Research Article

Human Simulated Intelligent Control with Double-Direction Dead-Zone Compensation for Joint Motion Control of a Large-Sized Boom System

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Joint motion control of a 52-meter-long five-boom system driven by proportional hydraulic system is developed. It has been considered difficult due to strong nonlinearities and parametric uncertainties, the effect of which increases with the size of booms. A human simulated intelligent control scheme is developed to improve control performance by modifying control mode and control parameters. In addition, considering the negative effects caused by frequent and redundant reverse actions of the proportional valve, a double-direction compensation scheme is proposed to deal with the dead-zone nonlinearity of proportional valve. Sinusoidal motions are implemented on a real boom system. The results indicate that HSIC controller can improve control accuracy, and dead-zone nonlinearity is effectively compensated by proposed compensation scheme without introducing frequent reverse actions of proportional valve.

1. Introduction

Large-sized boom system driven by proportional hydraulic system is widely applied for material handling and earth moving in construction, mainly due to high forces to weight ratio and large working space which can be well reflected by construction machinery such as truck mounted concrete pump, telescopic handler, and crane. However, manual operation has been the main operation mode, with which not only construction quality and efficiency are limited, but additional assistance from workers is also needed, which may increase the probability of unpredictable personal injury. Therefore, automation of the large-sized boom system is required. In [1], a semiautomatic control system for hydraulic excavator is developed to simplify operation mode. With respect to truck mounted concrete pump, an automatic pouring approach is proposed for the repetitive tasks [2]. To accomplish semiautomated assembly of facade panels, an automatic control system of an upgraded telescopic handler is proposed [3].

Joint motion control is the basis for accomplishing automation of the boom system. Unfortunately, various nonlinearities and parametric uncertainties induced by

mechanical structures and hydraulic components may deteriorate control performance seriously. Therefore, joint motion control becomes more difficult due to these factors. In [4], the adaptive robust control (ARC) is proposed for hydraulic robot arms with taking uncertain nonlinearities and strong coupling among various hydraulic actuators into account. A fixed cascade controller with an adaptive dead-zone compensation scheme to overcome nonlinearities is proposed [5, 6]. Time-delay control with a compensator is adopted to achieve a straight-line motion of a real heavy-duty excavator under working speed conditions [7]. A discontinuous projection based adaptive robust controller is proposed for the motion control of single-rod hydraulic actuators regulated by proportional directional control valve in [8]. In [9], a new control approach called adaptive switching learning PD control is proposed for trajectory tracking of robot manipulators with faster convergence rate.

In addition, the severity of these problems appears to increase with the size of booms. In this work, joint motion control of a 52-meter-long five-boom system of truck mounted concrete pump is studied, but fewer research works have been focused on. In [10], a closed-loop detection

and open-loop control strategy is proposed for single joint control to avoid exciting vibration, and it is evaluated through computer simulation. Real-time, simple and effective control strategy is essentially required for joint motion control in practice. Human-simulated intelligent control (HSIC) as an intelligent control strategy has been used to solve complex control problems. In [11, 12], HSIC has been applied to accomplish the swinging-up and handstand control of cart double pendulum system. To attenuate unwanted vibration of magneto rheological suspension system of a passenger car, HSIC controller is designed according to the expert's experience [13]. In this work, motion control level and self-turning level are developed for joint motion control of the large-sized boom system.

Dead-zone in proportional valve due to spool overlap is a hard nonlinearity which can lead to performance degradation of the controller and instability in the closed-loop system. To overcome the negative effects of the dead-zone nonlinearity, many works based on modern techniques have been published. Direct dead-zone inverse is used to compensate the negative effects of the dead-zone nonlinearity in [3, 6, 8, 14–17], but discontinuous control law will be established by adopting this scheme. In [18], a soft compensation scheme to avoid the discontinuity is proposed, but an error will be introduced in compensation process. In [19], a smooth inverse dead-zone approach is adopted in adaptive control for nonlinear systems. An adaptive fuzzy sliding mode controller is developed for an electrohydraulic system subject to an unknown dead-zone input [20, 21]. In this work, a compact double-direction compensation approach is developed for the unknown dead-zone of the hydraulic proportional valve, which avoids redundant and frequent reverse actions of the valve which lead to vibration and instability. Moreover, lower computation cost is required for implementation.

2. Electrohydraulic Control System

In this work, electrohydraulic proportional control system for each joint works according to the proportional flow rate regulation principle, in which the flow rate through proportional valve is controlled by appropriate control signal setting. The electrohydraulic control system is divided into six identical groups. Figure 1 shows the control principle of one joint including the so-called load-independent valve, which indicates that the output flow rate has no relation to external load but is just proportional to spool displacement. In this figure, q_1 and q_2 are the incoming flow rate within each chamber, p_1 and p_2 are the pressures inside the two chambers of the cylinder, A_1 and A_2 are the effective areas of the two chambers and $A_2/A_1 = \varepsilon$, m is the total mass of piston and load referred to piston, y is the piston displacement, θ is the measured joint angle, and F is the external force.

Neglecting external leakage and deformation of the fluid inside the chamber, the flow rate continuity equation can be derived as

$$q_1 = A_1 y' + C_{ip} (p_1 - p_2), \quad (1)$$

in which C_{ip} is the internal leakage coefficient.

FIGURE 1: Electrohydraulic control principle.

The balance of forces on the piston leads to the equation of motion when the piston extends:

$$p_1 A_1 - p_2 A_2 = m y'' + b y' + F, \quad (2)$$

in which b is the combined viscous coefficient.

Defining the pressure drop across the load as $p_L = p_1 - \varepsilon p_2$ and neglecting the relatively small friction force, (2) can be rewritten as

$$p_L A_1 = m y'' + F. \quad (3)$$

The continuity flow rate equations of the two orifices inside the proportional valve can be expressed, respectively:

$$\begin{aligned} q_1 &= C_d w x \sqrt{\frac{2(p_s - p_1)}{\rho}}, \\ q_2 &= C_d w x \sqrt{\frac{2p_2}{\rho}}, \end{aligned} \quad (4)$$

in which C_d is the discharge coefficient, w is the valve orifice area gradient, x is the effective displacement of the spool without taking the dead-zone nonlinearity into account, p_s is the supply pressure of pump, and ρ is the density of hydraulic oil.

Considering that the internal leakage is much less compared with the input or output flow rate of the chamber, it can be approximately deduced that

$$\frac{q_1}{q_2} = \sqrt{\frac{p_s - p_1}{p_2}} \approx \frac{A_1}{A_2} = \frac{1}{\varepsilon}. \quad (5)$$

FIGURE 2: Block diagram of joint motion control.

Therefore, p_1 and p_2 are solved according to (5) and the definition of p_L :

$$\begin{aligned} p_1 &= \frac{\varepsilon^3 p_s + p_L}{1 + \varepsilon^3}, \\ p_2 &= \frac{\varepsilon^2 (p_s - p_L)}{1 + \varepsilon^3}. \end{aligned} \quad (6)$$

Defining $\Delta p = p_s - p_1$, according to the characteristic of load-independent valve it can be deduced that Δp is a constant, and Δp has been determined by maker. Since the dynamics of the valve are fast enough, the spool of proportional valve can be approximately expressed as follows:

$$x = ku, \quad (7)$$

in which u represents the effective control signal and k is the amplification gain. Combining (1), (3), (4), (6), and (7) together and taking the definition of Δp into account, u can be expressed as

$$u = K \left(A_1^2 y' + C_{ip} m y'' + C_{ip} A_1 \varepsilon^2 (\varepsilon - 1) \Delta p + C_{ip} F \right), \quad (8)$$

in which $K = \sqrt{2\Delta p p} / 2C_d w \Delta p A_1 k$.

3. Design of Control System

Figure 2 shows the control block diagram of joint motion control. In this work, the feed-forward control term shown in this figure is applied to reduce the steady state error, and control signal u_d corresponding to the desired joint trajectory can be calculated according to (8), in which the piston displacement y is uniquely determined according to (9) which has already been given in [13]

$$y = \sqrt{l_1^2 + l_2^2 - 2l_1 l_2 \cos(\theta)}, \quad (9)$$

in which l_1 and l_2 represent the length between the joint and two endpoints of the cylinder, respectively.

3.1. HSIC Controller. In view of strong nonlinearities and parametric uncertainties, an HSIC controller is formulated and implemented for joint motion control. The basic idea of HSIC is to imitate people's behavior and is composed of three levels including direct control level, parameter correction level, and task adjustment level. In this work, the first two levels are adopted. The direct control level is to determine

FIGURE 3: Characteristic models based on the error state.

the appropriate control mode according to the dynamic characteristics, and the parameter correction level is to adaptively adjust the control parameters by imitating the expert's experience. e represents the error between the desired angle θ_d and the actual angle θ . Figure 3 shows the characteristic models based on the error state, and special dynamic behavior shown in the box corresponds to a quadrant. In the figure, e_1 , e_2 , and e_3 are the threshold values of error, and e_{s1} and e_{s2} are the threshold values of the derivation of error. The details are illustrated in the following.

3.1.1. Direct Control Level. Characteristic models are constructed in direction control level according to the error state:

$$\Phi_D = \{\varphi_{D1}, \varphi_{D2}, \varphi_{D3}, \varphi_{D4}\}, \quad (10)$$

in which φ_{D1} : $0 \leq \text{abs}(e) < e_1$ and $0 \leq \text{abs}(e') < e_{s1}$; φ_{D2} : $0 \leq \text{abs}(e) < e_2$ without including φ_{D1} ; φ_{D3} : $e_2 \leq \text{abs}(e) < e_3$; φ_{D4} : $e_3 \leq \text{abs}(e)$. Function $\text{abs}(\cdot)$ means the absolute value of the variable in bracket.

Correspondingly, the control modes in direction control level are set as

$$\Psi_D = \{\psi_{D1}, \psi_{D2}, \psi_{D3}, \psi_{D4}\}, \quad (11)$$

in which ψ_{D1} is zero output mode; ψ_{D2} is derivation control mode; ψ_{D3} is proportional plus derivation control mode; ψ_{D4} maintains output mode.

Production system is applied to describe the structure of each level in HSIC controller. The reference rules in direction control level are expressed as

$$\Omega_D = \{\omega_{D1}, \omega_{D2}, \omega_{D3}, \omega_{D4}\}, \quad (12)$$

in which ω_{D1} : IF φ_{D1} , THEN ψ_{D1} ; ω_{D2} : IF φ_{D2} , THEN ψ_{D2} ; ω_{D3} : IF φ_{D3} , THEN ψ_{D3} ; ω_{D4} : IF φ_{D4} , THEN ψ_{D4} .

3.1.2. Parameter Correction Level. In this level, the characteristic models are chosen as follows:

$$\Phi_P = \{\varphi_{P1}, \varphi_{P2}, \varphi_{P3}, \varphi_{P4}, \varphi_{P5}, \varphi_{P6}\}, \quad (13)$$

in which φ_{P1} : $0 \leq \text{abs}(e') < e_{s1}$ and $ee' \geq 0$; φ_{P2} : $0 \leq \text{abs}(e') < e_{s1}$ and $ee' \leq 0$; φ_{P3} : $e_{s1} \leq \text{abs}(e') < e_{s2}$ and $ee' \geq 0$; φ_{P4} : $e_{s1} \leq \text{abs}(e') < e_{s2}$ and $ee' \leq 0$; φ_{P5} : $e_{s2} \leq \text{abs}(e')$ and $ee' \geq 0$; φ_{P6} : $e_{s2} \leq \text{abs}(e')$ and $ee' \leq 0$.

According to the control modes aforementioned, parametric correction law is mainly designed for proportional coefficient k_p and differential coefficient k_d , and it is given more carefully in the following:

$$\Psi_P = \{\psi_{P1}, \psi_{P2}, \psi_{P3}, \psi_{P4}, \psi_{P5}, \psi_{P6}, \psi_{P7}, \psi_{P8}\} \quad (14)$$

in which ψ_{P1} : $k_p(t_i) = k_p(t_{i-1})$, ψ_{P2} : $k_p(t_i) = \text{MIN}(k_p(t_{i-1})/k_1, k_{p\max})$, ψ_{P3} : $k_p(t_i) = \text{MAX}(k_{p\min}, k_p(t_{i-1})k_1)$, ψ_{P4} : $k_p(t_i) = a_1 + b_1(1 - \text{sech}(e(t)c_1))$, ψ_{P5} : $k_d(t_i) = k_d(t_{i-1})$, ψ_{P6} : $k_d(t_i) = \text{MIN}(k_d(t_{i-1})/k_2, k_{d\max})$, ψ_{P7} : $k_d(t_i) = \text{MAX}(k_{d\min}, k_d(t_{i-1})k_2)$, and ψ_{P8} : $k_d(t_i) = a_2 + b_2/(1 + \exp(e(t)d_2c_2))$. Coefficients $k_1, k_2, k_{p\min}, k_{p\max}, k_{d\min}, k_{d\max}, a_1, b_1, c_1, a_2, b_2, c_2$, and d_2 are all positive constants.

The corresponding reference rules about parametric correction are given as

$$\begin{aligned} \Omega_P \\ = \{\omega_{P1}, \omega_{P2}, \omega_{P3}, \omega_{P4}, \omega_{P5}, \omega_{P6}, \omega_{P7}, \omega_{P8}, \omega_{P9}, \omega_{P10}\}, \end{aligned} \quad (15)$$

in which ω_{P1} : IF φ_{D1} or φ_{D4} , THEN ψ_{P1} and ψ_{P5} ; ω_{P2} : IF φ_{D2} and (φ_{P1} or φ_{P2}), THEN ψ_{P1} and ψ_{P7} ; ω_{P3} : IF φ_{D2} and φ_{P3} , THEN ψ_{P1} and ψ_{P5} ; ω_{P4} : IF φ_{D2} and (φ_{P4} or φ_{P5}), THEN ψ_{P1} and ψ_{P8} ; ω_{P5} : IF φ_{D2} and φ_{P6} , THEN ψ_{P1} and ψ_{P6} ; ω_{P6} : IF φ_{D3} and (φ_{P1} or φ_{P2}), THEN ψ_{P1} and ψ_{P7} ; ω_{P7} : IF φ_{D3} and φ_{P3} , THEN ψ_{P4} and ψ_{P5} ; ω_{P8} : IF φ_{D3} and φ_{P4} , THEN ψ_{P4} and ψ_{P8} ; ω_{P9} : IF φ_{D3} and φ_{P5} , THEN ψ_{P2} and ψ_{P5} ; ω_{P10} : IF φ_{D3} and φ_{P6} , THEN ψ_{P3} and ψ_{P6} .

3.2. Dead-Zone Compensation. Figure 4 shows the nonsymmetric dead-zone nonlinearity of the proportional valve, in which u_{act} represents the actual control signal outputting to the valve, k_r and k_l represent the slopes, and D_r and D_l represent the breakpoint values.

According to (7), the mathematic model of dead-zone can be expressed as

$$u_c = \begin{cases} u_{\text{act}} - D_r & u_{\text{act}} \geq D_r \\ 0 & D_l < u_{\text{act}} < D_r \\ u_{\text{act}} - D_l & u_{\text{act}} \leq D_l \end{cases} \quad (16)$$

in which u_c represents the sum of control signals generated by feed-forward controller and HSIC controller, respectively.

Actually, the accurate breakpoints cannot be acquired in time by the adopted algorithms including fuzzy logic,

FIGURE 4: Characteristic of nonsymmetric dead-zone.

FIGURE 5: Direct dead-zone inverse compensation.

neural network, and adaptive control because of the parametric uncertainty and external environment. Assume that the ranges of D_r and D_l are known: $D_{r1} \leq D_r \leq D_{r2}$ and $D_{l1} \leq D_l \leq D_{l2}$. To offset the deleterious effect of the dead-zone nonlinearity in closed-loop control, a direct dead-zone inverse compensation approach shown in Figure 5 is generally adopted when rigor control performance is not required. In this figure, $D_{r\text{-est}}$ and $D_{l\text{-est}}$ represent the estimates of D_r and D_l , respectively.

In practice, redundant and frequent reverse actions of proportional valve are inevitably introduced by adopting direct dead-zone inverse compensation scheme, which may deteriorate control performance and even lead to severe vibration. In order to offset the adverse effect, we proposed a double-direction compensation scheme shown in Figure 6. In this figure, the dead-zone compensation is composed of two parts which represent different implementation direction and are shown by dotted line and solid line, respectively, and the same compensation principle is applied in each part which is composed of three sections. In the following, the first part represented by dotted line is taken for illustration.

In section one, when $u_c \geq 0$, the direct dead-zone inverse compensation approach aforementioned is adopted for compensation.

In section two, a variable Dep_r , named “depth” is defined and determined according to (17). By adding the depth Dep_r , the reverse action of the valve will be delayed as u_c decreases. Quadratic function is established to describe the relationship between u_c and u_{act} in this section. In this manner, it can be guaranteed that the orifice is completely closed gradually

FIGURE 6: Double-direction dead-zone compensation.

before reverse action is implemented, so that vibration can hardly be excited:

$$\text{Dep}_r \leq D_{r1} - D_{r\text{-est}}. \quad (17)$$

In section three, reverse action is implemented at first. Then, a quick compensation mode is implemented to avoid introducing or enlarging the system error, in which the integral function with respect to u_c is applied to track the trajectory of direct dead-zone inverse approach quickly but without introducing discontinuity.

According to the description above, the mathematical expression of this part is given as follows:

$$u_{\text{act}} = \begin{cases} u_c + D_{r\text{-est}} & u_c \geq 0 \\ D_{r\text{-est}} - \frac{(u_c - \text{Dep}_r)^2}{\text{Dep}_r} + \text{Dep}_r & 0 > u_c \geq \text{Dep}_r \\ u_c + D_{l\text{-est}} + U_r & u_c < \text{Dep}_r, \end{cases} \quad (18)$$

in which $U_r = \text{MAX}(\int (u_c - \text{Dep}_r) dt, \text{Dep}_r) - \text{Dep}_r$.

In the second part, another depth Dep_l can be determined according to

$$\text{Dep}_l \geq D_{l2} - D_{l\text{-est}}. \quad (19)$$

The mathematical expression of the dead-zone compensation for the second part can also be given in the same way:

$$u_{\text{act}} = \begin{cases} u_c + D_{l\text{-est}} & u_c \leq 0 \\ D_{l\text{-est}} - \frac{(u_c - \text{Dep}_l)^2}{\text{Dep}_l} + \text{Dep}_l & 0 < u_c \leq \text{Dep}_l \\ u_c + D_{r\text{-est}} + U_l & u_c > \text{Dep}_l, \end{cases} \quad (20)$$

in which $U_l = \text{MIN}(\int (u_c - \text{Dep}_l) dt, \text{Dep}_l) - \text{Dep}_l$.

FIGURE 7: The experimental setup.

4. Control Results and Discussion

To evaluate control performances of proposed control schemes, joint motion control is implemented on a real 52-meter-long five-boom system shown in Figure 7. The proposed control schemes are implemented on the original 16-bit controller without assistance from any other equipment in real time. Joint angle is measured by means of two space angle sensors mounted on the neighboring booms, respectively, and the measured angles are stored in a PC via a CAN-bus recorder. The sampling period is set to 20 ms in practice.

Sinusoid expressed as (21) is taken as the target trajectory for evaluation, and two sinusoidal motions with different period T are implemented in experiments. Moreover, conventional PID control is also adopted for comparison, which is widely used in industrial process for its satisfactory performance and simple structure, and its control coefficients are optimized by using the standard Ziegler-Nichols closed-loop tuning approach. And the proposed double-direction dead-zone compensation scheme is applied to deal with dead-zone nonlinearity of proportional valve. We take the first joint for illustration, the results of which are shown in Figure 8. According to Figure 8(a), the error obtained by HSIC controller is almost within ± 0.15 degrees, and its maximum is reduced to 43% of the maximum error obtained by PID control. In Figure 8(b), the maximum error obtained by HSIC controller is also reduced to 52% compared with PID control. These results indicate that, by modifying control action to adapt to the dynamic behavior of the boom system, the joint control accuracy can be improved, so that HSIC is verified to be effective for joint motion control of such a large-sized boom system. Because of the advantage of the openness of HSIC, more excellent control schemes or correction strategies can be included in HSIC to modify control action further:

$$\theta_{\text{in}} = \sin\left(\frac{2\pi t}{T}\right). \quad (21)$$

Figure 9 shows control signal u_c and corresponding actual control output u_{act} obtained by double-direction dead-zone compensation module. According to this figure, dead-zone nonlinearity is offset in the moment of reverse action, and frequent reverse actions of the valve in regions

FIGURE 8: Experimental results of the sinusoidal trajectory. (a) $T = 30$ s and (b) $T = 45$ s.

FIGURE 9: Effect of the double-direct dead-zone compensation.

indicated by the red boxes are obviously avoided, which will be introduced by adopting direct dead-zone inverse compensation and lead to vibration and instability. Therefore, it is stated that double-direct dead-zone compensation not only overcomes the negative effect of dead-zone nonlinearity but also improves the capacity of resisting disturbance and stability of the control system without introducing redundant reverse actions, so that this scheme is suitable for dead-zone compensation of proportional valve. And the compensation performance lies on the parameters Dep_r and Dep_l . If they are set too large, the control accuracy may be compromised,

and the capacity of resisting disturbance will be reduced if the smaller values are determined. So these parameters need to be determined according to characteristics of the control object and control requirements established.

5. Conclusions

In this work, the joint motion control of a 52-meter-long five-boom system is studied, which is the basis of achieving automation of the boom system. Because of the strong nonlinearities and parametric uncertainties, it becomes more difficult. Feed-forward control is adopted to reduce the steady state error. A human simulated intelligent controller is developed to improve the adaptive ability of the system by modifying control mode and parameters. For the dead-zone nonlinearity of proportional valve, a double-direction dead-zone compensation scheme is proposed to avoid vibration and instability caused by frequent reverse actions of the valve. Experiments are implemented on a real boom system to demonstrate the effectiveness of the proposed control schemes. Tracking results indicate that HSIC can improve control accuracy, and the effective reverse actions are guaranteed by proposed compensation scheme without introducing redundant and frequent ones, so that the capacity of resisting disturbance of the control system is also improved.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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