Development of Parallel Two-wheel Vehicle with Lower Gravity Center of Vehicle Body

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Abstract: This paper presents an advanced parallel two-wheel vehicle which has lower gravity center of vehicle body. The gravity center is assigned at the lower position than the wheel axis. Therefore, the vehicle has a structure of the pendulum, and enables the vehicle body with the passenger to always keep the stable posture, even if the vehicle is in the power-off or control-off condition. And, 2-DOF joystick which has operation with back-and-forth direction and rotation is applied to the proposed vehicle. The elderly or handicapped passenger can operate easily the vehicle by this joystick. Moreover, in order to suppress the sway of the vehicle body as a pitching oscillation while driving the vehicle, the sway suppression control system with an active mass damper system is proposed in this paper. The control system is designed by a backstepping method. The effectiveness of the proposed sway suppression control system with the active mass damper system is verified by the experiments using the proposed parallel two-wheel vehicle with lower gravity center.

1 INTRODUCTION

In recent years, barrier-free society is advancing, and welfare environment has been gradually improved. There have been many studies about an electric wheelchair as welfare device for elderly and people with lower physical ability. And the demand for the electric wheelchair will be increased in the future. In addition, many researches and developments have also focused on personal vehicle with low energy requirement, which is required to support for short-distance transport, (Hun-ok Lim and Tamai, 2008), (Y. Ueno and Kitagawa, 2009) and (Y. Noda and Terashima, 2010).

The typical electric wheelchair is driven with four-wheel which is composed of the front casters and the rear wheels. However since these wheelchairs have the large turning radius, it is difficult to pass through in a narrow passage. A parallel two-wheel vehicle with structure of inverted pendulum, which has small turning radius, has been developed in recent years, (M. Sasaki, 2005). In such vehicle, an inverted pendulum control system is used for standing the vehicle stably by only using the two driving parallel wheels, (C. Nakagawa and Hirayama, 2011) and (Karkoub and Parent, 2004). Therefore, the gravity center of the vehicle body is higher than the wheel axis, and the vehicle is moved by tilting the vehicle body forward or backward by moving the gravity center. However, when the vehicle is in the power-off or control-off conditions which are caused by a breakdown in the vehicle, the vehicle cannot keep the standing posture. Moreover, we consider the vehicle with the passenger sitting which can be used by elderly and handicapped people. In this case, since the gravity center is lower than the standing posture, the larger action of the passenger's upper body is required for operating the vehicle. The elderly or handicapped passenger is difficult to the large action in the vehicle.

Therefore, we propose a parallel two-wheel vehicle with safety and easy operation which can be used by elderly and handicapped people. The proposed vehicle has the lower gravity center of the vehicle body as shown in Figure 1. The position of the gravity cen-

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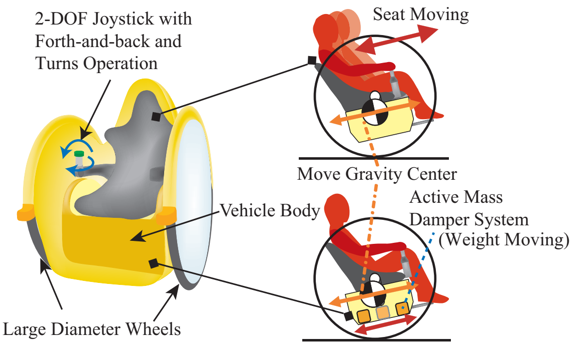

Figure 1: Illustration of parallel two-wheel vehicle with lower gravity center of vehicle body.

ter of this vehicle with the passenger sitting is lower than the wheel axis, since the vehicle has the large diameter wheels and the battery, the actuators and the control devices are placed under the wheel axis. Therefore, it allows the vehicle to always stand stably without electric power supply. And, 2-DOF joystick with both back-and-forth and rotating operation is applied to the proposed vehicle for operating easily.

However, when the passenger rides on the proposed vehicle, the vehicle body is leaned by moving the gravity center due to sit the passenger on the vehicle. In the proposed vehicle, the posture of the vehicle body with the passenger is compensated by adjusting automatically the position of the seat with the passenger. Moreover, in order to suppress the sway of the vehicle body as a pitching oscillation while driving the vehicle, the sway suppression control system of the vehicle body is proposed in this paper. In the control system, an active mass damper system is installed inside the vehicle body. In the design of the control system, the vehicle body dynamics is modeled using Lagrange equation of motion. And, the control system of the sway suppression control with the active mass damper system is designed by a backstepping method, (Fu and Zao, 2007).

The effectiveness of the proposed sway suppression control system is verified by the experiments using the proposed parallel two-wheel vehicle with lower gravity center.

2 OVERVIEW OF PARALLEL TWO-WHEEL VEHICLE WITH LOWER GRAVITY CENTER

The parallel two-wheel vehicle with lower gravity center consists of two large diameter driving wheels and the vehicle body where the passenger sits on as shown in Figure 1. The power-supply, the control devices and actuators are set on the lower position than the wheel axis. As a result, the gravity center of this vehicle with the passenger sitting on the seat is lower than the wheel axis. Therefore, it allows the vehicle to always stand stably without electric power supply. A conventional parallel two-wheel vehicle has high gravity center of the vehicle body with the passenger, and an inverted pendulum control system is applied for stably standing the vehicle body and operating the vehicle by moving the gravity center of the passenger. However, in the proposed vehicle, it is difficult to operate the vehicle by moving the gravity center of the passenger, because the passenger sits on the seat and the gravity center of the vehicle body is in low position. Therefore, the vehicle is operated by the 2DOF joystick. This joystick has both forth-and-back and rotating operation. It is easy to operate the vehicle, since the movement of the joystick is the same as the movement of the vehicle.

When the passenger sits on the seat of the vehicle, the vehicle body is leaned statically by moving the gravity center of the vehicle body with the passenger. For compensating the vehicle body leaning, the seat positioning control system is installed to the seat supporting structure in the vehicle body. And, while the vehicle moving, the vehicle body is swayed by the acceleration of the vehicle. In order to suppress the vehicle body swaying, the sway suppression control system with the active mass damper is proposed in this paper. The sway suppression control system is located on the bottom of the vehicle body.

The parallel two-wheel vehicle with lower gravity center developed by the present authors is shown in Figure 2, and the specification of the developed two-wheel vehicle is shown in Table 1.

Figure 2: Photos of developed two-wheel vehicle.

Table 1: Specification of Vehicle.

Size (m)	W0.90×D1.20×H1.70
Mass (kg)	137
Wheel Diameter (mm)	1041
Driving Power (W)	300
Driving Voltage (V)	24

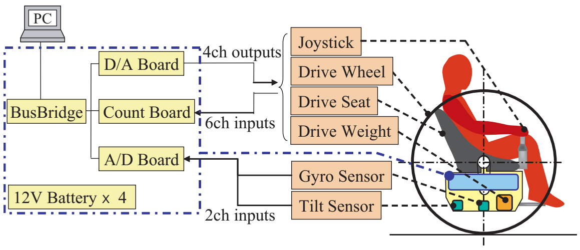

Figure 3: Control system of vehicle.

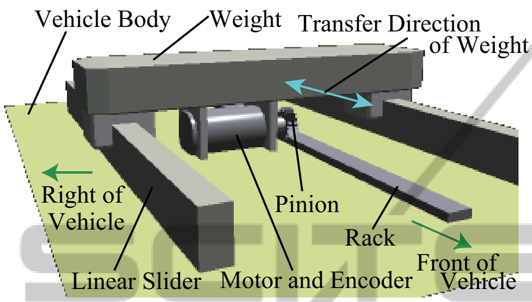

Figure 4: Active mass damper system.

2.1 Control Apparatus of Vehicle

The construction of the control system in the parallel two-wheel vehicle is shown in Figure 3. The right and left wheels are rotated by DC servomotors, reducers and pulley mechanisms. The rotations of the wheels are detected by the rotary encoders fitted on the servomotors. The forth-and-back tilting and rotating of the joystick are detected by the rotary encoders fitted on each axis. The weight in the active mass damper system is transferred by DC servomotor, reducer, rack-and-pinion and linear sliders as shown in Figure 4. The position of the weight is detected by the rotary encoder fitted on the servomotor. The seat transfer system is also the same mechanism as the active mass damper system. The tilting angle and the angular velocity of the vehicle body are detected by the tilt sensor and the gyro sensor respectively located on the bottom of the vehicle body.

The signals detected by the encoders, the tilt and the gyro sensors are collected into the PC through the A/D and the counter boards. Then, the input signals are added to the servomotors through the D/A board.

3 SEAT POSITIONING CONTROL SYSTEM

When the passenger sits on the seat of the vehicle, the vehicle body is leaned by moving the gravity center of the vehicle body with the passenger. For compensating the vehicle body leaning, the seat positioning control system is proposed in this paper. The vehicle body's posture leaned statically can be compensated by the proposed control system. The proposed seat positioning control system is shown in Figure 5. In the control system, the tilting angle of the vehicle body is detected by the tilt sensor, and the angular velocity is detected by the gyro sensor. The control input to the servomotor system is generated by PD control which consists of P control to the error between the reference angle and the tilting angle of the vehicle body, and D control to the angular velocity of the vehicle body. The servomotor system consists of the velocity feedback control system. The seat is transferred by driving the servomotor. By moving the seat with the passenger, the posture of the vehicle body is compensated. In PD control, the proportional and derivative gains are given as 0.08 and 0.05 respectively by adjusting in the experiments.

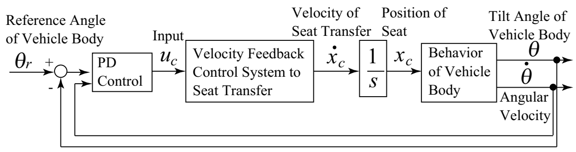

Figure 5: Block diagram of seat positioning control.

This control system works only static condition of the vehicle, because of the compensation to the vehicle body's posture leaned statically. Therefore, while driving the vehicle, the seat is secured on the vehicle body.

4 SWAY SUPPRESSION CONTROL SYSTEM OF VEHICLE BODY

While the vehicle moving, the vehicle body is swayed as a pitching oscillation by the acceleration of the vehicle driving. In order to suppress the vehicle body swaying, the sway suppression control system with an active mass damper is proposed in this paper. For designing the control system, the dynamics of the pitch angle to the vehicle body swaying is modeled by using Lagrange equation of motion. Then, the control system is designed by an backstepping method, (Fu and Zao, 2007).

4.1 Modeling Behavior of Vehicle Body

The illustration about the vehicle body behavior is shown in Figure 6. In Figure 6, m_1, m_2 and m_3[kg] are the masses of the vehicle body, the weight in the active mass damper system and the passenger, respec-

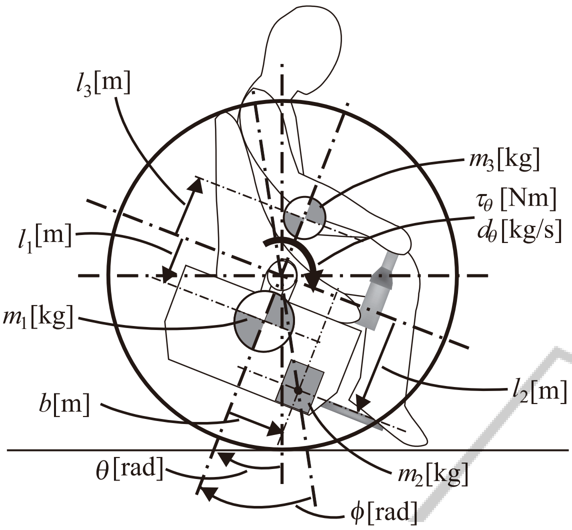

Figure 6: Model parameters concerned with vehicle body behavior.

tively. And, l_1, l_2 and l_3[m] show the height of the gravity centers of the vehicle body, the weight in the active mass damper system and the passenger, respectively. θ[rad] is the pitch angle between the line connecting the gravity center of the vehicle body with the wheel axis and the vertical axis. ϕ[rad] is the angle between the line connecting the gravity center of the weight with the wheel axis and that connecting the gravity center of the vehicle body with the wheel axis. b[m] is the horizontal position of the weight on standing the vehicle body. τ_θ[Nm] and d_θ[kg/s] are the external torque around the wheel axis which is occurred by driving the wheels and the viscosity damping coefficient to rotate the wheels, respectively. Lagrange equation of motion to the behavior of the vehicle body is shown as

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\theta}}\right) - \frac{\partial T}{\partial \theta} + \frac{\partial D}{\partial \dot{\theta}} + \frac{\partial U}{\partial \theta} = \tau_\theta, \qquad (1)$$

where T is the kinetic energy, U is the potential energy, and D is the dissipative energy. These energies are represented as

$$T = \frac{1}{2}(m_1 l_1^2 + m_2(l_2^2 + b^2) + m_3 l_3^2)\dot{\theta}^2 + \frac{1}{2}m_2\dot{b}^2, \quad (2)$$

$$D = \frac{1}{2}d_\theta\dot{\theta}^2, \qquad (3)$$

$$U = m_1 g l_1(1-\cos\theta) + m_2 g\sqrt{l_2^2+b^2} \cdot (1-\cos(\theta-\phi)) + m_3 g l_3 \cos\theta, \quad (4)$$

where g[m/s^2] is the gravity acceleration. By substituting Eqs.(2)-(4) into Eq.(1), the differential equation can be obtained as

$$A_{11}\ddot{\theta} + A_{12}\dot{\theta} + A_{13}\sin\theta - m_2 g b \cos\theta = \tau_\theta, \qquad (5)$$

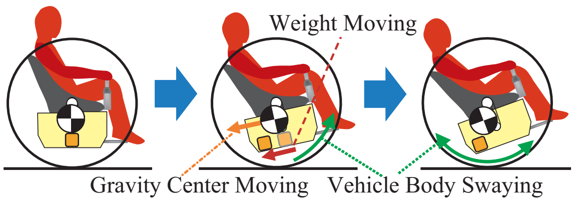

Figure 7: Experiment procedure for model parameters identification.

Table 2: Model parameters.

Notation	Values	Notation	Values
l_1(m)	0.382	m_1(kg)	110.30
l_2(m)	0.380	m_2(kg)	26.70
l_3(m)	0.628	m_3(kg)	60.25
d_θ(kg/s)	32.325		

where

$$A_{11} = m_1 l_1^2 + m_2(l_2^2 + b^2) + m_3 l_3^2,$$
$$A_{12} = 2m_2 b\dot{b} + d_\theta,$$
$$A_{13} = (m_1 l_1 + m_2 l_2 - m_3 l_3)g.$$

4.2 Identification of Model Parameters

In the model of the behavior of the vehicle body in previous section, τ_θ is the external torque such as the torque generated by the servomotors for driving wheels. The tilt angle θ and the angular velocity $\dot{\theta}$ of the vehicle body and the position b and the velocity $\dot{b}$ of the weight are the state variables in the model. The mass m_1, m_2 and m_3, the height l_1, l_2 and l_3, and the viscosity damping coefficient d_θ are the constant parameters. The mass m_1, m_2 and m_3 are measured by a weigher. The parameters l_1, l_2, l_3 and d_θ are the unknown parameters, and are identified by minimizing the error between the vehicle body's behaviors in the experiment and the simulation. In the experiment for the identification, the weight in the active mass damper system is moved as shown in Figure 7. Then, the vehicle body is swayed by moving the weight. The parameters are searched by minimizing the error using a downhill simplex method, (Nelder and Mead, 1965).

As the result, the identified model parameters are shown in Table 2. And, the comparison between the behaviors of the vehicle body in the experiment and the simulation are shown in Figure 8. (a) and (b) show the position and the velocity of the weight transfer in the active mass damper system. (c) and (d) show the tilt angle and the angular velocity of the vehicle body. In Figure 8(c) and (d), the solid lines are the experimental results, and the broken lines are the simulation results. As seen from Figure 8, the sway of the vehicle body can be represented precisely by the proposed

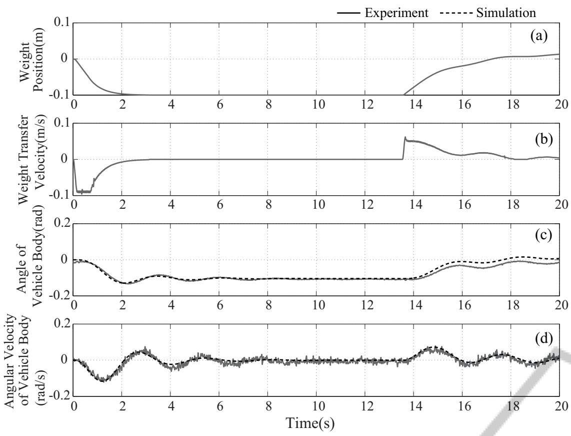

Figure 8: Comparison between behaviors of vehicle body in experiment and simulation.

model. The proposed model is used for designing the sway suppression control system with the active mass damper system.

4.3 Design of Sway Suppression Control based on Backstepping Method

For designing the sway suppression control system, the model of behavior of the vehicle body is simplified by linearization. It is assumed that the sway angle of the vehicle body is small. Eq.(5) is linearized as

$$A_{11}\ddot{\theta}+A_{12}\dot{\theta}+A_{13}\theta-m_2gb=\tau_\theta. \tag{6}$$

Moreover, Eq.(6) is transformed to the spring-mass-damper system as

$$M(b)\ddot{\theta}+C(b,\dot{b})\dot{\theta}+K\theta=d(\tau_\theta)+u(b), \tag{7}$$

where

$$\begin{aligned}
&M(b)=m_1l_1^2+m_2(l_2^2+b^2)+m_3l_3^2,\\
&C(b,\dot{b})=2m_2b\dot{b}+d_\theta,\\
&K=(m_1l_1+m_2l_2-m_3l_3)g,\\
&d(\tau_\theta)=\tau_\theta,\ u(b)=m_2gb.
\end{aligned}$$

In Eq.(7), τ_θ is the external torque to sway the vehicle body, and is treated as the disturbance d in the control system design. b is the weight position, and the torque m_2gb generated by the weight position is treated as the control input u to stabilize the posture of the vehicle body.

Here, the model in Eq.(7) is a LPV (Linear Parameter Varying) system, and has two control variables θ and $\dot{\theta}$. Many control approaches to the LPV system have been proposed in the previous studies. In this paper, the backstepping method as one of the control approaches to the nonlinear system is applied to design the sway suppression control. It enables to design easily to the LPV or nonlinear system with small number of the control variables. The backstepping method is the control design which a Lyapunov function is sequentially-constructed with respect to each state variable based on the formal Lyapunov function as

$$U_{(x,y)}=V_{(x)}+\frac{1}{2}\left[y-\mu_{(x)}\right]^2, \tag{8}$$

where x is the state variable, y is the controlled variable. In this paper, the state variable x consists of $(\theta,\dot{\theta})$, and the controlled variable y is the tilt angle θ of the vehicle body. At first step, based on Eq.(8), the candidate Lyapunov function to the state variable θ in the model is given as

$$V_1=\frac{1}{2}z_1^2=\frac{1}{2}(y-y_r)^2, \tag{9}$$

where y_r is the reference tilt angle of the vehicle body. z_1 is the error between the actual and reference tilt angles. In this paper, the reference y_r is 0, because the purpose of the control system is to stabilize the vehicle body on the standing posture. Thus, the Lyapunov function in Eq.(9) is represented as

$$V_1=\frac{1}{2}z_1^2=\frac{1}{2}y^2. \tag{10}$$

The time derivative of Eq.(10) is shown as

$$\dot{V}_1=z_1\dot{z}_1=y\dot{y}. \tag{11}$$

Here, since $\dot{z}_1=\dot{y}=\dot{\theta}$, Eq.(9) is replaced as

$$\dot{V}_1=z_1\dot{z}_1=y\dot{y}=y\dot{\theta}. \tag{12}$$

By arranging $\dot{\theta}$ as

$$\dot{\theta}=-c_1y,(c_1\geq 0), \tag{13}$$

Eq.(12) is represented as

$$\dot{V}_1=-c_1y^2\leq 0. \tag{14}$$

Therefore, it can be obtained as a certain Lyapunov function. Here, the ideal state variable α to the state variable $\dot{\theta}$ is introduced as

$$\alpha=-c_1y. \tag{15}$$

At second step, the error z_2 between the actual state $\dot{\theta}$ and the ideal state α is shown as

$$z_2=\dot{\theta}-\alpha. \tag{16}$$

Based on Eq.(8), the candidate Lyapunov function V_2 involving the state z_2 is given as

$$V_2=V_1+\frac{1}{2}z_2^2. \tag{17}$$

The time derivative of Eq.(17) is shown as

$$\dot{V}_2=\dot{V}_1+z_2\dot{z}_2. \tag{18}$$

Here, by substituting Eqs.(15) and (16) into Eq.(12), the time derivative of the Lyapunov function $\dot{V}_1$ is represented as

$$\dot{V}_1 = y(z_2 + \alpha) = y(z_2 - c_1 y) = -c_1 y^2 + y z_2. \quad (19)$$

And, the model shown in Eq.(7) and the time derivative of the ideal state α in Eq.(15) are substituted into the time derivative of Eq.(16) as

$$\begin{aligned} \dot{z}_2 &= \ddot{\theta} - \dot{\alpha} = \ddot{\theta} + c_1 \dot{y} \\ &= \frac{1}{M(b)} u(b) - (\frac{C(b,\dot{b})}{M(b)} - c_1)\dot{\theta} - \frac{K}{M(b)}\theta, \quad (20) \end{aligned}$$

where the disturbance d is removed from the model of Eq.(7) for substituting. By substituting Eqs.(19) and (20) into Eq.(18), $\dot{V}_2$ is represented as

$$\begin{aligned} \dot{V}_2 &= -c_1 y^2 + y z_2 + z_2 \dot{z}_2 = -c_1 y^2 + z_2(\dot{z}_2 + \theta) \\ &= -c_1 y^2 + z_2(\frac{1}{M(b)} u(b) \\ &\quad -(\frac{C(b,\dot{b})}{M(b)} - c_1)\dot{\theta} - (\frac{K}{M(b)} - 1)\theta). \quad (21) \end{aligned}$$

By arranging $-c_2 z_2$ as

$$\begin{aligned} -c_2 z_2 &= \frac{1}{M(b)} u(b) - (\frac{C(b,\dot{b})}{M(b)} - c_1)\dot{\theta} \\ &\quad -(\frac{K}{M(b)} - 1)\theta, \ (c_2 \geq 0), \quad (22) \end{aligned}$$

$\dot{V}_2$ is represented as

$$\dot{V}_2 = -c_1 y^2 - c_2 z_2^2 \leq 0. \quad (23)$$

Therefore, the state variables θ and $\dot{\theta}$ are in Lyapunov stability by constructing the Lyapunov function V_2. By substituting Eqs.(15) and (16) into Eq.(22), the control input u is obtained as

$$\begin{aligned} u(b) &= (C(b,\dot{b}) - M(b)(c_1 - c_2))\dot{\theta} \\ &\quad +(K - M(b)(1 + c_1 c_2))\theta. \quad (24) \end{aligned}$$

Since $u(b) = m_2 g b$ as shown in Eq.(7), the weight position b with the state feedback law is derived as

$$\begin{aligned} b &= \frac{1}{m_2 g}\{(C(b,\dot{b}) - M(b)(c_1 - c_2))\dot{\theta} \\ &\quad +(K - M(b)(1 + c_1 c_2))\theta\}. \quad (25) \end{aligned}$$

The block diagram of the sway suppression control system with the active mass damper system designed by the backstepping method is shown in Figure 9. The input torque u is calculated using the tilt angle measured by the tilt sensor and the angular velocity measured by the gyro sensor. The weight position b is obtained from the input torque, and is realized by the weight positioning system in the active mass damper system. In the feedback control law, b is given from the weight position measured by the rotary encoder fitted on the servomotor in the active mass damper system. $\dot{b}$ is derived as the approximate derivative of the weight position b measured by the rotary encoder. In this paper, since it is assumed that the response of the weight positioning system in the active mass damper system is much faster than that of the behavior of the vehicle body, the dynamics of the weight positioning system is not considered.

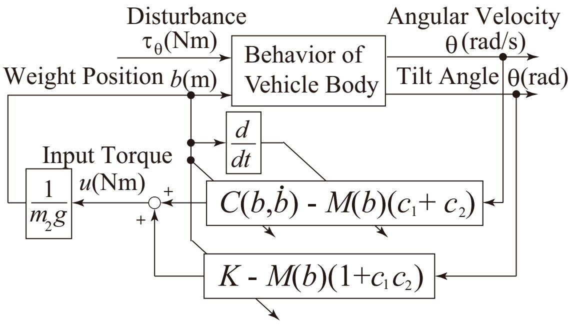

Figure 9: Block diagram of sway suppression control system.

In this sway suppression control system, the tilt angle θ and the angular velocity $\dot{\theta}$ of the vehicle body are converged fast with increasing the design parameters c_1 and c_2 as seen from Eq.(23). However, the control system with increased design parameters is influenced by the noise in the signal measured by the sensors. Moreover, it is hard to track the actual weight position in the active mass damper system to the ideal weight position generated by the control law in the sway suppression control system. In this paper, the design parameters are adjusted by performing the experiments and the simulations, and obtained as $c_1 = 2.37$ and $c_2 = 2.37$.

5 EXPERIMENTAL RESULTS

In order to verify the effectiveness of the proposed sway suppression control system with the active mass damper system, the experiments using the proposed parallel two-wheel vehicle are performed. In the experiments, the behavior of the vehicle body with the sway suppression control system is compared with that without the sway suppression control system. In the vehicle without the sway suppression control system, the weight in the active mass damper stays at the center of vehicle body. The experimental results are shown in Figure 10. In Figure 10, (a) shows the input voltage added to the servomotor for driving the wheel. (b) and (c) show the position and the velocity of the weight transfer in the active mass damper,

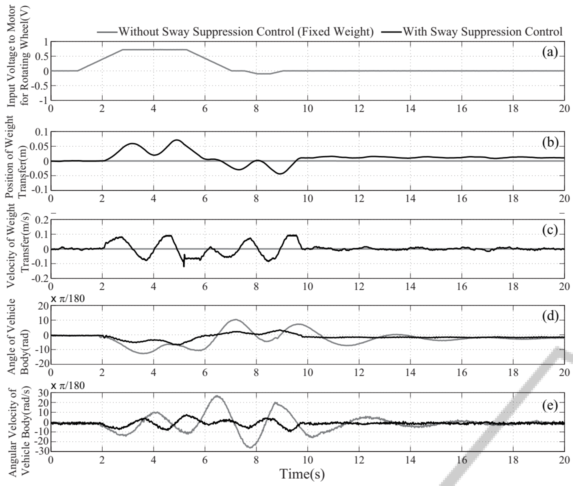

Figure 10: Experimental results.

respectively. (d) and (e) show the tilt angle and the angular velocity of the vehicle body, respectively. In Figure 10(b)-(e), the black and the gray lines are the experimental results by the vehicle with and without the sway suppression control, respectively. As seen from Figure 10(d), the sway angle of the vehicle body in the vehicle without the sway suppression control is over $12.0\times\pi/180$(rad). On the other hand, the sway angle in the vehicle with the sway suppression control can be suppressed within $7.0\times\pi/180$(rad). Moreover, the residual vibration of the sway after driving the wheel also suppressed by the proposed sway suppression control. Therefore, the proposed sway suppression control is useful to the safety aspect of the parallel two-wheel vehicle with lower gravity center.

6 CONCLUSIONS

In this paper, the novel parallel two-wheel vehicle with lower gravity center has been proposed. It has the advantage of safety than the conventional two-wheel vehicle with an inverted pendulum structure, because the proposed vehicle has the stable structure which the gravity center of the vehicle body is lower than the wheel axis. And, in order to suppress sway of the vehicle body while driving the vehicle, the sway suppression control system with the active mass damper system is proposed. The control system is designed by the backstepping method. By the experiments using the proposed two-wheel vehicle, it has been shown that the proposed sway suppression control system is effective in the sway suppression of the vehicle body, and useful for the safety aspect to the proposed two-wheel vehicle.

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