Robust stabilizing control for oscillatory base manipulators by implicit Lyapunov method

Yufei Guo · Baolin Hou · Shengyue Xu · Ruilin Mei · Zhigang Wang · Van Thanh Huynh

Abstract Oscillatory base manipulator (OBM) is a kind of mechanical system suffering from unexpected base oscillations. The oscillations affect tremendously system stability. Various control methods have been explored, but most of them require measurement or prediction of the oscillations. This study is concerned with a novel OBM—the autoloader, which is used in modern, autonomous main battle tanks. The base oscillation of the autoloader is hard to be obtained in practice. Furthermore, control synthesis for autoloaders is complicated with intrinsic payload uncertainty and actuator saturation. To address these issues, a novel robust control scheme is proposed in this work relying on the implicit Lyapunov method. Moreover, a novel two-degree-of-freedom manipulator operating on a vibrating base is constructed to realize the proposed control. To the best of the authors’ knowledge, this is the first study considering both control and hardware implementation for the OBM-like autoloaders. Simulation and experiment results demonstrate that, although without prior information of the base oscillation, the proposed controller exhibits good robustness against the base oscillation and payload uncertainty.

Keywords Autoloaders · Oscillatory base manipulators · Implicit Lyapunov function · Robust control · Self-tuning PD control

List of symbols

| Symbol | Description |
|--------|-------------|
| $\theta_3$ | Angle of the revolving part of the autoloader |
| $\theta_{1x}$ | Roll base oscillation |
| $\theta_{1z}$ | Pitch base oscillation |
| $B_1$ | Oscillatory base |
| $B_2$ | Lifting part of the autoloader |
| $B_3$ | Revolving part of the autoloader |
| $C_2$ | Centroid of $B_2$ |
| $C_3$ | Centroid of $B_3$ |
| $g$ | Gravitational acceleration |
| $J_3$ | Inertia moment of the revolving part with respect to the joint |
| $L_1, L_3$ | Geometric parameters as shown in Fig. 3 |
| $m_2$ | Mass of the lifting part |
| $m_3$ | Mass of the revolving part (including the payload) |

Y. Guo (✉) · S. Xu · R. Mei · Z. Wang
Key Laboratory of Metallurgical Equipment and Control Technology of Ministry of Education, School of Machinery and Automation, Wuhan University of Science and Technology, Wuhan 430081, China
e-mail: guoyufei_1985@163.com

Y. Guo
Precision Manufacturing Institute, Wuhan University of Science and Technology, Wuhan 430081, China

B. Hou
School of Mechanical Engineering, Nanjing University of Science and Technology, Nanjing 210094, China

V. T. Huynh
School of Engineering, Deakin University, Geelong, VIC 3217, Australia
OXYZ Inertial coordinate, Y axis is along with the gravitational direction

\( oxz \) Non-inertial coordinate, namely the base fixed coordinate

\( y_1 \) Shake base oscillation

\( y_2 \) Position of the lifting part

1 Introduction

Main battle tank (MBT) plays an important role in the modern military field. Considerable efforts have been made to improve the performance of MBT, one of which is the use of the autoloader. The autoloader is used to automatically complete the loading task [1]. Compared with manual loaders, the autoloader is advantageously small, lightweight, and responsively fast. The use of the autoloader not only brings in a higher firing rate, but also improves the mobility of the tank.

However, controlling the autoloader is problematic because it usually operates in harsh working conditions [2]. When the MBT is driven off-road at high speed, its chassis—the mounting base of the autoloader, will be oscillated due to the uneven terrain, as shown in Fig. 1. The recoil force of the tank gun will also cause serious chassis oscillation. Chassis oscillation brings great challenges to the control of autoloaders. Existing autoloaders mostly adopt the traditional control method, which lead to low control accuracy and have poor robustness against base oscillation.

Controlling autoloaders subject to chassis oscillation can be considered as regulating oscillatory base manipulators (OBMs) [3,4]. Many mechanical systems belong to this category, such as macro/micro-manipulators, space manipulators, UVM (underwater vehicle manipulator), and offshore cranes. A macro/micro-manipulator consists of a rigid link, a flexible link, and a base whose vibration mainly comes from the structural vibration of the flexible link [5,6]. For the space manipulator, because it works in a weightless environment, the very small collision between the manipulator and the captured object will also lead to base oscillation [7,8]. For UVM and offshore cranes, the base oscillation is mainly vehicle vibration caused by ocean currents and waves [9–15].

Table 1 demonstrates classification methods and types of OBMs. Firstly, it can be classified according to origination of the oscillations, i.e., whether the oscillation comes from internal interaction (Type 1) or external disturbance (Type 2). Autoloader, offshore crane, and UVM belong to the first type, while space manipulator and macro/micro-manipulator belong to the second type. Secondly, OBMs can be classified according to prior information about the oscillation, i.e., whether information about the base oscillation can be obtained in advance. Except for autoloaders, base oscillations of other OBMs can all be measured or predicted in practice. Thirdly, OBMs can be classified according to their working task spaces, that is, in the inertial or non-inertial frame. Autoloader belongs to the latter type, and other OBMs belong to the former type. For simplicity, each type of OBM is named in this study, as shown in Table 1. Letters (A, B, C) and numbers (1, 2, 3) indicate the classification method and type, respectively. For instance, the autoloader is an A2-B2-C2 type OBM.

For an OBM control problem, a typical question is ‘how to counteract influence of the base oscillation.’ For A1-type OBM, the base oscillation can be suppressed directly, and various methods have been developed [16–21]. For example, Yang et al. proposed an adaptive suppression control for macro/micro-manipulator, in which the flexible vibration is suppressed by a bonded fiber composite actuator [16]. Wongratanaphn and Cole used impedance control to damp out the flexible vibration [17]. For space manipulator, Xu et al. adopted adaptive reactionless control to ensure the attitude stability of the base (spacecraft) [18]. Wang and Xie used the velocity estimation (or velocity reference) of the spacecraft as dynamic compensation of base oscillation [19]. For A2-B1 OBM, the nonlinearity caused by base oscillation is mainly compensated by its measurement or prediction [22–35]. For example, Londhe et al. developed an estimation disturbance compensator for UVM to suppress external disturbances caused by ocean currents [23]. Tang et al. developed a dynamic feedforward compensator for UVM to suppress external disturbances caused by ocean currents [26]. For offshore crane, Kuchler et al. proposed a prediction algorithm for active compensation of ship vertical motion [32]. Schaub developed a novel sensor based on inertial measurement unit and then proposed two active compensation strategies for ship oscillation [33]. Toda and Sato proposed robust control to compensate the ship oscillation, assuming that the frequency range of the base oscillation is known in advance [34,35].
However, the above-mentioned methods for OBM regulation are not suitable for autoloaders (A2-B2-C2-type OBM). The reasons are as follows.

(i) For autoloaders, since the base oscillation comes from external excitation, it cannot be directly suppressed or eliminated.

(ii) The base oscillation caused by uneven terrain and gun recoil is difficult to be measured or predicted in practice.

(iii) Due to working in the non-inertial frame, the base oscillation of the autoloader will produce additional nonlinear coupling forces, such as centrifugal force and Coriolis force. This will also bring difficulties to the autoloader’s control.

(iv) Most of the above methods for OBMs do not consider payload uncertainty. This uncertainty, however, is inevitable for the autoloader because it usually operates objects with varying masses.

(v) The saturation problem should also be considered in the control of autoloaders.

To address the control problem of the autoloader, this study has made the following contributions:

(i) An uncertain dynamic model of the autoloader subject to the base oscillation is established, in which the base-oscillation-induced nonlinearity is treated as an unknown external disturbance. The established model is a general model, which is also applicable for other OBMs.

(ii) A novel stabilizing control method of the above uncertain dynamic model is proposed, which maintains accurately positioning control of the autoloader without prior information about the base oscillation.

(iii) Both simulation and hardware implementation are established to evaluate performance of the proposed control. As far as we know, this is the first experimental study on the control of autoloaders.

The proposed control is a Lyapunov function-based control [36,37]. Compared with existing control methods, it has the following merits:

(i) The Lyapunov function is easy to be obtained and solved. However, other Lyapunov controls often accompany with difficulties in the design of the

| Classification method | Type | Macro/micro-manipulator | Space-manipulator | UVMs | Offshore cranes | Autoloader of MBT |
|-----------------------|------|-------------------------|-------------------|------|----------------|-------------------|
| A-origination of oscillations | 1-internal interactions | ✓ | ✓ | ✓ | ✓ | ✓ |
|                        | 2-external disturbances | ✓ | ✓ | ✓ | ✓ | ✓ |
| B-oscillations available | 1-yes | ✓ | ✓ | ✓ | ✓ | ✓ |
|                        | 2-not | ✓ | ✓ | ✓ | ✓ | ✓ |
| C-working task space   | 1-inertial frame | ✓ | ✓ | ✓ | ✓ | ✓ |
|                        | 2-non-inertial frame | ✓ | ✓ | ✓ | ✓ | ✓ |
Lyapunov function, especially for complex systems.

(ii) The proposed control has a simple structure and is easy to be implemented in practice. It is actually a varying-gain PD control, and the gain is automatically tuned according to the system state. However, other robust control methods (such as $H_\infty$ control and high-order sliding mode control) are difficult to be implemented because of their high-order structures.

(iii) It also has a unique advantage that the control vector is always norm-bounded.

The organization of the paper is as follows. In Sect. 2, the autoloader system’s configuration, as well as the uncertain dynamics model, is presented with some analysis. Section 3 presents development of the control design and stability analysis. Section 4 provides the experimental implementation and results. Section 5 concludes the paper.

2 Preliminaries

In this section, a full-scale autoloader system is proposed, and its uncertain dynamic model under the influence of real base oscillation is established. Notations of system parameters and frames used for the development are also mentioned in the sequel.

2.1 System configuration

A novel 2-DOF (degree of freedom) autoloader was designed in our previous research, as presented in Fig. 2.

As shown, the autoloader mainly consists of three parts: a transfer device, a lifting device, and a mounted frame. The automatically loading process of the autoloader has four steps. Firstly, the transfer device grasps an object, e.g., the ammunition object. Then, the lifting device hoists the object to a pre-specified position. Thirdly, the transfer device flips the object to make it aligned with a succeeding device. Finally, the object is loaded into the succeeding device. From the perspective of positioning control, only the motions of the second and third steps need to be taken into account. Moreover, the base oscillation also needs to be considered in the control design. Therefore, the autoloader with the base can be simplified as a 3-link apparatus. In addition, the base mainly suffers from three directions of oscillations in practice, as shown in the figure.

2.2 Dynamic model

Based on the above analysis, a simplified model for the autoloader with the oscillatory base is established, as shown in Fig. 3.

The dynamics of the simplified model can be derived using the second Lagrange equation. Firstly, for simplicity, the base oscillation is divided into three parts: the pitch, shake, and roll oscillations, respectively. Furthermore, as mentioned before, the tank chassis has much greater inertia than the autoloader, so the base oscillation’s effect is regarded as the uncertain perturbation of the autoloader, but the autoloader’s reaction on the base is ignored. Finally, regarding $q = [y_2, \theta_3]^T$ as the system’s state variables, the uncertain dynamical model of the autoloader with base oscillation is established as follows:

$$H(q)\ddot{q} + C(q, \dot{q})\dot{q} + G(q) = S + U$$ (1)

where $H(q)$ denotes the inertia matrix, the Coriolis/centripetal force term is $C(q, \dot{q})\dot{q}$, the gravity force is presented using $G(q)$, the control needed to be constructed is $U$, and $S$ denotes the uncertain perturbation from base oscillations. (Payload uncertainty could also be included into this term.) Moreover, each term of Eq.
Robust stabilizing control for oscillatory base manipulators

Fig. 3 Simplified model of the autoloader with oscillatory base

(1) is as follows:

\[
H(q) = \begin{bmatrix}
m_2 + m_3 & m_3L_3\cos\theta_3 \\
m_3L_3\cos\theta_3 & m_3L_3^2 + J_3
\end{bmatrix}
\]

\[
C(q, \dot{q}) = \begin{bmatrix}
-m_3L_3\sin(\theta_3)\dot{\theta}_3^2 \\
0
\end{bmatrix}
\]

\[
G(q) = \begin{bmatrix}
(m_2 + m_3)g \\
m_3gL_3\cos\theta_3
\end{bmatrix}, \quad U = \begin{bmatrix}
U_1 \\
U_2
\end{bmatrix}
\]

where \(U_1\) and \(U_2\) are control force/torque of the lifting part and revolving part, respectively. \(S = [S_{1z}, S_{2z}]^T, [S_{1p}, S_{2p}]^T\) and \([S_{1r}, S_{2r}]^T\) denote uncertain perturbation caused by the shake, pitch, and roll oscillations, respectively.

\[
S_{1z} = -(m_2 + m_3)\ddot{y}_1
\]

\[
S_{2z} = -m_3L_3\cos\theta_3\ddot{y}_1
\]

\[
S_{1p} = -(m_2L_1 + m_3L_1 + m_3L_3\cos\theta_3)\ddot{\theta}_{1z}
\]

\[
-2m_3L_3\sin\theta_3\dot{\theta}_{1z}\dot{\theta}_3
\]

\[
-(m_2y_2 + m_3y_2 + m_3L_3\sin\theta_3)\ddot{\theta}_{1z}
\]

\[
-((m_2 + m_3)g\cos\theta_1z - (m_2 + m_3)g)
\]

\[
S_{2p} = -(J_3 + m_3L_3^2 + m_3L_1L_3\cos\theta_3)
\]

\[
+m_3L_3y_2\sin\theta_3\dot{\theta}_{1z}
\]

\[+
2m_3L_3\sin\theta_3\dot{\theta}_{1z}\ddot{y}_2 + (m_3L_1L_3\sin\theta_3
\]

\[-m_3L_3y_2\cos\theta_3)\dot{\theta}_{1z}^2
\]

\[-(m_3gL_3\cos(\theta_{1z} + \theta_3) - m_3gL_3\cos\theta_3)
\]

\[
S_{1r} = (m_2 + m_3)y_2\dot{\theta}_{1x}^2 + m_3L_3\sin\theta_3\dot{\theta}_{1x}^2
\]

\[+(m_2 + m_3)g\cos\theta_2
\]

\[-(m_2 + m_3)g
\]

\[
S_{2r} = m_3y_2\dot{\theta}_{1x}^2 + m_3L_3\sin\theta_3\dot{\theta}_{1x}^2
\]

\[-m_3gL_3\cos\theta_{1x,\cos}\theta_3
\]

\[+m_3gL_3\cos\theta_3
\]

Generally, the dynamical model described by Eq. (1) possesses peculiar properties of robotic manipulators [38]. Below are the properties that will be used in the development of the main results of this paper.

**Property 1** \(H(q)\) is a symmetrical matrix, and it is uniformly bounded from above and below, i.e.,

\[
m^2 \leq \|z^TH(q)z\| \leq M^2
\]

where \(m\) and \(M\) are positive constants, and \(z\) is an arbitrary vector. In this paper, \(\|\cdot\|\) denotes the vector or matrix’s Euclidean norm.

**Property 2** \(H(q)\) also satisfies the condition that its partial derivative is uniformly norm bounded, i.e.,

\[
\left\| \frac{\partial H(q)}{\partial q_i} \right\| \leq D, \quad D > 0
\]

**Property 3** The following matrix:

\[
N(q, \dot{q}) = \dot{H}(q) - 2C(q, \dot{q})
\]

\[,\text{ is an antisymmetric matrix. Specially, for any vector} z, \text{ there is}
\]

\[
z^TN(q, \dot{q})z = 0
\]

**Property 4** In practice, \(S\) satisfies the following bounded conditions:

\[
\|S\| \leq S_0, \quad S_0 > 0
\]

**Property 5** The control input should also be bounded in norm in practice. That means

\[
\|U\| \leq U_0, \quad U_0 > 0
\]
Property 6 There exists a positive constant $g_0$ such that
\[
\|\frac{\partial P(q)}{\partial q}\| = \|G(q)\| \leq g_0
\] (7)
where $P(q)$ denotes gravitational energy of the system.

Remark 1 The control problem of this study can be stated as: Construct a control $U$ for system (1) affected by unpredictable perturbations $S$ of Eq. (5) so that the closed-loop controlled system is robustly stabilized and simultaneously satisfies the given constraint shown in Eq. (6).

3 Main development

In this section, a robust control based on the implicit Lyapunov function is proposed to solve the above-stated problem. Moreover, for stability analysis of the closed-loop, uncertain system is carried out. In addition, for the purpose of comparison, a traditional constant PD controller with a gravity compensator is also presented.

3.1 Constant PD control

For system (1), we can introduce a controller as
\[
U = \tau + G(q)
\] (8)
where $\tau = -k_p(q_d - q) - k_d q$. The $q_d$ is the setting reference point. For simplicity, we set $q_d = 0$ here. Then, the system (1) can be rewritten as
\[
H(q)\ddot{q} + C(q, \dot{q})\dot{q} = S + \tau
\] (9)

Theorem 1 Consider the closed-loop system (9), if $S = 0$, there exists a PD control $\tau$ with constant gains ($k_d > 0$ and $k_p > 0$) can realize the stabilization of the system.

Proof Let’s choose a candidate Lyapunov function as the following:
\[
V = \frac{1}{2} \dot{q}^T H(q) \dot{q} + \frac{1}{2} \dot{q}^T k_p q
\] (10)
Since $H(q)$ and $k_p$ are positive definite, we can conclude that the Lyapunov function $V$ is also positive definite. Then, differentiating the function $V$ along the closed-loop system trajectory, we can obtain
\[
\dot{V} = -\frac{1}{2} \dot{q}^T k_d \dot{q} \leq 0
\] (11)
Obviously, the derivative of Lyapunov function is negative at nonzero state. Hence, according to LaSalle’s theorem, it can be concluded that above closed-loop system is asymptotically stable at $(q, \dot{q}) = (q_d, 0)$.

The proof is completed. $\square$

3.2 Implicit Lyapunov control

If $S \neq 0$, the above control is no long valid. We introduce a novel varying-gain PD control $\tau$ [36], the gains of which are as follows
\[
k_d = \alpha H, \quad k_p = \beta I
\] (12)
\[
\alpha = \frac{3U_0^2}{8V}, \quad \beta = \frac{\sqrt{\beta}}{M}
\] (13)
\[
V = \frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \beta q^2 + \frac{1}{2} \alpha q^T H q
\] (14)
where $V$ is the candidate Lyapunov function, which is the implicit function of the state variables. In addition, $I$ is the identity matrix.

Remark 2 $\tau$ together with the gravity compensation term constitutes the robust control of the system (1). However, in practice, the gravity compensation usually requires extra real-time calculations. Furthermore, since the gravity term $G(q)$ is also norm bounded, it can be treated as part of $S$. Then, for (9), we obtain $U = \tau, S = S - G(q)$.

Theorem 2 Consider the closed-loop system (9) with $S \neq 0$, there exists a robust controller, whose gains $k_d$ and $k_p$ (12)(13) are functions of $V$ (14) and can realize robust stabilization of the system.

Proof We will estimate the bound of $V$ firstly. By Eq. (13), we have
\[
\|\alpha q^T H q\| \leq \alpha \sqrt{(\dot{q}^T H \dot{q}) q^T H q}
\leq \frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \alpha^2 q^T H q
\leq \frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \beta q^2
\] (15)
Combining Eq. (14) and inequality (15), we further obtain the bound of $V$ as follows
\[
\frac{1}{4} (\dot{q}^T H \dot{q} + \beta q^2) \leq V \leq \frac{3}{4} (\dot{q}^T H \dot{q} + \beta q^2)
\]  
(16)

Obviously, as a real variable, $V$ is global positive definite for any $(q, \dot{q}) \neq (0, 0)$.

Secondly, we will evaluate the derivative of the Lyapunov function. By differentiating the function $V$, we obtain
\[
\dot{V} = \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{1}{2} \dot{q}^T H \dot{q} + \beta q^2 + \frac{1}{2} \dot{q}^T H \dot{q}
\]
(17)

with
\[
\dot{\alpha} = -\frac{\alpha}{2V} \dot{V}, \quad \dot{\beta} = -\frac{\beta}{V} \dot{V}
\]  
(18)

Then, rewrite Eq. (11) as
\[
\ddot{q} = H^{-1} (-k \dot{q} - k_\theta \dot{q} - C \dot{q} + S)
\]
(19)

By substituting Eqs. (12), (18) and (19) into Eq. (17), we obtain
\[
\dot{V} \left(1 + \frac{1}{2} \frac{\beta}{\sqrt{V}} q^2 + \frac{1}{4} \frac{\alpha}{V} \dot{q}^T H \dot{q}\right) = -\alpha \left(\frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{\beta}{2} q^2 \right)
\]
(20)

According to Property 3, we further rewrite Eq. (20) as
\[
\dot{V} \left(1 + \frac{1}{2} \frac{\beta}{\sqrt{V}} q^2 + \frac{1}{4} \frac{\alpha}{V} \dot{q}^T H \dot{q}\right) = -\alpha \left(\frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{\beta}{2} q^2 \right)
\]
(21)

Moreover, due to the vector of the Coriolis/centripetal forces $C \dot{q}$ satisfies the following condition [38]
\[
C \dot{q} = \dot{H} \dot{q} - \frac{1}{2} \left(\frac{\partial}{\partial \dot{q}} (\dot{q}^T H \dot{q})\right)^T
\]
(22)

Equation (21) can be further rewritten as
\[
\dot{V} \left(1 + \frac{1}{2} \frac{\beta}{\sqrt{V}} q^2 + \frac{1}{4} \frac{\alpha}{V} \dot{q}^T H \dot{q}\right) = -\alpha \left(\frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{\beta}{2} q^2 \right)
\]
(23)

For the first term of Eq. (23), by Cauchy inequality and inequality (3), we have
\[
\alpha \left(\frac{1}{2} \dot{q}^T H \dot{q} + \frac{1}{2} \alpha \dot{q}^T H \dot{q} + \frac{\beta}{2} q^2 \right)
\]
(24)

For the second term of Eq. (23), by Cauchy inequality and inequality (3), we have
\[
\frac{1}{4} \alpha \left(\frac{\partial}{\partial \dot{q}} (\dot{q}^T H \dot{q})\right)^T q^T
\]
(25)

Moreover, according to Eq. (16) and inequality (2), we obtain
\[
q^2 \leq \frac{1}{\beta} (\dot{q}^T H \dot{q} + \beta q^2) \leq \frac{4}{\alpha^2 M} V
\]
(26)

\[
q^2 \leq \frac{1}{m} (\dot{q}^T H \dot{q}) \leq \frac{1}{m} (\dot{q}^T H \dot{q} + \beta q^2) \leq \frac{4}{m} V
\]
(27)

By inequalities (26) and (27), we can rewrite inequality (25) as
\[
\frac{1}{4} \alpha \left(\frac{\partial}{\partial \dot{q}} (\dot{q}^T H \dot{q})\right)^T q^T \leq \frac{2\sqrt{2} D V^{3/2}}{m\sqrt{M}}
\]
(28)

For the third term of Eq. (23), by Cauchy inequality and inequalities (2) and (5), we obtain
\[
\|S^T (\dot{q} + \frac{1}{2} \alpha \dot{q})\| \leq S_0 \sqrt{\|\dot{q}^T + \frac{1}{2} \alpha \dot{q}^2\|^2}
\]
(29)

\[
\leq S_0 \sqrt{\frac{5}{4} (q^2 + \alpha^2 q^2)}
\]

\[
\leq S_0 \sqrt{\frac{5}{4} \left(\frac{1}{m} \dot{q}^T H \dot{q} + \frac{\beta}{M} q^2\right)}
\]

\[
\leq S_0 \sqrt{\frac{5}{4} \left(\dot{q}^T H \dot{q} + \beta q^2\right)} \leq S_0 \sqrt{\frac{5}{m} V}
\]
Substituting Eq. (24), inequalities (28) (29) into (23), we introduce the notation $B$, and then, we can obtain
\begin{equation}
B \dot{V} \leq -\sqrt{3}U_0 V^{1/2} + \frac{2 \sqrt{2} D}{m \sqrt{M}} V^{3/2} + S_0 \sqrt{\frac{5}{m}} V^{1/2}
\end{equation}
(30)

where
\begin{equation}
B = 1 + \frac{1}{2} \frac{\beta}{V} q^2 + \frac{1}{4} \frac{\alpha}{V} q^T H q.
\end{equation}
(31)
For $B$, by Eqs. (14) and (30), we have
\begin{equation}
B = \frac{1}{2} \frac{\alpha}{\dot{q}} H \dot{q} + \beta q^2 + \frac{3}{4} \alpha^q T H q)
\end{equation}
(32)
Moreover, according to Eq. (15), we have
\begin{equation}
\alpha \dot{q}^T H q \geq -\frac{1}{2} \dot{q}^T H \dot{q} - \frac{1}{2} \beta q^2
\end{equation}
(33)
Substituting inequality (33) into Eq. (32), we can rewrite $B$ as
\begin{equation}
B \geq \frac{1}{2} \left( \frac{1}{2} \dot{q}^T H \dot{q} + \beta q^2 + \frac{3}{4} \left( -\frac{1}{2} \dot{q}^T H \dot{q} - \frac{1}{2} \beta q^2 \right) \right)
= \frac{1}{2} \left( \frac{1}{2} \dot{q}^T H \dot{q} + \frac{3}{4} \beta q^2 \right) \geq 0
\end{equation}
(34)
It is clear that $B$ is positive for any $(q, \dot{q}) \neq (0, 0)$. Hence, the inequality (31) can be rewritten as
\begin{equation}
\dot{V} \leq -\frac{1}{B} \left( \frac{\sqrt{3} U_0}{2 \sqrt{2} M} - \frac{2 \sqrt{2} D}{m \sqrt{M}} V - S_0 \sqrt{\frac{5}{m}} \right) V^{1/2}
\end{equation}
(35)
If $U_0$ and $S_0$ are set to satisfy the following inequality, the global negative definiteness of $\dot{V}$ can be obtained.
\begin{equation}
U_0 > 2 \sqrt{\frac{10 M}{3 m} S_0} + \frac{8 D}{\sqrt{3 m}} V(q_0, \dot{q}_0),
\end{equation}
(36)
where $(q_0, \dot{q}_0)$ is the initial state of the system.

The proof is completed.

\textbf{Remark 3} As mentioned above, the control input $U$ is the function of $V$, and $V$ is the implicit function of system state variables. Therefore, there is a problem in practical application, that is, how to solve $V$.

By substituting Eqs. (13) into (14), we obtain
\begin{equation}
16V^2 = 8\dot{q}^T H \dot{q} V + 3U_0^2 q^2 + \frac{2 \sqrt{6} U_0}{\sqrt{M}} \dot{q}^T H q V^{1/2}
\end{equation}
(40)
Introducing the following notations
\begin{equation}
x = V^{1/2}, \quad \xi = 2 \sqrt{2} \dot{q}^T H \dot{q},
\end{equation}
(41)
\begin{equation}
\eta = \frac{2 \sqrt{6} U_0}{\sqrt{M}} \dot{q}^T H q, \quad \gamma = \sqrt{3} U_0 \| q \|,
\end{equation}
we can rewrite Eq.(40) as
\begin{equation}
16 \xi^2 - \xi^2 \gamma^2 - \eta x - \gamma^2 = 0
\end{equation}
(42)
\textbf{Theorem 4} [37]. For the equation (42), if the coefficients satisfy
\begin{equation}
\xi^2 + \gamma^2 > 0, \quad |\eta| \leq |\xi \gamma|,
\end{equation}
\begin{equation}
it must have a unique positive real root.
\end{equation}
It is clear that for any $(q, \dot{q}) \neq (0, 0)$, we have
\begin{equation}
\xi^2 + \gamma^2 = 8\dot{q}^T H \dot{q} + 3U_0^2 q^2 > 0,
\end{equation}
\begin{equation}
\eta^2 = 24 \frac{U_0^2}{M} (q^T H q)^2 \leq 24 U_0^2 (q^T H q)^2 = \xi^2 \gamma^2
\end{equation}
Obviously, the implicit function (40) satisfies the condition (43). Therefore, we can obtain a unique positive real value of $V$ by solving equation (42). Then, $k_p, k_d$, and $U$ can also be obtained in turn.

\textbf{Remark 4} The proposed control can be used independently or in combination with the gravity compensator.

\textbf{Remark 5} Property 5 only considers the constraint of the norm of $U$. However, in the practical saturation problem, engineers are more concerned with the constraint of the input force/torque $U_1$ and $U_2$. The saturation function (block) can be added after $U_1$ and $U_2$ to maintain this constraint in practice.
Therefore, the block diagram of the proposed control can be depicted as shown in Fig. 4. The \( (q_{1d}, \dot{q}_{1d}) \) is the setting point, which is set as \((0, 0)\) in this study. IL control means the implicit Lyapunov control. The optional blocks (gravity compensator and saturation block) are represented by dot curves. In addition, \( V \) can be obtained by solving the implicit function using the Newton iterative method. \( V_0 \) is the initial value of \( V \), which is set as \( V_0 = 1 \) in this study. \( V_t \) is the real-time value of \( V \), and \( V_{t-1} \) is the value of the previous time. In the iterative process, the value of the previous time of \( V \) is considered as the initial approximation.

4 Performance evaluation

To evaluate the performance of the proposed control, simulations for the full-scale autoloader (as shown in Fig. 2) were carried out. To further verify the control’s performance, a hardware system was developed and experiments were carried out.

4.1 Simulations and result analysis

Three groups of simulations are carried out, which are named Group s1, Group s2, and Group s3, respectively, as shown in Table 2. In particular, Group s1 is used to evaluate the proposed control’s robustness against base oscillations. The base oscillation data are from our previous research. Group s2 further evaluates its robustness to payload uncertainty. Group s3 is a contrast group, where various control technologies/methods are considered.

IL (IL+G) denotes the proposed implicit Lyapunov control (+G means gravity compensator). The control parameters are \( U_0 = 1400, M = 50.19 \). PD+G means constant-gain PD control with gravity compensator whose control gains are \( k_p = 1674.0, k_d = 289.86 \).

Table 2 Simulation groups

| Group no. | Case no. | Control method | Base oscillation | Payload uncertainty | Saturation function |
|-----------|----------|----------------|------------------|---------------------|-------------------|
| Group s1  | Case s1.1 | IL             | Shake            | N/A                 | N/A               |
|           | Case s1.2 | IL             | Pitch            | N/A                 | N/A               |
|           | Case s1.3 | IL             | Roll             | N/A                 | N/A               |
|           | Case s1.4 | IL             | N/A              | N/A                 | N/A               |
| Group s2  | Case s2.1 | IL             | Shake            | +20%                | N/A               |
|           | Case s2.2 | IL             | Shake            | -20%                | N/A               |
|           | Case s2.3 | IL             | Shake            | +20%                | Y                 |
| Group s3  | Case s3.1 | IL             | Shake            | +20%                | N/A               |
|           | Case s3.2 | IL             | Shake            | +20%                | Y                 |
|           | Case s3.3 | IL+G           | Shake            | +20%                | N/A               |
|           | Case s3.4 | IL+G           | Shake            | +20%                | Y                 |
|           | Case s3.5 | PD+G           | Shake            | +20%                | N/A               |
|           | Case s3.6 | PD+G           | N/A              | N/A                 | N/A               |
Moreover, for simplicity, only shake oscillation is considered in Group s2 and Group s3. In addition, the system parameters are as follows: $m_2 = 10$ kg, $m_3 = 40$ kg, $L_3 = 0.074$ m, $J_3 = 5$ kg $\cdot$ m$^2$, $L_1 = 0.5$ m. The gravitational acceleration is $g = 9.8$ m/s$^2$. The initial state vector is $(0.5, 0, -0.524, 0)$, and the terminal vector state is set as $(0, 0, 0, 0)$.

Group s1: This group includes four simulation cases, namely case s1.1, case s1.2, case s1.3, and case s1.4. The first three cases consider the shake, pitch, and roll base oscillation, respectively. The fourth case is the comparative case, without considering the base oscillation. In addition, neither gravity compensator nor saturation function is used in this group.

Simulation results are presented in Figs. 5, 6 and 7. The black solid line corresponds to case s1.1, the red dash line corresponds to case s1.2, the blue dash-dot line corresponds to case s1.3, and the orange dot line corresponds to case s1.4.

Figures 5 and 6 depict the time history of the autoloader’s motion. Obviously, in all cases, the controller successfully steers the autoloader from the given initial state to the terminal state. However, the setting times are different, which are 0.810s, 0.713s, 0.689s, and 0.689s for cases s1.1, s1.2, s1.3, and s1.4, respectively. In comparison, it takes the most time under the shake oscillation case and the least time under roll oscillation case. The motion response under the roll oscillation almost overlaps with that without oscillation. It can also be seen that the velocity curve of the lifting part under shake oscillation presents a non-smooth characteristic. To sum up, we can conclude that the shake oscillation has the greatest impact on the motion of the autoloader, followed by pitch oscillation and roll oscillation.

Figure 7 depicts the control response of this group. The first and second subfigures describe the control force of the lifting part $U_1$ and the control torque of the rotating part $U_2$, respectively. Units of $U_1$ and $U_2$
are $N$ and $Nm$, respectively. The third subfigure plots the time history of the control vector’s norm, i.e., $\|U\|$. It is obvious that the control input is always bounded in norm. The lower subfigure is the time history of the Lyapunov function value, which decreases monotonically in the case of roll and no oscillation, but non-monotonically in the case of shake and pitch oscillation. The reason may be that the estimation of $U_0$ and $S_0$ is not conservative enough. Nevertheless, the system motion responses still show that the proposed control has good robustness against base oscillations.

**Group s2**: This group includes three other simulation cases, namely cases s2.1, s2.2, and s2.3, which correspond to the cases of no payload uncertainty, positive payload uncertainty, and negative payload uncertainty, respectively. In all cases, shake base oscillation is considered. Case s2.1 is actually case s1.1 in Group s1, which is used as a control case in this group. The positive and negative payload uncertainties refer to the increase and decrease in inertia parameters of the revolving part by 20%.

Simulation results are shown in Figs. 8, 9 and 10. Similarly, the black solid line, the red dash line, and the blue dash-dot line correspond to cases s2.1, s2.2, and s2.3, respectively.

The motion response of the lifting part and revolving part is plotted in Figs. 8 and 9, respectively. It can be seen that the proposed control successfully leads to the robust stabilization of the system. The setting times in the three cases are 0.810s, 0.999s, and 0.787s, respectively. Obviously, the increase in payload uncertainty will lead to the increase in setting time. It can also be found that the greater the payload uncertainty, the greater the overshoot.

The control response is shown in Fig. 10. The upper two subfigures illustrate $U_1$ and $U_2$, respectively, while the lower two show the time history of $\|U\|$ and $V$, respectively. It is clear that the control inputs under no payload uncertainty and negative payload uncertainty are always bounded in norm, but that under positive payload uncertainty is not. In all cases, the Lyapunov function curves are non-monotonically decreasing. The reason is similar to Group s1. Nevertheless, the motion response proves that the proposed control is robust not only to base oscillation but also to payload uncertainty.

**Group s3**: This group includes six simulation cases. In the first five cases, both the base oscillation and positive payload uncertainty are considered, but different control strategies are used. Case s3.1 is actually case s2.2 of Group S2, where the IL control is used. In case s3.2, the IL control with saturation function (added on $U_1$ and $U_2$) is used. The saturation value is set as ±1000. Case s3.3 uses the IL control with a gravity compensator. Case s3.4 uses the IL control with a gravity compensator and saturation functions. Case s3.5 and case s3.6 use traditional constant-gain PD control with gravity compensator. In addition, case s3.6 is the control case of case s3.5, in which neither base oscillation nor payload uncertainty is considered.
Simulation results are shown in Figs. 11, 12 and 13. The black solid line and black dash-dot line correspond to cases s3.1 and s3.2, respectively. The red solid line and red dash-dot line correspond to cases s3.3 and s3.4, respectively. The blue solid line and blue dash-dot line correspond to cases s3.5 and s3.6, respectively.

Figures 11 and 12 show the motion response of the lifting and revolving parts, respectively. Obviously, four IL-based controls (with or without gravity compensator/saturation function) have successfully realized the stabilization of the autoloader. The setting times are 0.999s, 1.068s, 1.060s, 1.560s, and 1.222s for cases s3.1, s3.2, s3.3 and s3.4, respectively. It is clear that the control takes more time when the control force/torque is restricted by the saturation function. It can also be seen that the constant-gain PD control is unable to stabilize the system when considering base oscillation and payload uncertainty.

Figure 13 describes the control response. The upper two subfigures describe the control force/torque of the lifting part and revolving part, respectively. The third subfigure shows the control vector’s norm. The lower subfigure is the time history of $V$. It can be seen that the control force/torque is limited within a given range by the saturation function. The control vector’s norm is also limited within a certain range. In addition, the Lyapunov function curve still exhibits non-monotonically decreasing characteristics.

In addition, the real oscillations used in this group are derived from our previous works on vehicles’ maneuverability, i.e., simulations in the ADAMS/ATV software. The upper, middle, and lower subfigures of Fig. 14 present the shake, pitch, and roll oscillation, respectively.
Robust stabilizing control for oscillatory base manipulators

4.2 Hardware configurations

The developed reduced-scale model of the autoloader is a manipulator system shown in Fig. 15. It mainly consists of three mechanical components: a lifting part, a revolving part, and a mounted frame. Two servo motors are used to drive the lifting and revolving parts. Digital encoders are installed at the rear of both motors to sensor motion of the motors. Readings from the encoders are then converted into motion feedback of the autoloader. In addition, planetary gear reducers are used in both the lifting part and the revolving part. Above servomotors, encoders and reducers are all provided by MAXON company.

Two digital positioning controllers (EPOS2 controllers provided by MAXON) and a personal computer (Lenovo 32-bit) are used in the experiment. The master–slave communication technology is adopted to realize the simultaneous parallel control of the lifting part and the rotating part. Two EPOS2 controllers communicate as slaves, and the PC communicates as master. In addition, the CANopen communication protocol is used to transmit and receive data between the master (PC) and slave (EPOS2 controller).

A three-loop control strategy is embedded in the EPOS2 controller, including current, position, and speed loop. In our experiment, only the built-in current control loop is used. The position and velocity loops are replaced by our self-built one. The current loop adopts proportional–integral (PI) control, and the self-built loops adopt the implicit Lyapunov control. The two control loops are connected through the torque constant of the servo motor, as shown in Fig. 16. According to the control’s boundary property (6) and the linear relationship between motor torque and current, we can estimate the max desired current of the lifting and revolving motors as the following

\[
c_{d1max} = \frac{U_0}{(i_1 T_{c1})}, \quad c_{d2max} = \frac{U_0}{(i_2 T_{c2})}
\]

(44)

where \(i_1, i_2\) are the reduction ratio of the lifting and revolving reducer, respectively; \(T_{c1}, T_{c2}\) are the torque constant of the lifting and revolving motor, respectively. These estimations are probably on the conservative side. In addition, the real-time control program is developed and implemented in the LabVIEW programming environment.

The hardware system of the developed reduced-scale model is presented in Fig. 17. As shown in the figure, the system is mounted on an electromagnetic vibration table (provided by ETS Solution Beijing Ltd.), which can provide vibration to simulate the shake base oscillation.

The system parameters are as: \(m_2 = 5\ \text{kg}, \ m_3 = 1\ \text{kg}, \ J_3 = 0.01\ \text{kg} \cdot \text{m}^2, \ L_3 = 0.05\ \text{m}, \ m = 0.01, \ M = 8.\) The gravitational acceleration is \(9.8\ \text{m/s}^2.\) The initial state of the system is set as \((0.144, 0, -0.524, 0)\), and the terminal state is \((0, 0, 0, 0)\). For the lifting part and

Fig. 13 Control response of Group s3

Fig. 14 Base oscillation of simulations

\[\text{Fig. 13 Control response of Group s3}\]

\[\text{Fig. 14 Base oscillation of simulations}\]
4.3 Experiment results and analysis

Based on the above hardware system, two groups of experiments were carried, namely Group e1 and Group e2, respectively, as shown in Table 3. Group e1 is to certify the control’s performance under base oscillation, and Group e2 is to further certify the control’s performance under payload uncertainty. In addition, only shake base oscillation is considered in these experiments.

*Group e1*: This group includes three experimental cases. Cases e1.2 and case e1.3 correspond to the scenarios of real and sinusoid base oscillations. Case e1.1 is a comparative case, in which the base oscillation is not considered. Similar to the simulation, the real base oscillation here also comes from our previous research. The amplitude and frequency of sinusoidal oscillation are set to 10 mm and 36 rad/s, respectively.

Experiment results are depicted in Figs. 18, 19 and 20. The black solid line corresponds to case e1.1, and the blue dash-dot and red dash lines correspond to case e1.2 and e1.3, respectively.

Figures 18 and 19 present the time history of the motion of the lifting part and revolving part, respectively. It is clear that the manipulator system successfully converges to the terminal state in finite time. The setting times for the three cases are 2.915 s, 3.01 s, and 3.38 s respectively. This proves that the proposed control yields good robustness against the base oscillations.
### Table 3  Experiment groups

| Group no. | Case no. | Oscillation | Payload uncertainty |
|-----------|----------|-------------|---------------------|
| Group e1  | Case e1.1 | N/A         | N/A                 |
|           | Case e1.2 | Real shake  | N/A                 |
|           | Case e1.3 | Sinusoid shake | N/A               |
| Group e2  | Case e2.1 | N/A         | N/A                 |
|           | Case e2.2 | N/A         | 0.4 kg              |
|           | Case e2.3 | Real shake  | 0.4 kg              |

Moreover, it can be found that, for the case of the base oscillation, especially the sinusoidal base oscillation, it takes more time than the case without the base oscillation. It is also found that the velocity curve under base oscillation has an obvious non-smooth characteristic.

Figure 20 shows the control response of this group of experiments. The upper two subfigures show the current response of the lifting and revolving motors. The actual motor current is illustrated by dot curves. It is clear that the actual motor current can track the desired current in a very short time. The lower figure presents the time history of the Lyapunov function. The function curves are non-monotonically decreasing. It can also be found that the current of non-monotonic time is greater than that of monotonic time. The reason for this non-monotonicity is probably the same as that in the simulation. The values of $S_0$ and $U_0$ are roughly estimated, which may be probably on the conservative side. Nonetheless, results still proved the robustness of the proposed control to base oscillation.

**Group e2:** There are three more cases of experiments in this group, which are named as cases e2.1, e2.2, and e2.3, respectively. In case e2.2, only the payload uncertainty is considered. In case e2.3, both the payload uncertainty and base oscillation (real shake) are considered. Case e2.1, which is actually the case e1.1 in Group e1, is a contrast case. The payload uncertainty
here refers to a mass block (0.4 kg) at the end of the revolving part of the manipulator.

Experiment results are shown in Figs. 21, 22 and 23. Similarly, the black solid, red dash, and blue dash-dot lines are corresponding to cases e2.1, e2.2, and e2.3, respectively.

Figures 21 and 22 show the motion response of the manipulator. It can be seen that the controllers successfully steer the manipulator from the given state to the terminal state for all three cases. The setting times of cases e2.1, e2.2, and e2.3 are 2.915s, 3.01s, and 3.311s, respectively. Obviously, it takes a longer time under payload uncertainty than when neither base oscillation nor payload uncertainty is considered. It takes the most time for the cases considering both the base oscillation and payload uncertainty. Moreover, it can also be seen that the payload uncertainty will affect the overshoot of the system.

Figure 23 depicts the control response of experiment in this group. The upper two subfigures show the current tracking response of the lifting and revolv-
ing motor, respectively. Similar to Group e1, the actual motor current is plotted as dot curves. Obviously, the PI control of the current loop successfully yields accurate and fast trajectory tracking of the current. The boundedness of the control input can also be deduced from the current response. The lower subfigure is the time history of the Lyapunov function, which is also non-monotonically decreasing. Nevertheless, however, the above results still prove that the proposed control has good robustness not only to base oscillation but also to payload uncertainty.

In addition, Fig. 24 describes the base oscillation response in the experiment, and the upper and lower subfigures correspond to cases e1.3 and e2.3.

5 Conclusion

In the presence of chassis oscillation and payload uncertainty, this paper has proposed a robust stabilizing control scheme for the MBT autoloader. The proposed control method has been shown to be novel and offers advantages compared with existing researches. It has the ability to deal with uncertain chassis-oscillation-induced perturbations without the predictions or measurements of the chassis oscillations. It also has exhibited good robustness against payload uncertainty. Moreover, the proposed control has always been bounded in norm. Results of simulations and hardware experiments have further proved that the proposed control has offered good performance in practice.

The proposed control method can be used in not only autoloaders but also other OBMs. However, results show that the Lyapunov stability condition in this study is conservative. In some cases, although the Lyapunov function is non-monotonically decreasing, the controller still exhibits good robustness. In addition, it is clear that the controller needs more time to stabilize the system in the experiment than in the simulation. One reason may be that the influence of hardware network on control performance is not considered. To improve the control performance in the experiment, consideration should be invested on improvement of hardware circuits and solution algorithms. As such, theoretical analysis of the controller’s stability and research on network-induced phenomena need to be further studied.

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Declarations

Conflict of interest The authors have no conflicts of interest to declare that are relevant to the content of this article.

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