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Hierarchical Disturbance/Uncertainty Estimation and Attenuation for Integrated Modeling and Motion Control: Overview and Perspectives

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***Abstract*—Advanced motion control with higher precision and faster dynamic response is emerging as an enabling technique for higher performance mechatronic systems. In this article, a systematic hierarchical disturbance/uncertainty estimation and attenuation (HDUEA) motion control framework is presented for cutting-edge and higher-precision mechatronic systems under various working conditions to counteract the most deleterious factors, including a wide operating range, nonlinear characteristics, unmatched/multiple disturbances, and uncertainties. This article will review the state-of-the-art of basic components for modeling, analysis, and motion control of mechatronic systems. Built on this, the promising framework of HDUEA consisting of three layers is elaborated. The first layer is on feature analysis and classification of nonlinearities, disturbances, and uncertainties. The second layer focuses on modeling and identification of modelable nonlinearities, observation, and reconstruction of estimatable disturbance/uncertainties. In the last layer, a composite hierarchical control strategy includes nonlinearity cancellation-based nonlinear feedback, feedforward disturbance compensation, and robust feedback suppression of unmodelable/unobservable nonlinearities, disturbances and uncertainties. Based on this HDUEA framework, the challenges faced by motion control systems are resolved comprehensively. Finally, the article ends with a discussion of open problems and perspectives in this research area.**

***Index Terms*—Attenuation, compensation, disturbance-/uncertainty, estimation, modeling, motion control, nonlinearity.**

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I. Introduction

ADVANCED motion control with higher precision and faster dynamic response is emerging as a significant enabling technique for ever-increasing high-performance mechatronic systems [1], [2], [3], [4], [5], [6], [7]. Subsequently, advanced motion control technique has been extensively used in a variety of higher performance mechatronics systems, including micro-/nano-scale motion systems (such as data storage devices, scanning electron microscopes, semiconductor manufacturing tools, and minimally invasive surgical robots) [2], [3]. The key performance specifications of motion control systems in these applications mainly include satisfactory *tracking bandwidth* and *tracking precision* [8], which are evident in high-end servo products.

However, motion performance is generally severely affected by various nonlinearities, external disturbances, and system uncertainties [1], [9], [10]. Toward this end, robustness against system uncertainties and nonlinearities together with disturbance rejection performance are crucial for motion performance improvement. These factors pose huge challenges for academics and industrial practitioners in advanced mechatronic system control design. How to obtain fast and satisfactory dynamic and static performance, superior disturbance rejection ability, strong adaptability to operation condition variation, and control parameter optimization/self-tuning ability has become key academic and technique problems in motion control theory and engineering [9], [11]. In conclusion, to obtain satisfactory performances for higher-precision mechatronic systems, there are mainly three control problems to be solved, which are briefly introduced as follows.

1) *Wide Operation Range:* With the increasing production efficiency and quality requirements, many modern mechatronic systems are demanded to work in a wide domain of operation, in particular, the performance in small tracking error operation becomes much more common and important than before. For example, in the idle condition of automobile engines, the actual speed fluctuates near the target idle value [78]. Another example is that

when a robot performs continuous contour machining, the vast majority of working conditions belong to small error position tracking control. However, it is quite difficult for traditional controllers based on smooth state feedback to balance the performances of both the large and small tracking error working conditions.

2) *Nonlinear Characteristics:* To be specific, there are various nonlinear characteristics in mechatronic systems, including friction, backlash, hysteresis, saturation, hydraulic nonlinearity, spring nonlinearity, torque nonlinearity, etc [12]. Under the traditional motion control framework, most of these nonlinearities are dealt with by default robustness of the controller without directly and fully exploiting the key features and information of these nonlinear characteristics, which can cause considerable degradation in motion control performance. This fact will be verified later in this article (see *Example 1*).

3) *Multiple Disturbances/Uncertainties:* In addition to nonlinear characteristics, the control performance of mechatronic systems also suffers a great deal of multiple disturbances and uncertainties, including parameter uncertainty and structural uncertainty in the model, drift, load disturbance, environmental disturbance, etc [9], [10], [13], [14], [15], [16]. From the point of view of signal characteristics, these disturbances can be characterized as variables with bounded derivatives, harmonic, step, slope or higher-order variables with respect to time, uncertain norm bounded variables, neutral stable system output, and other types of disturbance models. At the same time, the multisource disturbance in these systems has matching and mismatching characteristics. The complexity and diversity of multiple disturbances and uncertainties and the relentless pursuit of high-quality motion control performance make the traditional control method that accounts for a single disturbance inadequate for the high-precision control specification. This will be demonstrated later in *Examples 2 and 3*.

In what follows, the basic philosophy and limitations of the widely applied motion control approaches, including proportional-integral-derivative (PID) control and robust control (RC), are presented through a specific example. It should be highlighted that most other control approaches in motion control systems exploit similar principles to attenuate disturbances and uncertainties [9]. Consider the tracking error dynamics of a typical second-order mechatronic systems, given by

$$\begin{aligned}\dot{e}_1(t) &= e_2(t)\\ \dot{e}_2(t) &= u(t) + f(e(t)) + d_{\text{ex}}(t) + d_{\text{un}}(e(t), t)\end{aligned} \tag{1}$$

where $e_1(t)$ is the tracking error, $e_2(t)$ is the derivative the tracking error, and $e(t) = [e_1(t), e_2(t)]^T$. $f(e(t))$ is generally a state-dependent nonlinear function (which might be discontinuous) that represents the impacts of nonlinear characteristics. $d_{\text{ex}}(t)$ denotes the external disturbance, and $d_{\text{un}}(e(t), t)$ is again a state-dependent function that represents the uncertainty of the model. In traditional motion controllers, it is common to lump nonlinearities $f(e(t))$, external disturbance $d_{\text{ex}}(t)$, and model uncertainties $d_{\text{un}}(e(t), t)$ together, formulating a lumped disturbance as $d_l(t) = f(e(t)) + d_{\text{ex}}(t) + d_{\text{un}}(e(t), t)$. Toward this end, a PID controller is usually designed as

$$u_{\text{pid}}(t) = k_p e_1 + k_d e_2 + k_i \int_0^t e_1(\tau) d\tau. \tag{2}$$

The robustness principle underlying the PID controller is that the integral action works like an adaptive mechanism to search, approximate, and counteract the lumped disturbance [17], which indicates that $k_i \int_0^t e_1(\tau) d\tau$ approaches $d_l(t)$ when t tends to infinity. Clearly, the PID controller can only asymptotically compensate for the effects of the lumped disturbance with a constant steady-state value rather than time-varying values, which may be caused by the continuous movement or complex disturbed situation of a mechatronic system.

On the other hand, RC has also been extensively used for the motion control of mechatronic systems. For the above second-order system, a standard robust controller is designed as

$$u_{\text{rc}}(t) = k_1 e_1 + k_2 e_2. \tag{3}$$

Taking the lumped disturbance as input and the tracking error as output, it is possible to describe the input–output relationship by a transfer function $T_{\text{ed}}(s)$, which is in terms of feedback control gains k_1 and k_2. RC is quite attractive due to its performance-orientated design and optimization property in the sense that the design goal is to solve k_1 and k_2 from some optimal or suboptimal problems, e.g., searching k_1 and k_2 such that $\|T_{\text{ed}}(s)\|_{2/\infty} < \gamma$, where γ is a performance metric [19]. Note that in general, the higher the control gains k_1 and k_2, the smaller the norm value of $\|T_{\text{ed}}(s)\|_{2/\infty}$. This indicates that the robustness of an RC system is usually achieved by high-gain control. This inevitably brings about conservative performance or high control energy. The key idea of both PID and RC approaches is to utilize feedback regulation to suppress the impact of disturbances and uncertainties, which are difficult to balance in both large (transition operation) and small (static operation) tracking error working conditions.

In summary, the main issues for traditional controllers in achieving fast and high-precision motion control performance are listed as follows.

1) *Dynamics Modeling:* Existing control approaches cannot make full use of the rich prior information/knowledge of the controlled plant, including the model of nonlinear characteristics, disturbances of the external environment, and uncertainties of the system model.
2) *Strategy Analysis:* When making decisions of control strategies, it lacks systematic and comprehensive analysis on the originality (from the plant, the sensor, or the actuator), signal characteristics, uncertainty quantification, error propagation mechanism, and action channels (matched or unmatched), etc.
3) *Controller Design:* There lacks of dedicated motion controllers which are tailored to balance operation conditions of both large and small tracking errors, and to compensate or attenuate nonlinearities, disturbances and uncertainties in a complete and unified framework.

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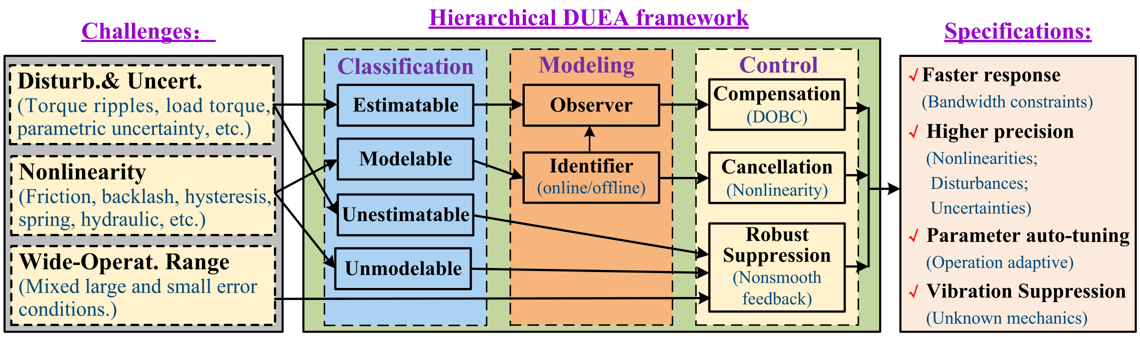

Fig. 1. Philosophy diagram of the HDUEA motion control framework.

In this article, a comprehensive *overview* and *perspectives* on advances in motion control systems in terms of disturbance/uncertainty estimation and attenuation (DUEA) is provided. It is well known that considerable harmful factors, including a wide operating range, nonlinear characteristics, and multiple disturbances/uncertainties, seriously reduce the precision and control bandwidth of motion control systems. However, to the best of our knowledge, most published articles focus on dealing with part of rather than all of these impacts throughout the full closed-loop motion control system. Existing approaches cannot satisfy a higher performance requirement when the control system suffers from more than one (or all) of these issues. In particular, there is no comprehensive framework for integrated modeling, estimation, and motion control that provides in-depth design and analysis tools and improves motion performance at a systematic level. Consequently, in this article, we advocate a hierarchical DUEA (HDUEA) control framework that can give insights to readers with new concepts, guidelines, design, and analysis tools for higher-performance motion control systems. The nature of this study is between a survey and a perspective article.

II. HDUEA Framework

In this article, to tackle the three key factors that include a wide operating range, nonlinear characteristics, and unmatched/multiple disturbances and uncertainties that affect high precision motion control of mechatronic systems, a systematic composite HDUEA motion control framework is developed that consists of nonlinear characteristic modeling, state/disturbance estimation, nonlinearity cancellation, disturbance compensation, and advanced feedback control with some fundamental theoretic developments including estimation and compensation of unmatched disturbance and multiple disturbances. The proposed solutions contribute to improving the dynamic and static performances of mechatronic systems against disturbances and uncertainties, and to provide mechatronic systems with more features, including parameter auto-tuning with adaptation to different operation conditions, resonance suppression, etc.

Basic Principle of HDUEA Framework: The basic idea of the proposed composite HDUEA motion control framework is shown in Fig. 1. First, in addition to classical dynamic modeling techniques for a linear time/frequency domain system, modelable nonlinear characteristics including friction, backlash, hysteresis, etc., are accordingly identified and embedded (depending on the analysis of the working conditions and the performance specifications of the motion) in the model of the mechatronic system. Moreover, another influential factor, estimatable uncertainties/disturbances, is analyzed and classified with their unique features, and estimated (by corresponding observers chosen based on signal characteristic analysis) and embedded into the mechatronic system model as well. Second, as shown in Fig. 1, after classification analysis, identification and observation, in the control design stage, three different control actions can be integrated to form a composite HDUEA strategy, i.e., using feedback linearization to cancel modelable nonlinearities, using disturbance feedforward compensation to reject observable disturbances, and using advanced feedback control to suppress unmodelable nonlinearities and unobservable disturbances. In most applications, the reference command planning and feedforward techniques are generally required to realize fast and smooth transition, precise tracking, and vibration suppression, which are omitted here by taking them as a default control design step. Under the proposed framework, it is especially important to re-establish the stability since the well-established separation principle is inapplicable for general nonlinear systems, and to give system performance analysis such as robustness analysis on how much uncertainties can be tolerated under the framework of the disturbance estimation based control strategy. Hence, different nonlinear control tools must be delicately employed and combined to prove the closed-loop system stability, such as constructing new Lyapunov functions, introduction of input-to-state stability and other nonlinear control tools, etc. On the other hand, the disturbance observer (DOB) design is quite different due to the presence of modelable nonlinearities as well as multiple features of disturbances. The proposed HDUEA motion control structure is given in Fig. 2. In what follows, the key elements for the proposed HDUEA motion control framework will be presented.

III. Modeling, Identification, and Cancellation of Nonlinear Characteristics

In addition to dynamic modeling of mechatronic systems, nonlinearities including friction, backlash, hysteresis, spring nonlinearity, etc., can be modeled and identified accordingly

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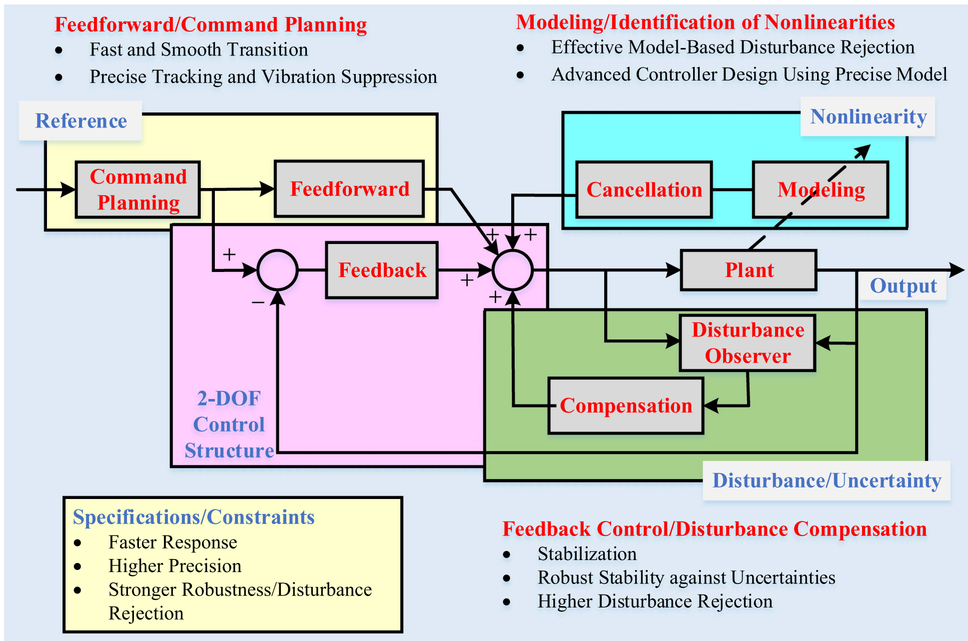

Fig. 2. HDUEA motion control structure.

and then embedded into the mechatronic system model in a more general way for controller design. These nonlinearities appear on different occasions and have different influences on the motion control performance of mechatronic systems [18].

To facilitate the presentation of the subsequent parts, a benchmark with a third-order dynamics of a generic nonlinear mechatronic systems is considered and given by

$$\begin{cases} \dot{\xi}_1 = \xi_2 \\ \dot{\xi}_2 = b\xi_3 + f_1(\xi_1, \xi_2) + d_1 \\ \dot{\xi}_3 = g(\xi_1, \xi_2, \xi_3)\, u + f_2(\xi_1, \xi_2, \xi_3) + d_2 \end{cases} \tag{4}$$

where ξ_1, ξ_2, and ξ_3 are system states, d_1 and d_2 denote the unmatched and matched external disturbances, b is a parameter known or unknown, $f_1(\xi_1, \xi_2)$, $f_2(\xi_1, \xi_2, \xi_3)$ and $g(\xi_1, \xi_2, \xi_3)$ are linear or nonlinear, continuous or discontinuous functions (the nonlinear characteristics might be discontinuous), representing system dynamics known or unknown. It should be noted that most mechatronic systems can be described by (4). For example, in electromechanical systems, the motor rotation angle θ, the speed ω, and the current on the q axis i_q can be taken as the quantities of three states ($\xi_1 = \theta, \xi_2 = \omega, \xi_3 = i_q$). The nonlinearities of the backlash and friction can be characterized by $f_1(\xi_1, \xi_2)$. Taking the piezoelectric positioning stages as another example, the output displacement x, velocity v and total charge q in the piezoelectric ceramic actuator can be taken as the three state quantities ($\xi_1 = x, \xi_2 = v, \xi_3 = q$). The hysteresis nonlinearity exists in $f_2(\xi_1, \xi_2, \xi_3)$, b and g of the general model are two constants in the above two systems. However, the input gain g can sometimes be a function in terms of the states and inputs of the system; e.g., in electrohydraulic servo-control systems, it depends on the physical characteristics of the flow rate in the hydraulic system [20]. The dynamic description and key features of some typical nonlinear characteristics are presented as follows.

A. Modeling of Nonlinear Characteristics

1) Friction Nonlinearity: Friction is a kind of complicated nonlinear physical phenomenon, which exists widely in various mechatronic systems, such as a robotic manipulator, machine tool, hydraulic actuator, and harmonic driver [21]. For precision motion control systems, the existence of friction causes stick slip motion, limit cycle oscillation, tracking error, and other adverse factors [22]. So far, there exist dozens of mathematical friction models, which can be divided into static and dynamic friction models.

The static friction model is simple in structure and intuitive for parameter identification, but it cannot describe the dynamic behaviours of friction. The most commonly used static friction model is the Stribeck model, which describes the relationship between torque and relative velocity between two contacting objects. The mathematical expression of Stribeck friction model is described by [23]

$$F_f = \left(F_c + (F_s - F_c)\, e^{-(v/v_s)^2}\right) \mathrm{sgn}(v) + Bv \tag{5}$$

where F_s is the static friction, F_c is Coulomb friction, v is relative velocity, v_s is the Stribeck velocity, and B is the viscous friction coefficient. F_s, F_c, v_s, B are usually the parameters to be identified.

On the contrary, the dynamic friction model can describe the friction characteristics comprehensively, but is complex in structure and difficult to identify parameters. A generic typical dynamic friction model is the LuGre model, where the total friction force is the summation of the elastic friction produced by the contact surface. It describes multiple frictions like Coulomb friction, variable static friction, viscous friction, Stribeck friction, friction lag, etc. Consequently, it is complete and adequate to reflect the real friction phenomenon. The LuGre friction model is usually expressed as [23]

$$\begin{cases} F_f &= \sigma_0 z + \sigma_1 \frac{dz}{dt} + Bv \\ \frac{dz}{dt} &= v - \frac{|v|}{g(v)} z \\ \sigma_0 g(v) &= F_c + (F_s - F_c)\, e^{-(v/v_s)^2} \end{cases} \tag{6}$$

where σ_0 is the bristle stiffness coefficient, σ_1 represents the bristle damping coefficient, B denotes the viscous friction coefficient, and z is the average deformation of the bristle. B, F_c, F_s, v_s are the static parameters to identify and σ_0, σ_1 are the dynamic parameters to identify.

2) Backlash Nonlinearity: In a mechatronic system with transmission mechanisms (such as gears, ball screws, and drive shaft, etc.), the backlash is deemed one of the most crucial nonlinearities that affect the dynamic performance and static precision of motion control systems [24], [25]. The limit cycle caused by backlash nonlinearities may even exacerbate the instability of closed-loop control systems [26]. In general, the backlash phenomenon can be described with the classic dead-zone model as [27], [28]

$$\tau(t) = \begin{cases} k[\Delta\theta(t) - \alpha] + c\Delta\dot{\theta}(t), & \Delta\theta(t) > \alpha \\ 0, & |\Delta\theta(t)| < \alpha \\ k[\Delta\theta(t) + \alpha] + c\Delta\theta(t), & \Delta\theta(t) < -\alpha \end{cases} \tag{7}$$

where k is the stiffness coefficient of the transmission component, α represents the backlash gap size, c denotes the damping coefficient of the transmission component, τ is the transmitted torque, and $\Delta\theta$ represents the angular difference between the driving side and the driven side. The dead-zone model here expresses the relationship between the transmission torque and the angular difference. In addition, the dead-zone model can also be used to characterize the dead-zone effects in many mechanical actuators. For example, a servo motor only rotates when the input voltage reaches a certain value [29], and the reconstruction precision of three-phase currents is largely degraded by the dead-zones in space vector PWM [30].

3) Hysteresis Nonlinearity: Hysteresis nonlinearity widely exists in the fields of sensors, piezoelectric ceramics, ferromagnets, semiconductor materials, and smart materials. Its main characteristics are nonsmoothness. Without precise modeling and compensation, hysteresis nonlinearity may lead to undesirable control performance, such as undesirable inaccuracies in positioning and tracking control applications [31], and even instability, such as oscillation and chaos.

The hysteresis characteristics of piezoelectric ceramics refer to the nonlinear relationship between the input of the driving voltage and the output of the displacement. If this relationship can be adequately tackled, it will greatly promote the development of piezoelectric ceramics in the fields of high-precision manufacturing and micromachining precision operation. At present, there are a number of ways to model the nonlinear characteristics of hysteresis, which can be mainly divided into the differential-equation-based models and operator-based models [32] and will be discussed in the following.

The first kind of typical representative is Bouc–Wen hysteresis model, which has the advantage of having fewer parameters to adjust, and can be used to precisely describe the hysteric nonlinear system. The mathematical expression of Bouc–Wen model is given as follows [33]:

$$\dot{u} = \alpha k \dot{v}_{\text{in}} - \beta \left|\dot{v}_{\text{in}}\right| u - \gamma \dot{v}_{\text{in}} |u| \tag{8}$$

where v_{in} is the input voltage, u denotes the output variable of hysteric, k is the gain, α, β, and γ are adjustable parameters to determine the amplitude and shape of hysteresis loop.

The operator-based representative type is Prandtl–Ishlinskii (PI) model, which has the following expression as [34]

$$u(v_{\text{in}}) = P[v_{\text{in}}](t) = p_0 v_{\text{in}}(t) + \int_0^{\Lambda} p(r) F_r[v_{\text{in}}](t) dr \tag{9}$$

where $p(r) \geq 0$ is the density function for the operator, p_0 is a constant, and r is the threshold to construct the basic play operator $F_r[v_{\text{in}}](t)$ given as

$$F_r[v_{\text{in}}](t) = \begin{cases} \max\left(v_{\text{in}}(t) - r, u(t_i)\right), & \text{for } v_{\text{in}}(t) > v_{\text{in}}(t_i) \\ \min\left(v_{\text{in}}(t) + r, u(t_i)\right), & \text{for } v_{\text{in}}(t) < v_{\text{in}}(t_i). \\ u(t_i), & \text{for } v_{\text{in}}(t) = v_{\text{in}}(t_i) \end{cases} \tag{10}$$

Although the PI model is relatively complex as compared with Bouc–Wen model (8) based on differential-equations, the main advantage is that its inverse model has analytical expression, which is very beneficial for control design.

4) Other Nonlinearities: Besides the above mentioned typical nonlinearities, there are also many other common nonlinearities, such as spring, and hydraulic, which have considerable adverse impacts mechatronic systems, as shown in Fig. 3. For example, an electronic throttle valve equipped with a stiff spring introduces a nonlinear spring torque to the control system. Without precise modeling and compensation, it can compromise the precision or even the stability of the system. Hydraulic nonlinearity often appears in hydraulic fluid power systems, which is expressed as a nonlinear relationship between liquid flow rate and pressure. In a fuel quantity actuator, the complex hysteresis characteristics of sensors and actuators lead to nonlinearity and multivalue mapping between output and input variables.

These nonlinearities will be characterized and deemed as modelable or unmodelable nonlinear characteristics in the proposed HDUEA framework.

B. Identification/Cancellation of Nonlinear Characteristics

In order to adequately exploit the rich knowledge and key features of the nonlinear characteristics in mechatronic systems, various system identification methods have been proposed. From the perspective of information exploration, these methods can be classified into two groups. One is a date-driven paradigm to identify the model of the full nonlinear characteristics including model structure and parameters; that is, it focuses on how to get as accurate as possible a mathematical model of the system with pure available measurements. The other is a model-based paradigm that aims to identify model parameters with given model structure.

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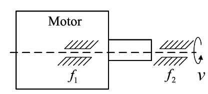
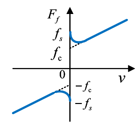
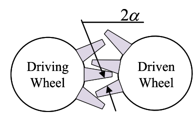
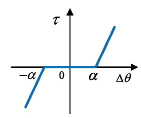
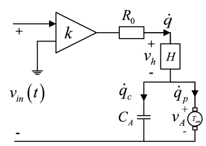
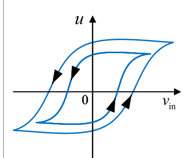
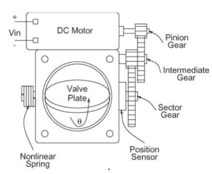
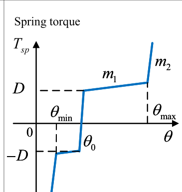
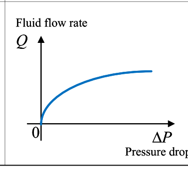
Nonlinearity	Physical characterization	Principle curve	Mathematical expression
Friction			Stribeck model : $F_f(v) = (f_c + (f_s - f_c)e^{-(v/v_s)^2})\mathrm{sgn}(v) + Bv$
Backlash			Dead-zone model : $\tau(\Delta\theta) = \begin{cases} k(\Delta\theta(t) - \alpha) & \Delta\theta(t) > \alpha \\ 0 & \lvert\Delta\theta(t)\rvert < \alpha \\ k(\Delta\theta(t) + \alpha) & \Delta\theta(t) < -\alpha \end{cases}$
Hysteresis			Prandtl-Ishlinskii model : $u(v_{in}) = P[v_{in}](t) = p_0 v_{in}(t) + \int_0^\Lambda p(r)F_r[v_{in}](t)dr$
Spring	 Electronic throttle valve		Signum function model : $T_{sp}(\theta) = \begin{cases} D + m_1(\theta - \theta_0), & \theta_0 < \theta < \theta_{max} \\ -D - m_1(\theta_0 - \theta), & \theta_{min} < \theta < \theta_0 \end{cases} = m_1(\theta - \theta_0) + D\,\mathrm{sgn}(\theta - \theta_0)$
Hydraulic			Orifice flow equation model : $Q(\Delta P) = C_d A\sqrt{\frac{2\Delta P}{\rho}}$

Fig. 3. Characteristics of typical nonlinearities in mechatronic systems.

Consequently, identification tools are broadly divided into two categories including traditional identification methods and intelligent identification methods. The traditional methods focus on utilizing traditional modeling tools in control systems society such as impulse response [35], least square method [36], [37], maximum likelihood method [38], etc., for nonlinear characteristics modeling. These traditional methods have some common defects. For example, the input signal is generally required to be known, the system structure and parameters cannot be determined at the same time, and it is easy to fall into the local optimal solution and miss the global optimal one for the nonconvex optimization problem. On the other hand, as complementary to traditional methods, intelligent identification methods with a data-driven property including neural network [39], [40], genetic algorithm [41], [42], [43], fuzzy logic [44], particle swarm optimization [45], [46], [47], [48], etc., have been extensively investigated for nonlinear characteristics identification, and applied to model identification in induction motors, linear motors, permanent magnet synchronous motors (PMSM), robotic systems, excitation systems, magnetorheological fluid dampers, hydraulic turbine governing systems, flight control systems, etc. It is worth mentioning that, Liu et al. [40] proposed a novel B-spline wavelet neural network identification method, which solves the challenging problem of balancing the precision of identification and computational complexity. Besides, depending on the specific situation of the real plant, the identify modes can be basically divided into offline or online. Although the latter has better adaptability, it also puts higher requirements for hardware computation and parameter separability. The advantages and disadvantages of the prevalent nonlinearity identification methods for mechatronic systems are shown in Table I.

Remark 1: The precise modeling and identification of nonlinearities are essential for favourable motion control performance without resorting to high gain means, because the indifferentiable properties of many nonlinearities make them difficult to be effectively attenuated [49]. Meanwhile, it should be noted that even for these modelable nonlinearities, modeling errors due to model deviations and identification errors are inevitable, which can be regulated online by adaptive methods before using robust techniques [50].

IV. Estimation/Compensation of Multiple/Unmatched Disturbances and Uncertainties

A. Refined Estimation/Compensation of Multiple Disturbances and Uncertainties

Higher precision motion control of mechatronic systems is critical but quite challenging due to the complicated and intrinsically nonlinear mechatronic dynamics, complex uncertainties, and disturbances. These disturbances and uncertainties are originated from a variety of sources including the mutations of driven load, working environments, and the electrical/mechanical parts inside the motor bodies. Specifically, the nonlinear components

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TABLE I
ADVANTAGES AND DISADVANTAGES OF NONLINEARITY IDENTIFICATION METHODS FOR MECHATRONIC SYSTEMS

Capability	Traditional Methods			Intelligent Methods			
	Impulse Response	Least Square Method	Maximum Likelihood Method	Neural Network	Genetic Algorithm	Fuzzy Logic	Particle Swarm Optimization
Noise Sensitivity	High	Moderate	Moderate	Low	Low	Low	Low
Algorithm Complexity	Low	Low	Moderate	High	High	High	Moderate
Model Adaptivity	Low	Low	Low	High	High	High	Low
Computation Burden	Low	Low	Moderate	Large	Large	Large	Low
Nonlinear Dynamics	×	×	×	√	√	√	√

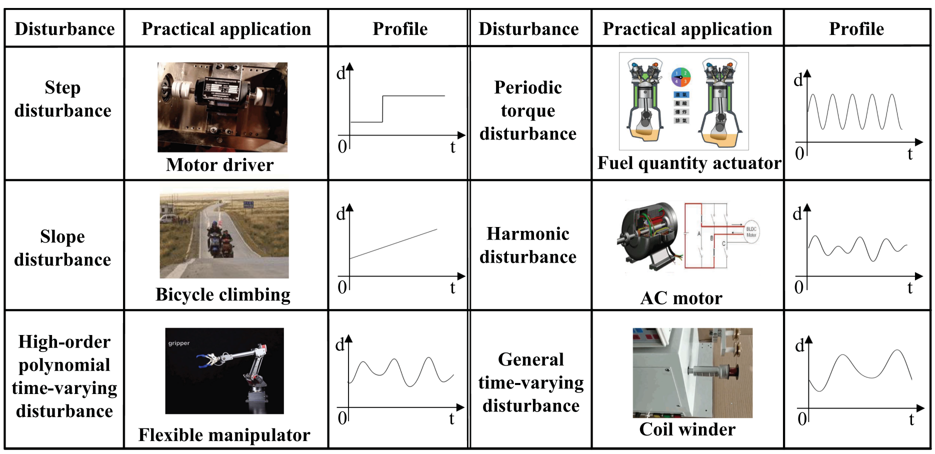

Fig. 4. Profiles of typical disturbances in mechatronic systems.

which are not included in the nominal model of the mechatronic system, can also be treated as part of "lumped" disturbances. Thereby, one particularly important task of controller design for mechatronic systems is to improve robustness against such "lumped" disturbances.

To this end, disturbance/uncertainty estimators can be used to estimate the composite impacts of this lumped disturbances including the nonlinearities, and the compensation action can be implemented based upon the estimation. This is actually the motivation behind classic disturbance observer-based control (DOBC) and active disturbance-rejection control (ADRC) techniques, where an integrator chain (linear or nonlinear) dynamic system is considered as the nominal model while all the ignored system dynamics are estimated by DOB, extended state observer (ESO) or other kinds of disturbance estimators [51], [52], [53]. However, most of these conventional disturbance estimation and compensation methods can only deal with a single kind of disturbance effectively, e.g., linear DOB for constant or slow-changing disturbances, harmonic DOB for periodic disturbances, etc. Thus, such schemes can hardly achieve superior performance in some working conditions.

First, in practice, a number of nonlinearities, such as friction, backlash, spring, and hydraulic nonlinearities may change with system states which track their desired references dynamically. When the working condition is complex and the states change globally, these nonlinear terms will not satisfy the convergence conditions (i.e., boundedness of nonlinearities) of these disturbance estimators. Second, the model uncertainties caused by the system parameter perturbations, such as the large-scale variations of load inertia, are also changing dramatically with the state during the dynamic process. Third, the disturbances faced by mechatronic systems generally show diverse features, and the profiles of some typical disturbances are shown in Fig. 4. Specifically, the actual disturbance may be a superposition of several different forms, which can hardly be estimated with the desired precision by baseline DOBs.

In conclusion, while it may be straightforward to lump all nonlinearities, uncertainties, and disturbances together for estimation and compensation, but the design of DOBC and related methods based on such a linear nominal model is probably not able to guarantee satisfactory control performance and even stability. As a typical example, the promising application of the gantry positioning system in [54] has shown that lumping all disturbances together exhibits poor control precision. To be more intuitive, we consider the following practical examples.

Taking the PMSM drive system as an example, there surely exist multiple types of disturbances and uncertainties caused by measurement errors and dead-time effects, load and cogging

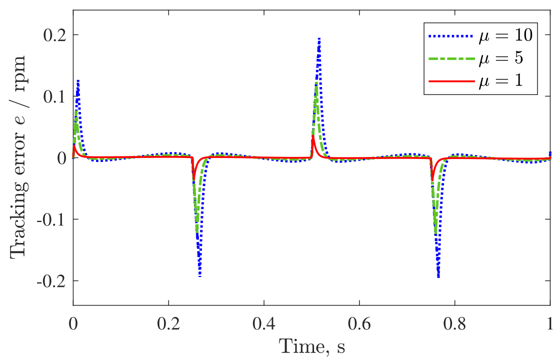

Fig. 5. Capability of DOBC for handling friction nonlinearity under different operation conditions (See *Example 1*).

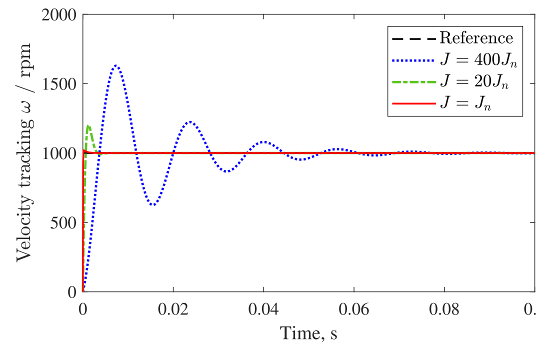

Fig. 6. Capability of DOBC for handling model uncertainties under different perturbed conditions (See *Example 2*).

torques, and parameter perturbations [9]. In addition, transmission friction is also a major nonlinearity that affects its motion control performance.

Example 1: Consider a first-order PMSM speed regulation system subject to friction nonlinearity and load disturbance as

$$\dot{e} = bi_q - \mu F_f(\omega) - T_L(t) - \dot{\omega}_{\text{ref}} \tag{11}$$

where ω, i_q are the system velocity state and control input, respectively; $F_f(\omega) = 10^{-3}[(36 + 20e^{-(\omega/20)^2})\,\text{sgn}(\omega) + 13\omega]\text{N}\cdot\text{m}$ is the nonlinear friction torque in terms of system velocity, and μ is an uncertain coefficient; Besides, $T_L(t) = 0.1\sin(\pi t)\ \text{N}\cdot\text{m}$ is the smooth disturbance here. For this system, an DOBC law is designed with a linear DOB as

$$\begin{aligned} i_q^* &= -k(\omega - \omega_{ref}) - \hat{F}_f/b \\ \dot{z} &= -l(z + le + i_q^*),\ \hat{F}_f = z + le \end{aligned} \tag{12}$$

where z is the observer state, k and l are the controller and observer gains to be designed. In this example, we set $k = 18$ and $l = 160$, and the different friction torques with $\mu = 1, 3, 5$ are tested here. The response curves of the output velocity under different operation conditions are provided in Fig. 5 . Although the smooth disturbance $T_L(t)$ can be well attenuated by DOB, but the tracking precision degrades severely when the velocity ω crosses zero. This is mainly due to the complex nonlinear characteristics of friction at zero crossing, which is difficult to accurately estimate by a linear DOB.

Example 2: Likewise, we still consider the PMSM speed regulation system with controller (11), which is subjected to severe model uncertainties (inertia perturbations) as

$$\dot{e} = b_0 i_q + (b - b_0)i_q - \dot{\omega}_{ref} \tag{13}$$

where $b = K_t/J$ containing the perturbed inertia as compared with nominal $b_0 = K_t/J_n$. Since the equivalent disturbance $d = (b - b_0)i_q$ is relevant with the control input, it changes dramatically during the dynamic processes. The control parameters here are selected as $k = 18$ and $l = 1200$.

The output velocity curves under DOBC scheme when faced by different degrees of inertia perturbations are provided in Fig. 6. Clearly, the control accuracy is severely degraded by the uncertainties of the model. What is worse, the stability margin of the closed-loop system has decreased significantly along with the increase of inertia perturbation.

Consequently, the disturbance rejection ability and robustness of the closed-loop system under DUEA can be significantly increased if the (known) nonlinear and other dynamics could be fully exploited in control design [55]. This motivates the development of nonlinear DUEA approaches for general nonlinear systems [56], [57].

Among many advanced robust and/or adaptive control strategies, the DUEA approach shows great potential and benefit in compensating for the influences of model uncertainties and disturbances in motion control systems [58], [59], [60]. A notable feature of the DUEA is that the robustness of the closed-loop system is achieved while retaining its nominal control performance. Another important merit of DUEA is that it could entirely compensate for the effects of nonvanishing disturbances within the system if they can be precisely estimated. Nevertheless, it is generally not possible to attenuate all these disturbances/uncertainties in the entire control loop by using the above control methods because of the intricate distribution and the complex features. Most of the servo-control methods merely deliver partial attenuation tools for disturbances/uncertainties generated in a certain single kind of disturbances instead of multiple sources of disturbances.

For this part, we aim to analyze and estimate the multiple observable uncertainty/disturbance including small parametric uncertainty, and external disturbance, etc. Then, according to the characteristics of the disturbance signal, e.g., constant disturbance, slope disturbance, acceleration disturbance, harmonic disturbance, composite time-varying disturbance, etc., different DOBs are adequately developed and embedded in the HDUEA motion control framework as shown in Fig. 2. To conclude, in this new HDUEA framework, multiple disturbances are modeled, estimated, and compensated by constructing different kinds of composite DOBs and compensation control terms accordingly, which helps to significantly enhance the disturbance rejection ability. The basic idea underlying refine estimation/compensation of multiple disturbances and uncertainties is demonstrated via the following example.

Example 3: For simplicity and intuitiveness of the analysis, the PMSM speed-loop system suffered from multiple disturbance and uncertainty with distinctive forms from internal or external sources, $d(t)$ is considered here as

$$\dot{x}_1 = b_0 u + d(t)$$
$$d(t) = d_{\mathrm{tv}}(t) + d_{\mathrm{h}}(t) \tag{14}$$

where $d_{\mathrm{tv}}(t)$ denotes a generic time-varying disturbance that includes parametric uncertainties, unmodeled dynamics, etc [10]. It can be characterized by a time series polynomial, which is represented as

$$d_{\mathrm{tv}}(t) = a_0 + a_1 t + a_2 t^2 + \Delta(t) \tag{15}$$

with a_0, a_1, and a_2 are unknown constant parameters, and $\Delta(t)$ is the residual unmodeled term, which is assumed to be third order differentiable. It is worth noting that, as a special case of (14), $d(t) = a_0 + \Delta(t)$ or $d(t) = \Delta(t)$ is widely adopted to represent the lumped load torque in motion control systems [10], [61] or the gust wind disturbance in flight control systems [62], [63], which is extensively investigated in the conventional disturbance rejection methods.

Then, $d_{\mathrm{h}}(t)$ represents the harmonic-type disturbance with unknown magnitude b and phase c, including cogging torque, flux harmonics, etc [10]. It can be described by

$$d_{\mathrm{h}}(t) = b\sin(wt + c) \tag{16}$$

with w is the frequency that needs to be acquired, which is extremely important for the harmonic disturbance-rejection.

To achieve an observer-oriented model, two types of disturbances are extended as the new system states as

$$x_2 = d_{\mathrm{tv}}(t),\ x_3 = \dot{d}_{\mathrm{tv}}(t),\ x_4 = \ddot{d}_{\mathrm{tv}}(t)$$
$$x_5 = d_{\mathrm{h}}(t),\ x_6 = \dot{d}_{\mathrm{h}}(t).$$

On this basis of original model (15) and the extended states, the following state-extended model can be derived as

$$\begin{aligned}
\dot{x}_1 &= x_2 + x_3 + b_0 u \\
\dot{x}_2 &= x_3, \dot{x}_3 = x_4, \dot{x}_4 = \Delta^{(3)}(t) \\
\dot{x}_5 &= x_6, \dot{x}_6 = -\omega^2 x_5 \\
y &= x_1.
\end{aligned} \tag{17}$$

By regarding the high-order item $\Delta^{(3)}(t)$ as a residual perturbation, a modified Luenberger observer can be straightforwardly designed for system (17) as follows:

$$\begin{aligned}
\dot{\hat{x}}_1 &= \hat{x}_2 + \hat{x}_5 + b_0 u + \lambda_1(\hat{y} - y) \\
\dot{\hat{x}}_2 &= \hat{x}_3 + \lambda_2(\hat{y} - y) \\
\dot{\hat{x}}_3 &= \hat{x}_4 + \lambda_3(\hat{y} - y) \\
\dot{\hat{x}}_4 &= \lambda_4(\hat{y} - y) \\
\dot{\hat{x}}_5 &= \hat{x}_6 + \lambda_5(\hat{y} - y) \\
\dot{\hat{x}}_6 &= -\omega^2 \hat{x}_5 + \lambda_6(\hat{y} - y)
\end{aligned} \tag{18}$$

where λ_i, $i = 1, \ldots, 6$ are the observer gains to be designed. Based on the disturbance estimation $\hat{d} = \hat{x}_2 + \hat{x}_5$, the unknown disturbance can be adequately estimated and the estimation error

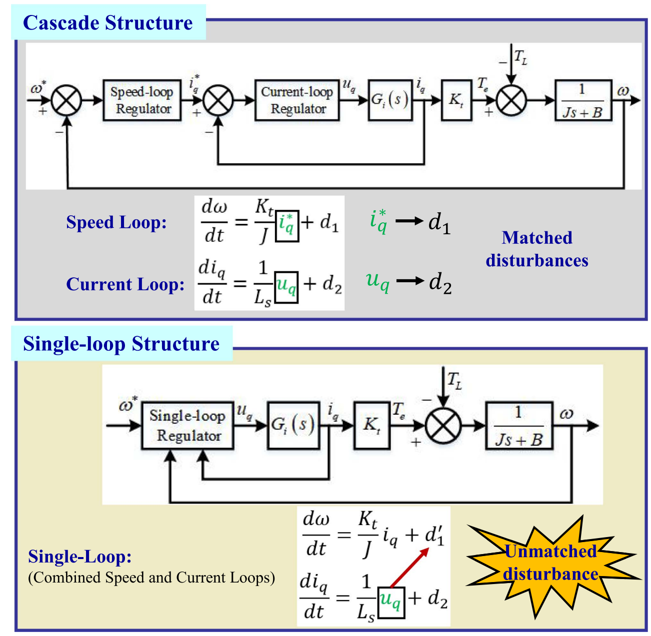

Fig. 7. Illustration of unmatched disturbance in single-loop speed regulation of PMSM servo systems.

is generally affected by the perturbation $\Delta^{(3)}(t)$ and the selection of observer gains λ_i.

Remark 2: For a more general form of disturbance $d(t)$ with nth order time series polynomial, as well as harmonic components with multiple frequencies $\omega_j, j = 1 \ldots, n$, the extended system model (13) and the related observer (14) can be generalized accordingly. Furthermore, the prior nonlinearity obtained by identification can also be embedded into the observer for a refined disturbance estimation [10], [49].

B. Unmatched Disturbance Compensation Control

Despite the excellent properties of DUEA approaches, most of these approaches are merely able to deal with matched disturbances but are quite vulnerable to unmatched ones. Unmatched disturbance, which indicates that the uncertainties/disturbances pose impacts on the system via channels different from the control inputs. Unmatched disturbances exist extensively in many engineering control systems. Taking the single-loop speed regulation problem of PMSMs in Fig. 7 as an example, the load torque disturbance is a typical unmatched disturbance because it acts on the system through the speed equation, which is different from the voltage control input that changes the current directly rather than the speed. There are extensively similar phenomena in mechatronic systems, such as the MAGLEV suspension air-gap control system [64]. The mismatching property between control input and disturbance/uncertainty poses great challenges to active compensation design in the sense that the disturbance/uncertainty appear in different channels from the control input and thus are not able to be compensated directly. In this section, we will introduce three effective unmatched disturbance compensation control approaches.

1) Disturbance Compensation Gain Construction Approach: Without loss of generality, a single-input-single-output (SISO) system with unmatched disturbance/uncertainty is considered as follows:

$$\dot{x} = Ax + b_u u + b_d f(x, d(t), t)$$
$$y_m = C_m x, y_o = c_o x \tag{19}$$

where $x \in \mathbf{R}^n$, $u \in \mathbf{R}$, $d \in \mathbf{R}$, $y_o \in \mathbf{R}$, and $y_m \in \mathbf{R}^r$ are system state, control input, external disturbance, measurable outputs, and controlled output, respectively. To facilitate controller development, we define the lumped disturbance as an extended state, given by [65]

$$x_{n+1} = f(x, d(t), t) \tag{20}$$

then the extended system is derived as

$$\dot{\bar{x}} = \bar{A}\bar{x} + \bar{b}_u u + Eh(t), y_m = \bar{C}_m \bar{x} \tag{21}$$

where $\bar{x} = [x^T, x_{n+1}]^T, h(t) = \frac{df(x,\omega(t),t)}{dt}$, and $\bar{A}$, $\bar{b}_u$, E, and $\bar{C}_m$ are new system matrices which can be derived accordingly.

Remark 3: System (19) gives a general description of a SISO system with unmatched disturbance/uncertainty [65]. It includes the motion control system (4) as a special case. Extension (20) is a common approach to deal with lumped disturbances and uncertainties in ADRC.

Suppose that (A, b_u) is controllable and $(\bar{A}, \bar{C}_m)$ is observable. For system (21), an ESO is constructed as follows:

$$\dot{\hat{\bar{x}}} = \bar{A}\hat{\bar{x}} + \bar{b}_u u + L(y_m - \hat{y}_m)$$
$$\hat{y}_m = \bar{C}_m \hat{\bar{x}} \tag{22}$$

where $\hat{\bar{x}} = [\hat{x}^T, \hat{x}_{n+1}]^T$, $\hat{x}$ and $\hat{x}_{n+1}$ represent the estimates of the state variable $\bar{x}$, x, and x_{n+1} in (21), respectively. L is the observer gain to design.

The *unmatched disturbance compensation control law* is designed as [65]

$$u = K_x x + K_d \hat{d} \tag{23}$$

or

$$u = K_x \hat{x} + K_d \hat{d} \tag{24}$$

where K_x and K_d are feedback control gain and feedforward *disturbance compensation gain* satisfying

$$K_d = -[c_o(A + b_u K_x)^{-1} b_u]^{-1} c_o (A + b_u K_x)^{-1} b_d. \tag{25}$$

Under some mild assumptions about disturbance, it has been proven that the offset due to the unmatched disturbance/uncertainty in (19) can be attenuated from the output channel if the observer gain and the control gain are adequately selected [65]. The approach on construction of disturbance compensation gain has been extended to deal with a class of affine nonlinear systems with unmatched disturbances in [66], [67].

2) Dynamic Sliding Surface Construction Approach: A second-order system with unmatched disturbance is considered here, and given by

$$\dot{x}_1 = x_2 + d(t)$$
$$\dot{x}_2 = a(x) + b(x)u$$
$$y = x_1 \tag{26}$$

where x_1 and x_2 are system states, u is the control input, $d(t)$ is the unmatched disturbance, and y is the controlled output.

A DOB in [68] is first introduced for the estimation of disturbances for (26), given by

$$\dot{p} = -lg_2 p - l[g_2 lx + f(x) + g_1(x)u]$$
$$\hat{d} = p + lx \tag{27}$$

where $\hat{d}$, p, and l are the disturbance estimate, the observer state, and the observer gain, respectively, $f(x) = [x_2, a(x)]^T, g_1(x) = [0, b(x)]^T$, $g_2 = [1, 0]^T$ are vector fields or vectors.

A *dynamic sliding surface* integrating the unmatched disturbance estimation (27) is defined in [69], and given by

$$\sigma = x_2 + cx_1 + \hat{d} \tag{28}$$

where $c > 0$ is a designable control parameter. The resultant dynamic sliding model control law is designed as

$$u = -b^{-1}(x)\left[a(x) + c(x_2 + \hat{d}) + k\text{sgn}(\sigma)\right] \tag{29}$$

with k the switching gain.

It has been shown that if the switching gain and the observer gain are properly selected, the closed-loop system under the control law (29) is then asymptotically or practically stable with some mild assumptions on the unmatched disturbance [69]. In this part, we present the basic idea of dynamic sliding mode control to unmatched disturbance compensation via a simple second-order system. This idea has been further extended for higher-order linear or nonlinear systems, and even by using nonlinear sliding mode control techniques that achieve finite-time stability [70]. A further result on invariant manifold-based output feedback sliding mode control has also been developed for unmatched disturbance rejection in [71].

V. Attenuation: Nonsmooth Feedback Control

A. Motivation

As clearly discussed in the framework of HDUEA, it is impossible for the aforementioned modeling, identification/observation, feedforward cancellation/compensation techniques to completely eliminate all the influences of nonlinearities, disturbances, and uncertainties because some of them might be difficult to characterize/quantify due to their lack of representable features. However, these unmodelable nonlinearities and unestimatable disturbances/uncertainties still cause considerable adverse effects for higher precision motion control of mechatronic systems. On the other hand, with the increasing demand for quality and production efficiency, energy or environmental requirements, etc., many mechatronic systems need to work in "small deviation" working condition which is close to the extreme optimal operation point. For instance, when an automobile engine is idling, it is necessary to keep

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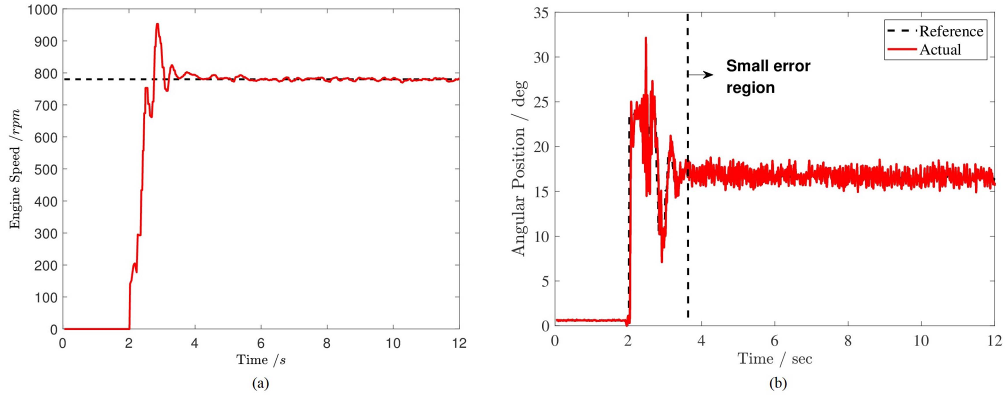

Fig. 8. Schematic diagram of engine idling condition. (a) Engine idle speed curve. (b) Fuel quantity actuator position curve.

it running at a constant low speed without additional load. To this end, the actual engine speed fluctuates around the target idle value. This is a typical speed-tracking control problem with small deviations near the equilibrium point. These can be observed from the idle speed curve of the engine and the actuator position profile of the quantity of fuel in Fig. 8(a) and (b). For traditional smooth feedback control methods, the greater the deviation, the larger the control gains will be. Thus, it is difficult to maintain a good balance between convergence performance and disturbance rejection ability.

In this section, we advocate a nonsmooth feedback control mechanism. As a powerful nonlinear robust feedback control approach, nonsmooth feedback control provides a practical and effective solution to mitigate the undesirable impacts of these unmodelable nonlinearities and unestimatable disturbances/uncertainties. In addition, nonsmooth feedback control exhibits a unique and promising property of "small deviation and equivalent large control gain," which provides the possibility to balance a wide range of operation conditions from small tracking error to large one. For example, in the low-speed section of servo systems, there widely exists multiaxis interpolation motion, which demands high trajectory accuracy, small tracking error, and repeated positioning error. In such cases, linear feedback algorithms like PID control strategy, can hardly keep balance among rapid response, small overshoots, and strong disturbance robustness. The piece-wise linear feedback control, as a commonly used improved method, often requires many adjustment parameters and lacks stability analysis under complex circumstances.

B. Properties

Nonsmooth feedback control has been rapidly developed with advances of the finite-time homogeneous theory [72], [73] and the finite-time Lyapunov stability theory [74], [75]. It is continuous but nondifferentiable with respect to system states [76]. As compared to smooth control, nonsmooth control exhibits faster convergence performance in a region close to the equilibrium point, as well as better robustness and stronger disturbance rejection ability. In what follows, we will explain the advantages with examples.

1) Property 1–Faster Convergence Rate Near the Equilibrium Point: The nonsmooth control is actually a kind of time-optimal control approach, which can force the closed-loop system to converge in finite time. In most existing control methods, the fastest convergence rate of the closed-loop system is asymptotic, indicating that the closed-loop system errors only converge to zero when time goes to infinity.

For example, considering a first-order error system $\dot{e} = u$, where e denotes the tracking error, and u represents the control input. A nonsmooth controller is designed as $u_1 = -k\,\mathrm{sgn}(e)|e|^{\alpha}, 0 < \alpha < 1$ and a smooth controller is designed as $u_2 = -ke$, where $k > 0$ is the control gain. It can be obtained that under the nonsmooth controller u_1, e converges to the equilibrium point in a finite time of $\frac{|e(0)|^{1-\alpha}}{(1-\alpha)k}$, while under the smooth controller u_2, e can only converge to zero exponentially. The finite-time convergence property of nonsmooth feedback control is attribute to the utilization of the fractional power state feedback. When the tracking error is small around the equilibrium point, the nonsmooth controller generally has a relatively larger control gain than the smooth controller, thereby ensuring a faster convergence performance. In practical applications, the small error band can be defined according to the actual requirements of the application, and the tracking error can be normalized to facilitate the design and analysis of the controller.

2) Property 2–Stronger Robustness and Disturbance Rejection Ability: The nonsmooth feedback control technique exhibits stronger disturbance rejection ability in practical applications. To be specific, again we consider a first-order error system $\dot{e} = u + d(t)$ for example, where e represents the tracking error, u denotes the control input, and $d(t)$ is the external disturbance bounded by $|d(t)| \le d^*$, which is supposed to be difficult to deal with by cancellation and compensation. A nonsmooth controller is designed as $u_1 = -k\,\mathrm{sgn}(e)|e|^{\alpha}, 0 < \alpha < 1$ and a smooth controller is designed as $u_2 = -ke$, with $k > 0$ the control gain. With straightforward calculations, the bound of the steady-state tracking error under the nonsmooth controller is governed by $|e(\infty)| \propto (d^*/k)^{\frac{1}{\alpha}}$, while that under the smooth controller is given by $|e(\infty)| \propto d^*/k$. To obtain a better disturbance rejection

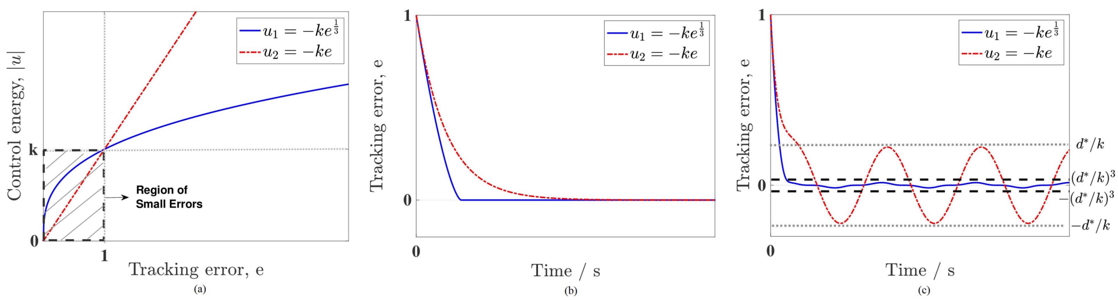

Fig. 9. Performance comparisons between the nonsmooth controller and the smooth controller: (a) Control quantity curve, (b) Error curve without external disturbance, (c) Error curve with external disturbance.

TABLE II
PROPERTY COMPARISONS BETWEEN NON-SMOOTH CONTROL AND OTHERS

Capability	Non-smooth control	Discontinuous control (e.g. SMC)	Continuous control (e.g. PID)
Convergence Rate	Finite-time	Finite-time	Exponential
Implementation Complexity	Moderate	Easy, Chattering	Easy
Anti-disturbance Ability	Good	Strong	Moderate

ability, the control gain k is generally chosen such that $k > d^*$ in practical applications. When using the nonsmooth controller, except for the tuning parameter of the controller gain k, one has an additional design of freedom, that is, the fractional power α, which can be adjusted so that $(d^*/k)^{\frac{1}{\alpha}}$ is much smaller than d^*/k.

The simulation results of the two different controllers are shown in Fig. 9 ($k = 5, \alpha = 1/3$), which demonstrates that in a small error region, the nonsmooth controller has larger control effort than the smooth controller, i.e., the feature of ‘smaller deviation and equivalently larger control gain’. As shown in Fig. 9(b) and (c), when the tracking error is small, the nonsmooth controller can not only achieve faster convergence rate and smaller steady-state error, but also suppress the influence of the unmodelable nonlinearity and unobservable disturbance better. Comparisons of properties between nonsmooth control and other typical control mechanisms, including discontinuous control [e.g. sliding mode control (SMC)] and continuous control (e.g. PID control) are summarized in Table II.

C. Relationship Between Nonsmooth Control, Discontinuous Control, and Continuous Control

From the aspects of control structures, nonsmooth control is actually a kind of control mechanism between discontinuous (e.g., bang–bang) control and continuous (e.g., PD) control, and it offers the dual advantages of strong robustness against disturbances and chattering elimination, supported by adequate computing power. Recall the nonsmooth controller $u_{ns} = -k\,\mathrm{sgn}(e)|e|^{\alpha}, 0 < \alpha < 1$ for a first-order system. When selecting $\alpha = 0$, the control law becomes $u_{dc} = -k\,\mathrm{sgn}(e)$, which clearly forms a discontinuous controller. If we take $\alpha = 1$, the control law then becomes $u_{dc} = -ke$, which turns into a continuous controller.

Remark 4: The traditional RC belongs to the pure feedback suppression method, which is included as an important layer of the proposed HDUEA framework. It should be noted that RC provides a worst-case manner for dealing with disturbances/uncertainties that produces conservativeness for performance improvement. To attenuate multisource disturbances/uncertainties and nonlinearities, the control gains of RC should be assigned particularly high. This inevitably brings about performance conservativeness or high control energy [52]. Compared to RC, HDUEA can be considered a ‘refined’ RC method, which provides a promising approach to trade off between nominal performance and robustness.

VI. OPEN PROBLEMS AND PERSPECTIVES

In spite of significant progress has been made for HDUEA techniques in motion control systems, there are still many open issues to be addressed in the future:

1) *Predictive Control With Disturbance Preview:* Although real-time disturbance can be estimated by DOBs for generalized HDUEA controllers, predictive control methods such as model predictive control and optimal control can further utilize disturbance information in the future for better prediction and performance improvement [79]. For this reason, the HDUEA techniques that further utilize the disturbance preview remain an area ripe for exploration.

2) *Disturbance Rejection With Synchronous Filtering:* As revealed in [80], there exists an inherent contradiction between the antidisturbance ability and the noise sensitivity during the design of DUEA framework. Generally, high observer gains are needed in DOBs for a considerable bandwidth to achieve fast convergence, which can make it sensitive to measurement noise. Therefore, the design and synthesis of HDUEA techniques for motion control systems, simultaneously with noise filtering, remains a hot topic that waits to be researched.

3) *Safety–Critical Disturbance Rejection Control:* To ensure accuracy and safety in practical servo applications, some variables must be precisely constrained within specific ranges [82]. For instance, owing to the randomness of wind speed, large fluctuations in motor speed can result in over-voltage and unstable power transmission on the bus side in wind turbine systems [81]. Furthermore, in the single-loop control structure of a PMSM system, the q-axis current becomes a system state. If the q-axis current is not appropriately constrained by the controller design, excessive current can pose a threat to the circuit safety of PMSMs [79]. Therefore, the design and synthesis of HDUEA techniques with constrained problems, remains an open issue that needs to be addressed.

4) *Disturbance Rejection Control With Time-Delay:* With the development of motion control technology, the visual servoing control, which aims to regulate the motion of a platform by using the visual signal from a camera, has to be applied into many unmanned autonomous systems [83]. However, the relatively long image acquisition and processing times compared to the control period result in a significant time delay, which restricts the bandwidth of the motion control system. To this end, the implementation of HDUEA techniques that further considers time-delay characteristic is another research direction to be explored.

5) *Robust Temporal Logic Motion Control:* Existing motion control systems primarily focus on reference tracking or stabilization. However, achieving a high level of autonomy in robotics and autonomous systems requires new motion control methods to accomplish more complex objectives while maintaining safety conditions [84]. Temporal logics, which effectively characterize complex tasks with strict deadlines, hold promise as an enhancement to the existing HDUEA framework in the future.

VII. Conclusion

Facing an ever-increasing demands on higher-precision motion control of mechatronic systems, an HDUEA motion control framework has been presented in this article. The proposed framework has provided a comprehensive and systematic solution to deal with challenging factors in higher-precision motion control, such as wide operating range, key nonlinear characteristics, unmatched/multiple disturbances and uncertainties. Three layers including analysis and classification, identification and observation, and cancellation/compensation/suppression of nonlinearities, disturbances and uncertainties have been advocated. Some typical application examples have also been conducted to demonstrate the usefulness of the proposed HDUEA framework. The benefits and efficacy of this promising motion control framework has been demonstrated by applying it to some typical mechatronic systems, such as a PMSM in [10] and [77], as well as a fuel quantity actuator in [78]. Finally, some open problems and perspectives concerned with the HDEUA framework design of motion control systems have been discussed in detail.

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