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ACPI Control Theory

Zhezhao ZENG[1,2]
[1](Hunan Automation Control Technology Co., LTD, Hunan, China)
[2](Changsha University of Science and Technology, Hunan, China)

ABSTRACT: *To address the issue of stabilizing each gain in a PI (Proportional-Integral) controller, this paper propo-ses a Zeng's stabilization rule (ZSR) based on a single speed factor and an ACPI (Auto-Coupling PI) control theory method.The core concept of this method is to incorporate the internal dynamics of a known or unknown nonlinear system and an external bounded disturbance into a total disturbance, thereby enabling the mapping of said nonlinear system onto a linear disturbance system. By applying PI control law, we establish a controlled error system that utilizes the total disturbance as excitation. To ensure optimal dynamic characteristics and robust stability of the controlled error system, we have designed PI stabilization rules and an ACPI control law based on a single speed factor, derived from the characteristic equation of the critical damping system. These methods are both practical and elegant, with significant scientific value and broad applications in the field of control engineering.*

KEYWORDS - *ACPI control, Nonlinear system, speed factor, Zeng's stabili-zation rules (ZSR)*

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I. INTRODUCTION

The fundamental principle of Proportional Integral (PI) control is to devise control strategies based on the deviation between the actual output value of controlled objects and the desired value (the control target). With proper regulation of PI gains, a stable closed-loop control system can effectively achieve its intended objective. As such, PI control methodology has been widely adopted [1]-[3]. So far, PI stabilization methods are mainly divided into two categories: classical stabilization methods for linear systems [1]-[3] and modern optimization methods for nonlinear systems [4]-[13]. Both classical stabilization methods and modern optimization techniques have consistently demonstrated the objective reality of poor gains robustness, which PI stabilization has always posed a formidable challenge in the realm of control engineering, significantly constraining the potential control capabilities of PI control theory.

To address the issue of gain stabilization in PI controllers, Chinese scholar Professor Zeng Zhezhao has conducted extensive research [14],[15]. Through the analysis of the physical properties of PI control systems, he discovered that the dimensionless proportional gain makes PI controller can only provide the control force of the **generalized displace-ment** to controlled system, while any first-order system requires the input control force of the **generalized speed**, and any second-order system requires the input control force of the **generalized acceleration**, and so on. It has been demonstrated that there exists a dimensional inconsistency in control force between a PI controller with **dimensionless proportional gain** and any controlled system. Moreover, the independent proportional gain and integral gain of a PI controller ensure that the proportional control force and integral control force operate independently, resulting in an incongruous control mechanism during the process. In order to rectify the deficiencies of PI control theory, it is imperative to scientifically define the dimensional attribute of proportional gain and dispel the erroneous notion of dimensionless proportional gain and mutually independent gain.

The main research contents of this paper are arranged as follows. Section II examines the physical characteristics and issues of PI control systems, scientifically defines the dimensional properties of proportional gain, and resolves the dimensional conflict between control force from PI controllers and input control force controlled systems. In Section III, the model of PI control system is established. Based on the characteristic equation of a critical damping system, Zeng's stabilization rule (ZSR) and ACPI control force model with a single speed factor are developed. The stabilization model of the speed factor is investigated and the robustness of the ACPI control system is analyzed. The efficacy and feasibility of ACPI control theory are validated through simulation experiments in Section IV. Section V is the conclusions.

II. PHYSICAL CHARACTERISTICS AND CHALLENGES OF PI CONTROL SYSTEM

For the purpose of analysis, it is assumed that both the expected output (r) and actual output ($y = x_1$) of the controlled system are physical variables of **generalized displacement**. Therefore, the tracking error ($e_1 = r - y$) is also a physical variable of **generalized displacement**, while the integral of error ($e_0 = \int_0^t e_1 d\tau$) represents physical variable of **generalized displacement·second** and differential of error ($\dot{e}_1 = \dot{r} - \dot{y}$) denotes physical variable of **generalized speed**.

2.1 Physical Characteristics of the PI Controller

The PI control force is defined as follows:

$$b_0 u = k_p(e_1 + e_0/T_i) = k_p e_1 + k_i e_0 \quad (1)$$

where, $b_0 u$ represents the control force, the k_p and k_i corres-pond to the proportional gain and integral gain respectively. $k_i = k_p/T_i$ and T_i is the integral time constant with dimension of second.

Using a dimensionless proportional gain (k_p) in PI controller ensures that the control force ($b_0 u$) only has the same dimensions as the **generalized displacement**, which is represented by each term in ($e_1 + e_0/T_i$).

2.2 Physical Characteristics of the PI Controller

Since PI controller is only suitable for controlling first-order systems, a first-order system is considered as follows.

$$\begin{cases} \dot{x}_1 = f(x_1, \xi) + b_0(d + u) \\ y = x_1 \end{cases} \quad (2)$$

where, u and y represent the input and output, x_1 represents a state, $f(x_1, \xi)$ represents known or unknown nonlinear model, b_0 is the control coefficient, ξ is the time-varying model parameter set, and d is the external bounded disturbance.

As y represents a physical variable of **generalized displacement**, $\dot{x}_1$ represents a physical variable of **generalized speed**. In accordance with the principle of dimensional symmetry, the internal dynamics $f(x_1, \xi)$, external disturbance ($b_0 d$), and input control force (b_0u) of the controlled system (2) are all physical variables of the **generalized speed**.

2.3 Dimensional Conflict Issue in PI Control Systems

According to the analysis in Section 2.1, using dimension-less proportional gain will ensure that the PI control force only has the same unit as the **generalized displacement**. However, based on the analysis in Section 2.2, there is a dimensional conflict between PI control force and input control force of any first-order system since it has unit of **generalized speed.**

2.4 Incongruous Control Mechanism of PI Control Force

Since k_p and T_i are two independent variables, there is uncertainty in the surface relation ($k_i = k_p/T_i$) between k_p and k_i, indicating that k_p and k_i are also objective independent variables. In the control process, the mutual independence of k_p and k_i causes the proportional control force ($k_p e_1$) and integral control force ($k_i e_0$) to be mutually independent as well, resulting in an uncoordinated PI control mechanism.

2.5 The Concept of Resolving Dimensional Conflict and Incongruous Control Mechanism

The analysis above demonstrates that the dimensionless proportional gain k_p is the fundamental cause of dimensional inconsistency between PI controller and controlled system. Therefore, by defining the dimension of proportional gain as 1/s, the PI control force can possess the dimension of generalized speed and satisfy the dimensional requirement of input control force for any first-order system, thus resolving the issue of dimensional conflict in PI control systems.

When the k_p has a dimension of 1/s, it can be inferred from the $k_i = k_p/T_i$ that the k_i has a dimension of 1/s^2. If we define a speed factor z_c with a dimension of 1/s, then $k_p \propto z_c$, and $k_i \propto z_c^2$. The z_c not only determines the dimensions of k_p and k_i, resolving the dimension conflict problem in PI control systems, but also establishes an internal relationship ($k_i \propto k_p^2$) between k_p and k_i, solving the discordant control mechanism between $k_p e_1$ and $k_i e_0$.

The aforementioned analysis demonstrates that the z_c inherently links $k_p e_1$ and $k_i e_0$ to generate an auto-coupling proportional-integral (ACPI) control force, which possesses distinct physical significance.

III. ACPI CONTROL THEORY

3.1 Model Mapping of Controlled System

In order to facilitate analysis, the total disturbance is deno-ted as $w = f(x_1, \xi) + b_0 d$. Then, system (2) can be represen-ted as a linear disturbance system:

$$\begin{cases} \dot{x}_1 = w + b_0 u \\ y = x_1 \end{cases} \tag{3}$$

where, $w = f(x_1, \xi) + b_0 d$, and $|w| \leq \varepsilon_0$.

An effective controller designed based on system (3) can be applied to effectively control system (2), as they are equivalent mappings of each other.

3.2 PI Control System

It can be derived from the system (3): $\dot{e}_1 = \dot{r} - \dot{y} = \dot{r} - w - b_0 u$. Based on Equation (1), the PI control system can be established in the following manner:

$$\begin{cases} \dot{e}_0 = e_1 \\ \dot{e}_1 = \widehat{w} - k_p e_1 - k_i e_0 \end{cases} \tag{4}$$

where, $\widehat{w} = \dot{r} - w$ represents the compound total disturbance, and $|\widehat{w}| \leq \varepsilon_1$。

As the PI control system (4) is essentially an error system that responds to the compound total disturbance, it can be classified as a causal system. Based on $E_0(s) = s^{-1} E_1(s)$, by applying Laplace transform to system (4), the PI control system can be represented in the complex frequency domain as follows:

$$E_1(s) = \frac{s}{s^2 + k_p s + k_i} \widehat{W}(s) \tag{5}$$

From system (5), the transfer function of the PI control system can be defined as follows:

$$H(s) = \frac{E_1(s)}{\widehat{W}(s)} = \frac{s}{s^2 + k_p s + k_i} \tag{6}$$

3.3 ACPI Control System

To ensure the good robust stability and dynamic response characteristics of the PI control system (6), it is imperative to scientifically stabilize both $k_p$and k_i.

As any critical damping system exhibits stable behavior and favorable dynamic response characteristics, it is desirable for the PI control system (6) to be a second-order critical damping system. To achieve this, its characteristic equation can be formulated as: $(s + z_c)^2 = s^2 + 2z_c s + z_c^2$.

The stabilization rule proposed by Zeng (ZSR) based on system (6) can thus be formulated as follows:

$$\begin{cases} k_i = z_c^2 \\ k_p = 2z_c \end{cases} \tag{7}$$

where, z_c represents the speed factor with a dimension of 1/s.

z_c-based ZSR (7) not only scientifically defines dimensions of the k_p and k_i, but also serves as an internal coupling factor between them. This theoretically ensures that the PI control system (6) is a critical damping system with excellent dynamic response characteristics and stability, thus making it an ACPI control system:

$$H(s) = \frac{E_1(s)}{\widehat{W}(s)} = \frac{s}{(s + z_c)^2} \tag{8}$$

As z_c is solely associated with the dynamic speed of the controlled system rather than its dynamic model, it is theoretically ensured that the ACPI control system (8) remains stable and exhibits good model robustness when z_c is greater than zero. The ZSR (7) and ACPI control system (8) based on z_c represents a novel control theory, which is referred to as ACPI control theory.

In accordance with the ZSR (7), the ACPI's proportional and integral control forces can be derived as follows:

$$\begin{cases} b_0 u_p = 2z_c e_1 \\ b_0 u_i = z_c^2 e_0 \end{cases} \tag{9}$$

The control force of ACPI, as per formula (9), is:

$$b_0 u = b_0(u_p + u_i) = z_c^2 e_0 + 2z_c e_1 \tag{10}$$

Where, $|u| \leq u_m$, and u_m is the maximum input amplitude of the controlled system.

The formula (10) implies that the ACPI control force increases with z_c, resulting in a stronger control ability for the first-order system (2) or (3).

ACPI control theory is a model-independent method as z_c solely relies on the dynamic speed of the controlled system, rather than its dynamic model.

3.4 Analysis of ACPI Control System

Theorem 1. Assume that the combined total disturbance is bounded: $|\hat{w}| \le \varepsilon_1$, then the steady-state error of ACPI control system is bounded as well:$|e_1(\infty)| < \varepsilon_1/z_c$, and the system exhibits strong resilience to combined total disturbance.

Proof: According to ACPI control system (8), its unit impulse response can be derived as follows:

$$h(t) = \dot{h}_1(t) = (1 - z_c t)e^{-z_c t}\varepsilon(t) \tag{11}$$

where, $h_1(t) = te^{-z_c t}\varepsilon(t)$, and $\varepsilon(t)$ represents a unit step function.

From system (8), the time domain model of tracking error can be written as:

$$e_1(t) = h(t) * \hat{w}(t) = \int_0^t h(\tau)\hat{w}(t-\tau)d\tau \tag{12}$$

when $|\hat{w}| \le \varepsilon_1$, and $|e_1(t)| \le \varepsilon_1 \int_0^t |h(\tau)|d\tau$, then steady-state error can be expressed as:

$$|e_1(\infty)| \le \varepsilon_1 \int_0^\infty |h(\tau)|d\tau \tag{13}$$

Based on formula (11), $h(t) \ge 0$ for $0 < t \le 1/z_c$, and $h(t) < 0$ for $1/z_c < t < \infty$, hence, we have

$$\int_0^\infty |h(\tau)|d\tau = \int_0^{1/z_c} h(\tau)d\tau - \int_{1/z_c}^\infty h(\tau)d\tau \tag{14}$$

$\because \int_0^\infty h(\tau)d\tau = \int_0^{1/z_c} h(\tau)d\tau + \int_{1/z_c}^\infty h(\tau)d\tau = H(0) = 0$, according to formula (14), it can be obtained that: $\int_0^\infty |h(\tau)|d\tau = 2\int_0^{1/z_c} h(\tau)d\tau = 2h_1(1/z_c) = \frac{2}{ez_c} < \frac{1}{z_c}$, where $e \approx 2.718$ is the Napierian base, put it into formula (13), the steady-state error is:$|e_1(\infty)| < \varepsilon_1/z_c$. *Proof completed*.

From *proof of theorem 1*, it is known that the steady-state error of ACPI control system is inversely proportional to the z_c. Hence, increasing the value of the z_c will improve the control accuracy and anti-disturbance robustness.

3.5 Intrinsic Relationship between z_c and T_i

Given that ACPI comes from PI, hence, the z_c of ACPI is intrinsically related to the T_i of PI. From $k_i = k_p/T_i$, and formula (7), it can be obtained that: $z_c = 2/T_i$. Since z_c is only related to T_i, and unrelated to the model of the controlled system, hence the ACPI control system (8) is a **critical damping system** that has good robust stability and dynamic responsiveness. Its physical significance is that the greater the value of z_c, the stronger the control force of ACPI becomes. As a result, the ability to control system (2) or (3) is enhanced, then steady-state control precision will be improved and anti-disturbance ability will be strengthened accordingly, and vice versa. However, discussing the value of z_c in isolation from the controlled system holds no practical significance. The ACPI control theory aims to stabilize the z_c by integrating the dynamic speed of system (2) or (3).

3.6 External Relationship between z_c and the Controlled System

Let τ_0 and $2/\tau_0$denote the characteristic time and the dynamic speed of the controlled system (2) or (3), respectively, in accordance with the $z_c = 2/T_i$. The smaller τ_0 is, the faster system (2) or (3) operates; otherwise, it operates slower. To ensure sufficient control power for the ACPI controller over the controlled system (2) or (3), it is necessary to set z_c greater than the $2/\tau_0$ of the system (2) or (3), that is

$$z_c = 2/T_i > 2/\tau_0 \tag{15}$$

Based on the inequality (15), set the speed factor of the ACPI as $z_c = 2\alpha/\tau_0$, where$1 < \alpha \le 10$ represents acceleration factor. Set dynamic process time as t_r, and $t_r = 10\tau_0$, then the stabilization model of the speed factor based on t_r is:

$$z_c = 20\alpha/t_r \tag{16}$$

where, $1 < \alpha \le 10$, and t_r represents the dynamic process time of the controlled system.

The formula (16) reflects the external relationship between the speed factor of ACPI controller and the dynamic process time of the controlled system(2). Its physical significance is that the dynamic speed of the controlled system gets faster, it is required to have a greater value of the speed factor in ACPI controller, so that the ACPI controller can have great enough control force on the controlled system. Since altering the value of speed factor can alter the control force of ACPI controller, it indicates that ACPI controller can control first-order systems with various dynamic models.

IV. SIMULATIONS

4.1 Description of the Controlled Systems

Two numerical simulation systems are given in this section to illustrate the effectiveness of the proposed ACPI control scheme.

System 1: Consider a nonlinear system as follows:

$$\begin{cases} \dot{x}_1 = sin(x_1) + d + u \\ y = x_1 \end{cases} \tag{17}$$

where, u and y represent the input and output, x_1 represents a state of system (17).

Set $w = sin(x_1) + d$, then, ***System 1*** can be represented as a linear disturbance system:

$$\begin{cases} \dot{x}_1 = w + b_0 u \\ y = x_1 \end{cases} \tag{18}$$

where, $b_0 = 1$.

System 2: Consider a non-affine nonlinear time-varying system as follows:

$$\begin{cases} \dot{x}_1 = a_1(t)sin(x_1) + b_1(t)u^3 + b_2(t)u^2 \\ \qquad +usin(u) + d \\ y = x_1 \end{cases} \tag{19}$$

where, u and y represent the input and output, x_1 represents a state of system (18), $a_1(t) = 1 + 0.1sin(2t)$, $b_1(t) = 0.1 + 0.01sin(t)$, and $b_2(t) = 0.5 + 0.05cos(t)$ are time-varying coefficients of system (18) respectively.

Set $w = a_1(t)sin(x_1) + b_1(t)u^3 + b_2(t)u^2 + usin(u) + d - b_0 u$, then, ***System 2*** can be represented as a linear disturbance system:

$$\begin{cases} \dot{x}_1 = w + b_0 u \\ y = x_1 \end{cases} \tag{20}$$

where, $b_0 = 1$.

Despite the distinct dynamic characteristics of the two systems, they can be transformed into a unified form of linear disturbance system.

Let the external disturbance of the two systems be as follows:

$$d = \begin{cases} 1, 10s \leq 12s \\ 0,\ t < 10s\ or\ t > 12s \end{cases} \tag{21}$$

and the initial state of both systems is $x_1(0) = 0$.

4.2 ACPI Controller

Set $t_r = 1$s, and $\alpha = 4$, then, $z_c = 20\alpha/t_r = 80$/s, and ACPI controller as follows:

$$u = (z_c^2 e_0 + 2z_c^2 e_1)/b_0 \tag{22}$$

where, $b_0 = 1$, and $z_c = 80$/s.

4.3 Simulation Experiments

Let the expected output of both systems be a unit step trajectory: $r = 1$. Consider that the state output of any system is unlikely to undergo mutation, therefore, a well-planned transition process is implemented to achieve the desired output by means of a low-pass filter as follows

$$H(s) = 1/(T_0 s + 1) \tag{23}$$

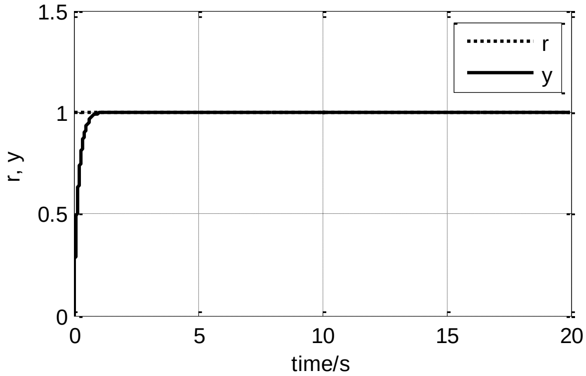
where $T_0 = 0.2t_r = 0.2$s.

Let the integral step be 0.001s. The following simulation experiments use ACPI controller with the same speed factor.

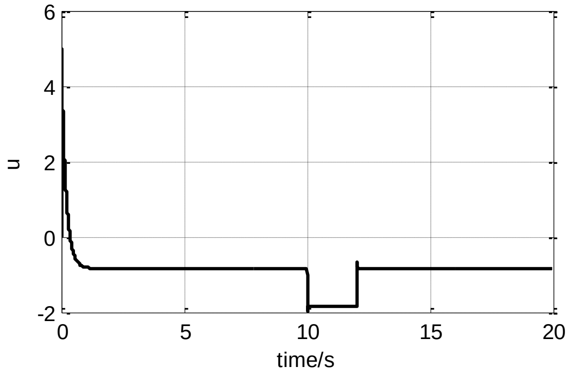
Simulation 1: Unit step tracking for System 1

The unit step tracking control results for the System 1 are depicted in Fig.1.

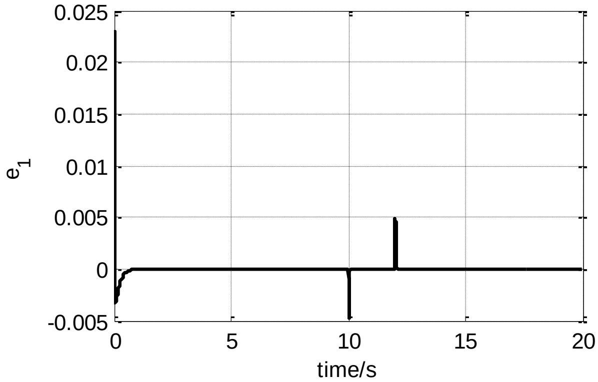
Fig. 1 demonstrates that ACPI control method achieves steady-state within 1.0 second, with a maximum steady-state error less than 1.0e-11 after 4 seconds. The restoration of the disturbed state to a steady-state can be achieved within 0.1 seconds, as depicted in Fig. 1 (e). It shows that ACPI control system has strong anti-disturbance capability.



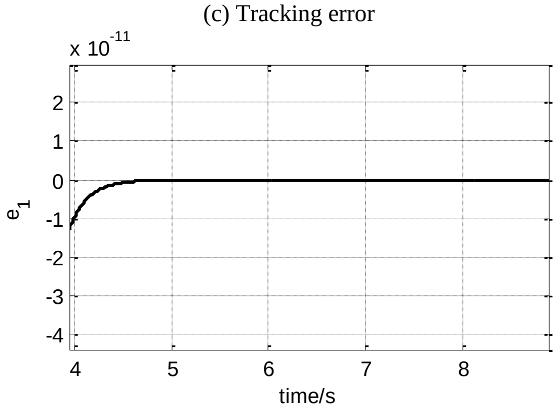
(a) Tracking trajectory



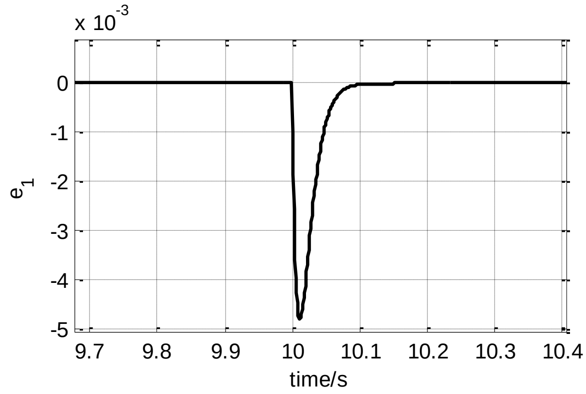
(b) Control input



(c) Tracking error



(d) Steady-state error



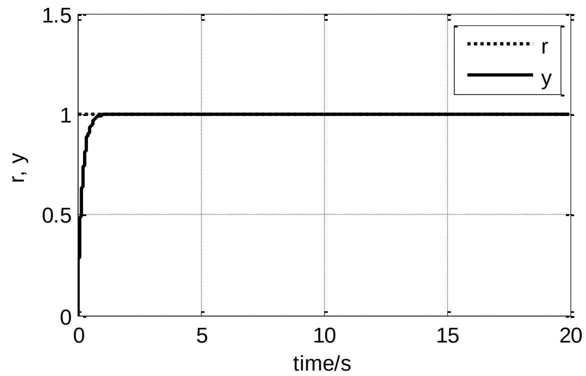
(e) Anti-disturbance result

Fig.1 Control results for System 1

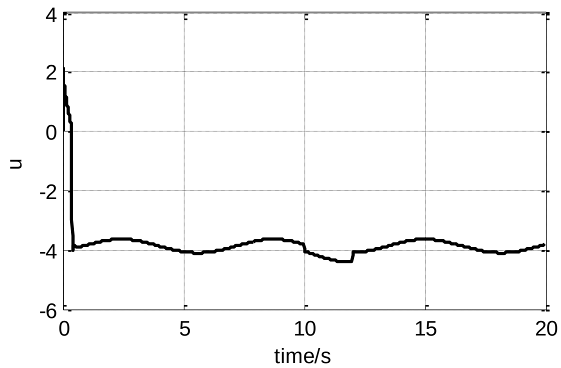
Simulation 2: Unit step tracking for System 2

The unit step tracking control results for System 2 are depicted in Fig.2.

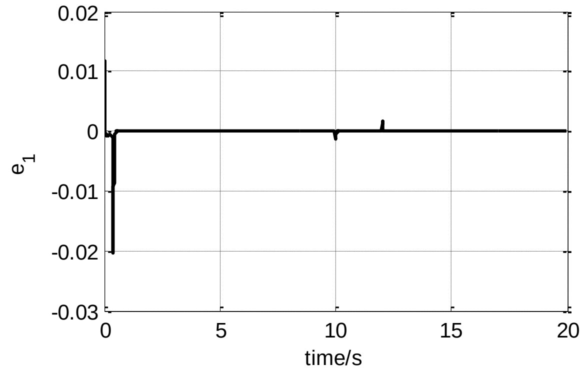
Fig. 2 demonstrates that ACPI control method achieves steady-state within 1.0 second, with a maximum steady-state error less than 4.0e-5 after one second. The restoration of the disturbed state to a steady-state can be achieved within 0.2 seconds, as depicted in Fig. 2 (e). It shows that ACPI control system has strong anti-disturbance capability.



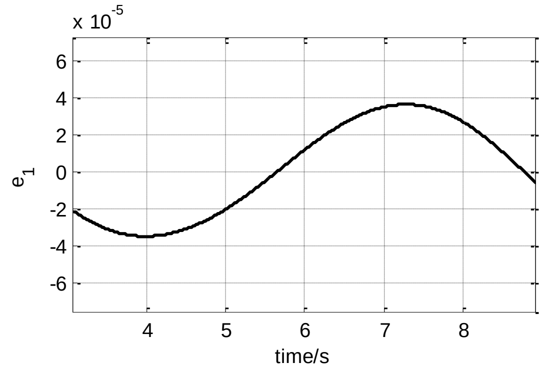
(a) Tracking trajectory



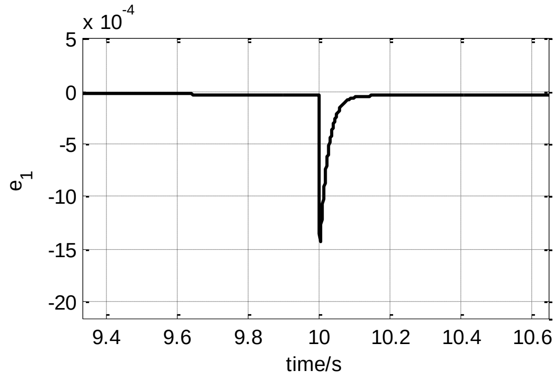
(b) Control input



(c) Tracking error



(d) Steady-state error



(e) Anti-disturbance result

Fig.2 Control results for System 2

The simulation results above demonstrate that the ACPI controller, with a consistent speed factor, can achieve satisfactory control outcomes for two systems with distinct dynamic models. Therefore, the ACPI controller exhibits excellent versatility. Since the ACPI controller possesses a single speed factor for stabilization, it is capable of controlling various linear or nonlinear systems, known or unknown, with comparable dynamic speeds by adjusting the value of speed factor. Furthermore, as the speed factor solely pertains to the controlled object's dynamic speed rather than its dynamic model, implementing ACPI control on diverse complex nonlinear systems becomes convenient.

V. CONCLUSION

The main features and innovations are as follows:

1) The stabilization problem of PI controller is scientifically solved by Zeng's stabilization rules based on a single speed factor. Its innovative point lies in defining the unit of proportional gain according to the speed factor, establishing an internal relationship between k_p and k_i, correcting theoretical defects of dimensionless k_p and mutually independent gains.

2) The ACPI controller, which is based solely on z_c, requires stabilization of only one variable. As a result, the ACPI control system has a simple structure, low computational requirements and is highly practical.

3) The ACPI control system is a critically damped system that relies solely on the dynamic speed of the controlled system, rather than its dynamic model. Therefore, in theory, the model robustness and anti-disturbance robustness of the ACPI control system are guaranteed.

The scientific conclusion has been reached that the upper bound of steady-state error in an ACPI control system is inversely proportional to the z_c. The steady-state accuracy and anti-disturbance ability of the ACPI control system can be precisely controlled.

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ZENG Zhe-zhao was born in Hunan, China, in 1963. He received the B.S. and the M.S. degree in physics from the Xiangtan University and Tsinghua University, China, in 1987 and 1989, respectively. He received the Ph.D. degree in circuit and system from Hunan University, China, in 2008.
He is currently a Professor II of automatic control in the Changsha University of Science and Technology. He has been working on PID, SMC and ADRC for nearly 15 years. By analyzing the physical properties of PID control systems, he found that dimensionless proportional gain is the root cause of poor PID gain robustness and anti-disturbance robustness. Based on a single speed factor, Zeng's stabilization rule is proposed, and ACPID control theory is formed, which successfully solves the PID stabilization problem and creates the development direction of control theory based on dimensional matching principle. E-mail: 508984293@qq.com.

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