Abstract: In this paper, we propose a feedforward compensation method to enhance the performance of the lateral guidance system of heavy vehicles on automated highway systems (AHS). The feedforward controller blends well with the linear robust feedback controller. The proposed feedforward compensation is motivated by the analogy between a vehicle lateral control system on curved roads and a mechanical system with Coulomb friction. The sliding mode controller for vehicle lateral guidance is shown to inherently involve a feedforward compensation term. Performance of a tractor-semitrailer combination vehicle under linear robust feedback control is used as the base line to show by simulations the enhancement of performance by fixed gain and adaptive feedforward controllers. The tracking performance is further improved by sliding mode control. In view that the sliding mode controller is more complex than the linear controller, the combination of the linear feedback controller and the feedforward controller is at a mid-point between ease of implementation and control performance. Copyright © 2002 IFAC

Keywords: Automated Highway Systems, Vehicle Control, Lateral Control, Feedforward Compensation, Nonlinear Control, Sliding Mode Control, Heavy Vehicles, Tractor-Semitrailer
to focus research on AHS (Shladover, 1991). The Automated Highway Systems proposed by PATH is a hierarchial control system with four control layers: network layer, link layer, planning layer and regulation layer (Varaiya, 1993). In the regulation layer, controls are achieved at the vehicle dynamics level by the longitudinal controller and lateral controller. In recent years, heavy vehicles are considered to be primary candidates for deployment of AHS. This paper concentrates on the lateral control of heavy vehicles using a tractor-semitrailer combination as an example.

Researchers in PATH conducted extensive studies on the lateral control of passenger vehicles (Patwardhan et al., 1997b; Patwardhan et al., 1997a; Peng and Tomizuka, 1993; Pham et al., 1994). In 1993, PATH initiated research program for heavy vehicles and since then there has been active research and steady progress covering modeling (Chen and Tomizuka, 1995; Tai and Tomizuka, 1998), system analysis (Wang and Tomizuka, 1998), controller designs (Wang and Tomizuka, 1999; Wang and Tomizuka, 2000; Tai and Tomizuka, 1999; Tai and Tomizuka, 2000) and experimentation (Hingwe et al., 1999). The model of a heavy vehicle system is nonlinear and it has model uncertainties including both unmodeled dynamics and parametric uncertainties.

There have been mainly two independent approaches to the lateral control of heavy vehicles: linear robust control and nonlinear robust control. Wang and Tomizuka (1998) designed a lead-lag controller based on the linear analysis and a linear robust $H_\infty$ controller assuming a first order dynamics of the steering subsystem (Wang and Tomizuka, 1999). Tai and Tomizuka (1999) designed a Sliding Mode Controller (SMC) to deal with model uncertainties. SMC involves a switching function or a saturation function, which have adverse effects on performance. To improve the performance robustness and passenger comfort, they also proposed an Adaptive Robust Control (ARC) scheme. ARC maintained high gain nature at a reasonable level and achieved robust performance and passenger comfort by adapting the coefficients of the tire cornering stiffness while guaranteeing the transient performance and stability. There have also been some experimental results reported for linear controllers (Hingwe et al., 1999). However, there has been no experimental results reported for nonlinear controllers.

Usually, nonlinear controllers make use of more detailed knowledge of the dynamics of the plant than linear controllers, and intuitively they are supposed to perform better than linear controllers. However, nonlinear controllers are more costly in terms of implementation and sometimes it is impractical to implement them at all. On the other hand, there are rich design methodologies and software tools available for analysis and design of linear controllers.

In this paper, we propose to use a feedforward compensator to augment the linear robust feedback controllers for reduced lateral tracking errors while sustaining a reasonable customer comfort. This is motivated by the observation that a centrifugal force to a vehicle lateral control system is a Coulomb friction force to a mechanical system. Two feedforward compensators are designed: a fixed-gain feedforward compensator and an adaptive feedforward compensator. Furthermore, it is recognized that nonlinear controllers such as a sliding mode controller for lateral control of vehicles inherently involve a feedforward compensation term and linear feedback terms inside its boundary layer when a saturation function is used instead of a sign function for the reduction of chattering.

The organization of this paper is as follows. In the next section, the motivation of using feedforward compensation is presented and two feedforward compensators are proposed. In section 3, simulation results of a linear robust feedback controller with each of the two feedforward compensators are presented and they are compared with that of the same linear robust feedback controller. Nonlinear controllers are analyzed and their relation to feedforward compensators are studied in section 4. Conclusions are given in section 5.

2. LINEAR ROBUST FEEDBACK CONTROLLER WITH FEEDFORWARD COMPENSATION

In this section, we will give a brief review of the linear model of a tractor-semitrailer vehicle system, and propose a feedforward compensation method for performance enhancement.

2.1 Linear model of a tractor-semitrailer vehicle system

The linear control model of a tractor-semitrailer vehicle system in the road coordinate system is given by

$$M_r \ddot{q}_r + D_r \dot{q}_r + K_r q_r = F_r \delta_f + E_1 \dot{e}_d(t) + E_2 \ddot{e}_d(t),$$

$$y_s = \begin{pmatrix} 1 & d_s & 0 \end{pmatrix}^T q_r,$$

where

$$q_r = \begin{pmatrix} y_r & \dot{e}_r & \ddot{e}_f \end{pmatrix}^T.$$  (2)

The matrices, $M_r$, $D_r$, $K_r$, $F_r$, $E_1$ and $E_2$, are as given in the Appendix B of (Tai, 2001). In the above generalized coordinates, $y_r$ is the lateral
displacement of the tractor’s center of gravity, $\varepsilon_r$ is the tractor’s yaw error measured in the road coordinate system and $\varepsilon_f$ is the articulation angle. The system input is the tractor’s front wheel steering angle $\delta_f$ and the system output $y_s$ is the lateral tracking error of the virtual sensor located at the look-ahead distance of $d_s$. In Eq. (1), $\dot{\varepsilon}_d$ and $\dot{\varepsilon}_d$ are the desired tractor’s yaw rate and rate of change of the desired yaw rate, respectively.

In linear control designs they are treated at disturbances to the system. $\dot{\varepsilon}_d$ is related to the vehicle speed $V_x$ and the road curvature $\rho$ by

$$\dot{\varepsilon}_d = V_x \rho.$$  \hspace{1cm} (3)

Define a $6 \times 1$ state vector as

$$x_r = \begin{pmatrix} q_r \\ \dot{q}_r \end{pmatrix}. \hspace{1cm} (4)$$

Then the linear state-space model of the vehicle lateral control system can be obtained as

$$\dot{x}_r = \begin{pmatrix} 0 & I \\ -M_{r}^{-1}K_{r} & -M_{r}^{-1}D_{r} \end{pmatrix} x_r + \begin{pmatrix} 0 \\ M_{r}^{-1}F_{r} \end{pmatrix} \delta_f + \begin{pmatrix} 0 \\ M_{r}^{-1}E_1 \end{pmatrix} \dot{\varepsilon}_d + \begin{pmatrix} 0 \\ M_{r}^{-1}E_2 \end{pmatrix} \ddot{\varepsilon}_d. \hspace{1cm} (5)$$

It is noted that the system matrices in Eqs. (1) and (5) depend on the vehicle speed, tire cornering stiffness and load configuration on the semitrailer. Details of this aspect are documented in (Wang and Tomizuka, 1998).

Notice that the desired yaw rate, $\dot{\varepsilon}_d$, appears in the linearized model. In designing linear controllers, the $\dot{\varepsilon}_d$-related terms are treated as disturbances coming from the road, and linear controllers are designed so that they reject not only the road disturbances but also other disturbances such as wind gust. In the presence of such disturbances, the steady-state tracking error is directly affected by the linear gain of the controller. The larger the linear gain, the smaller the steady-state tracking error. On the other hand, a larger linear gain may excite unmodeled dynamics and may induce oscillations.

2.2 Controller design

By examining the sources of the road disturbances and the ways they enter the model equations, we find the following analogy between the lateral dynamics of a vehicle system and the dynamics of a mechanical system with Coulomb friction.

When a vehicle is on a straight road, $\dot{\varepsilon}_d$ is zero, and therefore there is no road disturbance. When the vehicle is negotiating a curve and turning left, there are centrifugal forces acting at the centers of gravity of the tractor and of the semitrailer. These centrifugal forces point right and their magnitude is proportional to $V_x \dot{\varepsilon}_d$ as shown in Fig. 1(a). To be precise, the centrifugal forces are proportional to $V_x \dot{\varepsilon}_1$ and $V_x \dot{\varepsilon}_2$, respectively. But, at steady state, we have $\dot{\varepsilon}_1 = \dot{\varepsilon}_2 = \dot{\varepsilon}_d$, and hereafter in this paper, we only consider the steady state case. When the vehicle is turning right, the centrifugal forces are proportional to $V_x \dot{\varepsilon}_d$ and they point left as shown in Fig. 1(b). In other words, the centrifugal forces are proportional to $-V_x \dot{\varepsilon}_d$ and are discontinuous.

Recall that, for mechanical systems with Coulomb friction forces (see Fig. 2), the Coulomb friction forces are proportional to $-\dot{x}$ and discontinuous. That is, the centrifugal forces of a vehicle negotiating a curve are analogous to the Coulomb friction forces.

For a mechanical system with Coulomb friction, if they can be obtained or estimated, adding a feedforward term to compensate for the disturbances (the friction forces) is a practical and efficient approach. Motivated by this, we propose to add a feedforward compensator to a linear feedback control system to attenuate the road disturbances as shown in Fig. 3. We take the input to the feedforward compensator as $V_x \dot{\varepsilon}_d$. As such, the feedforward compensator has a built-in switch to turn on and off the compensator based on needs. That is, when the vehicle is negotiating a curve, $\dot{\varepsilon}_d$ is nonzero and therefore the feedforward compensator is on, and when the vehicle is travelling on a straight section, $\dot{\varepsilon}_d$ is zero and therefore the switch is off.

A possible candidate for the feedforward compensator is the inverse dynamics of the system, from the road disturbances $V_x \dot{\varepsilon}_d$ to the output $y_s$. But, as we have learned, the vehicle dynamics has
model uncertainties and it may not exactly compensate for the disturbances as we expect. We propose to use a constant gain feedforward compensator with the constant equal to the inverse of the linear gain of the disturbance dynamics. This gain is a function of vehicle inertia and dimensional parameters as well as tire cornering stiffness. To account for the parametric uncertainties in the constant feedforward compensator, we introduce an adaptation to the constant gain based on the lateral tracking error at the c.g. of the tractor.

3. SIMULATION RESULTS

Simulation scenario is described by the desired trajectory and the longitudinal velocity profile. The desired trajectory is as shown in Fig. 4. It is the same as the actual test track in Crows Landing, a small town 90 miles south of Berkeley. The test track consists of three curved sections extended by two straight sections. The radii of the curved sections are 800 m. The longitudinal velocity profile is as given in the upper left plot of Fig. 5. It is taken from one of the experiments conducted at Crows Landing. During experiments, the longitudinal velocity is controlled by the driver. Given the desired trajectory and the velocity profile, the desired yaw rate, $\dot{\varepsilon}_d$, can be obtained by Eq. (3), and it is shown in the second plot of the left column of Fig. 5.

A linear robust feedback controller was designed based on the loop shaping technique, and the tracking performance under this controller was as shown by the solid lines in Fig. 5. Since the objective here is to show advantages of feedforward compensation, the details of the feedback controller as well as its design are omitted.

4. COMPARISON WITH NONLINEAR ROBUST CONTROLLERS

4.1 Vehicle lateral control model

The simplified nonlinear control model of tractor-trailer heavy vehicles is (Tai and Tomizuka, 1999)

$$M(q)\ddot{q} + c(q, \dot{q}) = 2C_{\alpha f}[1, l_f, 0]^T\delta_f$$

where the generalized coordinate $q$ is $q(t) = [\int_{0}^{t} V_r(\tau)d\tau, \varepsilon_1, \varepsilon_f]$ and $\delta_f$ is the control input, the tractor’s front wheel steering angle, and $M(q)$ and $c(q, \dot{q})$ are as given in Appendix A. It should be noted that the linear model given in Eq. (1) is obtained from Eq. (6) by introducing the small angle assumption for articulation angle and reformulating it in the road coordinate system (Tai and Tomizuka, 1999). Then, the output dynamics is given by

$$\dot{y}_s = \dot{V}_y + d_s\varepsilon_1 + V_x\dot{\varepsilon}_r$$

where $\dot{V}_y$ and $\dot{\varepsilon}_1$ can be obtained from Eq. (6) (Tai and Tomizuka, 1999) and they can be written as

$$\dot{V}_y = b_1(q)\delta_f + f_1(q, \dot{q})$$

and

$$\dot{\varepsilon}_1 = b_2(q)\delta_f + f_2(q, \dot{q})$$.

Then, the output dynamics given in Eq. (7) becomes

$$\ddot{y}_s = (b_1(q) + b_2(q))\delta_f + (f_1(q, \dot{q}) + f_2(q, \dot{q})) + V_x\dot{\varepsilon}_r = b_x(q)\delta_f + f_0(q, \dot{q}) + V_x\dot{\varepsilon}_r,$$

where

$$b_x(q) = b_1(q) + b_2(q),$$

$$f_0(q, \dot{q}) = f_1(q, \dot{q}) + f_2(q, \dot{q}).$$
where \( b_x(q) = b_1(q) + b_2(q) \) and \( f_0(q, \dot{q}) = f_1(q, \dot{q}) + f_2(q, \dot{q}) \). A straightforward algebraic computation shows that \( f_0(q, \dot{q}) \) contains \(-V_x \dot{\varepsilon}_d\) term, that is,

\[
f_0(q, \dot{q}) = f_x(q, \dot{q}) - V_x \dot{\varepsilon}_d.
\]  

(11)

This can also be recognized from the details of the matrix \( M \) and vector \( c \) given in Appendix A. Notice that each \( c(i) \) contains a term \( \ast_i V_x \dot{\varepsilon}_1 \) and the coefficient \( \ast_i \) is the same as the \((i, 1)\) element of \( M \), i.e., \( \ast_i = M(i, 1), i = 1, 2 \) and 3. So, Eq. (6) can be reformulated as

\[
M \begin{bmatrix} V_y + V_x \dot{\varepsilon}_1 \\ \dot{\varepsilon}_f \\ \dot{\varepsilon}_r \\ \delta_f \end{bmatrix} + c_1(q, \dot{q}) = 2C_{af} \begin{bmatrix} 1 \\ l_f \\ 0 \end{bmatrix} \delta_f.
\]  

(12)

From Eq. (12), it is easy to see that \( V_y \) actually contains \(-V_x \dot{\varepsilon}_1\) term. In the mean time, \( V_y + V_x \dot{\varepsilon}_1 \) is the lateral acceleration of the tractor’s center of gravity and Eq. (6) can be viewed as a force balance equation by Newton’s second law.

By combining Eqs. (10) and (11), we have

\[
\ddot{y}_s = b_x(q) \delta_f + f_x(q, \dot{q}) - V_x \dot{\varepsilon}_1 + V_x \dot{\varepsilon}_r
\]

\[
= b_x(q) \delta_f + f_x(q, \dot{q}) - V_x \dot{\varepsilon}_d
\]  

(13)

Recall that the nonlinear robust controllers, such as a sliding mode controller, involve a feedback linearization term. Therefore, the nonlinear controllers have the form of

\[
\delta_f = -\frac{1}{b_x(q)}(f_x(q, \dot{q}) - V_x \dot{\varepsilon}_d + \cdots) + \cdots
\]

\[
= \frac{1}{b_x(q)}(V_x \dot{\varepsilon}_d) - \frac{1}{b_x(q)}(f_x(q, \dot{q}) + \cdots) + \cdots.
\]  

(14)

Equation (14) suggests that the nonlinear controllers inherently have a feedforward compensation which corresponds to the fixed gain feedforward control. For comparison, Fig. 6 shows the simulation results of a sliding mode controller and the above mentioned three linear robust controllers. As we can see from the figure, the sliding mode controller offers the best tracking performance as expected. However, the implementation of nonlinear controllers is not trivial and much harder than that of linear controllers. Therefore, linear robust controllers with feedforward compensators are a mid-point between linear feedback controllers and nonlinear controllers from the view point of ease of implementation and control system performance.

5. CONCLUSIONS

An analogy between the vehicle lateral control system and a mechanical system with a Coulomb friction is identified. Motivated by the friction compensation practices for the latter, feedforward compensators, a fixed gain and an adaptive compensators, are designed to enhance the tracking performance of linear feedback controllers without sacrificing other performances. It is also pointed out that the nonlinear controllers actually involve the feedforward compensator corresponding to the fixed gain compensator. A linear robust feedback controller with a feedforward compensation of the road disturbance is a mid-point in terms of ease of implementation and control performances.

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REFERENCES


Appendix A. NONLINEAR CONTROL MODEL OF A TRACTOR-SEMITRAILER IN THE VEHICLE COORDINATE SYSTEM

The symbols are defined as follows: $m_1$: mass of the tractor; $m_2$: mass of the semitrailer; $I_{2y}^{2}$: moment of inertia of the tractor about its center of gravity (c.g.) along the vertical axis; $I_{2z}^{2}$: moment of inertia of the semitrailer about its c.g. along the vertical axis; $l_{11}$: distance from the tractor’s c.g. to its front axle; $l_{12}$: distance from the tractor’s c.g. to its rear axle; $l_{21}$: distance from the semitrailer’s c.g. to its rear axle; $l_{22}$: distance from the tractor’s c.g. to its front axle.

\[ M(1,1) = m_1 + m_2, \]
\[ M(2,1) = -m_2(d_{11} + d_{12} \cos \epsilon f), \]
\[ M(3,1) = -m_2d_{12} \cos \epsilon f, \]
\[ M(1,2) = -m_2(d_{11} + d_{12} \cos \epsilon f), \]
\[ M(2,2) = I_{2x}^{2} + I_{2y}^{2} + m_2(d_{11}^2 + d_{12}^2) + 2m_2d_{11}d_{12} \cos \epsilon f, \]
\[ M(3,2) = I_{2x}^{2} + m_2d_{12}^2 + m_2d_{11}d_{12}, \]
\[ M(1,3) = -m_2d_{12} \cos \epsilon f, \]
\[ M(2,3) = I_{2x}^{2} + m_2d_{12}^2 + m_2d_{11}d_{12}, \]
\[ M(3,3) = I_{2x}^{2} + m_2d_{12}^2; \]

\[ c(1) = (m_1 + m_2)V_2 \epsilon_1 + m_2d_{12} \epsilon_1 \sin \epsilon f + \frac{2}{V_2} (C_{af} + C_{ar} + C_{a1}) \epsilon_1 \]
\[ + \frac{2}{V_2} (l_{f1}C_{af} - l_{11}C_{ar} - (d_{11} + d_{12} + l_{r2})C_{a1}) \epsilon_1 \]
\[ - \frac{2}{V_2} (d_{11} + l_{r2}) \epsilon_1 + 2C_{ar} \epsilon_1. \]

\[ c(2) = -m_2(d_{11} + d_{12} \cos \epsilon f)V_2 \epsilon_1 + m_2d_{12}V_2 \epsilon_1 \sin \epsilon f + 2m_2d_{11}d_{12} \epsilon_1 \sin \epsilon f + m_2d_{12}d_{12} \epsilon_1 \sin \epsilon f + \frac{2}{V_2} (l_{f1}C_{af} - l_{11}C_{ar} - (d_{11} + d_{12} + l_{r2})C_{a1}) \epsilon_1 \]
\[ + \frac{2}{V_2} (l_{f1}C_{af} + l_{11}C_{ar} + (d_{11} + d_{12} + l_{r2})^2C_{a1}) \epsilon_1 \]
\[ + \frac{2}{V_2} (d_{11} + l_{r2})(d_{11} + d_{12} + l_{r2})C_{a1} \epsilon_1 + 2(d_{11} + d_{12} + l_{r2})C_{a1} \epsilon_1. \]

\[ c(3) = -m_2d_{12}V_2 \epsilon_1 \cos \epsilon f - m_2d_{12}V_2 \epsilon_1 \sin \epsilon f + m_2d_{11}d_{12} \epsilon_1 \sin \epsilon f + \frac{2}{V_2} (d_{11} + l_{r2})C_{ar} \epsilon_1 \]
\[ + \frac{2}{V_2} (d_{11} + l_{r2})(d_{11} + d_{12} + l_{r2})C_{a1} \epsilon_1 + 2(d_{11} + d_{12} + l_{r2})C_{a1} \epsilon_1. \]