

Safe Truck Platooning With String Stability and Energy Dissipation Analysis

Shaoyu Sun

Department of Control Science
and Engineering, Jilin University
Changchun, China
sysun22@mails.jlu.edu.cn

Shuyou Yu*

Department of Control Science
and Engineering, Jilin University
Changchun, China
shuyou@jlu.edu.cn

Hong Chen

College of Electronics and Information Engineering
Tongji University
Shanghai, China
chenhong2019@tongji.edu.cn

Abstract—Truck platooning can improve traffic efficiency and reduce driving workload, but maintaining both longitudinal string stability and lateral safety remains challenging, especially when trucks travel on curved roads. To address this issue, this paper proposes an integrated longitudinal and lateral control framework for truck platoons. First, a longitudinal truck-following model is established under the predecessor-following communication topology and constant time headway policy. An LQR-based longitudinal controller with predecessor acceleration feedforward is then designed to suppress spacing errors, velocity errors, and acceleration fluctuations. For lateral motion, a model predictive control-based controller is developed using the vehicle lateral dynamics and lane-keeping model to ensure path-following accuracy. In addition, the time headway is selected through frequency-domain string stability analysis. Monte Carlo simulations are conducted under different initial spacing and velocity errors, and an energy dissipation stability analysis is introduced to characterize the vehicle dynamic stability region. Co-simulation results based on TruckSim and Matlab/Simulink show that the proposed control framework can maintain longitudinal string stability, and accurate path following under the considered expressway driving scenario.

Index Terms—Truck platoon, string stability, model predictive control, energy dissipation stability

I. INTRODUCTION

Truck platooning has attracted increasing attention as an effective technology for improving traffic efficiency, reducing aerodynamic drag, and enhancing the safety of freight transportation [1, 2]. By using vehicle-to-vehicle (V2V) communication and automated control, multiple trucks can travel with short inter-vehicle distances while maintaining coordinated longitudinal motion. However, compared with passenger vehicles, heavy-duty trucks have larger mass, slower longitudinal response and stronger nonlinear tire characteristics. These characteristics make the safety control of truck platoons more challenging, especially under high-speed expressway driving conditions [3].

One of the key issues in truck platoon control is longitudinal string stability. If the spacing error or velocity disturbance is amplified along the platoon, small fluctuations of the leading truck may cause large oscillations in downstream trucks, which

can threaten driving safety and reduce traffic efficiency [4, 5]. To address this problem, various control strategies, such as PID control, optimal control, and model predictive control (MPC), have been applied to platoon longitudinal control [6]. Nevertheless, the selection of control parameters and time headway remains critical for ensuring that spacing errors are not amplified along the platoon. In addition to longitudinal string stability, lateral safety is also important for truck platoons operating on curved roads. When trucks pass through curved segments, the lateral velocity, yaw rate, tire force, and road curvature directly affect path-following accuracy and vehicle stability [7]. Therefore, only considering longitudinal spacing control is insufficient. A safe truck platoon control framework should simultaneously coordinate longitudinal following performance and lateral path tracking performance.

To address these issues, this paper develops an integrated longitudinal and lateral control framework for homogeneous truck platoons. For longitudinal control, a predecessor-following communication topology and a constant time headway policy (CTHP) are adopted. A linear quadratic regulator (LQR)-based controller with predecessor acceleration feedforward is designed to suppress spacing error, velocity error, and acceleration fluctuation. For lateral control, an MPC-based controller is designed based on the lateral dynamics and lane-keeping model to improve path-following performance while satisfying steering constraints.

Furthermore, this paper evaluates the stability of the proposed control framework from multiple perspectives. First, the time headway is selected through frequency-domain string stability analysis. Then, Monte Carlo simulations are conducted to statistically verify the string-stable performance under different initial spacing and velocity errors. Finally, an energy dissipation stability analysis is introduced to characterize the vehicle dynamic stability region with respect to yaw rate, lateral velocity, and longitudinal velocity.

The main contributions of this paper are summarized as follows:

- 1) A longitudinal truck-following model is established. Based on this model, a longitudinal controller is designed to improve spacing regulation, velocity tracking, and disturbance attenuation. An MPC-based lateral controller is developed based on the lateral dynamics and lane-keeping model.

This work is supported in part by the Natural Science Foundation of Jilin Province (No.20240402079GH), in part by the National Natural Science Foundation of China (No.62473167). (Corresponding author: Shuyou Yu, shuyou@jlu.edu.cn)

The controller is used to ensure accurate path following and lateral safety during curved-road driving.

2) A comprehensive stability evaluation framework is provided. Frequency-domain string stability analysis is used to select the time headway, Monte Carlo analysis is conducted to statistically verify spacing-error attenuation under different initial conditions, and an energy dissipation field is introduced to evaluate vehicle dynamic stability from the perspective of yaw rate, lateral velocity, and longitudinal velocity.

The structure of this paper is as follows: In Section II, the models and control objectives are provided. Controller design is provided in Section III. In Section IV, the paper presents and analyzes the simulation results. Finally, in Section V, a conclusion is provided.

II. PRELIMINARIES

A. Longitudinal Dynamics of Trucks

To characterize the truck-following behavior within the platoon, the longitudinal dynamics of the i^{th} truck can be described as [8]:

$$\begin{cases} \dot{s}_i = v_i^x \\ \dot{v}_i^x = \frac{1}{m_i} \left(\frac{\eta_{T,i}}{R_{e,i}} T_i - \frac{1}{2} C_{d,i} A_i \rho_i v_i^{x2} - m_i g f_i \right) \\ \dot{T}_i = -\sigma_i^{-1} T_i + \sigma_i^{-1} T_{des,i} \end{cases} \quad (1)$$

where terms of s_i and v_i^x are the longitudinal position and velocity of the i^{th} truck, respectively. Terms of m_i and $R_{e,i}$ denote the mass of the truck and the radius of rotation of the tire, and $\eta_{T,i}$ represents the mechanical efficiency of the transmission system. The term of A_i is the windward area, ρ_i denotes the air density, f_i denotes the rolling resistance coefficient, $C_{d,i}$ is the air resistance coefficient, and g denotes the gravitational constant. The term of σ_i denotes the time constant of the longitudinal dynamics, $T_{des,i}$ denotes the required control torque.

Assume that the aforementioned parameters are known, then the nonlinear control law is designed:

$$T_{des,i} = \frac{R_{e,i}}{\eta_{T,i}} \left(m_i u_i^x + m_i g f_i + \frac{1}{2} C_{d,i} A_i \rho_i v_i^x (2\sigma_i a_i + v_i^x) \right) \quad (2)$$

Combining (1) and (2), the nonlinear system can be transformed into its equivalent linear one:

$$\begin{cases} \dot{s}_i = v_i^x \\ \dot{v}_i^x = a_i \\ \dot{a}_i = -\sigma_i^{-1} a_i + \sigma_i^{-1} u_i^x \end{cases} \quad (3)$$

where the control input u_i^x is the desired acceleration, a_i is the acceleration of the i^{th} truck, σ_i is the time constant of longitudinal dynamics.

In this paper, the predecessor-following (PF) communication topology and a CTHP are adopted [9]. The longitudinal spacing error of the i^{th} truck can be defined as:

$$\tilde{e}_i^s = s_i - (s_{i-1} - d_0 - h \cdot v_i^x) \quad (4)$$

Define the velocity error of the i^{th} truck as the difference between the desired velocity and the actual velocity, it can be denoted as:

$$\tilde{e}_i^v = v_i^x - v_{i-1}^x \quad (5)$$

Define $\tilde{x}_i = [\tilde{e}_i^s \ \tilde{e}_i^v \ a_i]^T$, then the longitudinal error model for i^{th} truck can be expressed as:

$$\begin{cases} \dot{\tilde{x}}_i(t) = A_i \tilde{x}_i(t) + B_i u_i(t) + E_i a_{i-1}(t) \\ \tilde{y}_i(t) = \tilde{C}_i \tilde{x}_i(t) \end{cases} \quad (6)$$

with $A_i = \begin{bmatrix} 0 & 1 & h \\ 0 & 0 & 1 \\ 0 & 0 & -\sigma_i^{-1} \end{bmatrix}$, $B_i = [0 \ 0 \ \sigma_i^{-1}]^T$, $E_i = [0 \ -1 \ 0]^T$ and $\tilde{C}_i = \text{diag}(1, 1, 1)$.

Taking the sampling time as T_s , the term of (6) can be discretized as:

$$\begin{cases} \tilde{x}_i(k+1) = \tilde{A}_i \tilde{x}_i(k) + \tilde{B}_i u_i(k) + \tilde{E}_i a_{i-1}(k) \\ \tilde{y}_i(k) = \tilde{C}_i \tilde{x}_i(k) \end{cases} \quad (7)$$

where $\tilde{A}_i = \begin{bmatrix} 1 & T_s & h \cdot T_s \\ 0 & 1 & T_s \\ 0 & 0 & 1 - T_s \cdot \sigma_i^{-1} \end{bmatrix}$, $\tilde{B}_i = \begin{bmatrix} 0 \\ 0 \\ T_s \cdot \sigma_i^{-1} \end{bmatrix}$ and $\tilde{E}_i = \begin{bmatrix} 0 \\ -T_s \\ 0 \end{bmatrix}$.

B. Lateral Dynamics of Trucks

The lateral dynamics of the i^{th} vehicle in the platoon can be described by the following model:

$$\begin{cases} (m_i \dot{v}_i^y + m_i \dot{\psi}_i \dot{\psi}_i) = F_i^{yf} + F_i^{yr} \\ I_i^z \ddot{\psi}_i = a_i F_i^{yf} - b_i F_i^{yr} \end{cases} \quad (8)$$

where v_i^y and $\dot{\psi}_i$ are the lateral velocity and yaw angle rate of the truck. The term of I_i^z is the moment of inertia around z axis, a_i and b_i are the distances from front axle and rear axle to mass center. Terms of F_i^{yf} and F_i^{yr} are the lateral force of front wheel and lateral force of rear wheel, respectively.

C. Pacejka Tire Model

The ‘‘Magic Formula’’ tire model [10] is applied, and the corresponding tire force can be calculated by:

$$F(\chi) = D \sin(C \arctan(B\chi - E(B\chi - \arctan(B\chi)))) \quad (9)$$

where B , C , D and E represent the stiffness, shape, peak, and curvature factor, respectively. The input variable χ is the slip angle, and the output variable $F(\chi)$ is the lateral tire force.

D. Lane-Keeping Model

Denote the heading error of vehicle i relative to the centerline as:

$$e_i^\psi = \psi_d - \psi_i \quad (10)$$

where ψ_i and ψ_d are the actual heading angle and the tangential angle of the desired lane, respectively.

Denote lateral position error from the mass center of the i^{th} truck to the center of lane as e_i^y . Then one has:

$$\begin{cases} \dot{e}_i^y = v_i^x \dot{\psi}_i - v_i^y - L\dot{\psi}_i \\ \dot{e}_i^\psi = \dot{\psi}_d - \dot{\psi}_i \end{cases} \quad (11)$$

where $\dot{\psi}_d$ is the desired yaw rate, i.e.,

$$\dot{\psi}_d = \frac{v_i^x}{R} \quad (12)$$

The term of R is the radius of the road and the term of L is the specified look-ahead distance.

Combining (8) and (11), the lateral dynamics of the i^{th} truck can be described as:

$$\begin{cases} \dot{v}_i^y = -v_i^x \dot{\psi}_i + \frac{1}{m_i} (F_i^{yf} + F_i^{yr}) \\ \ddot{\psi}_i = \frac{1}{I_i^z} (a_i F_i^{yf} - b_i F_i^{yr}) \\ \dot{e}_i^\psi = \dot{\psi}_d - \dot{\psi}_i \\ \dot{e}_i^y = v_i^x e_i^\psi - v_i^y - L\dot{\psi}_i \end{cases} \quad (13)$$

Define $\hat{x}_i = [v_i^y \ \dot{\psi}_i \ e_i^\psi \ e_i^y]^T$, $u_i^y = \delta_i$ as control variable and $\omega_i^y = [v_i^x \ R]$ as disturbance variables. Taking the sampling time as T_s , the vehicle lateral dynamics can be expressed as:

$$\begin{cases} \hat{x}_i(k+1) = f_y(\hat{x}_i(k), u_i^y(k), \omega_i^y(k)) \\ \hat{y}_i(k+1) = \hat{x}_i(k+1) \end{cases} \quad (14)$$

E. Control Objectives

(1) The longitudinal control objective of the truck platoon is to track the longitudinal velocity of the predecessor, and to maintain the desired safe distance between the front and rear trucks:

$$\begin{cases} \text{minimize} & \|v_i^x - v_{i-1}^x\|_2^2 \\ \text{minimize} & \|s_{i-1} - s_i - d_0 - h \cdot v_i^x\|_2^2 \end{cases} \quad (15)$$

(2) String stability: When a truck platoon is subject to unknown disturbances, the spacing errors are not amplified along the platoon:

$$\|G_i^s(s)\| = \left\| \frac{\tilde{e}_i^s(s)}{\tilde{e}_{i-1}^s(s)} \right\| \leq 1 \quad (16)$$

where $G_i^s(s)$ is the transfer function of the longitudinal spacing error. Terms of $\tilde{e}_i^s(s)$ and $\tilde{e}_{i-1}^s(s)$ are the related Laplace transform of $\tilde{e}_i^s(t)$ and $\tilde{e}_{i-1}^s(t)$, respectively.

(3) The lateral control objective is that trucks in the platoon run along the lane:

$$\begin{cases} \text{minimize} & \|e_i^\psi - 0\|_2^2 \\ \text{minimize} & \|e_i^y - 0\|_2^2 \end{cases} \quad (17)$$

III. CONTROLLER DESIGN

A. Longitudinal Controller

A longitudinal controller based on the LQR method is designed for each following truck. For the i^{th} following truck, the discrete-time longitudinal error model is given by

$$\tilde{x}_i(k+1) = \tilde{A}_i \tilde{x}_i(k) + \tilde{B}_i u_i^x(k) + \tilde{E}_i a_{i-1}(k) \quad (18)$$

where

$$\tilde{x}_i(k) = [\tilde{e}_i^s(k) \ \tilde{e}_i^v(k) \ a_i(k)]^T \quad (19)$$

Here, $\tilde{e}_i^s(k)$, $\tilde{e}_i^v(k)$, and $a_i(k)$ denote the spacing error, velocity error, and acceleration of the i^{th} truck, respectively. The term $u_i^x(k)$ denotes the desired acceleration command of the i^{th} truck, and $a_{i-1}(k)$ denotes the acceleration of its immediate predecessor.

The LQR controller is designed by minimizing the following quadratic performance index:

$$J_i = \sum_{k=0}^{\infty} [(\tilde{x}_i(k))^T \tilde{Q}_i \tilde{x}_i(k) + (u_i^x(k))^T \tilde{R}_i u_i^x(k)] \quad (20)$$

where $\tilde{Q}_i$ and $\tilde{R}_i$ are the state weighting matrix and control weighting matrix, respectively. They are selected as:

$$\tilde{Q}_i = \text{diag}(q_i^s, q_i^v, q_i^a) \quad \tilde{R}_i = r_i^u \quad (21)$$

For the discrete-time system in (18), the optimal feedback gain K_i can be obtained by solving the following discrete algebraic Riccati equation:

$$\tilde{P}_i = \tilde{A}_i^T \tilde{P}_i \tilde{A}_i - \tilde{A}_i^T \tilde{P}_i \tilde{B}_i (\tilde{R}_i + \tilde{B}_i^T \tilde{P}_i \tilde{B}_i)^{-1} \tilde{B}_i^T \tilde{P}_i \tilde{A}_i + \tilde{Q}_i \quad (22)$$

Then, the LQR feedback gain is calculated as:

$$K_i = (\tilde{R}_i + \tilde{B}_i^T \tilde{P}_i \tilde{B}_i)^{-1} \tilde{B}_i^T \tilde{P}_i \tilde{A}_i \quad (23)$$

Considering the predecessor acceleration information available from the communication topology, the final longitudinal control law is designed as:

$$u_i^x(k) = -K_i \tilde{x}_i(k) + k_i^f a_{i-1}(k) \quad (24)$$

where k_i^f is the feedforward gain.

B. Lateral Controller

An MPC method is designed to ensure accurate path-following performance. For truck i , the sequence of lateral control inputs over the prediction horizon is defined as:

$$u_i^y(k) = \{u_i^y(k|k), u_i^y(k+1|k), \dots, u_i^y(k+N_p^y-1|k)\} \quad (25)$$

where N_p^y denotes the prediction horizon.

The MPC-based lateral control problem can be formulated as the following finite-horizon optimization problem.

Problem 1

$$\underset{u_i^y(k)}{\text{minimize}} \quad J_i^y(\hat{x}_i(k), u_i^y(k)) \quad (26a)$$

$$\text{s.t.} \quad (26b)$$

$$\hat{x}_i(k+j+1|k) = f_y(\hat{x}_i(k+j|k), u_i^y(k+j|k)) \quad (26c)$$

$$\hat{y}_i(k+j|k) = \hat{x}_i(k+j|k) \quad (26d)$$

$$\hat{y}_i(k|k) = \hat{y}_i(k) \quad (26e)$$

$$u_{i,\min}^y \leq u_i^y(k+j|k) \leq u_{i,\max}^y \quad (26f)$$

$$\hat{x}_i(k+N_p^y|k) = 0 \quad (26g)$$

where

$$\begin{aligned} J_i^y(\hat{x}_i(k), u_i^y(k)) &= \sum_{j=0}^{N_p^y-1} \|\hat{x}_i(k+j|k)\|_{\hat{Q}_i}^2 \\ &+ \sum_{j=0}^{N_p^y-1} \|u_i^y(k+j|k)\|_{\hat{R}_i}^2 \end{aligned} \quad (27)$$

In the objective function J_i^y , terms of $\hat{Q}_i$ and $\hat{R}_i$ are the weight matrices, (26f) is the control constraints. Then, constraint (26g) is relaxed into a soft constraint, and the cost function in Problem 1 is reformulated as follows:

$$\begin{aligned} J_i^y(\hat{x}_i(k), u_i^y(k)) &= \sum_{j=0}^{N_p^y-1} \|\hat{x}_i(k+j|k)\|_{\hat{Q}_i}^2 \\ &+ \sum_{j=0}^{N_p^y-1} \|u_i^y(k+j|k)\|_{\hat{R}_i}^2 \\ &+ \|\hat{x}_i(k+N_p^y|k)\|_{\hat{P}_i}^2 \end{aligned} \quad (28)$$

where $\hat{P}_i$ is the terminal penalty matrix.

IV. SIMULATION AND RESULTS

The truck platoon consists of a leading truck and three following trucks. A co-simulation platform composed of TruckSim and Matlab/Simulink is developed. All the trucks are homogeneous with consistent parameters. The related parameters are as shown in Table I.

TABLE I
THE PARAMETERS OF TRUCKS

Parameters	Value	Parameters	Value
$\tilde{Q}_i$	diag (5, 1, 0.1)	$\tilde{R}_i$	1
T_s	0.01 s	σ_i	0.25 s
a_i	1.5 m	b_i	3.5 m
m_i	18000 kg	N_p^y	10
$\hat{Q}_i$	diag (50, 1000, 700, 100)	$\hat{R}_i$	3500

A. String Stability Analysis

Based on the CTH policy, the spacing error transfer function from the predecessor truck to the following truck can be derived as:

$$G_i^s(s) = \frac{\tilde{e}_i^s(s)}{\tilde{e}_{i-1}^s(s)} = \frac{k_i^f s^2 + k_i^v s + k_i^s}{\sigma_i s^3 + (1 + k_i^a) s^2 + (k_i^v + h k_i^s) s + k_i^s} \quad (29)$$

It can be observed from (29) that the time headway h directly affects the denominator of the transfer function, and thus has an important influence on the string stability of the platoon.

According to the frequency-domain string stability criterion, the spacing error should not be amplified along the platoon. Therefore, the following condition should be satisfied:

$$\|G_i^s(s)\|_\infty = \sup_{\omega>0} |G_i^s(j\omega)| \leq 1 \quad (30)$$

Equivalently, the magnitude of $G_i^s(j\omega)$ should not exceed 0 dB over the entire frequency range.

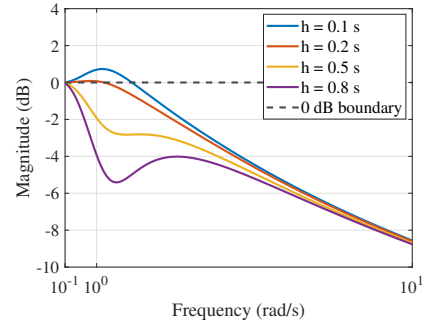


Fig. 1. Bode plots of string stability transfer functions under different time headways.

The time headway is selected according to

$$h^* = \min h, \quad \text{s.t.} \quad \sup_{\omega>0} |G_i^s(j\omega)| \leq 1 \quad (31)$$

The Bode plots under different time headways are used to compare the magnitude responses. As shown in Fig. 1, if the magnitude curve exceeds the 0 dB boundary, the corresponding time headway cannot guarantee string stability. Otherwise, the spacing error disturbance will not be amplified along the platoon. In this paper, the term of h is selected as 0.8 s.

B. Simulation and Analysis of Truck Platoon

Fig. 2 presents the simulation results of the truck platoon under the expressway driving scenario. As shown in Fig. 2(a), the maximum road curvature is 0.0025. Fig. 2(b) shows that all trucks can follow the desired driving path smoothly. Fig. 2(c) shows the longitudinal velocity responses of the platoon, indicating that the proposed longitudinal controller can effectively coordinate the platoon speed. The corresponding longitudinal spacing errors are shown in Fig. 2(d). Moreover, no obvious amplification of spacing errors is observed along the platoon.

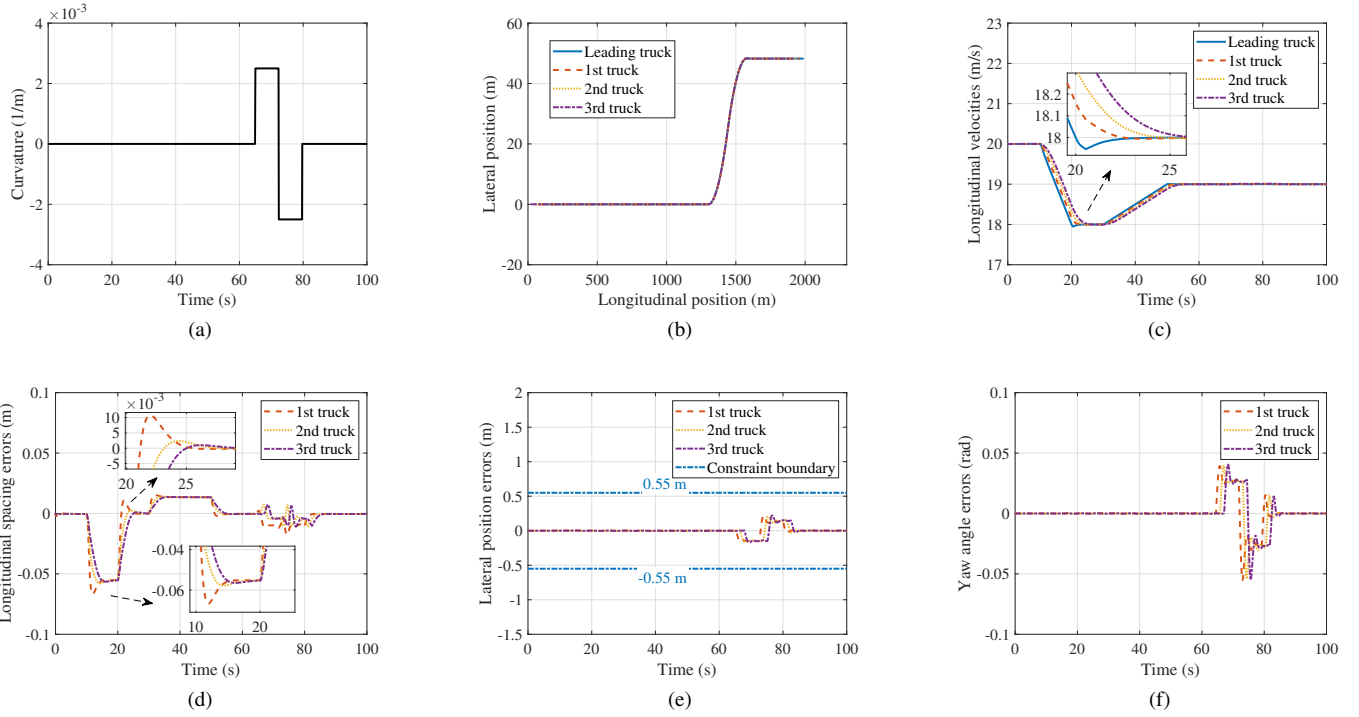


Fig. 2. Simulation results: (a) road curvature, (b) driving paths, (c) longitudinal velocities, (d) longitudinal spacing errors of following trucks, (e) lateral position errors of following trucks, (f) angular errors of following trucks.

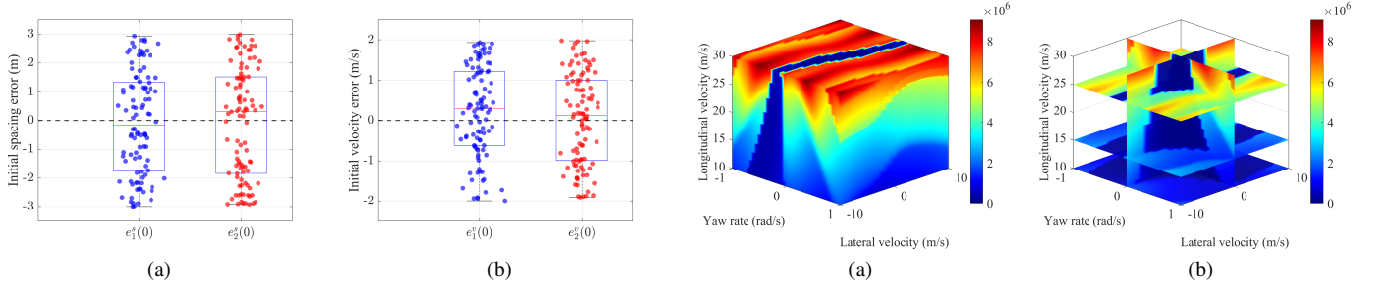


Fig. 3. (a) Distribution of initial spacing errors, (b) distribution of initial velocity errors.

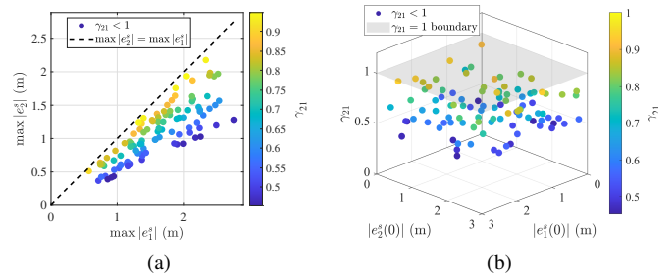


Fig. 4. (a) Maximum spacing error comparison, (b) results of string-stable Monte Carlo samples.

The lateral position errors and yaw angle errors are shown in Fig. 2(e) and Fig. 2(f), respectively. The lateral position errors of all following trucks remain within the prescribed

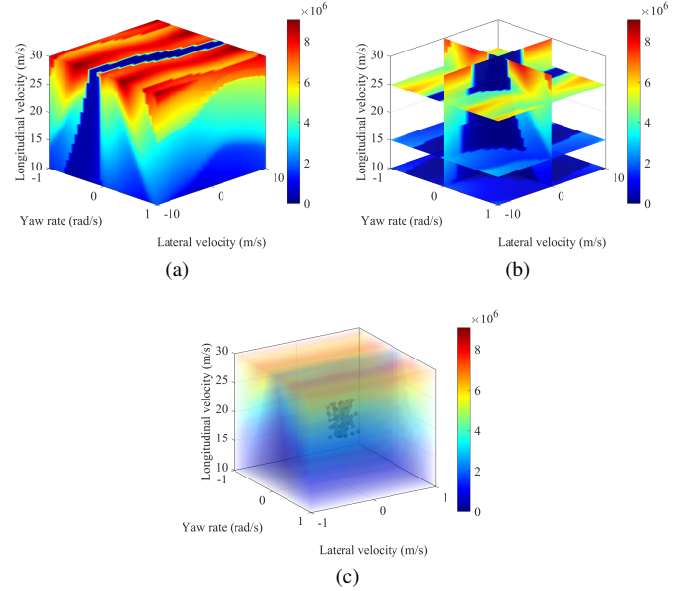


Fig. 5. Monte Carlo-based energy dissipation stability analysis: (a) three-dimensional energy dissipation field, (b) slice planes of the energy dissipation field, (c) distribution of Monte Carlo samples in the energy dissipation field.

safety boundary of ± 0.55 m, and the yaw angle errors are also maintained within a small range. These results indicate that the proposed control framework can maintain both longitudinal platoon string stability and lateral tracking safety under the considered expressway driving condition.

C. Monte Carlo-Based String Stability Analysis

To evaluate the string stability of the platoon under different initial conditions, a Monte Carlo analysis is conducted. One leading truck and two following trucks are considered. The spacing error of the first following truck is denoted as $e_1^s(t)$, and the spacing error of the second following truck is denoted as $e_2^s(t)$. To quantify whether the spacing error is amplified along the platoon, the following time-domain spacing-error amplification ratio is defined:

$$\gamma_{21} = \frac{\max_{t \in [0, T]} |e_2^s(t)|}{\max_{t \in [0, T]} |e_1^s(t)|}, \quad (32)$$

where T is the simulation duration. If $\gamma_{21} < 1$, the maximum spacing error of the second following truck is smaller than that of the first following truck, indicating that the spacing error is not amplified from the first following truck to the second following truck.

As shown in Fig. 3, the initial spacing errors are randomly selected from $[-3, 3]$ m and the initial velocity errors are randomly selected from $[-2, 2]$ m/s. Fig. 4(a) compares the maximum spacing errors of the two following trucks. All sampled points are located below the boundary line $\max |e_2^s| = \max |e_1^s|$, indicating that the spacing error is not amplified. Furthermore, Fig. 4(b) shows that all samples satisfy $\gamma_{21} < 1$, which further confirms that the proposed longitudinal controller can maintain string-stable performance under different initial conditions.

D. Monte Carlo-Based Energy Dissipation Stability Analysis

To evaluate the vehicle stability characteristics, the total dissipation of energy method proposed in [11] is employed. Fig. 5 presents the Monte Carlo-based energy dissipation stability analysis. Fig. 5(a) shows the three-dimensional energy dissipation field with respect to yaw rate, lateral velocity, and longitudinal velocity. It can be observed that the energy dissipation increases significantly when the yaw rate, lateral velocity, or longitudinal velocity becomes large, indicating that the vehicle tends to enter a more unstable dynamic region under severe lateral motion. Fig. 5(b) further shows the slice planes of the energy dissipation field, from which the distribution of high- and low-energy regions can be clearly observed. The low-energy region mainly appears around small yaw rates and small lateral velocities, corresponding to stable vehicle motion. Fig. 5(c) maps the Monte Carlo samples into the energy dissipation field, indicating that the vehicle states remain within a stable dynamic range.

V. CONCLUSION

This paper investigated the longitudinal and lateral control problem of truck platoons. A longitudinal controller was designed to ensure longitudinal string stability, while an MPC-based lateral controller was developed to ensure path-following accuracy. Simulation results showed that the following trucks can track the leading truck smoothly, with bounded spacing errors and no obvious error amplification along the platoon.

Meanwhile, the lateral position and yaw angle errors remain effectively constrained during curved-road driving. Furthermore, Monte Carlo analysis verified the effectiveness of the proposed controller.

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