Discrete time delay control for hydraulic excavator motion control with terminal sliding mode control

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ABSTRACT

It has been noted that the motion control of a hydraulic robotic excavator has some difficulties in excavator mechanical structure variations, disturbance, non-linearity of complex dynamics of the hydraulic manipulator, and so on. In this paper, a new discrete model-free robust controller is proposed for robotic excavator motion control based on the time delay control (TDC) combined with terminal sliding mode control (TSMC). The proposed controller formulation consists of TDC without acceleration information and TSMC with nonlinear desired error dynamics. Less computational effort and sensing signals are required with this controller, which can decrease the influence of noise by eliminating the acceleration information. Furthermore, the designed terminal sliding error dynamics provide robotic excavator motion control with high position tracking accuracy. The proposed controller was implemented on a 30 ton Volvo hydraulic robotic excavator, and its effectiveness was verified. The achieved end-effector position tracking accuracy was within 2 cm for a curved surface.

1. Introduction

In the field of hydraulic excavators, machine guidance systems have been developed and produced in the past several years [1,2]. The machine guidance system consists of sensors and display panels to track the poses of the excavator attachments and excavator machine body itself. With this information, the excavator operator can perform machine operations with higher precision and productivity than without the machine guidance system. However, the precision and productivity still depend on the operator’s skill, even with a machine guidance system, because the machine is controlled by an operator who determines actions based on such information from the excavator guidance system. Thus, to overcome the shortcomings of the excavator guidance system, construction equipment manufacturers have developed and launched semi-autonomous excavators [3,4]. Those semi-autonomous excavators can perform the end-effector trajectory tracking within 2–3 cm against the target surface. With that, they can provide improved productivity and reduced operator fatigue, especially for non-expert operators.

However, the motion control of a hydraulic robotic excavator manipulator is a highly challenging task in the development of an autonomous excavator system because there are excavator mechanical structure variations, non-linearity of complex dynamics of the hydraulic manipulator, and the disturbances. To solve these problems, many controllers for hydraulic manipulator systems have been proposed and investigated. As simulation studies, a fuzzy-proportional-integral (PI) soft-switch controller, which is a hybrid controller with fuzzy and PI control methods, was developed [5]. The improved particle swarm optimization (PSO) tuning-based proportional-integral-derivative (PID) controller was developed and achieved good position control performance in simulation [6]. As experimental studies, a precision of within 10 cm was obtained using PID and feed-forward compensator methods for the speed and acceleration of the target angle with rotary encoders and potentiometers [7]. PI control with an anti-windup algorithm was proposed for bucket tip position control utilizing electric proportional pressure reducing (EPPR) valves and a linear variable displacement transducer (LVDT) [8]. Cross-coupled control with PI control was proposed for robotic excavator motion control [9,10]. Online learning control for hydraulic excavator position control has also been investigated [11,12]. However, some of the aforementioned techniques [5–12] are limited in simulation and test rig. In other studies, even if the end-effector tracking performance was achieved within 2–3 cm, a system identification process was required or extensive tuning of the parameters was required.

As an alternative, time delay control (TDC), which can compensate for the non-linear and complex dynamics by time delay estimation (TDE) without system identification, has been proposed and investigated for robotic excavators. And its end-effector tracking performance

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was within 2–3 cm without extensive tuning of the parameters [13,14]. Furthermore, a new discrete time delay controller (NDTDC) has been proposed and investigated for hydraulic manipulators [15]. The benefit of this controller is that does not require acceleration information of TDC, which is a quadratic differential signal. This makes it more suitable for practical applications because it is free from the noise contained in the acceleration information.

In this paper, a new discrete model-free robust controller is proposed for hydraulic robotic excavator motion control. The proposed controller consists of NDTDC which is a TDC without acceleration information and TSMC with the desired error dynamics. With this controller, less computational effort and fewer sensing signals are required compared with traditional time delay control. And still, we don’t need a system model because of TDE of NDTDC. Furthermore, to get higher position tracking accuracy compared with the use of the NDTDC alone, the new TSMC is combined with NDTDC. The nonlinear sliding mode control called TSMC has been proposed that provides faster convergence than linear hyper-plane based sliding mode control [16–18]. This controller was implemented on a 30-ton Volvo hydraulic robotic excavator, as shown in Fig. 1, and experimentally verified the effectiveness. The achieved end-effector position tracking accuracy was within 2 cm for a defined curved surface without extensive tuning of the parameters.

2. Review of time delay control

2.1. Robot manipulator dynamics

The standard form of robot manipulator dynamics can be described as follows:

\[
\tau = \dot{\mathbf{M}}(\theta)\ddot{\theta} + \mathbf{C}(\theta, \dot{\theta}, \ddot{\theta}) + \mathbf{G}(\theta) + \mathbf{F}_r(\theta, \dot{\theta}) + \tau_d, 
\]

(1)

where \(\dot{\theta}, \ddot{\theta}, \dddot{\theta} \in \mathbb{R}^n\) denote the joint angle, joint angular velocity, and joint angular acceleration for each joint, respectively; \(\mathbf{M}(\theta) \in \mathbb{R}^{n \times n}\) is the generalized inertia matrix; \(\mathbf{C}(\theta, \dot{\theta}, \ddot{\theta})\) are terms from the Coriolis and centrifugal forces; \(\mathbf{G}(\theta)\) is the gravitational torque; \(\mathbf{F}_r(\theta, \dot{\theta})\) is the nonlinear friction of the manipulator; \(\tau_d \in \mathbb{R}^n\) is the disturbance torque; and \(\tau \in \mathbb{R}^n\) is the joint torque. The manipulator dynamics in Eq. (1) can be rewritten by introducing a constant diagonal matrix \(\mathbf{M}\) as follows: [19]:

\[
\tau = \mathbf{M}(\theta)\ddot{\theta} + \mathbf{N}(\theta, \dot{\theta}, \ddot{\theta}), 
\]

(2)

where \(\mathbf{N}(\theta, \dot{\theta}, \ddot{\theta})\) denotes the total sum of the nonlinear dynamics of the manipulator, which can be represented as follows: [19]:

\[
\mathbf{N}(\theta, \dot{\theta}, \ddot{\theta}) = \left( \dot{\mathbf{M}}(\theta) - \dddot{\mathbf{M}}(\theta) \right) \dddot{\theta} + \mathbf{G}(\theta) + \mathbf{F}_r(\theta, \dot{\theta}) + \tau_d. 
\]

(3)

2.2. Time delay estimation

Under the assumption that \(\mathbf{N}(\theta, \dot{\theta}, \ddot{\theta})\) is piecewise continuous and \(L\) is a small enough time, the value of \(\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta})\) at time \(t\) may be almost the same as \(\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta})\) at time \(t - L\) as follows: [19]:

\[
\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta}) \approx \mathbf{N}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L}. 
\]

(4)

Usually, the small time \(L\) is selected as the sampling period in discrete time control [20]. From the robot manipulator dynamics in Eq. (2), we can have \(\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta}) = \mathbf{N}(\theta, \dot{\theta}, \dddot{\theta})\|_{t - L}\) as follows:

\[
\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L} = \mathbf{N}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L} - \dddot{\mathbf{M}} \dddot{\theta} \|_{t - L}. 
\]

(5)

From Eqs. (4) and (5), we can define the time delay estimation of \(\mathbf{N}(\theta, \dot{\theta}, \dddot{\theta})\) as follows [19]:

\[
\dddot{\mathbf{N}}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L} = \dddot{\mathbf{N}}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L} - \dddot{\mathbf{M}} \dddot{\theta} \|_{t - L}. 
\]

(6)

2.3. TDC with linear desired error dynamics

The control objective of the TDC can be designed with the following error dynamics [19]:

\[
\dddot{e}_i + \mathbf{K}_d \dddot{e}_i + \mathbf{K}_p \dddot{e}_i = 0, 
\]

(7)

where \(e_i, \dddot{e}_i \in \mathbb{R}^n\) denotes the desired trajectory, \(\mathbf{K}_p \in \mathbb{R}^{n \times n}\) and \(\mathbf{K}_d \in \mathbb{R}^{n \times n}\) represent the diagonal gain matrices of decoupled PD controllers. In addition, with the introduced \(\dddot{\mathbf{M}}\) and \(\dddot{\mathbf{N}}\), the control torque \(\tau\) of TDC is formulated as follows:

\[
\tau = \dddot{\mathbf{M}} \dddot{u}_i + \dddot{\mathbf{N}}(\theta, \dot{\theta}, \dddot{\theta}) \|_{t - L}. 
\]

(8)

with

\[
\dddot{u}_i = \dddot{\theta}_i, 
\]

(9)

With the time delay estimation of \(\dddot{\mathbf{N}}\) in Eq. (6), Eq. (8) can be rewritten as follows:

\[
\tau = \dddot{\mathbf{M}} \dddot{u}_i \|_{t - L} - \dddot{\mathbf{M}} \dddot{\theta} \|_{t - L} + \dddot{\mathbf{u}}_i \|_{t - L}. 
\]

(10)

Combining Eqs. (10) with (9), the final TDC formulation can be obtained as follows [19]:

\[
\tau = \dddot{\mathbf{M}} \dddot{u}_i \|_{t - L} - \dddot{\mathbf{M}} \dddot{\theta} \|_{t - L} - \dddot{\mathbf{M}} \dddot{\theta} \|_{t - L} + \dddot{\mathbf{u}}_i \|_{t - L}. 
\]

(11)

Because of the TDE, the TDC is efficient, and its structure is very simple. Moreover, owing to each diagonal matrix \(\dddot{\mathbf{M}}, \mathbf{K}_p, \) and \(\mathbf{K}_d,\) the TDC can be designed as if it consists of \(n\) individual joint controllers.

3. Excavator manipulator dynamics

3.1. Excavator manipulator dynamics in joint space

Fig. 2 shows the excavator’s three-degree-of-freedom manipulator, with the boom, arm, and bucket as the end-effector. The travel and swing function is not included in this manipulator’s dynamics modeling.
In a previous study [13], the manipulator dynamics of Eq. (1) were transformed into the actuator space because of the non-linear relationship between the joint space and actuator space, as shown in Fig. 2. Therefore, as shown below, the transformed Eq. (12) for the actuator space was introduced and investigated for the robotic excavator manipulator motion control with TDC.

\[
F = M(t)\dot{l} + C(l, \dot{l}) + G(l) + F_r(l, \dot{l}) + F_d.
\]

(12)

where \(F \in \mathbb{R}^3\) denotes the forces acting on the cylinders for the boom, arm, and bucket; \(l \in \mathbb{R}^3\) denotes the actuator displacement; and \(F_d \in \mathbb{R}^3\) is the disturbance force on the actuators.

On the other hand, from the cylinder force balance equation, we can obtain the following relation:

\[
F = A_p P_a - A_p P_y,
\]

(13)

where \(A_p\) and \(A_y\) are the areas of both sides of the cylinder; and \(P_a\) and \(P_y\) are the pressures of both sides of the cylinder.

The relationship between effective actuator force \(F\) and joint torque \(\tau\) can be obtained by applying the principle of virtual work [15,21]:

\[
\tau^T d\theta = F^T d l,
\]

(14)

where \(d\theta\) and \(d l\) denote the incremental changes in the joint angle and actuator displacement, respectively. Therefore, from Eq. (14), we have

\[
\tau = J^T(\theta)F,
\]

(15)

where the Jacobian vector \(J(\theta)\) represents the synthetic equivalence relationship between the effective actuator force and joint torque. This is defined as follows:

\[
J(\theta) = \frac{d l}{d\theta}.
\]

(16)

By substituting Eq. (13) into (15), we can obtain a joint space dynamic equation as follows:

\[
\tau = J^T(\theta)F = J^T(\theta)(A_p P_a - A_p P_y).
\]

(17)

### 3.2. Hydraulic cylinder system stiffness

The hydraulic cylinder stiffness for the excavator should be checked before applying the new controller because the system stiffness affects the accuracy and stability of the control system. And from the previous study, the net stiffness \(K\) of the hydraulic cylinder system already studied and presented as follows:

\[
\frac{1}{K} = \frac{1}{K_0} + \frac{1}{K_{r}} + \frac{1}{K_{c1}} + \frac{1}{K_{c2}} + \frac{1}{K_{y}} + \frac{1}{K_{h}},
\]

(18)

where \(K_0\) denotes the hydraulic oil stiffness; \(K_r, K_{c1}, K_{c2}, K_{y}, K_{h}\) denote the piston rod axial stiffness, the cylinder large chamber barrel expansion stiffness, the cylinder small chamber barrel expansion stiffness, the metal pipe expansion stiffness, and the flexible hose expansion stiffness, respectively.

In this paper, we calculated the excavator boom cylinder system stiffness since the boom cylinder system is highly affected by the stiffness value. To calculate the boom cylinder system stiffness, main factors’ stiffness \(K_0, K_r, K_{c1}, K_{c2}, K_{y}, K_{h}\) were calculated as shown in Table 1. To obtain them, we used hydraulic system stiffness equations which were already established by Feng et al. [22]. And the boom cylinder specification which is the input data to calculate main factors’ stiffness is shown in Table 2.

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston diameter [m]</td>
<td>0.14</td>
</tr>
<tr>
<td>Piston rod diameter [m]</td>
<td>0.095</td>
</tr>
<tr>
<td>Piston rod length [m]</td>
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</tr>
<tr>
<td>Cylinder barrel outer diameter [m]</td>
<td>0.165</td>
</tr>
<tr>
<td>Cylinder barrel inner diameter [m]</td>
<td>0.14</td>
</tr>
<tr>
<td>Boom cylinder maximum stroke [m]</td>
<td>1.48</td>
</tr>
<tr>
<td>The metal pipe outer diameter [m]</td>
<td>0.035</td>
</tr>
<tr>
<td>The metal pipe inner diameter [m]</td>
<td>0.025</td>
</tr>
<tr>
<td>The length of metal pipe [m]</td>
<td>6.11</td>
</tr>
<tr>
<td>The flexible hose outer diameter [m]</td>
<td>0.005</td>
</tr>
<tr>
<td>The flexible hose inner diameter [m]</td>
<td>0.0025</td>
</tr>
<tr>
<td>The length of the flexible hose [m]</td>
<td>1.36</td>
</tr>
<tr>
<td>The hydraulic oil type</td>
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</tr>
<tr>
<td>The Elasticity modulus of steel [Pa]</td>
<td>2.06e+11</td>
</tr>
<tr>
<td>The Poisson ratio of steel</td>
<td>0.28</td>
</tr>
<tr>
<td>Working pressure [Mpa]</td>
<td>15</td>
</tr>
<tr>
<td>Boom stroke [m]</td>
<td>1</td>
</tr>
</tbody>
</table>

With the Eq. (18) and the main factors’ stiffness value, we can obtain the net stiffness \(K\) of the boom cylinder system as 2.55e7 N/m. With this value, we can review how much the system stiffness affects the system.

During the operation of making a surface with excavator, the \(\Delta P\) of boom cylinder chamber is less than 5 Mpa with 1.5 deg/s nominal angular velocity and for the arm the \(\Delta P\) is also less than 5 Mpa with 7 deg/s nominal angular velocity. And for the boom cylinder force deviation to act on the piston side with this pressure change will be 77 K. According to the formula of the spring stiffness \(K = \frac{F}{\Delta x}\), we can obtain the \(\Delta x = 0.003\ m\). When assuming the boom stroke is 1 m during pressure change, then the effectiveness from the boom cylinder stiffness will be 0.3 %. This is neglectable value, therefore the compressibility from hydraulic drive system stiffness will not be considered in this paper.

### 3.3. Dynamics of hydraulic actuator circuit

Fig. 3 shows the schematic of the open center-type hydraulic actuator circuit for the Volvo EC300D excavator. In addition, the dynamics of the open center-type hydraulic actuator circuit for the EC300D excavator was described in a previous study. The complete expressions can be found in Chang et al. [13].

Ignoring the compressibility of the system, from the continuity equation, the following relationships for junctions a, b can be obtained:

\[
\text{the flow at junction a, } Q_a = A_a \dot{l},
\]

(19)

\[
\text{the flow at junction b, } Q_b = A_b \dot{l},
\]

(19)

where \(\dot{l}\) is the actuator velocity. From the orifice flow equations, we have the following equation for the flow at each junction:

\[
Q = c_s A_{sp}(u) \sqrt{\Delta P},
\]

(20)

where \(\Delta P\) denotes the pressure drop in the orifice of the directional control valve; \(c_s\) is the coefficient of the flow equation; \(u\) is the displacement of each spool from each lever command, which is the excavator system input; and \(A_{sp}\) is the opening area of the directional control valve.

Eq. (20) is substituted into Eq. (19) and expanded with the Taylor series. This can be solved for \(P_s\) and \(P_e\), and the results can be substituted into Eq. (17). Then, we can obtain the joint torque, \(\tau\), as follows:

\[
\tau = J^T(\theta)(K_0 \dot{u} + K_1 \dot{l} + Q_{orc}).
\]

(21)
where $K_p$ and $K_v$ represent the coefficients for $u$ and $l$, respectively; $Q_{de}$ represents all the elements containing neither $u$ nor $l$.

From Eqs. (21) and (1), we can obtain the excavator manipulator dynamic model as follows:

$$u = (J^T(\dot{\theta}K_v)^{-1}(M(\dot{\theta})\dot{\theta} + G(\dot{\theta}) + Fr(\dot{\theta}, \dot{\theta}) + r_d) - (K_v)^{-1}K_l - (K_v)^{-1}Q_{de}.$$  

(22)

In addition, Eq. (22) can be rewritten by introducing $M_u$ and $N_u$ as follows:

$$u = M_u \dot{\theta} + N_u,$$  

(23)

where

$$M_u = (J^T(\dot{\theta}K_v)^{-1}M(\theta), N_u = (J^T(\dot{\theta}K_v)^{-1}(C(\dot{\theta}, \theta) + G(\dot{\theta}) + Fr(\dot{\theta}, \dot{\theta}) + r_d) - (K_v)^{-1}K_l - (K_v)^{-1}Q_{de}.$$  

4. Proposed new discrete time delay control with terminal sliding mode control scheme

4.1. Nonlinear desired error dynamics

In previous studies introduced and investigated the nonlinear desired error dynamics (NDDE) to decrease the position tracking error of TDC [17,20]. We propose the following NDE for the NDTDC:

$$e_{(a)} + K_p e_{(a)} + K_F \text{sgn}(e_{(a)})^a = 0,$$  

(24)

where

$$\text{sgn}(e) = [\text{sgn}(e_1), \cdots, \text{sgn}(e_n)]^T.$$  

All of the elements of $\alpha = \{\alpha_1, \alpha_2, \ldots, \alpha_n\}^T$ have positive values ($i.e.$., $\alpha_i > 0$ ($i = 1, 2, \ldots, n$)). Notice that the proposed NDE in Eq. (24) is also a decoupled error equation.

4.2. Discrete time delay control with NDE

Eq. (23) can be rewritten in the discrete form by integrating it over a sample period $L$ as follows: [15]

$$Lu_{k}= M_u(\dot{\theta}_{(k)} - \dot{\theta}_{(k-1)}) + \int_{(k-1)L}^{(k)L} N_u(k) dt.$$  

(25)

We can simplify Eq. (25) as follows:

$$Lu_{(k)} = M_u\dot{\theta}_{(k)} + N^1_{u(k)}.$$  

(26)

where

$$N^1_{u(k)} = -M_u\dot{\theta}_{(k-1)} + \int_{(k-1)L}^{(k)L} N_u(k) dt.$$  

The discrete dynamics shown in Eq. (26) can be rewritten by introducing a constant diagonal matrix $M_u$ as follows:

$$Lu_{(k)} = M_u\dot{\theta}_{(k)} + N^2_{u(k)},$$  

(27)

where

$$N^2_{u(k)} = (M_u - \bar{M}_u)\dot{\theta}_{(k)} + N^1_{u(k)}.$$  

Therefore, the NDTDC control input can be designed as follows:

$$Lu_{(k)} = \bar{M}_u u_{(k)} + \tilde{N}_u.$$  

(28)

where

$$u_{(k)} = K_p e_{(k)} + K_F \text{sgn}(e_{(k)})^a + \dot{\theta}_{d(k)}.$$  

In addition, based on the idea of time delay estimation, we can define $N(\dot{\theta}, \dot{\theta})_{(k)}$ as follows:

$$N(\dot{\theta}, \dot{\theta})_{(k)} = N^2(\dot{\theta}, \dot{\theta})_{(k-1)} = Lu_{(k-1)} - \bar{M}_u \dot{\theta}_{(k-L)}.$$  

(29)

Combining Eqs. (28) and (29) allows us to obtain the control model equation as follows:

$$Lu_{(k)} = Lu_{(k-L)} - \bar{M}_u \dot{\theta}_{(k-L)} + \tilde{M}_u u_{(k)}.$$  

(30)

Therefore, a new discrete time delay control with terminal sliding mode controller (NDTDC with TSNC) can be obtained as follows:

$$u_{(k)} = u_{(k-L)} + \bar{M}[K_p e_{(k)} + K_F \text{sgn}(e_{(k)})^a + \dot{\theta}_{d(k)} - \dot{\theta}_{(k-L)}],$$  

(31)

where $\bar{M} = L^{-1}\bar{M}_u$ is still a constant diagonal matrix as a control parameter. When $\alpha = 1$, the proposed controller becomes just a new discrete time delay controller. Additionally, if we assume that $\dot{\theta}_{(k-L)} \approx \dot{\theta}_{(k)}$, then we can obtain Wang’s control [15] as follows:

$$u_{(k)} = u_{(k-L)} + \bar{M}[1 + K_p e_{(k)} + K_F e_{(k)}].$$  

(32)
4.3. Properties of proposed controller

We can check the performance of the proposed controller, which includes the linear and nonlinear desired error dynamics according to the value of $\alpha_i$ $(i = 1, 2, \ldots, n)$. In a previous study, it was proven that if the following conditions are met for $\alpha_i$, a finite-time stable NDED can be obtained [18,20]:

$$0 < \alpha_i < 1.$$  (33)

With this condition, we can check the finite time convergence of the NDED, as shown in Fig. 4. And, the convergence speed of the NDED ($0 < \alpha_i < 1$) is faster than that of the linear DED ($\alpha_i = 1$), as shown in Fig. 4. Furthermore, we can see the convergence speed of NDED is getting faster by decreasing $\alpha_i$.

Therefore, we can see that the proposed controller, which is an NDTDC with NDED, has better performance than an NDTDC with linear DED such as Wang’s controller. In addition, the experimental results described in the following section support this conclusion.

4.4. Stability check of NDED

The stability check of the NDED can be done by considering the total energy of the NDED as a Lyapunov function candidate ($V$), as follows:

$$V_k = \sum_{i=1}^{n} \left[ \frac{1}{2} \mathbf{e}_i^T(k) + \frac{1}{2} K_{Fii} (\mathbf{e}_i(k) + 1)^{-1} \mathbf{e}_i^T(k) \right].$$  (34)

From Eq. (24), we can obtain the following:

$$(1 + K_{Di}) \mathbf{e}_i(k) + K_{Pi} \mathbf{e}_i(k)^T = 0,$$  (35)

$$(1 + K_{Di}) \mathbf{e}_i(k) + K_{Pi} \mathbf{e}_i(k)^T \mathbf{e}_i(k)^{\top} \mathbf{e}_i(k) = 0.$$  (36)

In addition, we can obtain the following from Eq. (36):

$$\dot{\mathbf{e}}_i(k) = - \frac{K_{Pi} \mathbf{e}_i(k)^T \mathbf{e}_i(k)}{(1 + K_{Di})}.$$  (37)

From Eqs. (35) and (37), we can obtain the time derivative of $V_k$ as follows:

$$\dot{V}_k = \sum_{i=1}^{n} \dot{\mathbf{e}}_i^T(k) \mathbf{e}_i(k) + \frac{1}{2} K_{Fii} \mathbf{e}_i(k)^T (1 + 1) \mathbf{e}_i(k)$$

$$= \sum_{i=1}^{n} \left[ - \frac{K_{Pi} \mathbf{e}_i(k)^T \mathbf{e}_i(k)}{(1 + K_{Di})} + \frac{1}{2} (1 + K_{Di}) \mathbf{e}_i^T(k) \right].$$  (38)

With positive $K_{Di}$ and $K_{Pi}$, the $V_k$ is $\leq 0$. Moreover, this implies that $(\mathbf{e}, \mathbf{e}) \equiv (0, 0)$ with Eq. (24). Therefore, the NDED is globally asymptotically stable according to LaSalle’s invariance principle.

4.5. Controller stability analysis

The TDE error exists due to the finite sampling time $L$. Substituting (28) into (27) and considering (29), then we can obtain TDE error as follows:

$$\mathbf{N}_k - \hat{\mathbf{N}}_k = \mathbf{N}_k - \mathbf{N}_k - \mathbf{M}_k \mathbf{u}_k - \hat{\theta}_k.$$  (39)

Now, define the TDE error $\mathbf{e}_k$ as follows:

$$\mathbf{e}_k = \mathbf{M}_k \mathbf{u}_k - \hat{\mathbf{N}}_k - \mathbf{M}_k \mathbf{u}_k - \hat{\theta}_k.$$  (40)

A combination of Eqs. (40) and (26)–(28) gives

$$\mathbf{e}_k = \mathbf{M}_k \mathbf{u}_k - \hat{\theta}_k.$$  (41)

Eq. (41) can be rewritten as follows with Eq. (40):

$$\mathbf{M}_k \mathbf{e}_k = [\mathbf{M}_k - \mathbf{M}_k] \mathbf{u}_k - [\mathbf{M}_k - \mathbf{M}_k] \hat{\theta}_k + (\mathbf{N}_k - \mathbf{N}_k) - \mathbf{M}_k \mathbf{u}_k - \mathbf{N}_k - \mathbf{N}_k.$$  (42)

Therefore, $\mathbf{e}$ is given by

$$\mathbf{e}_k = \mathbf{E} \mathbf{e}_k + \mathbf{E} \mathbf{u}_k + \mathbf{u}_k.$$  (43)

where,

$$\mathbf{E} = \mathbf{I} - \mathbf{M}_k^{-1} \mathbf{M}_k.$$  

$$\mathbf{u}_k = \mathbf{u}_k - \mathbf{u}_k.$$

$$\mathbf{n}_2 = \mathbf{M}_k^{-1} \mathbf{N}_k - \mathbf{N}_k.$$

In discrete time control, the value of $L$ is usually selected as the sampling period, which is a sufficiently small time. Therefore, $\mathbf{n}_1$ and $\mathbf{n}_2$ can be bounded. It has already been proven that the assumption of $\|\mathbf{E}\| < 1$ can be easily satisfied by the appropriate selection of $\mathbf{M}_k$ [13,20]. Therefore, the proposed controller is stable because $\lim_{k \to \infty} \mathbf{e}_k = 0$.

4.6. Compensator design

Fig. 5(a) shows the directional control valve’s dead band zone $\mathbf{b}$ which are the pilot pressure for cylinder extension and retraction, respectively. Therefore, the EPPR valve should be controlled from $\mathbf{b}$ which are matched with $\mathbf{b}$ and $\mathbf{b}$, respectively as shown in Fig. 5(b).

To compensate this, the control output from NDTDC with TSMC converted into the EPPR valve control signal through a dead band compensator as follows:

$$\mathbf{u}_k = \begin{cases} \mathbf{u}_k + \mathbf{b}_k', & \text{if } \mathbf{u}_k > 0 \\ \mathbf{u}_k - \mathbf{b}_k', & \text{if } \mathbf{u}_k < 0 \\ \mathbf{u}_k, & \text{if } \mathbf{u}_k = 0 \end{cases}.$$  (44)

where $\mathbf{u}_k$ is the output of dead band compensator, and $\mathbf{b}_k'$ and $\mathbf{b}_k'$ denote the dead band bounds of the $i$th input, respectively. The $\mathbf{u}_k$ is the output of NDTDC with TSMC.
5. Experimental study

5.1. Experimental setup

Fig. 6 shows the robotic excavator system that was built based on a Volvo EC300D excavator. A personal computer was used as a controller for the robotic excavator system. Position sensor was installed on each cylinder to obtain the feedback signal. To operate the directional control valve via the controller, EPPR valves were added on each side of the conventional directional control valves. And, a D/A converter (DAC) was installed between the controller and EPPR valves. This allowed the control inputs from the controller to be fed to the EPPR valves and control the directional control valves. The control algorithm which was deployed to the controller is executed at 100 Hz real time. For the communication between each unit, Controller Area Network (CAN) interface was used.

1. Controller: The Personal computer was used as a controller. All control algorithm was developed in the MATLAB simulink and compiled that to deploy in the controller with MATLAB xPC target toolbox. In the controller, CAN communication modules were added for communication with electric-joystick, sensors, and DAC. In normal operation, electric-joystick command by the operator will be transferred to EPPR valve control command. However, in autonomous mode, the EPPR valve control command will be generated by the controller itself.

2. EPPR Valve: To make the conventional excavator as into electro-hydraulic controlled system for robotic excavator, EPPR valves (Kawasaki) were installed on each side of conventional directional control valves.

3. DAC: The EPPR valve is controlled by currents. To generate control currents from the controller’s control signal, PLUS+1®SC micro-controller (Danfoss) was added. And PLUS+1®SC micro-controller support CAN communication, therefore this one is connected with the controller via CAN. All needed code which is to connect the controller and generating control currents were developed with PLUS+1®software.

4. Sensor: The position sensor is a string-type potentiometer (Clesesco, PT9CN) attached to boom, arm and bucket cylinder. This sensor is a heavy industrial type with CAN communication interface. Accuracy is ±0.1% and resolution is ±0.003% at full stroke which is 1400 cm.

5.2. Excavator kinematics

For the excavator kinematics, the DenavitHartenberg convention was used to make the coordinate system of the excavator, as shown in Fig. 2. To obtain the kinematics, each joint angle of the boom, arm, and bucket had to be calculated. The cylinder length information from the displacement sensors and geometric information of the excavator were used to calculate the joint angles. With the calculated joint angles, a homogeneous transformation matrix for the excavator could be calculated.

![Diagram of robotic excavator system](image-url)
as follows:

\[
\begin{align*}
H_4 &= H_2^2 H_3^3 H_4 \\
&= \begin{bmatrix} C_{123} & -S_{123} & 0 & L_1 C_1 + L_2 C_{12} + L_3 C_{123} \\ S_{123} & C_{123} & 0 & L_1 S_1 + L_2 S_{12} + L_3 S_{123} \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix},
\end{align*}
\]

where,

\[
\begin{align*}
H_2 &= \begin{bmatrix} C_1 & -S_1 & 0 & L_1 C_1 \\ S_1 & C_1 & 0 & L_1 S_1 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix},
\end{align*}
\]

\[
\begin{align*}
H_3 &= \begin{bmatrix} C_2 & -S_2 & 0 & L_2 C_2 \\ S_2 & C_2 & 0 & L_2 S_2 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix},
\end{align*}
\]

\[
\begin{align*}
H_4 &= \begin{bmatrix} C_3 & -S_3 & 0 & L_3 C_3 \\ S_3 & C_3 & 0 & L_3 S_3 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}.
\end{align*}
\]

Notice that \(C_i\) (\(i = 1, 2, 3\)) and \(S_i\) represent \(\cos(\theta_i)\) and \(\sin(\theta_i)\), respectively. \(C_{123}\) and \(S_{123}\) represent \(\cos(\theta_1 + \theta_2 + \theta_3)\) and \(\sin(\theta_1 + \theta_2 + \theta_3)\), respectively.

In the homogeneous transformation matrix, the bucket tip position \((O_{4x}, O_{4y})\) can be obtained as follows:

\[
O_{4x} = L_1 C_1 + L_2 C_{12} + L_3 C_{123},
\]

\[
O_{4y} = L_1 S_1 + L_2 S_{12} + L_3 S_{123}.
\]

5.3. Motion trajectory

To verify the proposed controller, a curved line was generated and tested. The desired bucket tip trajectory and velocity for this motion are shown in Figs. 8 and 9, respectively. We tested the inward and outward directions of the bucket tip motion so that we could verify the proposed controller performance in both directions of excavator movement. The bucket tip started to automatically move in the inward direction till the end point and then moved again in the outward direction till the start point.

To make this motion, the bucket tip trajectory was converted into each joint angle trajectory. For this motion, only the boom and arm joint angles were controlled. The generated desired joint angle trajectories are shown in Fig. 10. And, the desired joint angle velocity for boom and arm are shown in Fig. 11. The maximum joint angle velocity for the boom and arm are 1.47 deg/s and 6.6 deg/s, respectively.

5.4. Control system

Fig. 12 shows the control system block diagram for the robotic excavator. The generated joint desired trajectories are the reference commands for the proposed controller which is the NDTDC with TSMC. Each joint angle and angular velocity which are the control signal is calculated with a position sensor data. And, the controller output should be converted into the EPPR valve control command through a dead band compensator to compensate directional control valve dead band. Then, with the D/A converter, the EPPR valve control command is converted into an electrical signal to operate the EPPR valve. Finally, the controlled pilot pressure from an EPPR valve operates the directional control valve to cause individual movement in each link boom and arm.

5.5. Experimental results with different parameter sets

In the proposed controller, \(\dot{M}, K_p, K_v\), and \(\alpha\) are the tuning parameters. When we implemented this proposed controller on the robotic excavator, we tuned the parameters through the following procedure.
Experimental results were obtained with different parameter sets for the proposed control. The root mean square (RMS) error and maximum error for two joints (boom and arm) & the bucket tip position (z-axis and x-axis in excavator coordinate system) were the results, and are listed in Table 3. During the experiment, every case was done in the same condition.

1. Set $\alpha = 1/n$ and specify $K_D$, $K_P$ to make a stable linear DED. $K_D$ and $K_P$ are set as diag$[25, 25]$ and diag$[14, 14]$, respectively. (Pole located at $-25/14$ is stable.)

2. Tune only the diagonal gains of $M$ by increasing from a small value until the minimum tracking error is found. $M$ is tuned and set as diag$[0.07, 0.04]$ kg·m$^2$. 

(a) Desired bucket tip x-axis velocity  
(b) Desired bucket tip z-axis velocity

Fig. 9. Desired bucket tip velocity.

(a) Desired boom joint angle trajectory  
(b) Desired arm joint angle trajectory

Fig. 10. Desired joint angle trajectory.

(a) Desired boom joint angle velocity  
(b) Desired arm joint angle velocity

Fig. 11. Desired joint angle velocity.

Fig. 12. Control system block diagram.
3. Finally, tune the NDED by decreasing \( a_i \) (i = 1, 2) from 1 until the minimum tracking error is found. Experiment 1 to Experiment 5 (Table 1) show the tuning results. With the gradual reduction of \( a_i \), the RMS error decreased. However, we observed vibration from a certain level of \( a_i \) without a further decrease in the RMS error.

Fig. 13 shows the boom and arm joint tracking errors with various parameter sets. Experiment 1 (dash-dotted red line) involved the proposed controller without NDED, and Experiment 3 (solid black line) had the best results with the proposed controller with NDED by tuning \( a_i \) as \( \text{diag}(0.8, 0.8) \). In Experiment 5 (dashed blue line), we can observe vibration due to too much decrease in \( a_i \). This shows that if we properly tune the NDED by decreasing \( a_i \), the joint tracking performance can be improved.

The joint tracking error is directly reflected in the bucket tip trajectory performance. Fig. 14 shows the RMS error of the bucket tip position in the proposed controller’s results. Experiment 1 involved the proposed controller without NDED. We found better bucket tip trajectory tracking performances in Experiments 2 and 3, which used the proposed controller with NDED tuning by decreasing \( a_{1,2} \) to \( \text{diag}(0.9, 0.9) \) and \( \text{diag}(0.8, 0.8) \), respectively. Experiment 3 (\( a_{1,2} = \text{diag}(0.8, 0.8) \)) had the best results for the joint tracking performance. However, Experiments 4 and 5 showed worse results than Experiment 3 as a result of an excessive decrease in \( a_{1,2} \). In addition, the maximum error’s trend line was similar to the RMS errors.

After finding the best result by tuning the \( a_{1,2} \) value of the proposed controller, we compared this with Wang’s controller under the same condition. Table 4 lists the results of the comparison between Wang’s controller and the proposed controller (Experiment 3). The absolute maximum errors of Wang’s controller are 4.1380 cm (z-axis) and 1.9416 cm (x-axis). However, the proposed controller (Experiment 3)

---

**Table 3**

RMS error values with various parameter sets for proposed controller.

<table>
<thead>
<tr>
<th>No.</th>
<th>Kp</th>
<th>Kn</th>
<th>( \alpha_{1,2} )</th>
<th>RMS error</th>
<th>Max. error (Absolute value)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Boom angle</td>
<td>Arm angle</td>
</tr>
<tr>
<td>Experiment 1</td>
<td>25</td>
<td>14</td>
<td>1</td>
<td>0.9527°</td>
<td>0.1646°</td>
</tr>
<tr>
<td>Experiment 2</td>
<td>25</td>
<td>14</td>
<td>0.9</td>
<td>0.7882°</td>
<td>0.1361°</td>
</tr>
<tr>
<td>Experiment 3</td>
<td>25</td>
<td>14</td>
<td>0.8</td>
<td>0.628°</td>
<td>0.1137°</td>
</tr>
<tr>
<td>Experiment 4</td>
<td>25</td>
<td>14</td>
<td>0.7</td>
<td>0.791°</td>
<td>0.1336°</td>
</tr>
<tr>
<td>Experiment 5</td>
<td>25</td>
<td>14</td>
<td>0.6</td>
<td>0.9885°</td>
<td>0.1470°</td>
</tr>
</tbody>
</table>

**Table 4**

RMS error comparison between Wang’s controller and proposed controller.

<table>
<thead>
<tr>
<th>No.</th>
<th>Kp</th>
<th>Kn</th>
<th>( \alpha_{1,2} )</th>
<th>RMS Error</th>
<th>Max. Error (Absolute value)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Boom angle</td>
<td>Arm angle</td>
</tr>
<tr>
<td>Wang’s Controller</td>
<td>25</td>
<td>14</td>
<td>–</td>
<td>0.0968°</td>
<td>0.1576°</td>
</tr>
<tr>
<td>Experiment 3</td>
<td>25</td>
<td>14</td>
<td>0.8</td>
<td>0.628°</td>
<td>0.1137°</td>
</tr>
</tbody>
</table>

![Fig. 13. Comparison of joint tracking errors with various parameters for proposed controller.](image1)

(a) Comparison of boom joint tracking errors

(b) Comparison of arm joint tracking errors

![Fig. 14. Comparison of RMS error for bucket tip position error of proposed controller.](image2)

(a) Bucket tip z-axis RMS errors comparison

(b) Bucket tip x-axis RMS errors comparison

---
shows smaller maximum errors of 1.8188 cm (z-axis) and 1.4801 cm (x-axis). Moreover, the proposed controller (Experiment 3) shows a smaller RMS error. Fig. 15 shows the results of a comparison of the bucket tip trajectory tracking performances.

We also checked the bucket tip velocity for the proposed controller (Experiment 3) as shown in Fig. 16. We got maximum speed as about 0.47 m/sec for x-axis and 0.07 m/sec in z-axis.

6. Conclusions

A new NDTDC method for robotic excavator motion control proposed in this paper can easily compensate for the non-linearity of complex dynamics of hydraulic manipulators. And, this controller does not require the acceleration information needed by the traditional TDC, which makes it more suitable for practical applications as a result of fewer sensor requirements and less computational efforts. Furthermore, the injected terminal sliding error dynamics with tuned proper $\alpha$ results in high position tracking accuracy for robotic excavator manipulators compared with just the NDTDC. We implemented the proposed controller on a 30 ton Volvo hydraulic robotic excavator. The desired trajectory was a curved surface, and the achieved bucket tip position tracking accuracy was within less than 2 cm with tuned proper $\alpha$. Owing to the time-delay estimation of NDTDC, we could estimate the non-linearity of manipulator dynamics simply and effectively. However, it should be the future work to build compensator for high non-linearities from Coulomb friction and some uncertainty the hydraulic power circuit.

Conflict of interest

The authors confirm that there are no known conflicts of interest associated with this publication and there has been no significant financial support for this work that could have influenced its outcome.

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