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Motion-Control Techniques of Today and Tomorrow

MICHAEL RUDERMAN,
MAKOTO IWASAKI,
and WEN-HUA CHEN

A Review and Discussion of the Challenges of Controlled Motion

Motion-control technologies are at the core of multiple mechatronic products and applications. Wherever actuated motion takes place in machines and components, either position or force setting (or trajectory

tracking), or even a combination of both, are demanded from the control system. In high-performance mechatronic systems, including micro- and/or nanoscale systems, including micro- and/or nanoscale motion (such as data-storage devices, machine tools, manufacturing tools for electronics components, and industrial robots), the required specifications in motion performance, such as response/settling time and trajectory/settling accuracy, should be sufficiently achieved [1].

From an industrial perspective, the most common design objective for advanced motion control is achieving a good tracking/following performance in the presence of disturbances and modeling issues—the nominal plant and performance specifications [2]. As an applied field of control theory, motion control was established in the 1990s [3], [4], and it is being independently driven by contemporary

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Continued progress in integrated mechatronic design and novel actuator technologies has created new challenges for motion-control technologies.

research in robotics [5]; however, continued progress in integrated mechatronic design and novel actuator technologies has created new challenges for motion-control technologies.

Likewise, current requirements for reducing materials and energy use are pushing mechatronic systems toward sensor and actuator reduction (hence, feedback degradation and underactuation), and they are becoming more lightweight and, therefore, flexible (i.e., soft) structures. These trends burden robust and efficient motion control with some ineluctable theoretical and practical issues, including plant specifications, disturbance modeling, variable structure systems, identification and state estimation, and control robustness and adaptivity, among others.

In this article, we review the state of the art and discuss some recent and potential challenges for motion-control techniques and related developments. Our goal is to highlight the principal features and associated issues of controlled motion in advanced mechatronic systems and applications for interested members of the IEEE Industrial Electronics Society (IES). We refer to works published not only in IES journals and conferences but also those associated with broader research in motion-control topics and affiliated technologies. Beginning with the basic

principles of motion stiffness and ideal versus disturbed motion under control, we clarify the vibrational perturbations attributable to the structures and damping perturbations caused by friction on contact interfaces.

“Stiffness” of Motion: From Position to Force Control

In control-engineering practice, the so-called stiffness is often used to characterize the bandwidth of a closed-loop control system and, correspondingly, the time constants of its transient response. In that regard, a “stiff” control loop provides faster response times and its corner (i.e., bandwidth) frequencies lie farther to the right compared with a less “stiff” control system. This is a direct analogy to basic second-order mechanical systems with mass-spring-damper elements, for which the stiffness of the restoring spring determines the natural frequency and, therefore, dynamic characteristics of the system response.

Although the term *stiffness* sounds like something self-evident for mechanical structures and, in second-order dynamics, for closed-loop control systems as well, the so-called stiffness of controlled motion must be clarified. From the point of view of force balance, a controlled motion [here, with 1 degree of freedom (DoF) for the sake

of simplicity] can be seen as a generic (nonlinear) map,

$$F = h(X, t), \quad (1)$$

between the generalized force F , totally imposed on the motion system, and the resulting relative displacement expressed in the generalized coordinate X . Implicitly, (1) incorporates both the motion plant and motion-control system as well as the function of the time derivatives of displacement (of a necessary order). The stiffness of the controlled motion, then, can be defined as a partial derivative [3]:

$$K = \frac{\partial F}{\partial X}. \quad (2)$$

The operation point of a generally nonlinear plant, just like more advanced nonlinear (often hybrid and/or adaptive) controllers, can lead to a time-varying stiffness, thus resulting in $K(t)$. The concept of stiffness of the controlled motion has been used in [3] and [6] as particularly suitable for a uniform consideration of position (or velocity) and force control as well as for different combinations of both. A high robustness in the motion system causes the adjustable stiffness, and, as a result, versatile applications will be possible [6]. Quite often, to satisfy the varying requirements of a particular motion-control application, the controller should have variable stiffness while the system remains highly robust. Figure 1 illustrates the motion stiffness for different control cases.

To design an ideal position or, alternatively, velocity control, one must ensure the controlled motion system is insensitive to variation in the forces, which can be weakly known and rather unpredictable disturbances. That means that, for any deviation ∂F , an ideal position control inhibits any deviations in the desired position—that is, $\partial X \rightarrow 0$ —which implies an infinite stiffness $K \rightarrow \infty$. This is also in accordance with the internal model principle [7], which requires, for example, that an integral control action be incorporated to compensate for the constant force disturbances so that $\partial X \rightarrow 0$ as $t \rightarrow \infty$. Just as a

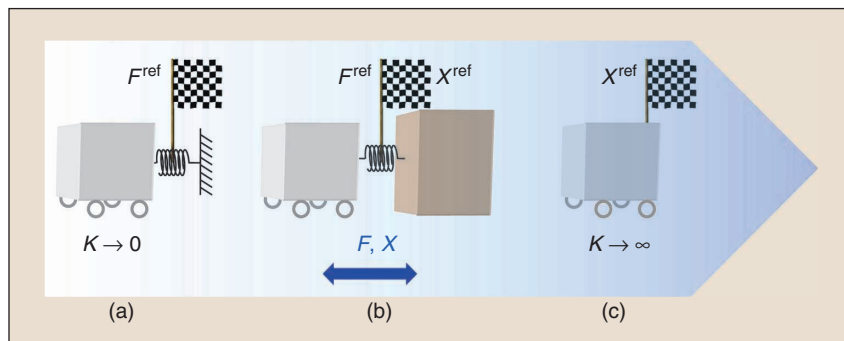


FIGURE 1 – The stiffness of controlled motion for the (a) force, (b) impedance, and (c) position control. The motion stiffness increases from the left to the right, while the flag represents the reference set value.

high-stiffness mechanical structure will deform less in response to the applied force, which is beneficial for high precision and, thus, position accuracy, a stiff motion-control design will ensure insensitive operation in the presence of external force disturbances.

Although an infinite static stiffness is provided by the integral-of-position-error state feedback, the so-called dynamic stiffness [8] should take into account the more specific behavior of foreseen disturbances. The static and dynamic stiffness terms used in [8] refer to infinitely high steady-state stiffness, due to the integral loop characteristics, and adjustable stiffness scalable for a particular disturbance frequency range. Generally, this requires a more elaborate and, often, adaptive control design.

However, control flexibility usually requires a tradeoff with some specific disturbance characteristics, such as modeling and robust performance. The latter point means a broader amplitude and frequency range of the exogenous signals, that is, reference trajectories and disturbances. Once the relative displacement state is available and used for control, a robust controller with high-gain feedback renders the motion stiffness respectively high. Therefore, with appropriately shaped closed-loop damping, the infinite-gain feedback design can theoretically approach an ideal position control with the desired infinite stiffness.

At the same time, the inherent actuator constraints make this theoretical realization impossible for practical applications, since the high-gain control actions, including integral-of-position-error feedback, may already hit the limits of the saturated actuator elements. A robust (in terms of performance) tracking controller with an explicit disturbance compensator (addressed in the “Control and Compensation Techniques in Motion Control” section) does not necessarily require an integral control action. This was shown in [4] for a high-accuracy positioning system and was explicitly analyzed in [9] for nonlinear friction-type disturbances.

In the same spirit of stiffness of the controlled motion [see (2)], an

Control flexibility usually requires a tradeoff with some specific disturbance characteristics, such as modeling and robust performance.

ideal force control should allow infinite deviation of position, that is, $\partial X \rightarrow \infty$, except when the force error is completely zero [6]. This implies the control stiffness tends to be zero, that is, $K \rightarrow 0$, under the nominal conditions of force regulation. From that point of view, an ideal force controller should drive the system to $\partial F \rightarrow 0$ through an induced relative motion, which will minimize the force error. That means that, to keep the total system force on some desirable balance value, $F(t) = \text{const}$.

Unlike pure position control, force control requires distinguishing between two principally different modes: 1) an unconstrained noncontact motion and 2) contact motion constrained by the system environment. In noncontact mode, a force control is basically an acceleration control [3]. Recall that the input of the robust motion controller should be the acceleration quantity [6]. Therefore, to realize versatile motion-control systems, for which the control stiffness can vary widely, the overall motion controller should have a double-cascade structure. The inner robust (against disturbances) acceleration loop allows accurate tracking of the desired acceleration reference, and the outer loop has the function of adjusting the motion stiffness by generating an appropriate acceleration reference.

This principle, explained in [6], was independently developed in robotics research [5], [10], [11] and has become standard in the robotics literature. It is also known as *feedback linearization*, which is often denoted (in robotics) as inverse dynamics control or computed torque control. The latter transmits coupled nonlinear manipulator dynamics into the well-known double integrator system [12]. An acceleration control (i.e., force control) during noncontact motion contains mainly a forward gain, which scales the force reference to an acceleration

quantity, provided the system inertia is known.

Except uncommon applications for which an exact acceleration measurement is available for feedback, a force control has no feedback loops during the noncontact motion. This appears quite naturally since having no contact with the environment means that force sensing is not returned from the tactile (end-effector) interface. Once the forced-controlled motion system comes in contact with environment, however, a closed mechanical loop arises through environmental stiffness and damping, which become the principal factors in the dynamics of a force-controlled system. With environmental damping that is typically insufficient and not ideally viscous, the force-controlled motion system becomes oscillatory at contact and, consequently, is repulsed back from the environment. However, the control loop, released from mechanical feedback, brings the system back into contact.

Repeated over and over, this hunting phenomenon [3] is well known in practical force-controlled robotics applications, and it was recognized in [13]. To overcome the hunting phenomenon, which can be seen as a local (i.e., transient) instability of the force control in the vicinity of environmental contact, a velocity feedback loop is usually inserted. However, when adjusted in this manner, the system can become unstable for small-velocity feedback gains, while the system response becomes remarkably slower for larger gain values [14].

Depending on the motion-control task, for instance, the position or force set point (or trajectory tracking), a large or, respectively, low stiffness must be achieved via a dedicated control design. At the same time, the motion-control problem can be specified in terms of obtaining the desired

A dedicated selection (or switching) strategy endows the entire control system with a hybrid behavior, including variable structures and logic-based switching.

mechanical impedance. Controlling the motion dynamics to respond as a second-order system yields

$$M_d(\ddot{X}_d - \ddot{X}) + D_d(\dot{X}_d - \dot{X}) + K_d(X_d - X) = F_c, \quad (3)$$

with the desired (apparent) mass M_d , desired damping D_d , and desired stiffness K_d . The latter is particularly characteristic of an impedance control, which can be seen as a tradeoff between an absolutely “stiff” positioning and absolutely “soft” force following (Figure 1). For the generalized dynamics (3) of an impedance-controlled motion system, the desired smooth motion reference is given by $X_d(t)$, while $F_c(t)$ is the contact force on interface with the environment.

For an ideal position control, with $X_d = 0$ for simplicity and $K_d \rightarrow \infty$, an external contact force, as disturbance, should not cause any displacement; that is, $X \rightarrow 0$. On the other hand, an ideal force control with $K_d \rightarrow 0$ will maintain the desired force while, at the same time, allowing for an unbounded displacement $X \rightarrow \infty$ of the apparent (i.e., virtual) mass with the shaped-as-desired damping. In other words, an impedance control is the general form of motion control [3], [5], since it is possible to turn the impedance control to both the position and force controllers

through the dedicated adjustment of control parameters.

The concept of impedance control was initially proposed by Hogan [15] and, since then, has been elaborated, especially in the robotics and control communities [11], [13], [16]. Particularly for safe and flexible human-machine interactions [17] and haptics [18], impedance control and its various extensions have gained remarkable recognition. With substantial progress in the sensing technologies and the lightweight, soft operation design of robotic and mechatronic systems, this remains a focus of active research. In robotics, for example, see the survey and references in Ajoudani et al. [19].

In the 1980s, it was recognized that both position and force cannot be controlled simultaneously; therefore, a hybrid position/force control was proposed in [20], based on an orthogonal decomposition of the task space. For multi-DOF motion control, a full-dimensional task space is split into the so-called position-controlled and force-controlled subspaces. A dedicated selection (or switching) strategy endows the entire control system with a hybrid behavior, including variable structures and logic-based switching. Today’s mature hybrid systems theory [21] provides increasing capabilities for

designing dedicated hybrid position/force controllers; for a recent example, see [22] and [23].

Ideal Versus Disturbed Motion Under Control

The relative motion under control is usually achieved with some feedforward and feedback regulators—both types designed, in large part, for a nominal system plant. Because of Newton’s laws of motion and the Lagrange–d’Alembert principle, the modeled relative motion always assumes a forwarding of the double integrator between the generalized force and relative displacement. At the same time, for multibody dynamics, which arise from multiple constructive elements in a motion system, the chain of multiple integrators appears. In this way, forward propagation of the input force (i.e., energy) results in increasing system dynamics (Figure 2). The additional control and actuator elements, located before the input force F , contain the coupled integrators, thus shaping overall system dynamics.

Although all modeled forward and feedback couplings make up the particular system behavior, determining the structure of signal flow and associated differential equations, the motion disturbances can appear as both exogenous quantities and/or functions of the system state, although, in the latter, with unknown/uncertain relationships and parameters. The structure depicted in Figure 2 represents a single-input, single-output (SISO) motion system, but it can also capture coupled single-input, multiple-output (SIMO) dynamics, provided the additional output channels are led out from the corresponding dynamic state variables. An example of such a SIMO system is a rotational robotic joint with gear elasticities (e.g., [16] and [24]), where both the motor and joint/link angular displacement appear as the output states of interest.

Motion systems with multiple input channels [i.e., multiple-input-multiple-output (MIMO)], are omitted here for the sake of simplicity.

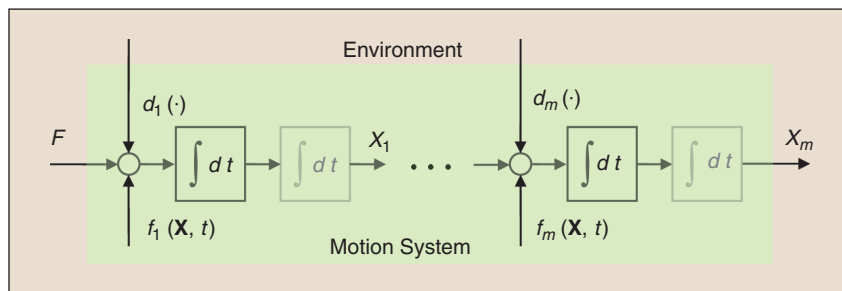


FIGURE 2 – A relative-motion system with m coupled double integrators actuated by the generalized force F and affected by nominal dynamics f and unknown internal and external (environmental) disturbances d .

Furthermore, depending on some of the unconstrained and uncoupled motion variables, the free integrators can vanish or be neglected in the modeling (see the light gray integrator blocks in Figure 2). An example of this is a linear damper (dashpot) element, within a chain of integrators, for which no inertial terms appear and the input force and output displacement are connected via the single integrator, according to Newtonian fluid dynamics. Another example is when the relative velocity of an unconstrained motion is the system output, thus resulting in elimination of one free integrator.

Since the disturbance signals can come from the environment—that is, outside of the motion system—their origins and impacts can be very different. Also, in terms of the matching condition (discussed in the “Matched Disturbance and uncertainty Attenuation” and “Mismatched Disturbance and Uncertainty Attenuation” sections), the disturbances can be matched, such as d_1 , which is the most simple case, or mismatched, such as generalization (i.e., $d_m \forall m \neq 1$). As a common feature, however, they mostly negatively affect the nominal system dynamics and, consequently, impair performance of the motion control or, in the worst case, even destabilize it.

Apart from the mechanical and often very pronounced vibrational and friction disturbances (addressed further in the “Vibrational Perturbation in Structures” and “Friction Perturbation on Interfaces” sections, respectively), other disturbing effects can appear in a closed-loop control system. Examples include the measured output bias or drift (e.g., [25]), disturbing effects due to control discretization [26], time-delay and communication disturbances in the loop [27], and parameter and load disturbances [28], [29], among others.

Assuming the vectors of the system state \mathbf{X} , nominal system dynamics f_i , disturbances d_i (Figure 2), and input force distribution $\mathbf{u}(\mathbf{X}, F)$, a generalized motion system can be approximated by

Since the disturbance signals can come from the environment—that is, outside of the motion system—their origins and impacts can be very different.

$$\begin{pmatrix} \dot{X}_1 \\ \ddot{X}_1 \\ \vdots \\ \dot{X}_m \\ \ddot{X}_m \end{pmatrix} = \begin{pmatrix} 0 & 1 & \cdots & 0 & 0 \\ & \frac{\partial}{\partial \mathbf{X}}(f_1 + d_1) & & & \\ & \vdots & & & \\ 0 & 0 & \cdots & 0 & 1 \\ & \frac{\partial}{\partial \mathbf{X}}(f_m + d_m) & & & \end{pmatrix} \begin{pmatrix} X_1 \\ \dot{X}_1 \\ \vdots \\ X_m \\ \dot{X}_m \end{pmatrix} + \begin{pmatrix} 0 \\ 1 \\ \frac{\partial}{\partial \mathbf{X}} \mathbf{u} \end{pmatrix} F. \quad (4)$$

The vector $\mathbf{u}(\cdot)$ of the force distribution maps the input force F to the lower-order motion variables X_i and \dot{X}_i with $i > 1$ (Figure 2) thus capturing zero dynamics in the system. The following remarks can be made for the linearized modeling (4) of the motion system.

- If the linearized motion system (4) has no zeros, the force coupling vector results in $\partial \mathbf{u} / \partial \mathbf{X} = 0$.
- The mismatched disturbances cannot be directly compensated by the control input force; see the discussion in the “Mismatched Disturbance and Uncertainty Attenuation” section. The computed (model-based) compensation of mismatched disturbances will be inherently affected by the impact of (multiple) integrators in the chain.
- If disturbances are not functions of the state and, hence, their partial derivatives do not exist, then the corresponding terms should be removed from the linearized system matrix of (4) and collected as an additional vector, thus approaching (14); see the “Measurement and Observation Techniques in Motion Systems” section.
- The state-independent disturbances, which appear as exogenous values, cannot destabilize the eigen-behavior of system (4) but significantly affect its transient and steady-state response.

The most challenging issues for advanced motion control are in the measurement or robust observation/estimation of various types of

disturbances, and their localization and isolation are a prerequisite for efficient compensation. This is necessary in addition to sufficiently accurate modeling that, however, may be subject to both parametric and structural uncertainties [30].

Vibrational Perturbation in Structures

Vibration in structures, caused by various types of mechanical resonances as well as electrical and/or physical linear/nonlinear properties, can deteriorate motion-control performances, especially for fast-response and high-precision applications [1]. To effectively design a motion controller (on the levels of force, torque, velocity, and/or position) against these vibrational properties, in general, precise modeling for and identification of the target systems should be indispensable for accurate model-based feedforward compensations and robust/adaptive feedback controllers.

Perturbation in the vibrational properties, however, essentially inheres in the actual target systems and should strongly affect higher motion-control performances. In mechatronic systems, the perturbation can appear as parameter/property variations in mechanical/electrical resonant frequency, damping/friction, actuator torque/force constants, and so on. In this section, practical knowledge of vibrational perturbation due to structured/unstructured uncertainties [30] is surveyed, with a focus on how this type perturbation occurs and how the effective modeling and controller should be designed.

Changes in Mechanical Structure

Changes and/or transformations in mechanical structures, such as posture/load of industrial robots, work arrangements of machine tools, and so on, directly vary the equivalent

moment of inertia and resonance frequency. An example is the linear trajectory motion for a six-axis industrial robot, where the equivalent moment of inertia, especially in the first or second axis, is drastically changed according to the posture [31]. This leads to perturbed-frequency Bode characteristics in the rigid mode gains and vibration mode characteristics (resonance/antiresonance frequencies and gains), resulting in deteriorating robot motion performances, including reduced accuracy in settling, trajectory, velocity, and so on.

In the case of the first or second axis, inertia generally varies up to three to four times, and this directly affects the control performance, such as bandwidth and stability of the feedback system as well as model-based feedforward compensation. To improve the motion-control performance against structural changes in the mechanism, precise modeling/simulation, based on kinematic analyses, variable structure control (e.g., variable filters [31]), adaptive algorithms, and robust control [32], may compensate for the variations. The CAD-supported kinematic analyses are also promising techniques for specifying the boundary of the perturbations, leading to robust controller design.

Scattered Characteristics in Mass-Production Systems

In the actual mass production of industrial mechatronic systems, the characteristics (of mechanical, electrical, and

physical properties) are essentially dispersed, which leads to scattered motion performance, because a controller is usually designed using a typical characteristic for a series of products. Figure 3(a) provides an example of scattered Bode gain diagrams for mass-produced galvano scanners for laser-drilling machines [1]. The scanner mechanism can be modeled by a multimass body system: a black line represents the gain characteristic in the nominal case, and the blue lines represent the scattered characteristics among numerous products. In Figure 3, the dispersion components are clearly observed in the rigid mode gain and frequency/gain of the primary, second, and third vibration modes. For this scanner positioning, the perturbation of some percentages leads to drastic performance deteriorations, with more than 10% error in the required specifications (settling time, accuracy, and so on); it is quite a different situation compared to the effects of perturbation due to changes in mechanical structure described earlier.

To ensure constant motion performance against the scattered characteristics, adaptive and/or robust control should be effective based on precise modeling and practical estimation in the perturbation boundary. An online autotuning algorithm [33], [34] is a promising method for performing the precise parameter selection in feedforward and feedback compensations as well as, from a practical viewpoint for industrial applications, saving time and labor.

Changes in Environmental Conditions

Environmental conditions in mechatronic systems, including temperature fluctuations, aged deteriorations (from hours to years), system delay components (dead-time and servo lag in the controller or time delay in communication lines), and unknown disturbances, drastically affect motion performance in a wide variety of actual motion operations. For example, the temperature fluctuations in mechanisms and/or electrical components can directly cause variations in the vibration modes (gain and frequency), damping and friction forces, and actuator torque/force constant due to the essential properties in mechanical/electrical materials (metals, magnets, and so on). Figure 3(b) shows an example of perturbed gain characteristics of the galvano mirror, with frequency variations $\delta = \pm 100$ Hz in the primary vibration mode due to temperature fluctuation during motion operations.

As a result of perturbations, the settling performances in positioning are clearly deteriorated, as shown in Figure 4 [35]. In Figure 4(a), as the positioning operations of galvano scanners mirror progress from 5 to 800 s, the overshoot responses gradually increase due to the temperature rising in the mechanism and actuator, which leads to changes in vibration mode and a decrease in the actuator torque constant. Figure 4(b), on the other hand, clarifies that the fluctuation in the actuator torque constant with a range of $\pm 1.5\%$ deteriorates

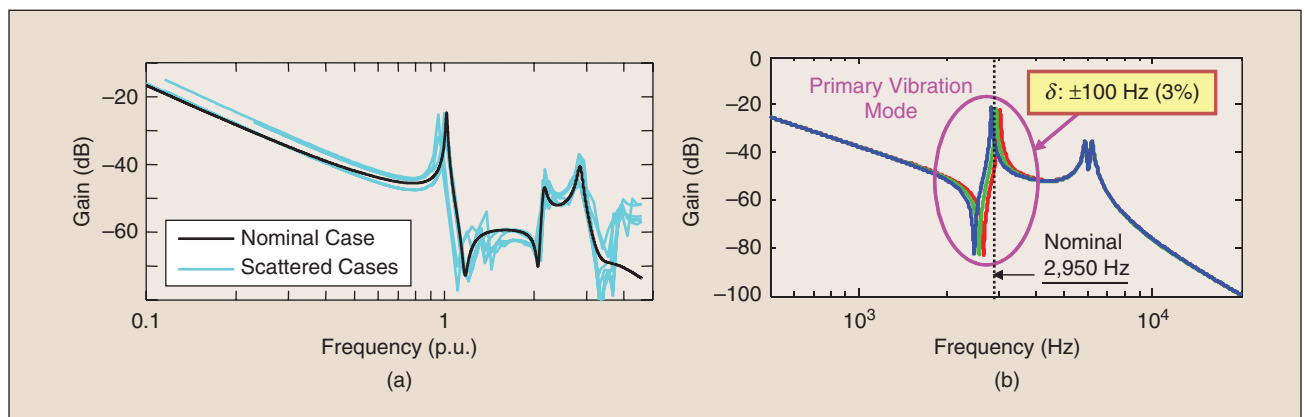


FIGURE 3 – (a) An example of scattered Bode gain characteristics among mass-produced galvano scanners for laser-drilling machines. (b) The gain diagrams of the plant system with frequency variations $\delta = \pm 100$ Hz in primary vibration mode due to temperature fluctuation during motion operations. p.u.: per unit.

the position-settling performances with overshoot and/or undershoot responses corresponding to the torque constant variation.

To compensate for environmental changes, online autotuning and/or adaptive control are also helpful techniques, especially for aging deteriorations in mechanisms and electrical components. Those algorithms, in addition, can be applied to cases caused by the machine-setting environment with different floor conditions, leading to a change in the resonance frequency of machine stand vibration. As recent globalized manufacturing chains are rapidly expanding, these types of autotuning/adaptation algorithms should be powerful enough to eliminate the various environmental perturbations [36].

The promising approaches and/or techniques—that is, practical precise modeling and controller design, robust and/or adaptive control, and optimization (including online autotuning)—should be applied to compensate for motion performance deteriorations due to the actual vibrational perturbation. In addition, these techniques may be used to create to diagnosis and/or fault-tolerance algorithms to prevent unexpected accidents [37].

Friction Perturbation on Interfaces

Motion-control systems are usually equipped with various types of contact interfaces, due to the part bearings, which move relative to each other. Exceptions, also encountered in mechatronic design, are the air and magnetic bearings; see [38] and [39], respectively, for an example and overview. Both technologies, though relevant and fairly challenging for design and motion control, are not directly associated with contact interfaces and related friction perturbations; therefore, we do not focus on them here.

Once a pair of mechanical parts is in loaded physical contact, the corresponding (generalized) friction force appears on the interface and acts in a direction opposite to the ongoing relative motion. The friction is present in all machines incorporating parts with

relative motion [40], and, although friction may be a desirable property, as it is for brakes, it is generally an impediment for servo control.

Not surprisingly, the volume of research dedicated to modeling, identifying, and compensating for the effects of friction is huge and cannot be satisfactorily summarized in this article. For an earlier and well-celebrated survey, please refer to [40]. A more recent seminal work [41], dedicated to friction-force dynamics, summarizes and highlights (as do the references therein) the most pronounced

phenomena associated with kinetic friction, which include the so-called presliding friction behavior, adhesion and creeping, transient friction lag, and stick-slip motion.

The appearance of friction forces on a contact interface, independent of the type of contact mechanism (e.g., slider, ball bearing, roller, and so on) can be imagined by inspecting the principal configuration depicted in Figure 5(a). A normal loaded (by force N) contact pair induces the kinetic friction force F , counteracting relative displacement with velocity \dot{X} . Sliding along each

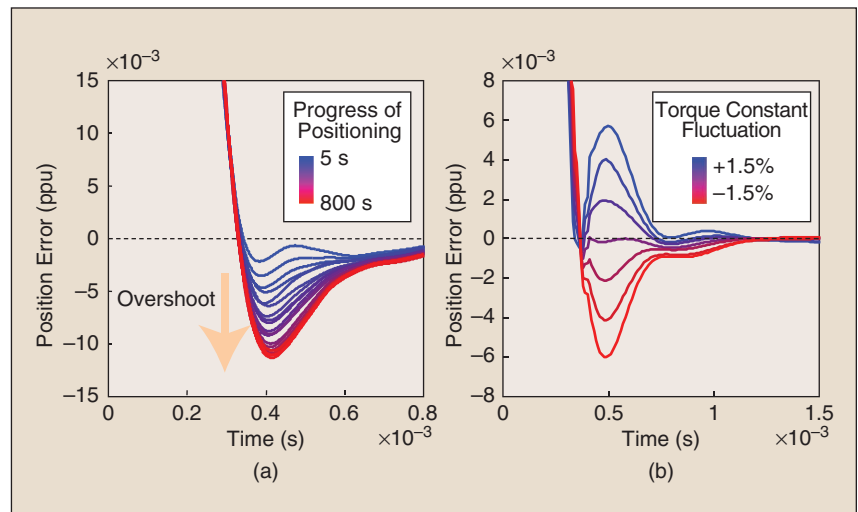


FIGURE 4 – (a) The changes in response of position error due to vibrational perturbation galvano mirror positioning progresses from 5 to 800 s: the temperature of mechanical/electrical components rises during the progress, leading to changes in vibration mode and actuator torque constant. (b) The deteriorations in settling performance due to fluctuation of the actuator torque constant: overshoot and/or undershoot responses clearly occur, corresponding to torque constant variation. ppu: position per unit.

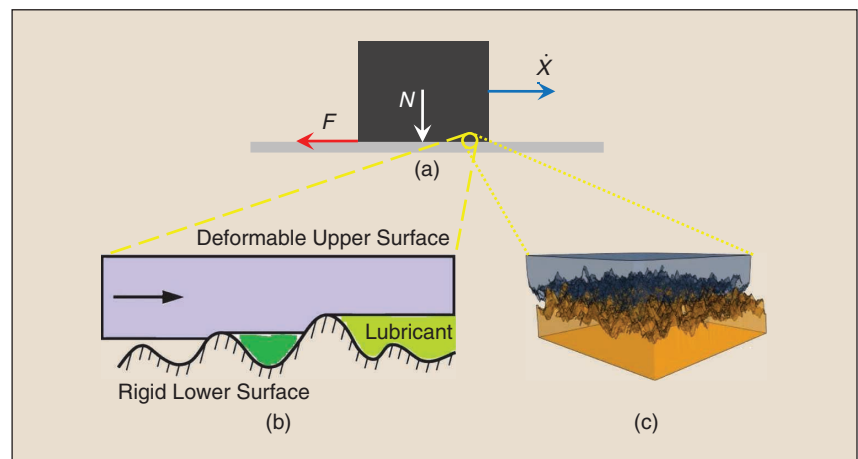


FIGURE 5 – The friction appearance of the contact interface. (a) A contact pair of moving bodies on the friction surface with normal load (N). (b) A schematic representation of the deformable upper surface moving along the rigid lower surface with lubricant penetration between asperities. (c) The typical topology of the rough surface pair in contact.

other, the outer surface layers deform elastoplastically [for the sake of simplicity, only one surface is drawn as deformable in Figure 5(b)], while the surface roughness provides irregular contact asperities, with lubricant medium distributed in between. The squeezed lubricated regions, marked by dark green color in Figure 5(b), are characterized by hydrodynamic pressure of the lubrication medium, which gives rise to an additional lift-up effect [41]. Despite the irregular landscape of the rough surfaces in contact [Figure 5(c)], a particular surface topology can be characterized by some average values of the asperities' height, distance, distribution, and elastoplastic characteristics.

Given an accurate and real-time-capable model of the friction forces, those can be compensated for (at least theoretically) in either a feedforward or feedback manner. This has been widely attempted in previous works since the kinetic friction, in the general consensus, constitutes a deterministic nonlinear map between the arising friction force and relative displacement rate as its cause; that is, $f: \dot{X}(t) \rightarrow F(t)$. However, despite notable progress in the understanding and modeling of friction phenomena, in most cases, the direct model-based compensation techniques fail when it comes to robust and practice-suitable motion control.

This is due to several reasons that are quite different in nature, most obviously the nonfeasibility of directly measuring friction on the contact interfaces. Since the friction force appears as a mismatched and internally

coupled disturbance (see Figure 2 and the related discussion on ideal versus disturbed motion), its proper decomposition and isolation within the total balance of forces almost always rely on some strong and often artificial assumptions, without sensor-based examination of the physical values. Therefore, the friction quantities in servo-controlled machines are usually obtained as residual values, out of the total dynamics balance, provided the other force components are properly identified and known based on the measurements.

The next obstacle in dealing with accurate friction properties lies in the (predominantly) spatial nature of friction, whereas the residual terms of the system and control dynamics are specified and handled in the time or frequency domains. An example of this misalignment in dynamics properties is a degraded-integral control performance when attempting to compensate for presliding friction [42]. This issue has been analyzed and experimentally demonstrated in [9]; see the example shown in Figure 6. Here, a well-tuned integral control part fails to ensure a fast settling behavior and results in a slow creeping of the controlled position response due to nonlinear presliding friction.

Finally, the kinetic friction in the motion-control system is subject to large uncertainties due to multiple, and often weakly known, internal and external factors, such as roughness, thermal and lubricant state of contacting surfaces, varying normal loads, dwell time, and wear, among others [43].

This fact requires an additional and, above all, fast estimation (or corresponding observation) of the transient and time-varying friction states and/or parameters. The latter can often be assumed as available, with the previously identified (i.e., nominal) values. Therefore, the classical adaptive (in parameters) strategies and algorithms are of less priority here, although they remain interesting for control tuning. This is in the case of changeable loads, thermal and lubrication conditions, or control initialization when set into operation. The necessity of observing the friction disturbances has been already shown in [44] and continues in current research, as has been demonstrated, for instance, for the absolute (encoder resolution) accuracy of positioning [43] and friction-torque estimation in robotic joints [24], [45].

Although a generalized empirical friction model structure, according to [41], requires the friction force to be a (nonlinear) map $F = \mathcal{F}(X, \dot{X}, z)$ of the relative displacement and some internal state vector equally, the dynamics of which are driven by $\dot{z} = \mathcal{G}(X, \dot{X}, z)$, another general friction force mapping can be postulated as well. For the relative displacement rate, assumed to be the main factor driving the dynamics of kinetic friction, one can write

$$F = f(\dot{X}, t), \quad (5)$$

where an explicit time argument captures the time-varying behavior and dynamic response of the friction force. In the most simple and well-known case of a static friction force, (5) transforms into $F = \alpha \dot{X} + \beta \text{sign}(\dot{X})$, with α and β as the linear viscous and corresponding Coulomb friction coefficients, respectively.

In the more general dynamic case of kinetic friction, a differential equation,

$$\dot{F} = g(F, \dot{X}, t), \quad (6)$$

can be assumed instead of (5). Note that (6) is compliant with the Dahl friction model [46] in its original differential form. The Dahl model can be credited as the first but still relevant approach for capturing transient

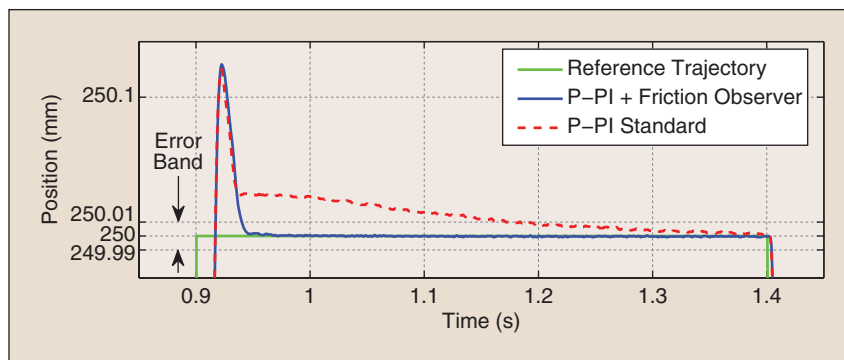


FIGURE 6 – The position response of the standard proportional–proportional–integral (P–PI) cascaded controller and that extended by an explicit friction observer (exemplified experimental evaluation on the ball-screw positioning stage from [9]).

presliding friction characteristics. If one assumes the total kinetic friction as some (general) superposition of the Coulomb- and viscous-type friction terms, then (5) and (6) can be concretized, as in the example provided in [43]. When doing this, the friction uncertainties will end up either in the lumped but time-varying friction parameters or unknown disturbance to be observed, as will be described later in this section.

Although disturbances due to unknown friction can be classified, more or less, by certain amplitude and frequency ranges or assumed to be correspondingly steady-state upper bounded, the disturbances due to uncertain friction are often highly dynamic and transient. This makes them particularly challenging for an appropriate estimation, as a convergence phase is required before an estimated friction state can be used instead of a real physical value. This is generally independent of the type of designed observer. For instance, following the modeling approach [43], the dynamics (6) of the friction force results in $g: \alpha \tau^{-1} \dot{X} - \tau^{-1} F + \beta d/dt \operatorname{sign}(\dot{X})$, where τ is the time constant of friction lag. Otherwise, the time-varying $\alpha(t)$, $\beta(t)$, and $\tau(t)$ parameters, which capture the uncertainties of friction behavior, would significantly affect the friction-force dynamics, and that coupling creates challenges for proper decomposition. It is also evident that the two-times Dirac delta function, due to the sign discontinuity of the modeled Coulomb friction, is additionally weighted by, in this case, the varying $\beta(t)$ parameter. That leads to an uncertain excitation of the force dynamics every time the sign of the relative velocity changes. Different approaches have been proposed to deal with force discontinuity at motion reversals and the fact that relative motion never starts or stops abruptly but, instead, within a certain (presliding) region of the relative displacement and, correspondingly, growing friction force.

This began with the Dahl model [46] and, since then, has flourished in the number of approaches with different modeling complexities and levels of details when capturing presliding

There are different types of observation techniques: state observation, disturbance observer, and joint state and disturbance estimation.

transitions. Along with several formulations based on nonlinear differential equations, the Maxwell-slip-type structures, equivalent to the distributed Prandtl-Ishlinskii hysteresis operator [47], turned out to be particularly suitable for shaping hysteresis in the presliding-force transitions. Further extensions of the principal Coulomb and viscous (including Stribeck effect) friction laws attempt to capture the stochastic aspects of rough surfaces, temporal and load-dependent relaxation, adhesion (also known as *stiction*), impact of dwell time and “warming up” effects, and others.

At the same time, despite the fact that more elaborate friction models can yield a better consonance with experimental data, it is the generality in the modeling of kinetic friction that remains, and perhaps will remain, one of the most appealing issues. Ensuring a sufficient tradeoff between generality and level of detail when describing the multifaceted phenomenon of kinetic friction should allow for an appropriate estimation and compensation of friction perturbations in controlled motion systems.

Measurement and Observation Techniques in Motion Systems

Measurement and estimation techniques are becoming more important in motion systems. They not only provide information required by control systems but also facilitate many other important tasks, including operating-status monitoring, fault detection and diagnosis, and finding changes in environment conditions. Sensors and physical devices are used to measure the state and workload of a dynamic system or sense the working/operational environment. With advances in sensing technologies, more physical quantities are now available for measurement, with increasing accuracy.

However, there is concern about cost, which is important for industrial mechatronic systems, particularly those in mass production. Significant effort is also being made to extract or infer as much information as possible from available measurements. Many estimation and observation techniques have been developed; examples include the development of sensorless control in motion systems [48], where some key feedback measurements are replaced by estimation techniques. More recently developed is the so-called virtual torsion sensor [24], with which an output encoder can be reduced in robotic joints with non-negligible elasticities. These could be considered soft sensor technologies, and they quite often rely on a good model or knowledge of a motion system and its operational environment of concern.

There are different types of observation techniques: state observation, disturbance observer (DOB), and joint state and disturbance estimation. In many applications, disturbance estimation techniques are used to estimate not just external disturbances but also the influence of system uncertainty in modeling, for example, due to the mismatch between a model and physical system or the change of a motion system as a consequence of the change of operational conditions and environments, including system faults. In the following sections, we discuss a number of existing and new observation technologies, focusing on disturbance estimation, as there exists a significant amount of literature on state observation (see [49] for an overview).

Linear Estimation Techniques

Many observation technologies are available for motion systems described by linear dynamics. The classic Luenberger state observer [50] design

technique for deterministic systems and Kalman filtering [51] for stochastic systems are well known methods for estimating unavailable system states from output measurements. The problem becomes slightly more complicated in the presence of external disturbances. A number of approaches have been developed for directly estimating the disturbance or jointly estimating the state and disturbance by augmenting the disturbance into the state so that a conventional Luenberger state observer or Kalman filtering technique can be used. Depending on the description of the motion-system model, transfer function and state-space approaches have been developed.

Transfer Function Approaches

The frequency-domain DOB was originally proposed in [52]. Suppose that a motion system is described by a transfer function $G(s)$. The basic idea in [52] is to obtain a disturbance estimate by filtering the differences between the control and calculated inputs using the inverse model of the nominal plant $G_n(s)$ through a low-pass filter $Q(s)$.

The basic diagram of the DOB is given in Figure 7(a), where $Q(s)$ is designed as a low-pass filter with unity gain. The relative degree of $Q(s)$ (i.e., the order difference between the numerator and denominator) is not less than that of the nominal plant $G_n(s)$ to ensure that $Q(s)G_n^{-1}(s)$ is implementable. One promising feature of the DOB technique is that it is able to estimate not only external disturbance but also the influence of uncertainty, the so-called lumped disturbance concept, illustrated by the diagram in Figure 7(a). Following block-diagram manipulation, it can be shown that

$$\hat{d} = Q(s)((G_n^{-1}(s) - G^{-1}(s))y + d).$$

This implies that the disturbance estimate consists of two parts: the external disturbance d and the mismatching between the physical system and its nominal model. Therefore, DOB techniques are able to estimate the influence of uncertainty; this is why they are also widely used as a robust control method, as discussed in the “Control and Compensation Techniques in Motion Control” section.

However, the original structure in [52] cannot handle the nonminimum phase systems, since the direct inverse of the nominal plant $G_n(s)$ brings unstable poles in $Q(s)G_n^{-1}(s)$. To this end, an improved and more generic version of DOB is given in [53]. It is depicted in Figure 7(b), where $M(s)$ and $N(s)$ take the following form:

$$M(s) = \frac{M_n(s)}{L_1(s)}, \quad N(s) = \frac{N_n(s)}{L_1(s)}. \quad (7)$$

The nominal plant is represented as $G_n(s) = M_n(s)/N_n(s)$, with $L_1(s)$ being a stable polynomial, and $Q(s)$ is designed as a low-pass filter, such that $M(s)Q(s)$ is a low-pass filter with unity gain. The key design parameter in this approach is the low-pass filter $Q(s)$, which depends on the frequency of the disturbance and uncertain dynamics to be estimated, the frequency of the sensor noise, and the system dynamics. Guidance for how to design the DOBs through this approach can be found in [54].

State-Space Approaches

Shortly after the state-observer technique (e.g., Luenberger observer [55]) was developed in the 1960s, Johnson [56], [57] extended it to estimate

external disturbance. This was realized by augmenting the system dynamics with a disturbance model so that the augmented system consists of both the system states and the disturbance (or the internal states of a disturbance dynamics). Therefore, the system states and disturbance can be simultaneously estimated by using an existing state-observation technique. Consider a SISO linear system subjected to unknown disturbances, given by

$$\begin{cases} \dot{x} = Ax + B_u u + B_d d, \\ y = Cx, \end{cases} \quad (8)$$

where $x \in R^n$, $u \in R$, $d \in R$, and $y \in R$ are the system states, control input, disturbance, and measurement, respectively. A , B_u , B_d , C , and D are the corresponding system matrices, which can be considered as a state-space realization of the nominal plant $G_n(s)$ in Figure 7(a). The disturbance is supposed to be approximately represented by the linear exogenous system

$$\dot{\omega} = S\omega, \quad d = H\omega, \quad (9)$$

where $\omega \in R^q$ and the pair (S, H) are known and observable. It can represent a wide range of disturbances including unknown constant, ramp, and periodic signals, such as sinusoid waves.

Combining the system dynamics (8) and disturbance dynamics (9), a composite system can be obtained:

$$\begin{cases} \dot{\bar{x}} = \bar{A}\bar{x} + \bar{B}u, \\ y = \bar{C}\bar{x}, \end{cases} \quad (10)$$

where $\bar{x} = [x^T, \omega^T]^T$, and the system matrices are given by

$$\bar{A} = \begin{bmatrix} A & B_d H \\ O_{q \times n} & S \end{bmatrix}, \quad \bar{B} = \begin{bmatrix} B_u \\ O_{q \times 1} \end{bmatrix}, \\ \bar{C} = [C \quad O_{1 \times q}].$$

Following a standard state-observer design procedure, the disturbance can be estimated by

$$\begin{cases} \dot{\hat{x}} = \bar{A}\hat{x} + \bar{B}u + K(y - \bar{C}\hat{x}), \\ \hat{\omega} = \bar{C}\hat{x}, \\ \hat{d} = H\hat{\omega}, \end{cases} \quad (11)$$

where K is the observer gain matrix, and hatted variables are the estimate of the corresponding variables. It is

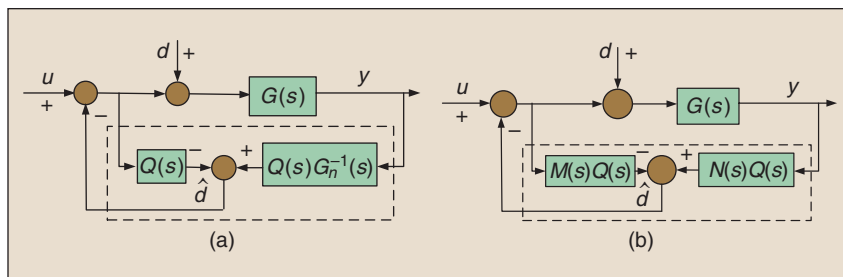


FIGURE 7 – (a) The classic Q -filter-based DOB, as in [52]. (b) A generic DOB structure.

easy to see that this approach could be extended to handle multiple disturbances and MIMO systems. For stochastic systems, Kalman filtering could be directly applied in a similar way.

This approach is basically joint state and disturbance estimation. In some applications, only the disturbance is of interest; for example, when all of the states are already available or only faults or environment changes (which can be modeled as a type of disturbance) are of concern, a disturbance estimation could be developed using the so-called functional observer concept in the state-space approach [50]. This also provides a counterpart of the DOB design in the transfer function approach described in the “Linear Estimation Techniques” section.

A state-space DOB is proposed in [58] using the functional observer concept:

$$\begin{cases} \dot{z} = Fz + Gy + Tu, \\ \hat{\xi} = z + Jy, \\ \hat{\omega} = \tilde{C}\hat{\xi}, \\ \hat{d} = H\hat{\omega}, \end{cases} \quad (12)$$

where z is the state variable of the observer, and F , G , T , and J are the gain matrices designed to satisfy

$$\begin{cases} W\bar{A} = FW + G\tilde{C} \text{ (Sylvester equation)}, \\ W = L - J\tilde{C}, \\ T = W\bar{B}, \\ F \text{ is stable}, \end{cases} \quad (13)$$

where W is the intermediate matrix, and L is chosen based on the order of the observer to be designed. From (13), one can prove that the disturbance estimation error converges to zero. Also in (13), the order of the DOB could be changed by using matrices with different dimensions, which is similar to choosing a different order of the Q filter in the transfer function approach. The design procedure can be found in [58]. The main difference between the functional DOB in (12) and (13) and the DOB in (10) and (11) is that it is much more flexible in choosing the order of the DOB. Essentially, the DOB in (10) and (11) is a full-order observer, whereas the one in (12) and (13) is a reduced-order observer, for which the order could be determined by a designer (subject to the existence condition in [58]).

Estimating states and/or disturbances for a motion system described by nonlinear dynamics is much more challenging.

Although there are similarities, the transfer function approaches and the state-space approaches have different features. For a further discussion of the relationships of the disturbance estimation to the transfer function and state-space approaches, please refer to [59] and [58].

Nonlinear Estimation Techniques

Estimating states and/or disturbances for a motion system described by nonlinear dynamics is much more challenging. In general, unlike for linear systems, there are no design methods for a nonlinear state observer for a general nonlinear system, although researchers have investigated developing nonlinear state observers for motion systems with a specific structure [60]. Consequently, it is even more difficult to design a joint state and disturbance observation for a nonlinear system. Therefore, the approach of augmenting the states with disturbances, widely used in linear systems, cannot be extended to nonlinear systems. However, there are several generic nonlinear DOB techniques for nonlinear systems.

Consider a nonlinear system, in companion form, described by

$$\begin{cases} \dot{x}(t) = f(x(t)) + g_1(x(t))u \\ \quad + g_2(x(t))d(t), \\ y(t) = h(x(t)). \end{cases} \quad (14)$$

It is assumed that $f(x)$, $g_1(x)$, and $g_2(x)$ are smooth functions in terms of the state x .

A number of techniques have been proposed to design nonlinear DOBs to estimate the disturbance d (e.g., [62]), and [49] provides an overview. A widely used method, sometimes also called *Chen's DOB* to distinguish it from other nonlinear DOB techniques, was proposed and evolved from an initial nonlinear DOB presented in [63] for dealing with friction in a robotic manipulator system. We introduce this nonlinear DOB technique here:

$$\begin{cases} \dot{z} = -l(x)g_2(x)z - l(x)[g_2(x)p(x) \\ \quad + f(x) + g_1(x)u], \\ \hat{d} = z + p(x), \end{cases} \quad (15)$$

where $z \in R$ is the internal state variable of the observer, and $p(x) \in R$ is an auxiliary variable to be designed. The nonlinear observer gain $l(x)$ is then determined by

$$l(x) = \frac{\partial p(x)}{\partial x}. \quad (16)$$

The nonlinear DOB asymptotically estimates the disturbance if the observer gain $l(x)$ is chosen such that

$$\dot{e}_d = -l(x)g_2(x)e_d \quad (17)$$

is asymptotically stable regardless of x , where $e_d = d - \hat{d}$ is the disturbance estimation error.

There are many ways to choose the nonlinear gain $l(x)$ and $p(x)$ such that (17) is stable and (16) is satisfied; a systematic approach was suggested in [64]. For a nonlinear system with a well-defined relative degree ρ from the disturbance to the output (i.e., the order difference between the highest derivative of the disturbance and the output), $p(x)$ is suggested as

$$p(x) = p_0 L_f^{\rho-1} h(x), \quad (18)$$

where $L_f^{\rho-1} h(x)$ denotes the Lie derivative defined as the $(\rho-1)$ th derivative of the function $h(x)$ along the function f ; see [65] for a basic discussion. Consequently, the nonlinear observer gain $l(x)$ is given by

$$l(x) = p_0 \frac{\partial L_f^{\rho-1} h(x)}{\partial x}. \quad (19)$$

Therefore, the whole design process boils down to the problem of choosing the parameter p_0 , which is a tuning parameter for the compromise between the convergence rate and sensitivity to noise. Once this parameter is chosen,

In addition to its simple design, a promising feature of the described method is that the controller design could be separated from the DOB design.

the nonlinear observer gain $l(x)$ and the auxiliary variable $p(x)$ are determined by (19) and (18), respectively; then, the nonlinear DOB is given by (15). This nonlinear DOB technique has been widely applied in motion and other mechatronic systems.

In addition to its simple design, a promising feature of the described method is that the controller design could be separated from the DOB design. This somehow extends the so-called separation principle of state-observer-based control design for linear systems. The separation principle, more formally known as the *principle of separation of estimation and control*, implies that the state-observer and controller designs could be distinct from each other.

First, a state feedback-control law is designed under the assumption that all of the states are available. A state observer is then designed to estimate the state. Finally, the feedback states are replaced by their estimates yielded by the state observer. In a similar fashion, a feedforward compensation could be designed under the assumption that the disturbance is measured (see the next section for design methods), and then it is replaced by its estimate yielded by the nonlinear DOB given in (15). Therefore, the control and DOB designs can be separated for disturbance attenuation and robustness of nonlinear systems.

This separation for nonlinear systems [66], [67] is realized due to the special feature of the proposed nonlinear DOB (15). That is, the convergence of the disturbance estimation is achieved irrespective of the state of the nonlinear systems. Consequently, Chen's nonlinear DOB can be combined with an arbitrary nonlinear control design method to improve its disturbance rejection or/and robustness under certain conditions. This feature makes the nonlinear DOB and

its control design framework (DOBC) attractive, thus creating a successful design method in the area.

Control and Compensation Techniques in Motion Control

In addition to traditional proportional-integral-derivative (PID) controllers or the like, Many advanced control methods have been developed and applied to motion-control systems, including model-predictive control, robust control, adaptive control, slide-model control, internal-mode control, output regulation, passivity-based control, and active-disturbance-rejection control. The amount of related control literature is too huge to be properly listed here; for some survey and overview sources, please refer to [1], [65], and [68]–[72].

In this section, we will focus on a new type of composite control method. This control mechanism, with a 2-DoF control structure, tries to exploit both feedback and feedforward strategies and combine them together to achieve high performance in tracking/regulation, good stability and disturbance rejection, and strong robustness. More specifically, a feedback-and-feedforward control strategy has been extensively discussed in [1], where a feedforward controller that directly generates a feedforward control signal based on reference/command is integrated with a feedback controller that is driven by the control error. Another 2-DoF robust control design has been shown for motion control of a servomotor [28] based on coprime factorization by the Youla–Kucera parametrization technique (see [73] for details).

Another widely used way to employ feedforward is to attenuate the influence of disturbance by generating a compensation based on disturbance measurement. However, since the disturbance or the influence of uncertainties may not be available (or not

measurable), the feedforward strategy is facilitated by the disturbance-estimation techniques introduced in the “Measurement and Observation Techniques in Motion systems” section. DOBC is one type of this composite control method. Thanks to the separation principle discussed earlier, a design procedure for developing a composite control scheme for motion systems is outlined.

- *Step 1:* Design a feedback controller to achieve stability and tracking/regulation performance without considering disturbances/uncertainties.
- *Step 2:* Develop a feedforward control strategy to reject disturbances or uncertainties under the assumption that they are measurable.
- *Step 3:* Create a DOB to estimate disturbance or the influence of uncertainties.
- *Step 4:* Integrate the feedback-and-feedforward strategy with the disturbance replaced by its estimate to constitute a composite controller and analyze its performance.

Design of a feedback controller in step 1 could be carried out with any suitable design methods, such as PID, model predictive control, and linear quadratic regulator, and several techniques for the DOB design in step 3 were introduced in the “Measurement and Observation Techniques in Motion Systems” section, depending on whether the dynamics system is linear or nonlinear.

In the following sections, we will discuss how to design the feedforward compensation to reduce the influence of disturbance and uncertainties in step 2. There are many disturbances and uncertainties in motion-control systems. For ac-motor-driving systems, these are the mechanical parameters, such as the changes in inertia; electrical parameters, including changes in stator resistance due to temperature; friction torque; load torque; cogging torque; flux-harmonic torque; distortion voltage; current-offset errors; skewed slot torque; and so on. Obviously, it is important to improve the disturbance rejection and robustness of the controlled motor-drive system.

Matched Disturbance and Uncertainty Attenuation

Disturbance and/or uncertainty satisfying the matching condition implies that the disturbance or effect of the uncertainty is applied in the same channel as the control input or, more precisely, the influence of a disturbance or uncertainty can be equivalent to the input channels in some way. Almost all of the current transfer-function-based composite design approach (e.g., DOB) has this assumption explicitly or implicitly. For a system described by a state-space model, such as (8), the matching condition is met when $B_d = B_u$ (or, more precisely, $B_d = B_u \Gamma$ for some Γ). Similarly, the matching condition is satisfied if $g_1(x) = g_2(x)$ in (14) for nonlinear systems.

Because the control input and the disturbance are in the same channel, the feedforward compensation strategy is quite straightforward. That is, the influence of the disturbance could be completely removed (at least theoretically) by generating a counteracting control action. Consider a composite controller given by

$$u = u_{fb} + u_{ff}, \quad (20)$$

where u_{fb} is control action calculated by a feedback controller designed in step 1, and u_{ff} denotes the feedforward control. Consequently, after replacing the true disturbance with its estimate, the corresponding composite control law in the presence of matched disturbance is given by

$$u = u_{fb} + u_{ff} = u_{ff} - \hat{d}, \quad (21)$$

where \hat{d} is the estimate of the lumped disturbance. It can be shown that the influence of the disturbance or uncertainty could be completely removed if the estimate yielded by a DOB approaches the real disturbances [64].

Mismatched Disturbance and Uncertainty Attenuation

Quite often, the disturbance and control do not occur in the same channel, so the matching condition is not satisfied; in general, this is true when considering the influence of

uncertainties as a part of disturbances. Mismatched disturbance and uncertainty widely exist in practical applications. In a servo control system, for example, the disturbance could be a load torque, but the control is the voltage applied to the motor so that they do not appear in the same channel. An electric driving system must generate corresponding current in the motor (which is a state of the motor) to counteract the external load torque. This implies that some states cannot approach to zero in the steady state in the presence of mismatched disturbances/uncertainties.

The key issue for compensation for the mismatched disturbance using a feedforward strategy is to find an action based on the estimate for general dynamic systems. This issue has been answered for not only linear but also nonlinear systems [74], [75], and a systematic approach to design disturbance compensation gain that is able to remove the influence of mismatched disturbance/uncertainty from the output, at least in the steady state has been proposed. For a linear system (8), the disturbance compensation gain is given by [74]

$$K_d = -[C(A + B_u K_x)^{-1} B_u]^{-1} \times C(A + B_u K_x)^{-1} B_d, \quad (22)$$

where K_x is the feedback control gain. It can be shown that this compensation gain reduces to $K_d = -1$ in the case of matched disturbance, as in (21). This is because $B_d = B_u$ in that case. Therefore, the matched case could be considered as a special case of the design for the mismatched case.

If all state variables are measurable, a reduced-order DOB (for example, the functional-observer-based design [58] discussed in the "Measurement and Observation Techniques in Motion Systems" section) can be used, and the following composite control law is employed

$$u = K_x x + K_d \hat{d}. \quad (23)$$

For unmeasurable state variables, a joint state and DOB could be employed. In this case, the composite control law is designed as

$$u = K_x \hat{x} + K_d \hat{d}, \quad (24)$$

where \hat{x} and \hat{d} are yielded by state and DOBs.

Now we consider a SISO nonlinear system (14). The disturbance can be removed from the output channel in the steady state by the following composite control law [75]:

$$u = \alpha(x) + \beta(x)v + \gamma(x)\hat{d}, \quad (25)$$

where \hat{d} is the disturbance estimate obtained by the nonlinear DOB,

$$\alpha(x) = -\frac{L_f^\sigma h(x)}{L_g L_f^{\sigma-1} h(x)},$$

$$\beta(x) = \frac{1}{L_g L_f^{\sigma-1} h(x)},$$

and σ is the input-to-output relative degree [65]. The disturbance compensation gain is designed by

$$\gamma(x) = -\frac{\sum_{i=1}^{\sigma-1} c_i L_p L_f^{i-1} h(x) + L_p L_f^{\sigma-1} h(x)}{L_g L_f^{\sigma-1} h(x)} \quad (26)$$

and

$$v = -\sum_{i=0}^{\sigma-1} c_i L_f^i h(x), \quad (27)$$

where c_i are the coefficients to be selected. Equation (22) reveals key challenges in developing a feedforward strategy to reject mismatched disturbances. Different from the matched disturbances, the compensation gain generally depends on the feedback control strategy designed in step 1 and the corresponding gain. This is also true for nonlinear cases, and the described result has been extended to other control strategies, such as slide-mode control. However, more research is still required to develop feedforward strategies for other types of feedback control design.

Conclusion

In this article, both long-known and recent issues in motion control were summarized and discussed. All types of motion control, such as position, force, and impedance control, rely on the same principles of motion stiffness. To make motion stiffness easily manageable for application-specific

tasks, a motion-control design should have an inner acceleration-control structure, ensuring that various disturbances and uncertainties are sufficiently compensated for, ideally providing the residual motion dynamics as a system with double integrators.

Despite considerable progress in more advanced and economical sensor technologies, it is often the inaccessibility of several process variables and cost factors of the hardware that require stable and robust observation and estimation techniques to be developed and deployed as soft (i.e., virtual) sensors. Our discussion in this regard was presented in two parts. First, we described both main sources of internal motion disturbances in the system: structures with associated vibrations and contact interfaces with associated friction. Then, some advanced observation techniques useful for motion systems were explained, together with control and compensation strategies. This overview may contribute to a better understanding of the research problems in motion control, which opens new opportunities for advanced mechatronic products and applications.

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