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An investigation on motor-driven power steering-based crosswind disturbance compensation for the reduction of driver steering effort

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An investigation on motor-driven power steering-based crosswind disturbance compensation for the reduction of driver steering effort

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This paper describes a lateral disturbance compensation algorithm for an application to a motor-driven power steering (MDPS)-based driver assistant system. The lateral disturbance including wind force and lateral load transfer by bank angle reduces the driver’s steering refinement and at the same time increases the possibility of an accident. A lateral disturbance compensation algorithm is designed to determine the motor overlay torque of an MDPS system for reducing the manoeuvring effort of a human driver under lateral disturbance. Motor overlay torque for the compensation of driver’s steering torque induced by the lateral disturbance consists of human torque feedback and feedforward torque. Vehicle–driver system dynamics have been investigated using a combined dynamic model which consists of a vehicle dynamic model, driver steering dynamic model and lateral disturbance model. The human torque feedback input has been designed via the investigation of the vehicle–driver system dynamics. Feedforward input torque is calculated to compensate additional tyre self-aligning torque from an estimated lateral disturbance. The proposed compensation algorithm has been implemented on a developed driver model which represents the driver’s manoeuvring characteristics under the lateral disturbance. The developed driver model has been validated with test data via a driving simulator in a crosswind condition. Human-in-the-loop simulations with a full-scale driving simulator on a virtual test track have been conducted to investigate the real-time performance of the proposed lateral disturbance compensation algorithm. It has been shown from simulation studies and human-in-the-loop simulation results that the driver’s manoeuvring effort and a lateral deviation of the vehicle under the lateral disturbance can be significantly reduced via the lateral disturbance compensation algorithm.

Keywords: crosswind; bank angle; disturbance compensation; motor-driven power steering (MDPS); human torque; overlay torque

Nomenclature

- $m$: vehicle mass (kg)
- $l_f$: distance from centre of gravity to the front axle of the vehicle (m)
- $l_r$: distance from centre of gravity to the rear axle of the vehicle (m)
- $v_x/v_y$: longitudinal/lateral velocity of the vehicle (m/s)

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\[
\dot{\psi} \quad \text{vehicle yaw rate (rad/s)}
\]
\[
F_{f/r,x} \quad \text{front/rear longitudinal tyre force (N)}
\]
\[
F_{f/r,y} \quad \text{front/rear lateral tyre force (N)}
\]
\[
\delta_f \quad \text{front steering angle (rad)}
\]
\[
I_z \quad \text{vehicle z-axis inertia (N m s}^2/\text{rad)}
\]
\[
F_{w/M_w} \quad \text{lateral force/yaw moment due to lateral disturbance (N/N m)}
\]
\[
\rho \quad \text{air density (kg/m}^3\text{)}
\]
\[
C_{f,i}/C_{n,i} \quad \text{i-axis air force/moment coefficient}
\]
\[
A \quad \text{reference area of the vehicle (m}^2\text{)}
\]
\[
t_f \quad \text{half the track width (m)}
\]
\[
J_s \quad \text{steering inertia (N m s}^2/\text{rad)}
\]
\[
b_s \quad \text{steering damping coefficient (N m s)}
\]
\[
K_s \quad \text{steering stiffness (N m/rad)}
\]
\[
N_g/N_m \quad \text{steering gear/motor reduction gear ratio}
\]
\[
y_r \quad \text{lateral position error (m)}
\]
\[
\psi_d \quad \text{desired yaw angle (rad)}
\]
\[
\dot{\psi}_d \quad \text{desired yaw rate (rad/s)}
\]
\[
y_r \quad \text{change in lateral position error (m)}
\]
\[
\Delta t \quad \text{change in time (s)}
\]
\[
\delta_{des} \quad \text{desired front steering angle (rad)}
\]
\[
L_p \quad \text{preview distance (m)}
\]
\[
\alpha_{\psi} \quad \text{yaw angle error (rad)}
\]
\[
C_{i,y} \quad \text{i-th lateral tyre force stiffness (N/rad)}
\]
\[
T_{align,i} \quad \text{i-th tyre aligning torque (N m)}
\]
\[
\xi_i \quad \text{i-th trail of the tyre (m)}
\]
\[
\alpha_i \quad \text{i-th tyre slip angle (rad)}
\]
\[
\lambda_i \quad \text{i-th tyre slip ratio}
\]
\[
\rho_{des} \quad \text{desired road curvature (1/m)}
\]
\[
T_m \quad \text{motor assist torque (N m)}
\]
\[
T_h \quad \text{human torque (N m)}
\]
\[
T_{overlay} \quad \text{motor overlay torque (N m)}
\]
\[
T_{assist} \quad \text{motor assist torque (N m)}
\]
\[
\delta_{ff} \quad \text{feedforward steering angle (rad)}
\]
\[
\delta_{SW,des} \quad \text{desired steering wheel angle of a driver model (rad)}
\]
\[
\delta_{SW,DM} \quad \text{steering wheel angle of a driver model (rad)}
\]
\[
\delta_{SW,DS} \quad \text{steering wheel angle of test data via a driving simulator (rad)}
\]
\[
T_{h,DM} \quad \text{steering torque of a driver model (N m)}
\]

1. Introduction

In recent years, lateral disturbance on vehicle stability has caused concern and has been met with great attention by researchers.[1] The main factors of lateral disturbance in actual driving are considered to be crosswind, which is usually experienced on bridges or near sea shores or roads with bank angles. Investigation of the crosswind stability of ground vehicles has been focused on railway vehicles.[2] In passenger cars, the lateral disturbance by crosswind is negligible at low speed, but at the high speed or strong crosswind, it can push the vehicle out of its lane or make it overturn. This lateral disturbance can reduce the driver's control in driving situations, and can be a direct threat to the safety of the driver above a certain grade.[3,4] The stability of the vehicle and the course deviation under strong crosswind were
investigated in previous research.\cite{5} The potential for accidents has increased over recent years as a result of the lighter vehicles for high fuel-efficiency and high vehicle speeds. Newly constructed roadways that are located on high embankments or viaducts also raise the potential for accidents, for wind flows are substantially increased in exposed routes.\cite{6} The lateral load from bank angles also leads to unwanted leaning motion towards the direction of the load.

Motor-driven power steering (MDPS) systems using motors to assist steering torque have become common in production vehicles. The MDPS is expected to replace conventional hydraulic steering systems. The motor-assisted torque can be controlled by the motor overlay torque within a physically feasible range.

In a previous research, a lateral control system with steering torque under lateral disturbance was examined using a steering–vehicle dynamics model without the driver’s steering torque.\cite{7} To investigate overall driver steering characteristics including the steering torque, steering system and vehicle dynamic system, the dynamics of each component system were unified in a state space equation. Section 2 includes the basic vehicle dynamic model under lateral disturbance, and characteristics of the lateral disturbance. The driver behaviour which reflects intent to keep the vehicle within the lane with a preview distance and the overall steering system including the motor assist torque characteristics and the tyre self-aligning torque characteristics will be described. The vehicle lateral position error dynamics, which are modified from the basic vehicle dynamics, will be covered in Section 2. Finally, the overall state space equation of the vehicle–driver–steering dynamics has been developed.

In order to improve driver’s steering behaviour under lateral disturbance, a lateral disturbance compensation system using the motor overlay torque as control input has been proposed. The control target is to get rid of the effect of the lateral disturbance in human steering behaviour, which is the same as a normal driving situation without the lateral disturbance. Linear analysis of the proposed system, ignoring the nonlinear characteristics in mild driving, straight driving or driving on a small-curvature road under lateral disturbance, has been conducted to choose the appropriate overlay torque control strategy. The estimated lateral disturbance is used to improve human steering behaviour in the feedforward control input.

A driver model which represents the behaviour of a driver under lateral disturbance has been developed. The driver model has been investigated via driving simulator tests with test subjects in various driving experiences. The steering behaviour of the driver model has been validated with the driving simulator test results.

The performance of the proposed control strategy has been confirmed by applying the developed driver model and the full vehicle model. The proposed strategy has also been implemented in a human-in-the-loop system with a full-scale driving simulator on a virtual test track (VTT).

2. Vehicle–driver–steering dynamic system

2.1. Vehicle body dynamics

A simplified 3-degree of freedom (DOF) vehicle model has been developed from a full vehicle model from previous research.\cite{8} The 3-DOF vehicle model represents longitudinal, lateral and yaw direction of vehicle motion with the tyre force and alignment torque of each wheel as shown in Figure 1. Roll and pitch motion of the actual vehicle are neglected in the planar model and the suspension dynamics including vertical tyre force are also neglected.

Each state in a 3-DOF vehicle model such as longitudinal velocity, lateral velocity and yaw rate is combined in nonlinear form. The dynamic equations of 3-DOF vehicle model under
lateral disturbance with the angle of incidence $\theta$ are

\begin{align*}
  m(\dot{v}_x - \gamma v_y) &= F_{f,x} + F_{f,x} \cos \delta_f - F_{f,y} \sin \delta_f - F_w \cos \theta, \\
  m(\dot{v}_y + \gamma v_x) &= F_{f,y} + F_{f,y} \cos \delta_f - F_{f,x} \sin \delta_f - F_w \sin \theta, \\
  I_z \dot{\gamma} &= l_f F_{f,y} \cos \delta_f - l_f F_{f,y} - l_f F_{f,x} \sin \delta_f + \frac{d}{2} (\Delta F_{f,x} + \Delta F_{f,x} \cos \delta_f) \\
  &- \sum_{i=1}^{4} T_{\text{align},i} + M_w. 
\end{align*}

\section{Lateral disturbance dynamics}

Wind forces acting on a vehicle were validated mathematically in previous research through numerous tests, including a wind tunnel test, full-size vehicle test and a flat plate test.[1,9] The forces on the vehicle due to crosswind are applied at the centre of gravity of the vehicle in the longitudinal, lateral and vertical axis. Three difference moments are generated by the forces, with roll, pitch and yaw representing each direction. The magnitude of the moment is determined by the magnitude of the force and the distance between the centre of gravity and the representative equivalent point of action. Crosswind force and moment are applied on the vehicle differently according to the shape of the vehicle.[10] Each force and moment is generated along the longitudinal, lateral and vertical axis by crosswind as shown in Figure 2.

The flow over a vehicle moving through still air is nominally symmetric about the vehicle’s plane of symmetry. Lift, pitching moment and drag are therefore the only aerodynamic components. The vertical force on a body close to the ground tends to lift the vehicle. The maximum cornering force of the tyre decreases with the accompanying reduction in load on the tyre. However, the effect is negligible for most vehicles. A pitching moment by the aerodynamic force promotes a tendency to oversteer or understeer with increasing vehicle speed or strong gust blown towards the heading direction of the vehicle. However, this effect is hardly noticeable to the driver.
Figure 2. Aerodynamic force and moments acting on a vehicle.

In driving in a crosswind, the flow around a vehicle becomes asymmetric. A lateral force, yaw moment and a rolling moment are generated. There are also the components of drag, lift and pitching moment, which are normally increased in the crosswind. From experience, it is known that only yaw moment and lateral force are dominant in a vehicle’s behaviour. General mathematical expressions of the force and moment are represented as follows:

$$F_{CW,i} = \frac{1}{2} \rho C_{f,i} v_t^2 A,$$

(2)

$$M_{CW,i} = \frac{1}{2} \rho C_{n,i} v_t^2 Al.$$

(3)

The crosswind coefficients in the $i$th direction, $C_{f,i}$, $C_{n,i}$, are determined by the reference shape of the vehicle. Each coefficient of the force and moment according to the airflow side slip angle which is described as $\beta_{cw}$ in Figure 2 is represented in Figure 3. The coefficients provided in CARSIM software have been used in this study.

The main disturbances from crosswind causing dangerous situations can be considered as yaw moment and lateral force. The lateral disturbance coming from the road bank angle shown in Figure 4 is represented in Equation (4). The yaw dynamics of the vehicle are not disturbed by the bank angle

$$F_{BANK} = mg \frac{\Delta z}{2t_f}.$$

(4)

Since a lateral disturbance exerting on a vehicle is hard to measure in actual driving, several methods for estimating the lateral disturbance are proposed.[11,12].

### 2.3. Steering dynamics

A mathematical representation of the steering dynamics around the king-pin represented in Figure 5 is described as the following equation:

$$J_s \ddot{\delta}_f = -b_s \dot{\delta}_f - K_s \delta_f + N_g N_m T_m + N_g T_h - T_{align},$$

(5)
where $\delta_f$ is the front steering angle, $N_g$ is the steering gear ratio, $N_m$ is the motor gear ratio, and $T_h$ is the driver’s torque. The driver’s torque is applied on the steering wheel in order to follow the desired path. The desired steering angle for tracking the desired path from the perception of the environment within the preview distance is described in Equation (6). The steering control law contains linear gains $k_1$ and $k_2$, which are combined with the lateral error in the current position and the error in the preview distance, respectively, as shown in Figure 6.

$$\delta_{des} = -k_1 y_r - k_2 e_L$$
$$= -k_1 y_r - k_2 (y_r + L_p e_\psi) .$$

(6)

The driver’s torque considering the human neuromuscular system delay is determined to follow the desired path with the difference of the desired steering angle and the current steering angle as shown in Figure 7. A representation of the steering torque originating from the steering
wheel angle for the steer-by-wire system has been studied in previous research.[13]

\[
T_h = \frac{k_T}{1 + \tau_T s} \left( \delta_f - \delta_{\text{des}} \right)
\]
\[
= \frac{1}{1 + \tau_T s} \left[ k_T \{ \delta_f + k_1 y_r + k_2 (y_r + L_p e_\psi) \} \right].
\]  

(7)

The time constant \( \tau_T \) is the time from the perception of the environment to the moment of applying steering torque. The time delay including the perception of the driving environment and the reaction of human muscular system can be evaluated by the driver’s steering reaction test as shown in Figure 8.[14]
\( \tau_d \) and \( \tau_N \) in Figure 8 are perception delay and muscular system delay, respectively. Through various human data, the value of the summation of the delays is evaluated to be 0.4 s. The time difference between the steering angle and the steering torque is shown in Figure 9.

The torque delay from the perception to the moment of steering torque can be represented as the following relation. The summation of the torque delay and the time difference between the steering angle and steering torque \( \tau_s \), which is shown to be 0.15 s in Figure 9, is equal to 0.4 s, which is the overall delay from perception to the moment of action on the steering angle. From this above relation, the time difference from the perception to the moment of action on the steering torque is 0.25 s.

\[ T_{\text{align}} = 2C_{i,y} \xi_i \alpha_i + \Delta T_{\text{align}}, \quad (8) \]

MDPS provides motor assist torque to reduce the driver's steering effort instead of the hydraulic assist torque of conventional hydraulic power steering systems. In general passenger cars, the assist torque characteristic is determined by the velocity of the vehicle and the steering torque of the driver. In the case of high-speed driving, the motor assist torque is small to prevent
the driver’s frequent steering behaviour, which can cause the vehicle to deviate easily from the original path at high speed.

In order to prevent a driver from feeling uncomfortable when the driver manoeuvres a vehicle, the direction of the motor assist torque has the same direction as the driver’s torque. These characteristics are shown in Figure 11.

Motor assist torque shows linear characteristics within a certain region. The proportional rate of the motor assist torque is shown to be small as the speed of the vehicle increases. The motor assist torque can be represented as follows:

\[ T_{\text{assist}} = k_d T_h + \Delta T_{\text{assist}}, \]

where \( k_d = f(v_x) \).

The torque overlaying to the motor assist torque can be allowable in a MDPS system as shown in Equation (5). The overlaid motor torque assists a driver in comfortable driving under lateral disturbance. The overlaid torque is determined through stability analysis of a vehicle–driver–steering dynamic system in the following section.

\[ T_m = T_{\text{assist}} + T_{\text{overlay}}. \]
2.4. Vehicle lateral error dynamics

Figure 12 shows the driver–vehicle system based on the preview distance of the driver. The driver intends to follow the desired path which comes from the driver’s perception of the driving environment.

The lateral position error is defined as the lateral distance between the centreline of the desired path and the centre of gravity of the vehicle. Yaw angle error is defined as the angle difference between the yaw angle of the vehicle and the angle of the tangent line on the desired path. The first derivative of the lateral position error and yaw angle error are represented as follows:[15]

\[
\begin{align*}
\Delta y_r &= v_y \Delta t + v_x e_\psi \Delta t, \\
\dot{y}_r &= v_y + v_x e_\psi, \\
\psi_d &= v_x \rho_{\text{des}} \Delta t, \\
\dot{\psi}_d &= v_x \rho_{\text{des}}.
\end{align*}
\]

Mathematical nonlinear tyre models have been studied widely so far, and various tyre formulae have been developed to represent the behaviour of actual tyres. The proposed tyre models represent the characteristics of the nonlinearities of the tyre forces according to the slip ratio and slip angle as shown in Figure 13.
The characteristics of the longitudinal/lateral tyre force are the summation of the force in the linear region and nonlinear region above certain value of the slip ratio and slip angle.

\[
F_{i,y} = C_{i,y} \alpha_i + \Delta f_{i,y}, \tag{13}
\]

\[
F_{i,x} = C_{i,x} \lambda_i + \Delta f_{i,x}.
\]

The tyre forces can be rewritten with the lateral position error and the yaw angle error in Equations (11) and (12) as shown in Equation (14).

\[
F_{f,y} = C_f \alpha_f + \Delta f_{f,y}.
\]

\[
F_{r,y} = C_r \alpha_r + \Delta f_{r,y}.
\]

Since the longitudinal velocity in a 3-DOF vehicle model in Equation (1) makes the tyre force equation in Equation (14) be nonlinear form, a 2-DOF bicycle model has been used instead of the 3-DOF model for linear analysis of an overall vehicle–driver–steering dynamic system. The 2-DOF bicycle model is designed with the assumptions of constant longitudinal velocity and the same slip angle between the left and right wheels. The lateral dynamics of the vehicle can be rewritten with respect to the lateral position error and yaw angle error with Equation (15).\cite{15}

\[
m \ddot{v}_y + m v_x \dot{\psi} = (2F_{f,y} + 2F_{r,y}) - F_w
\]

\[
I_\psi \ddot{\psi} = 2l F_{f,y} - 2l F_{r,y} - \sum_{i=1}^{4} T_{\text{align},i} + M_w,
\]

where \( F_w = F_{\text{cw},y} + F_{\text{bank}}, \)

\( M_w = M_{\text{cw},z}, \)

From Equations (5)–(15), the state equation of the vehicle–steering–driver system is represented as

\[
\begin{bmatrix}
y_r \\
\dot{y}_r \\
e_\psi \\
\dot{e}_\psi \\
\delta_f \\
\dot{\delta}_f \\
T_h
\end{bmatrix}
= \begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 & 0 \\
0 & A_{22} & A_{23} & A_{24} & A_{25} & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & A_{42} & A_{43} & A_{44} & A_{45} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 \\
A_{61} & A_{62} & A_{63} & A_{64} & A_{65} & A_{66} & A_{67} \\
A_{71} & 0 & A_{73} & 0 & A_{75} & 0 & A_{77}
\end{bmatrix}
\begin{bmatrix}
y_r \\
\dot{y}_r \\
e_\psi \\
\dot{e}_\psi \\
\delta_f \\
\dot{\delta}_f \\
T_h
\end{bmatrix}
+ \begin{bmatrix}
0 \\
0 \\
0 \\
0 \\
0 \\
0 \\
0
\end{bmatrix}
\]

\[
A
\]

\[
B
\]
\[
\begin{bmatrix}
0 & 0 \\
F_{21} & 0 \\
0 & 0 \\
0 & F_{42} \\
0 & 0 \\
0 & 0 \\
F \end{bmatrix} + \begin{bmatrix}
F_w \\
M_w \\
D_2 \\
D_4 \\
0 \\
0 \\
D_6 \\
\end{bmatrix} \rho_{des} + f_{nl}, \tag{16}
\]

where

\[
A_{22} = -\frac{2C_f + 2C_l}{mv_x} \quad A_{23} = \frac{2C_f + 2C_l}{m} \quad A_{24} = -\frac{2C_i l_f + 2C_i l_r}{mv_x} \quad A_{25} = \frac{2C_l}{m}
\]

\[
A_{42} = -\frac{2C_i l_f - 2C_i l_r}{I_z v_x} \quad A_{43} = \frac{2C_i l_f - 2C_i l_r}{I_z} \quad A_{44} = -\frac{2C_i l_f^2 + 2C_i l_r^2}{I_z v_x} \quad A_{45} = \frac{2C_i l_f}{I_z}
\]

\[
A_{62} = \frac{2C_l \xi}{v_x J_s} \quad A_{63} = -\frac{2C_l \xi}{J_s} \quad A_{64} = \frac{2C_l \xi l_f}{v_x J_s} \quad A_{65} = -\frac{2C_l \xi}{J_s}
\]

\[
A_{66} = -\frac{b_s}{J_s} \quad A_{67} = \frac{N_g k_a + N_g}{J_s} \quad A_{71} = \frac{1}{\tau_T} k_T (k_1 + k_2) \quad A_{73} = \frac{1}{\tau_T} k_T k_2 L_p
\]

\[
A_{75} = \frac{1}{\tau_T} k_T \quad A_{77} = -\frac{1}{\tau_T} \quad B_6 = \frac{N_g N_m}{J_s} \quad D_2 = -\frac{2C_i l_f + 2C_i l_r}{mv_x} - v_x
\]

\[
D_4 = -\frac{2C_i l_f^2 + 2C_i l_r^2}{I_z v_x} \quad D_6 = \frac{2C_i l_f \xi}{J_s} \quad F_{21} = \frac{1}{m} \quad F_{42} = \frac{1}{I_z}
\]

\[f_{nl} \text{ in Equation (16) contains the nonlinear terms in the tyre force, the tyre self-aligning torque and the assist torque. The nonlinearities of the equation are assumed to be neglected in the case of the small steering behaviour and vehicle side slip angle, which is shown in a straight driving or a curved road driving with the small curvature.}\]

3. Motor overlay torque control strategy for lateral disturbance compensation

A schematic diagram of the proposed lateral disturbance compensation algorithm is shown in Figure 14. When a vehicle is exposed to lateral disturbance, a driver tries to steer the vehicle to the desired path. In the case of straight driving, the driver steers the vehicle to the direction of the lateral disturbance in order to prevent the vehicle from deviating from the straight path, and the driver keeps imposing steady steering torque with high frequency compared to normal driving situations. In previous research, the linear controller of a steering–vehicle system for lateral control under lateral disturbance was proposed with feedback input based on linear quadratic (LQ) control theory and feedforward control input considering the lateral disturbance and the driver’s torque has been assumed to be zero.[7] In this part, the motor overlay torque control strategy has been proposed on the basis of human steering characteristics including steering torque and the vehicle dynamics by the analysis of the developed mathematical system model. Consequently, the proposed overlay torque input can be represented as a summation of the steering torque feedback input, which reduces the magnitude and the frequency of the human torque, and the feedforward steering torque for compensating for the tyre self-aligning torque generated by the lateral disturbance as follows:

\[u = T_{\text{overlay}} = -K \cdot T_h + T_{\text{overlay, ff}}. \tag{17}\]
3.1. Overlay torque input with human torque feedback

In order to reduce the magnitude of the human torque for keeping the desired path and the frequency of the steering behaviour, a feedback overlay torque of the human torque has been considered. The human torque can be represented as Equation (7) by using the state variables in Equation (16).

\[ y = T_h = Cx \]

\[ = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} y_r \ y_e \ v_c \ \dot{v} \ \psi \ \dot{\psi} \ \delta \ \dot{\delta} \ T_h \end{bmatrix}^T. \]

The steering torque can be measured from the torque sensor of the MDPS module, and the feedback overlay torque of the human torque can be represented as follows:

\[ u_{fb} = -K_{fb}T_h \]

\[ = -K_{fb}Cx \quad \text{where} \quad K_{fb} = \text{human torque feedback gain}. \]

Neglecting the nonlinear terms and road curvature shown in Equation (16) in the situation of straight driving with small steering, the overall vehicle–driver–steering system can be represented with feedback input as follows:

\[ \dot{x} = Ax + Bu_{fb} + FW \]

\[ = (A - BK_{fb}C)x + FW. \]  

(20)

From Equations (18) and (20), the transfer function of the human torque with the lateral disturbance as input is written as follows:

\[ G(s) = \frac{T_h(s)}{F_w(s)} = \frac{\sum_{i=1}^{7} n_i s^{i-1}}{\sum_{i=1}^{8} d_i s^{i-1}}. \]

(21)

In order to investigate the effect of the human torque feedback strategy on the overall vehicle–driver–steering system, the variation of the dominant pole location of the transfer function in Equation (21) according to the variation of the feedback gain, \( K_{fb} \), has been analysed as shown in Figure 15.

3.2. Feedforward overlay torque input

A driver tends to counter-steer towards the direction of the lateral disturbance on bridges or roads near the seashore where the wind is continuously blown in the same direction. In
straight driving, the angle difference between the direction of the vehicle and the direction of
the tire alignment becomes large compared with normal driving situations, due to the lateral
disturbance. Because of the angle difference, the reactive torque from the tire self-aligning
torque is generated and the driver imposes a steering torque to cancel out the reactive torque.
In order to assist the human torque by the reactive torque, the feedforward overlay torque
strategy has been proposed.

The magnitude and direction of the lateral disturbance are needed to calculate the tire self-
aligning torque generated by the lateral disturbance. Various methods to measure the lateral
disturbance by the lateral wind have been proposed so far,[9] but construction of measurement
systems on vehicles and real-time measuring in driving are very complicated. Estimation
methods of the lateral disturbance have been proposed as an alternative, such as direct force
and moment estimation methods [11,12] or indirect lateral disturbance estimation methods
including the bank angle.[16,17] By the use of the estimated force and moment, the tire
self-aligning torque has been calculated.

Equation (16) can be rewritten in reduced state space equation form with the steering angle
input as shown in Equation (22).

\[
x_r = \begin{bmatrix} y_r & \dot{y}_r & e_\psi & \dot{e}_\psi \end{bmatrix}^T,
W = [F_w \ M_w]^T,
\]
\[
\frac{d}{dr} \begin{bmatrix} y_r \ 
\dot{y}_r \\
 e_\psi \\
 \dot{e}_\psi 
\end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\
0 & A_{22} & A_{23} & A_{24} \\
0 & 0 & 0 & 1 \\
0 & A_{42} & A_{43} & A_{44} 
\end{bmatrix} \begin{bmatrix} y_r \\
\dot{y}_r \\
e_\psi \\
\dot{e}_\psi 
\end{bmatrix} + \begin{bmatrix} 0 \\
A_{25} \\
0 \\
A_{45} 
\end{bmatrix} \delta_f + \begin{bmatrix} 0 & 0 \\
F_{21} & 0 \\
0 & 0 \\
0 & F_{42} 
\end{bmatrix} \begin{bmatrix} F_w \\
M_w 
\end{bmatrix}.
\] (22)

The steering angle input in Equation (22) is the summation of the feedback angle for tracking
the desired path in Equation (6) and the feedforward steering angle for reducing the effect
of the lateral disturbance as follows:

\[
\delta_f = -k_1 y_r - k_2 e_L = -k_1 \cdot x_1 - k_2 (x_1 + L_p x_3) + \delta_{ff}
= -(k_1 + k_2) x_1 - k_2 L_p x_3 + \delta_{ff}
\]
\[ K_d x + \delta_{ff}, \]

where \( K_d = [k_1 + k_2 \ 0 \ k_2 \cdot L_p \ 0] = [k'_1 \ k'_2]. \) (23)

Substituting Equation (23) into Equation (22), the system can be rewritten as following form:

\[ \dot{x}_r = (A_r - B_r K_d) x_r + B_r \delta_{ff} + F_r W. \] (24)

Assuming the initial states are zero, Equation (24) can be reformulated by Laplace transform shown as follows:

\[
\begin{align*}
X_r(s) &= (sI - (A_r - B_r K_d))^{-1} \left\{ B_r \left( \frac{\delta_{ff}}{s} \right) + F_r \left[ \frac{\hat{F}_w}{s} \right] \right\} \\
&= -(A - B \cdot K)^{-1} \left\{ B \left( \frac{\delta_{ff}}{s} \right) + F \left[ \frac{\hat{F}_w}{s} \right] \right\} = 0.
\end{align*}
\] (25)

For a steady-state value of the lateral position error, which is the first state of the reduced states, the steady-state value of the reduced states can be represented as Equation (26) by the final value theorem:

\[
\begin{align*}
x_{ss} &= \lim_{t \to \infty} x(t) = \lim_{s \to 0} sX(s) \\
&= \lim_{s \to 0} s [sI - (A - B \cdot K)]^{-1} \left\{ B \left( \frac{\delta_{ff}}{s} \right) + F \left[ \frac{\hat{F}_w}{s} \right] \right\} \\
&= -(A - B \cdot K)^{-1} \left\{ B \left( \frac{\delta_{ff}}{s} \right) + F \left[ \frac{\hat{F}_w}{s} \right] \right\} = 0.
\end{align*}
\] (26)

The feedforward steering angle which makes the steady lateral position error go to zero can be calculated as shown in Equation (27). Each parameter of the equation including the cornering stiffness, and the distance from the centre of gravity of the vehicle to the front/rear axle is the nominal value of the normal driving condition.

\[
y_{r,ss} = \left( \hat{F}_w + 2 \tilde{C}_f \delta_{ff} \right) \left( \tilde{C}_r \tilde{l}_t - \tilde{C}_f \tilde{l}_t + \tilde{C}_r k'_2 \tilde{l}_t \right) - \left( \hat{M}_w + 2 \tilde{C}_f \delta_{ff} \tilde{l}_t \right) \left( \tilde{C}_r + \tilde{C}_f - k'_2 \tilde{C}_f \tilde{l}_t \right) = 0,
\]

\[
\delta_{ff} = \frac{-\hat{F}_w c_1 - \hat{M}_w c_2}{2 \tilde{C}_r c_1 + 2 \tilde{C}_f \tilde{l}_t c_2}. \] (27)

The feedforward steering angle calculated in Equation (27) shows high-frequency behaviour due to the estimated lateral disturbance. In order to prevent high-frequency manoeuvring from the estimated lateral disturbance, the random-walk Kalman filtering method has been proposed as shown in Equation (28). The variance of the process and measurement noise of the equation, \( Q_{\delta_{ff}} \) and \( R_{\delta_{ff}} \), are determined from actual tests.[18]

\[ \delta_{ff}(k + 1|k) = \delta_{ff}(k|k) + w(k) \quad w(k) \sim N(0, Q_{\delta_{ff}}), \] (28)

\[ z(k) = \delta_{ff}(k|k) + v(k) \quad v(k) \sim N(0, R_{\delta_{ff}}). \]

The response of the filtered signal is determined by the process noise and measurement noise of Equation (28). The small gain of the filter induces the actual signal to have low-frequency steady-state value. The calculated feedforward steering angle and the filtered feedforward steering angle under a 7–10 m/s crosswind are shown in Figure 16.
The tyre self-aligning torque generated by the lateral disturbance can be represented as Equation (29). The tyre self-aligning torque of the feedforward steering angle is determined by the slip angle of the feedforward steering angle. The slip angle in the equation can be calculated using the feedforward steering angle in Equation (27), the estimated lateral velocity which has been proposed by various methods, and the measured yaw rate that comes from the electric stability control module, which is a commonly equipped safety system. The feedforward overlay torque is represented by the nominal parameters of the cornering stiffness and the trail of the tyre as Equation (29).

\[
T_{\text{overlay}, \text{ff}} = \sum_{i=1}^{2} \hat{T}_{\text{align}, i}(\delta_{\text{ff}}) = 2\hat{C}_{i,y}\hat{\xi}\hat{\alpha}(\delta_{\text{ff}}). \quad (29)
\]

4. Lateral driver model under a crosswind disturbance

Existing models of driving behaviour are based mainly on visual cues within a preview horizon considering with human transport time delay properties.[14] A vehicle lateral position error, yaw angle error and the first derivatives of each error which are assumed to be obtained from a driver’s visual recognition are generally used to form a vehicle lateral error dynamics. The optimal solution of the driver’s command such as steering angle for minimising a cost function of lateral position error and yaw angle error is derived through the well-known Euler–Lagrange equation or standard LQ Riccati equation.[19,20] The driver’s lateral behaviour in the normal situation is quite well represented in the existing model, however, it is shown to be difficult to represent exactly the driving characteristics of a driver under the lateral disturbance with the four states models through the investigations via tests conducted on a driving simulator. To give additional flexibility to represent the driver’s lateral behaviour under the lateral disturbance, more consideration of higher derivatives of lateral position error and yaw angle error are required.

4.1. Driving simulator tests

The behaviour of a driver in crosswind is hard to investigate in real-world, for the difficulties of measuring lateral disturbance caused by the crosswind. Instead of a field test, an investigation via a driving simulator is the most widely used approach to develop a human driver model. In this study, a CARSIM vehicle model for a sedan type was implemented in the driving simulator. To investigate the behaviour of the driver in the crosswind, driving simulator tests
To verify crosswind sensitivity of test subjects, impulse crosswind of Netherlands Organization for Applied Scientific Research (TNO) test condition has been implemented in the driving simulator. The tests have been conducted at 80 km/h of vehicle speed. The impulse input type of crosswind disturbance, the event section is set as shown in Figure 17. The crosswind speed is assumed as 30 m/s, and the heading angle of the crosswind is assumed as 135° with respect to the heading direction of a vehicle. A distance between the starting place and the point at which the vehicle enters the event section has to be far enough in order to reach the target speed in the event section.

### 4.2. Design strategy of lateral driver model in a crosswind

A proposed driver model is developed in two steps. Desired steering wheel angle from path-error cognition is determined in the first step, and desired steering wheel torque for tracking the desired steering wheel angle is designed in the second step, which is the same process as a driver’s behaviour. State feedback control gains used in the controllers were calculated reversely from virtual manoeuvre profiles collected via driving simulator tests. An overall structure of the proposed driver model is described as Figure 18.
Table 2. Parameters for lateral driver model.

<table>
<thead>
<tr>
<th>$k_{\text{limb}}$</th>
<th>$j_{\text{limb}}$</th>
<th>$b_{\text{limb}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>2</td>
<td>50</td>
</tr>
</tbody>
</table>

A driver decides a desired steering wheel angle to correct deviation error from the desired course, and takes the action of a steering wheel through driver’s neuromuscular system. The physical limitations related to the time response of the driver are considered as the summation of the time delays on perception and neuromuscular system.

Desired steering wheel angle is determined as a form of state feedback with the states given in Equation (30) considering a perception delay. Lateral position error and the yaw angle error are defined as described in Equations (11) and (12), and Figure 12. In addition to the lateral position error, yaw angle error and the first derivatives of each error, the second derivatives and integral terms of each error are considered to give additional flexibility to represent the driver’s lateral behaviour under lateral disturbance. The desired steering wheel angle is rewritten as Equation (31) considering with the neuromuscular transport delay which is simplified with a second-order system.[14] Each parameter in Equation (31) represents the stiffness coefficient ($k_{\text{limb}}$), moment of inertia ($j_{\text{limb}}$), damping coefficient of a human limb system ($b_{\text{limb}}$), respectively, as shown in Table 2.

$$
\delta_{\text{SW,des}} = -K_{\text{sw}}x(t - \tau_N),
$$

$$
x = \left[ \int y_r \int e_\psi \ y_r \ e_\psi \ \dot{y}_r \ \dot{e}_\psi \ \ddot{y}_r \ \ddot{e}_\psi \right]^T, \quad (30)
$$

$$
\delta_{\text{SW,DM}}(s) = \frac{k_{\text{limb}}}{j_{\text{limb}}s^2 + b_{\text{limb}}s + k_{\text{limb}}} \cdot \delta_{\text{SW,des}}(s). \quad (31)
$$

Control gain vector in Equation (30) is determined to minimise a cost function of the error between a computed steering wheel angle of a proposed driver model and a steering wheel from driving simulator tests as shown in Equation (32). The optimal gain vector for minimising the cost function can be obtained using a well-known quadratic programming method.

$$
J = \int_{t_i}^{t_f} (\delta_{\text{SW,DS}} - \delta_{\text{SW,DM}})^2 \, dt. \quad (32)
$$

Steering angle of a proposed driver model is compared with driving simulator test results as shown in Figure 19. The steering angle from the driving simulator test under lateral disturbance is confirmed to be well represented with the steering wheel angle of the proposed driver model with consideration of a neuromuscular transport delay.

Desired steering torque is determined from the error between the steering wheel angle of the vehicle and the steering angle of the driver model. The desired torque is calculated with the first derivative and integral form of the error and each gain of the error as follows:

$$
T_{h,DM} = K_{t,P}e_{\text{SW}} + K_{t,I} \int e_{\text{SW}} \, dt + K_{t,D} \left( \frac{de_{\text{SW}}}{dt} \right), \quad (33)
$$

$$
e_{\text{SW}} = \delta_{\text{SW}} - \delta_{\text{SW,DM}}.
$$

A proposed driver model was validated with driving simulator test data of 14 subjects. Each simulator test was conducted in TNO crosswind simulator test condition described in Figure 17 with 100 km/h of vehicle speed. Simulation results show the behaviour of the driver model under the lateral force due to crosswind which is described in Figure 19(a). Figure 19(b)
Figure 19. Simulation results of a proposed driver model and driving simulator tests. (a) Lateral force due to crosswind. (b) Human torque. (c) Steering wheel angle. (d) Lateral distance error. (e) Yaw angle error.

and 19(c) shows the steering wheel torque and the steering wheel angle of the driver model and simulator tests. The steering characteristics of the actual driver in a crosswind were confirmed to be quite well represented with the driver model. As a result, the lateral distance error and yaw angle error of the driver model was shown to be quite similar to the mean value of the simulator tests as described in Figure 19(d) and 19(e).
Table 3. Parameters for vehicle model.

<table>
<thead>
<tr>
<th>m (kg)</th>
<th>I_z (kg m²)</th>
<th>l_f (m)</th>
<th>l_r (m)</th>
<th>C_f (N/rad)</th>
<th>C_r (N/rad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1824</td>
<td>4100</td>
<td>1.351</td>
<td>1.583</td>
<td>80,168</td>
<td>75,860</td>
</tr>
</tbody>
</table>

5. Simulation results

A simulation was conducted with a full vehicle model with the driver model developed in a previous section. It is known from a previous research that a 15 m/s crosswind can induce a vehicle to deviate from its lane, especially large-sided vehicle such as buses and trains travelling over 80 km/h.[4] Crosswind force and moment induced by 15 m/s wind velocity are applied to the sedan-type vehicle simulation model, which drives with a speed of 80 km/h. The parameters for the vehicle model are shown in Table 3.

Crosswind was imposed on the vehicle model 8 s after the simulation began as shown in Figure 20(a) and 20(b). Figure 21 shows the simulation results of the proposed lateral disturbance compensation algorithm under sudden crosswind. Figure 21(a) shows the human torque, and the course deviations which represent lateral distance error and yaw angle error are shown in Figure 21(b) and 21(c), respectively. A human torque feedback gain was chosen through the result of pole location analysis considered in a previous section to reduce both the magnitude and frequency of the human driver’s steering manoeuvre. When the compensation algorithm was not applied, the lateral course deviation exceeded 0.4 m at about 10 s as shown in Figure 21(b). The steering behaviour shows an overshoot tendency under sudden exposure to crosswind. In the case of the lateral disturbance compensation algorithm, it was ensured via the simulation results that lateral course deviation and human steering torque were reduced below 0.2 m and 0.5 N m as shown in Figure 21(a) and 21(b). From the simulation results, the performance of the proposed lateral disturbance compensation algorithm can be expected to show good performance in actual tests.

Figure 20. Lateral force and moment due to lateral disturbance. (a) Lateral force. (b) Yaw moment.
6. Evaluation of lateral compensation algorithm on a VTT

The proposed lateral compensation algorithm was implemented in real time and evaluated through the real-time human-in-the-loop simulation with a full-scale driving simulator on a VTT.

The configuration of the full-scale human-in-the-loop driving simulator is shown in Figure 22. The simulator consists of the following four parts: real-time simulation hardware, visual graphic engine, human–vehicle interface and motion platform. A host computer is utilised to execute a vehicle simulation programme and display current vehicle status. The
visual graphic engine projects the visual representation of the driving status of the vehicle and driving environment from real-time simulation via a beam projector onto a 100 in. screen. A human driver interacts with the virtual simulation provided by a virtual driving environment.
and the kinesthetic signal of the simulator body. The driver's responses are acquired through MDPS, brake pressure and throttle position sensors as shown in Figure 22. The motion platform installed under the vehicle body provides kinesthetic cues that come from real-time vehicle behaviour. An actual full-size brake system including vacuum booster, master cylinder, calipers, etc. is implemented in the simulator in order to provide the driver with feeling similar to an actual brake pedal. Steering angle, steering torque and motor torque are obtained from MDPS with controller area network (CAN) type signals. The reactive torque of the steering system is provided by MDPS motor torque that represents the actual tyre characteristics. The driving simulator was evaluated through the comparison of the test results using the driving simulator and actual vehicle test results in previous research.[21]

The test environment is the same as in the driver–vehicle numerical simulation, with 80 km/h vehicle speed in straight driving, and 15 m/s crosswind imposed on the vehicle 8 s after the test began. Figure 23(a) shows the human steering torque imposed on the steering wheel. The maximum human steering torque was shown in 9 s, or 1 s after the crosswind was applied. In the uncontrolled case, the maximum torque exceeds 2 N·m and fluctuates around 1 N·m. Figure 23(b) and 23(c) represent lateral course deviation and yaw angle error of the vehicle. The course deviation of the uncontrolled case exceeded 1 m at 10 s and crossed over to another lane, and the amplitude of the yaw angle error exceeded 1° after the crosswind was applied. The steering angle showed characteristics similar to the numerical simulation, remaining around 8° and keeping the vehicle in the same lane. In the compensation control case, the magnitude and frequency of human torque were significantly reduced compared with the uncontrolled case as shown in Figure 23(a). Both lateral course deviation and yaw angle error were reduced below 0.5 m and 1°, respectively, as shown in Figure 23(b) and 23(c). The steering wheel angle of the controlled case showed less oscillation around the steady-state steering angle compared with the uncontrolled case as shown in Figure 23(d). It has been shown that the compensation algorithm ensured more stable and safe driving even under the sudden exposure to crosswind.

To confirm the stochastic distribution of lateral distance error according to the vehicle speed and crosswind speed, a repeat test in crosswind speeds and vehicle speeds on a VTT were conducted. The test was conducted 10 times in each case. Figure 24 shows the stochastic

![Figure 24](http://example.com/figure24.png)

**Figure 24.** Stochastic distribution of lateral distance error. (a) 80 km/h vehicle speed, 10 m/s crosswind speed. (b) 80 km/h vehicle speed, 20 m/s crosswind speed. (c) 100 km/h vehicle speed, 10 m/s crosswind speed. (d) 100 km/h vehicle speed, 20 m/s crosswind speed.
distribution of lateral displacement with 80 and 100 km/h vehicle speed and 10 and 20 m/s crosswind speed. Each test result shows a significant error reduction even under the strong lateral disturbance.

Human steering torque under various lateral disturbances was investigated as shown in Figure 25. Each controlled case and uncontrolled case was shown in the figure with a normal straight driving situation without lateral disturbance. The frequency and magnitude of the human torque with the compensation control under sudden lateral disturbance were smaller than in the uncontrolled case, and were shown to be similar to the normal driving situation. This reveals that the proposed compensation algorithm could reduce the manoeuvring effort of the driver under lateral disturbance. From Figures 24 and 25, it was confirmed that the proposed compensation algorithm could improve vehicle stability and manoeuvrability, and at the same time prevent lane departure under lateral disturbance. It is expected that this study can contribute to decision-making on integrated torque overlay strategies as an application of MDPS.

7. Conclusion

A lateral disturbance compensation algorithm for application to MDPS assistant systems has been developed. In order to analyse an overall vehicle–driver–steering behaviour, vehicle lateral error dynamics, steering dynamics and a human driver model were considered. The steering dynamics with MDPS and the driver model considering the torque delay from perception and muscular system were investigated. The vehicle lateral error dynamics were derived from tyre force dynamics with position deviation from a desired path, and finally the unified state space equation including the above dynamics was investigated to represent the overall driving system. The proposed lateral disturbance compensation torque control law consists of the human torque feedback and the feedforward overlay torque. To investigate the appropriate control gain of the feedback motor overlay torque, the analysis of the vehicle–driver–steering
system was conducted. Feedforward overlay torque control algorithm was proposed to compensate for tyre self-aligning torque generated by the lateral disturbance. Estimated lateral force and yaw moment from the lateral disturbance were used to calculate the feedforward overlay torque.

To investigate the performance of a compensation control law in a closed loop simulation, a driver model which represents the behaviour of driver in a crosswind has been developed. The driver model has been designed based on driving simulator test data of 14 test subjects. Higher derivatives of lateral position error and yaw angle error are included in the vehicle lateral dynamics to give additional flexibility to represent driver behaviour under the lateral disturbance. Simulation results show the driver model can be adjusted to represent a range of drivers with different characteristics under the lateral disturbance, thus allowing the simulation assessment of the lateral disturbance compensation algorithm.

The performance of the compensation control law has been investigated via human-in-the-loop tests with a full-scale driving simulator on a VTT in real-time. From the simulation studies via a developed driver model and human-in-the-loop test results, it has been shown that vehicle lateral position error, heading angle error, and magnitude and frequency of the human torque, can be significantly improved compared to the uncontrolled situation. Since the compensation control algorithm proposed in the study has been investigated based on the estimated force and moment due to lateral disturbance, the compensation algorithm is expected to be applied on various driving situation not only crosswind but any lateral disturbance situation such as road geometrical disturbance.

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**References**


