Model Predictive Braking Control for Heavy-Duty Commercial Vehicles Considering Response Delay of Air-Brake

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ABSTRACT: Heavy-duty commercial vehicles (HDCVs) have specific characteristics different from passenger cars. This paper deals with one of the characteristics of HDCV, propagation delay by air of an air-brake system. It is related to the braking performance of the vehicle. Thus, it is necessary to reflect it appropriately in the control model. In this paper, we propose the model predictive braking controller with the model of HDCV, including the air-brake system. The effectiveness of the proposed method is demonstrated via numerical simulations with the full dynamics vehicle model and the realistic air-brake model.

KEY WORDS: Heavy commercial vehicle, Air brake, Model predictive control, Time delay system, Motion control

1. INTRODUCTION

The automatic driving of heavy-duty commercial vehicles (HDCVs) has been developed to improve the safety and efficiency of operation. On the other hand, HDCVs have some unique characteristics compared to passenger cars. In particular, Kaneko, et al. lists four items to be considered in HDCVs (9). First, there are a variety of number of wheels and axles. Second, vehicle dimensions and weight are larger than passenger cars. Third, the propagation delay of the actuator is large. Finally, the mass, moments of inertia, and center of gravity, etc. are changed widely depending on a cargo loading status.

As a study on the driving control of the vehicle, research using model predictive control (MPC) has been actively conducted. MPC is a control method to calculate control input by solving an optimization problem based on the prediction during a finite time horizon. MPC takes constraints explicitly in optimization problems, and there are research treating the braking torque limit or the road width as constraints. Besides, the following characteristics of HDCVs have been considered in MPC: the time delay due to actuators, sensors, and communication networks in HDCVs platoons, roll-over prevention by the high center of gravity, and vehicle model uncertainty.

On the other hand, the effect of the propagation delay of actuators is not considered well since the complexity of the fluid dynamics of an actuator. Air-brake systems are generally installed in HDCVs because HDCVs require a sizeable braking force compared with passenger cars. However, the time delay due to air propagation delay occurs, and it affects the braking performance of the vehicle. Though it is impossible to consider the full fluid dynamics model in MPC, we can implement an approximation of the propagation delay as a simple time delay model. In this paper, we propose a model predictive based braking controller that improves both vehicle stability and the path tracking performance by considering the propagation delay of braking pressure as well as an HDCV’s body dynamics. Although it is not clear whether the approximation works for the actual air-brake model. Hence, in this paper, we demonstrate the validity of the proposed controller by applied to the detailed HDCV model including the detailed air-brake model.

This paper is organized with five sections. Section 2 describes the vehicle model, including the air-brake system of a HDCV. In section 3, the proposed formulation of MPC is described. In section 4, we show the numerical simulation results using the proposed control method. Finally, the conclusion of this paper is described in section 5. This paper adopts the following notations. \( \| z \|_F = \sqrt{z^T P z} \) represents the norm of Frobenius. \( A > O \) denotes \( A \) is a positive definite matrix, and \( A \geq O \) denotes \( A \) is a semi-positive definite matrix. The subscripts of fl, fr, rl, and rr refer to front left, front right, rear left, and rear right, respectively. \( O_n \) is \( n \)-dimensional zero vector.

2. VEHICLE MODEL

2.1. Nonlinear Vehicle dynamics

In this section, the modeling of vehicle dynamics for controller design. Table 1 shows the description and symbols used to describe the vehicle model. 3DoF model of the vehicle motion in a plane, as shown in Fig. 1. In Fig. 1, \( x \) and \( y \) denotes the longitudinal and lateral positions in the vehicle frame respectively, \( X \) and \( Y \) denotes the longitudinal and lateral positions in the inertial frame respectively, \( \psi \) is the orientation of the vehicle called yaw angle. Vehicle kinematics are expressed as follows.
\[
\frac{d}{d\tau} \begin{bmatrix} X \\ Y \end{bmatrix} = \begin{bmatrix} V \cos(\psi + \beta) \\ V \sin(\psi + \beta) \end{bmatrix},
\]

(1)

where \(\psi\) is yaw angle with respect to the absolute coordinate, \(\beta = \tan^{-1}(v_y/V)\) is vehicle side-slip angle, \(V\) and \(v_y\) are vehicle velocity (approximately equal to longitudinal velocity), and lateral velocity in the vehicle frame, respectively, \(\gamma\) is yaw rate.

It is assumed that vehicle side-slip angle \(\beta\), yaw rate \(\gamma\), and front-wheel steer angle \(\delta\) are small, the equations of longitudinal, lateral and yaw dynamics in the vehicle frame are expressed as follows\(^{(13)}\):

\[
MV = F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr},
\]

(2a)

\[
M\ddot{y} = F_{yfl} + F_{yfr} + F_{yll} + F_{yyr} - MV\gamma,
\]

(2b)

\[
I_\zeta\ddot{\psi} = 2(l_1 F_{yfr} - l_1 F_{yfl}) + \frac{w_r}{2}(F_{xfl} + F_{xfr}) - \frac{w_l}{2}(F_{xrl} + F_{xrr}),
\]

(2c)

\[
V = \dot{s}_g, \quad \gamma = \psi_g, \quad \dot{\psi}_g = v_y + V\epsilon_{\psi}, \quad \dot{\epsilon}_{\psi} = \gamma - \epsilon_{\psi}^{ref},
\]

(2d-2g)

where \(M\) is total vehicle mass, \(l_1, l_r\) are distance from CoG to front and rear axle, \(I_\zeta\) is a moment of inertia of the vehicle body about the \(\zeta\)-axis, \(w_l, w_r\) are front and rear track width, \(F_{xij}, F_{yij}, \forall i \in \{r, l\}, \forall j \in \{r, l\}\) are longitudinal and lateral tire forces, \(s_d\) is vehicle running distance, \(\rho_i\) is the road curvature, \(\epsilon_{\psi} = y - \epsilon_{\psi}^{ref}\) and \(\epsilon_{\psi}^{ref}\) denote the vehicle orientation and position errors, respectively.

2.2 Longitudinal dynamics

To derive the longitudinal dynamics, we first identified the relationship between an brake pressure and a braking force using experimental data, as shown in Fig. 2. In Fig. 2, the experimental data is shown by blue points, and we identified the relation as a linear function depicted by a solid line. For each tire, the relation between the brake pressure \(P_{ij}\) and the longitudinal tire force \(F_{xij}\) is written as

\[
F_{xij} = -k_i P_{ij}, \quad \forall i \in \{r, l\}, \forall j \in \{r, l\},
\]

(3)

where \(k_i\) is a proportional constant of the relation from the brake pressure to the braking force characteristic. Assuming the good mechanical lubrication, we ignore the effects of mechanical and viscous friction of actuators. From Eq. (1a) and Eq. (3), the longitudinal dynamics is expressed as follows

\[
\dot{x}_{ion} = A_{ion}x_{ion} + B_{ion}u_{ion},
\]

(4)

2.3 Lateral dynamics\(^{(14)}\)

Assuming tire characteristics of left and right tire are equal and lateral tire forces \(F_{yi}\) are approximated as linear functions of tire side-slip angles \(\alpha_i\), lateral tire forces \(F_{yi}\) are written

\[
F_{yi} = -K_{yi}\alpha_i, \quad \forall i \in \{l, r\},
\]

(5)

where \(K_{yi}\) is cornering stiffness of front and rear tire. Within the small range of \(\alpha_i\), the following equations hold

\[
\alpha_l = \beta + \frac{l_r}{V} \gamma - \delta, \quad \alpha_r = \beta - \frac{l_l}{V} \gamma.
\]

(6a-6b)

From Eq. (1b), Eq. (1c), Eq. (5) and Eq. (6b), longitudinal dynamics are expressed as follows

\[
x_{lat} = A_{lat}x_{lat} + B_{lat}u_{lat} + E_{lat}P + M\psi_{lat}.
\]

(7)

where \(x_{lat} = [\epsilon_{\psi} \ \dot{\epsilon}_{\psi} \ \epsilon_{\psi}^{ref} \ \dot{\epsilon}_{\psi}^{ref}]^T\), \(u_{lat} = [\delta]\), \(\psi_{lat}\) and \(\epsilon_{\psi}^{ref}\) are the state and the control vectors, respectively. The coefficient matrices \(A_{lat}, B_{lat}, E_{lat}\) and \(M\) are
Air-brake system structure is shown in Fig. 3. This system consists of an air tank, a brake chamber, a solenoid, and a valve. The brake chamber is supplied with the compressed air through the relay valve. We assume that the supply source is always filled with compressed air. The air pressure is adjusted by opening and closing the valve by the command voltage applied to the solenoid. Relationship between time and pressure for step response is shown in Fig. 4. First order delay systems obtained by system identification as shown by solid line. Detailed air-brake model as shown by a broken line. From this result, in this paper, we assume that it has time delay due to the propagation delay by air and the time delay is modeled as a 1st order delay system

\[ \dot{p}_{ij} = -\frac{1}{\tau_i} p_{ij} + \frac{K_{Pl}}{\tau_i} u_{ij}, \forall i \in \{f, r\}, \forall j \in \{l, r\}, \]  

where \( \tau_i \) and \( K_{Pl} \) are time constant and a gain of the air-brake system, \( p_{ij} \) is the brake pressure, \( u_{ij} \) are command voltage. Thus, air-brake model is given by

\[ x_{air} = A_{air} x_{air} + B_{air} u_{air}, \]  

where \( x_{air} = [u_0, 0, u_r, u_l] \) are the state and the control vectors, respectively. \( A_{air}, B_{air} \) are coefficient matrices and are

\[ A_{air} = \begin{bmatrix} 1 & \tau_i & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & \tau_i \\ 0 & 0 & 0 & 1 \end{bmatrix}, \]  

\[ B_{air} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \]  

2.4 Air-brake model

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\[ A_{air} = \begin{bmatrix} 1 & \tau_i & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & \tau_i \\ 0 & 0 & 0 & 1 \end{bmatrix}, \]  

\[ B_{air} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \]  

2.5 State equation

The augmented system of longitudinal, lateral, and air-brake model is written by

\[ \dot{x} = Ax + Bu + Eu, \]  

where \( x = [x^T, x^T, x^T, x^T]^T \) and \( u = [u^T, u^T, u^T]^T \) are the state and the control vectors, respectively. \( A, B, E \) are coefficient matrices.

2.6 Friction circle

The magnitude of combined forces of driving forces \( F_{zi} \) and the lateral forces \( F_{zl,ij} \) acting on the tire and the road surface does not exceed the magnitude of the product of the coefficient of friction \( \mu_{ij} \) and the vertical wheel loads \( F_{zi} \). This relationship is called a friction circle and is expressed as follows

\[ F_{zi}^2 + F_{zl,ij}^2 \leq (\mu_{ij} F_{zi})^2, \quad \forall i \in \{f, r\}, \forall j \in \{l, r\}. \]  

2.7 Load transfer

The motions of braking, accelerating, and turning cause load transfer of vehicles. In this paper, vertical load \( F_{zl} \) is estimated as follows

\[ F_{zl} = W_{zf} + W_{lf} + W_{lat,b}, \]  

\[ F_{zr} = W_{zf} + W_{lr} - W_{lat,b}, \]  

\[ F_{zrl} = W_{zf} - W_{lr} + W_{lat,r}, \]  

\[ F_{zrr} = W_{zf} - W_{lr} - W_{lat,r}, \]  

where static load \( W_{zf}, \) longitudinal load transfer \( W_{lf}, \) lateral load transfer \( W_{lat,f} \) as

\[ W_{zf} = \frac{L}{L + l_x} m_s \pi^2 g \]  

\[ W_{zf} = \frac{l_y}{L + l_x} m_s g \]  

\[ W_{zl} = \frac{h_g}{L + l_x} m_s a_y \]  

\[ W_{lat,f} = \frac{m_s h_q + m_t r_w}{L + l_x} a_y \]  

\[ W_{lat,r} = \frac{m_s h_q + m_t r_w}{L + l_x} a_y, \]  

where \( L = l_f + l_r \) is wheelbase, \( m_s \) is vehicle sprung mass, \( m_{uf} \) is vehicle front and rear sprung mass, \( g \) the is the acceleration due to gravity, \( h_q \) is the height of CoG, \( r_w \) is the tire effective rolling radius.
3. PROPOSED CONTROLLER

The overall structure of our proposed control system is summarized in Fig. 5. The blocks in the controlled object are the most detailed model as we are available. Especially, we originally developed the air-brake model and pressure/force model. In this section, we describe the proposed controller. It consists of a feedforward controller and a model predictive control (MPC). The idea of MPC is to obtain a control input that minimizes a given objective function via solving a constrained finite-time optimization problem at each sampling time. The finite prediction is calculated based on the control model described in section 2. Based on this finite prediction, the behavior of a controlled object is optimized over a finite future.

3.1 Feedforward controller as steering angle input

From the steering angle \( \delta \) to the yaw rate \( \gamma \) in the steady-state circular, the feedforward steering angle \( \delta_{FF} \) is expressed by

\[
\delta_{FF} = (1 + A_{SF} V^2) L \rho_f
\]

where \( A_{SF} \) is stability factor and given as

\[
A_{SF} = \frac{M l_f K_f - l_i K_{yr}}{2 L^2 K_y R_{yr}}
\]

3.2. Objective function

In this paper, the objective function to be minimized is designed as follows

\[
J := \left\| \Delta x_{ref} \right\|_Q + \sum_{k=0}^{n-1} \left( \left\| \Delta x_{f+k} \right\|_Q + \left\| \Delta u_{f+k} \right\|_R \right) + \left\| e \right\|_P^2
\]

where \( \Delta x_i = x_i - x_i^{ref} \) is the error between the state \( x_i \) and the reference state \( x_i^{ref} \) at time \( t \). \( u_{i+k} \) is input of each step, \( \Delta u_{i+k} \) is slew rate on the input \( u \) is slack variable, \( Q \succ 0, Q_i \succ 0 \) and \( R \succ 0 \) is weight matrices, \( P \) is the weight coefficient of the slack variable, \( H \) is the prediction horizon.

3.3. Constraints

Pressure limit considering friction circle expressed by Eq. (11) is given

\[
0 \leq P_{ij} \leq \frac{1}{k_{by}} \sqrt{(\mu_i l_j)^2 - P_{ij}^2}
\]

Also, command voltage is limited as follows

\[
u_{\text{min}} \leq u_{ij} \leq u_{\text{max}}, \quad \forall i \in \{0, \ldots, n\}, \forall j \in \{l, r\}
\]

Furthermore, to avoid lane departure, lateral error \( e_y \) is limited as follows

\[
-\epsilon_{y\text{max}} \leq e_y \leq \epsilon_{y\text{max}}
\]

For avoidance of the infeasibility of the optimization problem, the constraint on state and input are relaxed by the slack variable \( \epsilon \) as follows:

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\[
\begin{align*}
\mathbf{u}_{\min} - \varepsilon \mathbf{v}_{\min} & \leq \mathbf{u}_{t+k|t} \leq \mathbf{u}_{\max} + \varepsilon \mathbf{v}_{\max}, \\
\mathbf{x}_{\min} - \varepsilon \mathbf{v}_{\min} & \leq \mathbf{x}_{t+k|t} \leq \mathbf{x}_{\max} + \varepsilon \mathbf{v}_{\max}, \\
\varepsilon & \geq 0,
\end{align*}
\]

where \( \mathbf{v}_{\min}, \mathbf{v}_{\max} \) are Equal Concern for Relaxation (ECR). If the ECR is zero, it means a hard constraint. The larger the ECR value, the more relaxed the constraint.

### 4. NUMERICAL SIMULATION

In this section, we verify the proposed system by co-simulation on MATLAB/Simulink-TruckSim. TruckSim vehicle model is built based on various experiments and represents the dynamic behavior of the real vehicle with high fidelity. To verify the effectiveness of the proposed system, TruckSim vehicle model is formulated as a controlled object in the joint platform as shown in Fig. 6. This truck model is a two-axis four-wheel, front-wheel steering, and considers the dynamic propagation delay of air-brake given as Eq. (8). Block diagram of MPC system is shown in Fig. 5. The steering wheel angle \( \delta \) is given as follows

\[
\delta = n_{\text{gear}} (\delta + \delta_{\text{FF}}),
\]

where \( n_{\text{gear}} \) is steering gear ratio, \( \delta_{\text{MPG}} \) and \( \delta_{\text{FF}} \) are front wheel steer angle input calculated by MPC and feedforward (FF) respectively. The brake pressure is converted to the braking torque as follows

\[
T_{ij} = r_w F_{ij} = -r_w k_b P_{ij}, \quad \forall i \in \{f, r\}, \forall j \in \{l, r\}.
\]

The simulation scenario is braking in a turn as shown in Fig. 7. The road surface environment is a split-\( \mu \) road. The road surface
Fig. 10 Torque (MPC w/ considering time delay: the proposed method)

(a) Front
(b) Rear

Fig. 11 Torque (MPC w/o considering time delay: the comparison method)

(a) Front
(b) Rear

Fig. 12 Vertical Forces

(a) MPC w/ considering time delay
(b) MPC w/o considering time delay

Fig. 13 Workload of vertical force of the rear right wheel

(a) MPC w/ considering time delay
(b) MPC w/o considering time delay
The truck was a circular path and the turning radius is \( r = 152.4 \) m. The initial vehicle speed \( V_0 \) is 70 km/h. The driving scenario is counterclockwise in the circular path, and the reference speed \( V_{\text{ref}} \) is 70 km/h during the first 2 seconds and 0 km/h after 2 s. Refractive state \( x_{\text{ref}} \) are set as \( x_{\text{ref}} = [x_{d}^T \; \nu_{\text{ref}} \; T_{\text{ref}}^T]^T \).

The control parameters are set as in Table 2. Initial input \( u_0 \) and initial state \( x_0 \) are set as \( u_0 = 0_{6} \) and \( x_0 = [0 \; V_0 \; 0_{3}^T]^T \) respectively. To verify the effectiveness of the proposed controller, we compared with the state of the control model the case of \( x = [x_{l_{\text{ion}}}^T \; x_{r_{\text{lat}}}^T \; x_{r_{\text{air}}}^T]^T \in \mathbb{R}^{10} \) and the case of \( x = [x_{l_{\text{ion}}}^T \; x_{r_{\text{lat}}}^T]^T \in \mathbb{R}^{6} \) as follows Eq. (4), (7), and (9). In the comparison method, we assume that the brake pressure \( P_i \) is proportional to the command voltage \( u_i \) as follows

\[
P_{ij} = K_p u_{ij}, \quad \forall i \in \{f, r\}, \forall j \in \{l, r\}.
\]  

The results of the simulation with or without considering the delay of the air-brake are shown in Fig. 8 to Fig. 13. Fig. 8 is the vehicle's driving trajectory in the absolute coordinates, Fig. 9(a) and (b) are time evolution of the vehicle speed, four wheels speed, the target vehicle speed. Time variation of the braking torque of four wheels in the case of with and without considering the time delay of the air-brake as shown in Fig. 10 and Fig. 11. The vertical forces of each wheel as shown in Fig. 12. Tire workload of the rear left wheel \( \eta_{rl} \) as shown in Fig. 13. \( \eta_{rl} \) is defined as

\[
\eta_{rl} = \frac{P_{l_{\text{err}}}^2 + P_{r_{\text{err}}}^2}{P_{l_{\text{err}}}^2}
\]  

and it satisfies \( \eta_{rl} \leq \mu_{rl} \) from the relationship Eq. (11). G-G diagram is shown as Fig. 14. G-G diagram is an expressed relationship between longitudinal acceleration \( a_x \) and lateral acceleration \( a_y \).

In Fig. 9(a), the smooth braking is achieved without the tire slip. On the other hand, in the case of without considering the time delay of air-brake of Fig. 9(b), the right rear wheel is skidded under braking. In both Fig. 10 and Fig. 11, the front wheel side exhibit a larger braking torque than rear wheels. Besides, at the start time of braking (2 s), the lower limit of the braking torque in the right wheels became larger than the lower limit of the braking torque in the left wheels. This is due to the load transfer represented as Eq. (12) during braking as shown in Fig. 12. In Fig. 12, unlike the 3DoF model as shown by solid line, the response of the TruckSim model as shown by a broken line is vibrational. It is due to model modeling error such as rolling and pitching. In Fig. 10 and Fig. 11, the rise time of the proposed method is faster because the proposed method explicitly considers the air propagation delay of the air-brake represented by Eq. (8). In the case of Fig. 11, the temporal delay is not explicitly considered as a constraint. Therefore, the delay of brake torques due to the influence of the modeling error occurs. Thus, the friction circle of the control model and the controlled object did not match by this propagation delay as shown in Fig 13. As show in Fig. 14. In the case of both, the acceleration on the center of gravity of the vehicle is similar. By relationship of friction circle, longitudinal acceleration and lateral acceleration is under the limit of the tire force as

\[
\sqrt{a_x^2 + a_y^2} \leq \mu g.
\]  

However, in the complex running scenario such as mu-slip road, it is insufficient in the relation of the Eq. (26) since \( \mu \) of each wheel is different value. Thus, considering about friction circle each wheel expressed as Eq. (11) is necessary. In the complex running scenario like Fig. 7, the effect of the delay of the actuator tends to occur. In the case of Fig. 10, considering the dynamic characteristics of the air-brake, a time delay is modeled, and the control input which predicted the propagation delay is calculated. From the above, the braking performance of both the stability and the lane following performance is improved by the model predictive control considering and the propagation delay of the air-brake.

5. CONCLUSION

In this study, we designed a controller that controls both the lane following performance and the running stability considering the propagation delay of the heavy-duty commercial vehicle based on model predictive control considering the vehicle limit performance and air-brake propagation delay. Besides, the usefulness of the control, considering the characteristics of the air-brake was verified through the braking motion simulation with the detailed full dynamics model, including the detailed air-brake model. As a future work, we will apply the proposed method to several types of HDCVs to verify the wide-range applicability. Moreover, it is required to build the estimation method of physical parameters, including a road friction coefficient, and to develop the compensation method for parameter variation.

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REFERENCES

(1) H. Hu, et al.: Plug and play distributed model predictive control for heavy duty vehicle platooning and interaction with passenger vehicles, 2018 IEEE Conference on Decision and Control.
(2) G. Guo, Q. Wang: Fuel-efficient en route speed planning and tracking control of truck platoons, IEEE Transactions on Intelligent Transportation Systems, Vol.20, No.8, p.3091-3103 (2018).

(3) T. Kaneko, et al.: Construction and Parameter Identification Method of Vehicle Dynamics Model for Steering Control System of Heavy Vehicle, Transactions of Society of Automotive Engineers of Japan, Vol.41, No.6, p. 1231-1236 (2010) (in Japanese).

(4) L. Li, et al.: Model predictive control-based efficient energy recovery control strategy for regenerative braking system of hybrid electric bus, Energy Conversion and Management, Vol.111, p. 299-314 (2016).

(5) R. Kianfar, et al.: Combined longitudinal and lateral control design for string stable vehicle platooning within a designated lane, 17th International IEEE Conference on Intelligent Transportation Systems (ITSC), p. 1003-1008 (2014).

(6) H. Wi, H. Park, and D. Hong.: Model Predictive Longitudinal Control for Heavy-Duty Vehicle Platoon Using Lead Vehicle Pedal Information, International Journal of Automotive Technology, Vol.21, No.3, p. 563-569 (2020).

(7) S. Lee, M. Kasahara, and Y. Mori: Roll Damping Control of a Heavy Vehicle under the Strong Crosswind, IFAC Proceedings, Vol.46, No.21, p. 219-224 (2013).

(8) H. Yuan, D. Zhang, and T. J. Gordon: Road vehicle roll-over prevention torque vectoring via model predictive control, 2017 36th Chinese Control Conference (CCC), p. 9401-9406 (2017)

(9) X. Zhang, et al.: Contour line of load transfer ratio for vehicle roll-over prediction, Vehicle System Dynamics, Vol.55, No.11, p. 1748-1763 (2017).

(10) M. Jalali, et al.: Model predictive control of vehicle roll-over with experimental verification, Control Engineering Practice, Vol.77, p. 95-108 (2018).

(11) A. Vahidi, A. Stefanopoulou, and H. Peng: Adaptive model predictive control for coordination of compression and friction brakes in heavy duty vehicles, International Journal of Adaptive Control and Signal Processing, Vol.20, No.10, p. 581-598 (2006).

(12) F.H.M., da Rocha, et al.: Model Predictive Control of a Heavy-Duty Truck Based on Gaussian Process, 2016 XIII Latin American Robotics, Vol.20, No.10, p. 97-102 (2016).

(13) R. Atiya, R. Orjuela, and M. Bassat: Combined longitudinal and lateral control for automated vehicle guidance , Vehicle System Dynamics, Vol.55, No.2, p. 261-279 (2014).

(14) H. Peng, and M. Tomizuka: Preview control for vehicle lateral guidance in highway automation, 1991 American Control Conference, p. 3090-3095 (1991).

(15) W. Cho et al.: Estimation of tire forces for application to vehicle stability control, IEEE Transactions on Vehicular Technology, Vol.59, No.2, p. 638-649 (2009).

(16) M. Abe: Vehicle handling dynamics: theory and application. Butterworth-Heinemann (2015).

(17) A. Bemporad, M. Morari, N. L. Ricker: Model Predictive Control Toolbox User's Guide, The mathworks (2010).