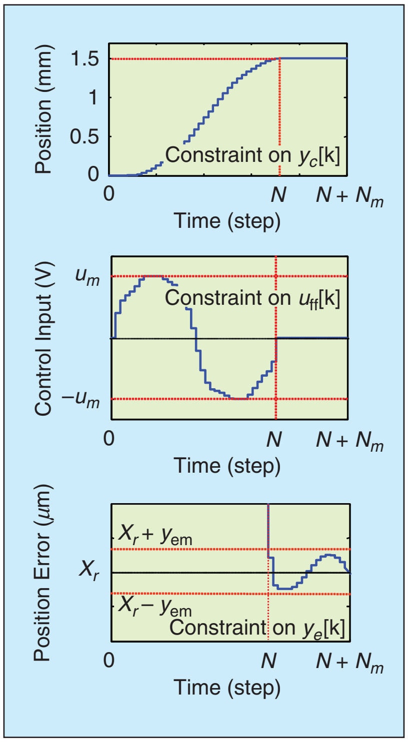

FIGURE 8 – Conceptual waveforms of constraints for position $y_c[k]$ of a nominal plant, motor torque $u_{ff}[k]$, and position error ($X_r - y_e[k]$) in a perturbed plant.

Actual calculation process in the LMI-based FSC design can be referred to in [24] and [47].

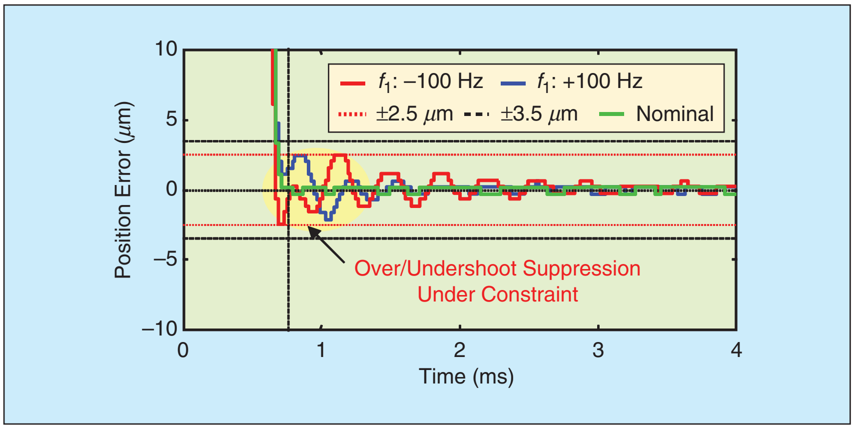

FIGURE 9 – Waveforms of position errors ($X_r - y$) for nominal and perturbed plant systems in step positioning with an amplitude of $X_r = 1.5$ mm under optimization by proposed LMI constraints: over/undershoot responses are suppressed within the constraint of $\pm y_{em} = 2.5$ μm (indicated by horizontal red dotted lines) and yet residual vibrations are still remaining due to a lack of disturbance suppression capability in the feedback controller.

$$\begin{cases} y_c[N] = X_r,\ \dfrac{dy_c}{dt}[N] = 0 & \text{settling constraint,} \\ -u_m < u_{ff}[k] < u_m,\ (k = 1, \ldots, N) & \text{constraint on input,} \\ X_r - y_{em} < y_e[k] < X_r + y_{em},\ (k = N, \ldots, N + N_m) & \text{constraint on perturbed plant output.} \end{cases} \quad (2)$$

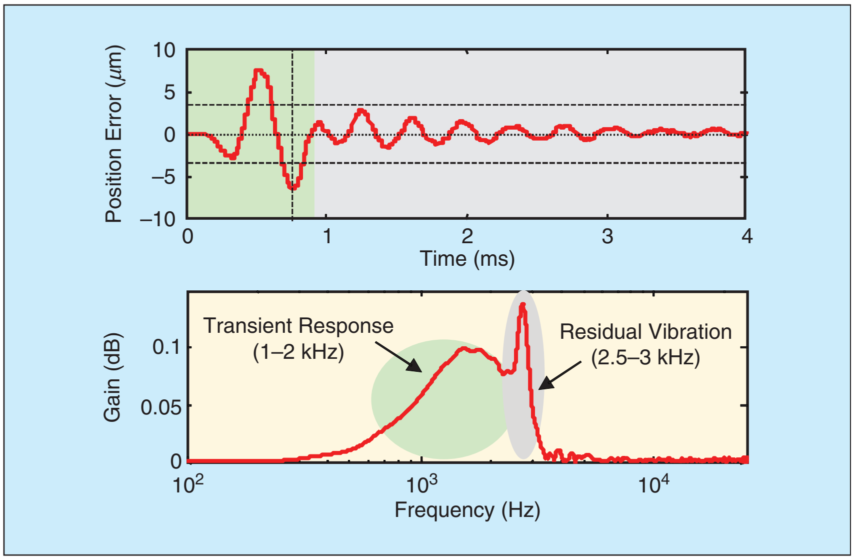

FIGURE 10 – Analysis of position error waveform in the case of a frequency variation of $\delta = +100$ Hz: frequency component at around 1.5 kHz is dominant during the transient period while residual vibration includes peak frequency of approximately 2.7 kHz.

From the above formulated constraints by LMI, the feedforward input has been redesigned to minimize the same input index J as in (1), where u_m is 80% of the rated motor torque, $y_{\mathrm{em}} = 2.5$ μm, and $N_m = 35$ have been given. Figure 9 shows the experimental waveforms by the proposed robust feedforward design under LMI constraints. From the figure, over/undershoot responses in cases with frequency variations are suppressed within the given constraint of $\pm y_{\mathrm{me}}$ (indicated by horizontal red dotted lines), and yet residual vibrations within the constraint are

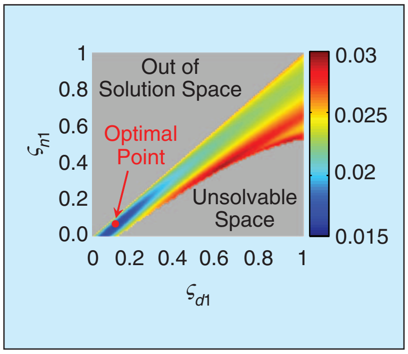

FIGURE 11 – Positioning index for parameters (ζ_{d1} and ζ_{n1}) of notch filter in the feedback compensator. The lower-right area corresponds to unsolvable space in optimization while the upper-left area is out of solution space.

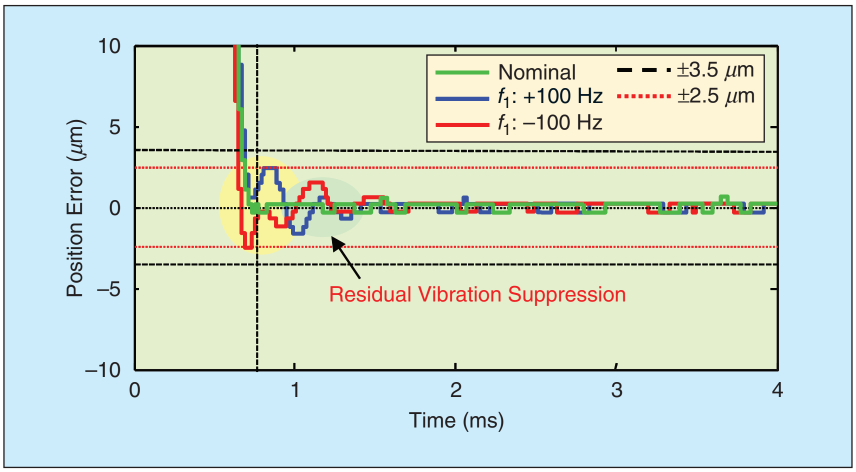

FIGURE 12 – Waveforms of position errors ($X_r - y$) for nominal and perturbed plant systems in step positioning with an amplitude of $X_r = 1.5$ mm: residual vibrations are effectively suppressed by applying an optimized notch filter.

still remaining. Those residual vibrations result in the deterioration of actual manufacturing performance and/or quality.

Because the residual vibrations cannot be inherently attenuated more than the constraint by feedforward design, the feedback compensator should be redesigned as a coordinate design with a feedforward compensator. In practice, the feedback compensator should be optimized by considering the frequency shaping in sensitivity characteristic for the corresponding frequency components of the residual vibration.

Coordinate Design Between Feedforward and Feedback Controllers

Figure 10 shows a waveform of position error e (input of feedback compensator $C(s)$ in Figure 5) and its frequency analysis, in the case of a frequency variation of $\delta = +100$ Hz in the primary vibration mode. From the frequency gain characteristic of e, the residual vibration mainly includes the component whose peak frequency is at approximately 2.7 kHz while the component around 1–2 kHz is dominant during the transient period. This analysis suggests that the improvement in sensitivity characteristic (i.e., disturbance suppression performance) at approximately 2.7 kHz leads to a residual vibration suppression.

The feedback compensator $C(s)$ has been originally designed by the following lead-lag phase filter and two notch filters, considering the disturbance suppression in the low-frequency range and robust stability at vibration modes:

$$C(s) = K_c \frac{s+\omega_{c1}}{s} \frac{s+\omega_{c3}}{s+\omega_{c2}} \prod_{k=1}^{2} \frac{s^2 + 2\zeta_{ni}\omega_{ni}s + \omega_{ni}^2}{s^2 + 2\zeta_{di}\omega_{ni}s + \omega_{ni}^2}, \tag{3}$$

where each notch filter stabilizes the primary mode and 2nd/3rd modes, respectively.

From the above frequency analysis for residual vibration, one of the notch filters (free parameters: ω_{n1}, ζ_{d1}, and ζ_{n1}) corresponding to the primary vibration mode is optimally parameterized to minimize the following settling performance index by an integral absolute error from the given feedforward input in 3.3 and the plant system with the frequency perturbation δ

$$S = \sum_{k=N-2}^{N+N_m} |X_r - y_e[k]|. \quad (4)$$

Here, the whole solution space of S has been numerically calculated by the all-search method with parameter ranges of $2,300 < \omega_{n1} < 2,900$ Hz, $0 < \zeta_{d1} < 1.0$, and $0 < \zeta_{n1} < \zeta_{d1}$. Figure 11 shows an example of the solution space for the parameters of ζ_{d1} and ζ_{n1} at the optimal $\omega_{n1} = 2.5$ kHz, where the cold color is much smaller in the index S than the warm color. The optimal parameters, as the result of coordinate design, have been determined as $\omega_{n1} = 2.5$kHz, $\zeta n1 = 0.06$, and $\zeta_{d1} = 0.12$, which correspond to the optimal point shown in Figure 11.

Figure 12 shows the experimental waveforms by coordinate design. From the figure, the residual vibrations are effectively suppressed under frequency variation by applying the optimized notch filter to a feedback compensator.

Soft computing techniques and/or computational intelligence can be an attractive alternative to conventional approaches to possess autonomous functions in the controller design.

Conclusions

This article surveyed the state of the art of high-precision motion control techniques for industrial applications, paying special attention to actual issues and relevant solutions in the 2-DoF controller design scheme.

Following the survey, an example of the practical applications of a robust 2-DoF controller was presented on the basis of the LMI-based FSC approach, to provide fast and precise positioning performance for the galvano scanner system with the frequency perturbation in the mechanism. In the proposed 2-DoF design, a novel concept to "coordinate design between feedforward and feedback controllers" was suggested to optimize the feedback compensator under the given feedforward input, to achieve a superior performance to the conventional 2-DoF controller design.

In the future, the authors plan to carefully examine the optimal feedback controller structure as well as its parameterization and should aim for the actual simultaneous optimization in both feedforward and feedback compensators to provide a novel 2-DoF controller design methodology.

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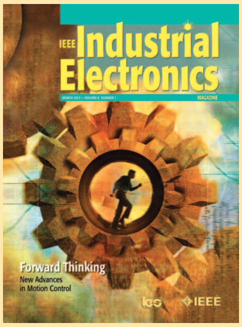

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