Omni-directional vehicle control based on body configuration

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Abstract
Purpose – The purpose of this paper is to develop a new wheel control scheme for wheeled vehicle with passive linkage mechanism which realizes high step-overcoming performance.
Design/methodology/approach – Developing wheeled vehicle realizes omni-directional motion on flat floor using special wheels and passes over non-flat ground using the passive suspension mechanism. The vehicle changes its body shape and wheel control references according to ground condition when it runs over the rough terrain.
Findings – Utilizing the proposed wheel control scheme, the slip ratio and the disturbance ratio of the wheel reduce when the vehicle passes over the step and its step-overcoming performance is improved.
Originality/value – The paper's key idea is modification of its kinematic model referring to the body configuration dynamically and using this model for wheel control of the vehicle. The controller adjusts the wheel control references when the vehicle passes over the rough terrain changing the body shape and reduces the slippage and the rotation error of wheels.

Keywords Control systems, Vehicle suspension systems, Robotics

Paper type Research paper

1. Introduction

In recent years, mobile robot technologies are expected to perform various tasks in general structured environment such as nuclear power plants, large factories, welfare care facilities, hospitals, and homes. However, there are narrow spaces with steps and slopes in such indoor environments, and it is difficult for general car-like vehicles to run around there.

Generally, the vehicle is required to have quick and efficient mobile function for effective task execution. The omni-directional mobility is useful for the tasks, especially in narrow spaces, because there is no holonomic constraint on its motion (Campion *et al.*, 1996; Ichikawa, 1995). Furthermore, the step-overcoming function is necessary when the vehicle passes over the general environment which has vertical gaps between two or plural flat floors. Thus, in order to run around general environment, the vehicle needs to equip both of functions.

In related works, various types of omni-directional mobile systems are proposed (legged robots, ball-shaped wheel robots, crawler robots, and so on). The legged robot (Endo and Hirose, 1999; Hirukawa *et al.*, 2005) can move in all directions and has high ability of passing over unevenness and large steps in natural environment. However, its energy efficiency is not so good because the mechanisms tends to be complicated and the robot need to use its actuators in order to only maintain its posture. Therefore, for the structured environment, these robots are over speck. The robot with ball-shaped wheels can run in all directions (Wada and Asada, 1999), however, it cannot run over the small gaps. The special crawler mechanism (Hirose and Amano, 1993) is also proposed for the omni-directional mobile robot, but which can climb over only small steps. Special wheeled robots can move all directions with simple mechanism same as a normal wheeled vehicle. However, there is a fatal demerit that the robots with the universal wheels can nearly pass over the unevenness of a ground in the force free directions (Ferriere and Campion, 1996). Thus, there is still a lack of well-adapted mobile system for both narrow spaces and irregular terrain operation.

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Therefore, we are developing a holonomic omni-directional vehicle with step-climbing ability (Yamashita *et al.*, 2001; Chugo *et al.*, 2003). Our vehicle utilizes new passive suspension system, which is suitable for steps in the structured environment, and it has seven special wheels with DC motors, which are controlled, respectively. Thus, the vehicle has the redundant actuations and the mobile system calculates the control reference of each wheel based on the kinematics model and controls each actuator. When our vehicle passes over the irregular terrain, the body shape changes by the ground condition and the kinematic model changes together. Therefore, it is required that our mobile system changes the kinematic model referring the modification of the body shape dynamically.

Many mobile vehicles, which have the passive suspension system, are developed (Bickler, 1993; Stone, 1996; Estier *et al.*, 2000). However, in many cases, these researches do not discuss the body configuration. Lamon already discusses the body configuration for 3D-odometry (Lamon and Siegwart, 2003), however, they do not discuss the kinematics model and control reference of each actuator. The unsuitable control reference, which is calculated by fixed kinematic model, will cause to make the vehicle takes unstable posture and the wheel slippage, and the performance of step-overcoming will become poor.

In general, mobile vehicle for rough terrain uses traction control scheme for wheel control. In our previous works, we discussed the wheel feedback control system considering with the output traction of each wheel (Chugo *et al.*, 2005a, b). However, it is difficult to coordinate the rotation velocities of each wheel using only the traction control, especially, when the change of body configuration is large.

Thus, we propose the wheel control method, which is according to the body shape. Our proposed method realizes that the mobile vehicle passes over the irregular terrain with stably posture by reducing the slippage and the rotation error of the wheels. Our control method improves the vehicle performance of passing over the irregular terrain. Our key idea is that our controller changes the kinematic model referring to the modification of the body shape dynamically. Using our proposed method, the controller can adjust the wheel control references based on this kinematic model when our vehicle passes over the rough terrain changing the body shape. We verify the performance of the proposed control method through the computer simulations and experiments.

This paper is organized as follows: we introduce the mechanical design and the controller of the vehicle in Section 2; we discuss the new proposed control method in Section 3; we show the results of computer simulations and experiments in Section 4; Section 5 is conclusion of this paper.

2. Mobile platform

2.1 Mobile mechanism

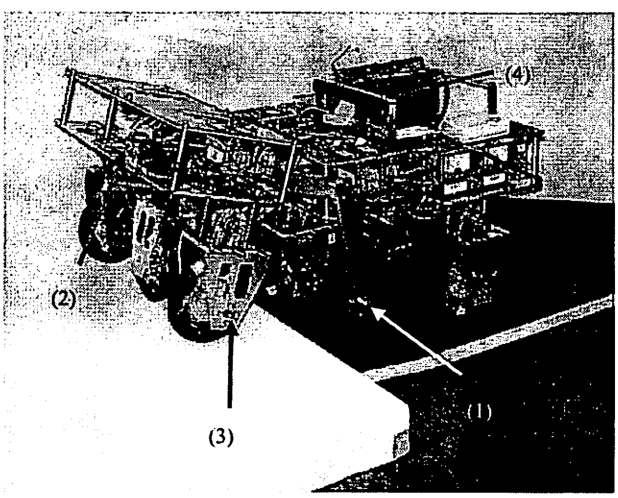
For practical use, it is important for the vehicle to design a reasonable mechanism that has no excessive performance. Generally, running on the structured terrain (Thianwiboon *et al.*, 2001), the energy efficiency of the wheeled mobile system is better than the other mobile mechanisms (e.g. legged or crawler type). Thus, we adopt the universal wheel for omni-directional mobile function. The universal wheel realizes the omni-directional mobility with the advantage of a wheeled system. Furthermore, when we design a mobile robot, there are two approaches to improve the step-climbing ability of the robot. One is to improve the step-climbing ability in all the directions equally, and the other is to improve the maximum ability to pass over steps in a fixed direction with simple mechanism. If the robot has a holonomic and omni-directional mobile ability, it can change its direction in front of the step. Therefore, to realize low cost system, we design the holonomic omni-directional wheeled robot which passes the step only in forward and backward direction.

Figure 1 shows the prototype vehicle system. The vehicle has seven wheels and each wheel is connected to single DC motor. The size of prototype vehicle is 750 mm (L) × 540 mm (W) × 520 mm (H) and its weight is 22 kg.

The mobile mechanism consists of seven special wheels with free rollers and a passive suspension system. The special wheel equips 12 cylindrical free rollers (Figure 2; Asama *et al.*, 1995) and realizes to generate the omni-directional motion using plural wheels arranged in the different direction and suitable wheel control.

Our mechanism utilizes new passive suspension system, which is more suitable for the step than general rocker-bogie suspensions as shown in Figure 3 (Chugo *et al.*, 2005a, b). The free joint point 1 is in the same height as the axle and this system helps that the vehicle can pass over the step smoothly when the wheel contacts it (Figure 4). No sensors and no additional actuators are equipped to pass over the irregular terrain.

Figure 1 Overview of the proposed mechanism



Notes: (1) is the passive joint 1 (pitch angle); (2) is the passive joint 2 (roll angle); (3) is the special wheel; and (4) is the control computer system (CPU and I/O card)

Figure 2 Special wheel

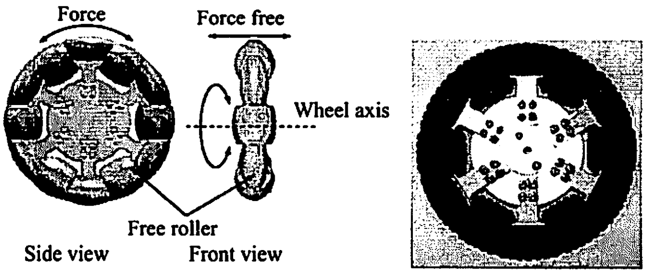


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Figure 3 New passive linkage mechanism

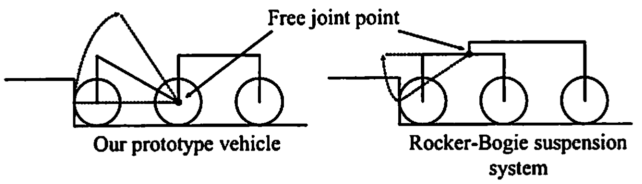
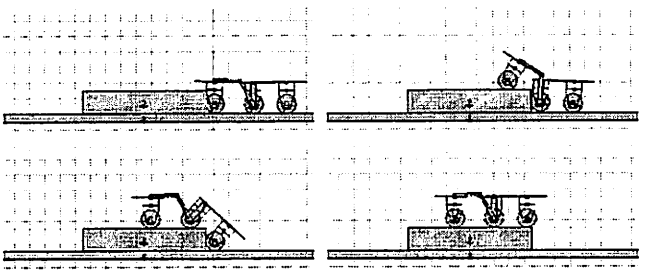


Figure 4 Step climbing



2.2 Controller

The control system is shown in Figure 5. The developed vehicle has seven motor drivers, two potentiometers each joint for the configuration angle and two tilt meters for the inclination of the body. Our vehicle has redundant actuation system using seven wheels and proportional-integral-derivations (PID)-based control system (Chugo *et al.*, 2005a, b) synchronizes the wheel rotation based on control reference, which is calculated by the kinematic model.

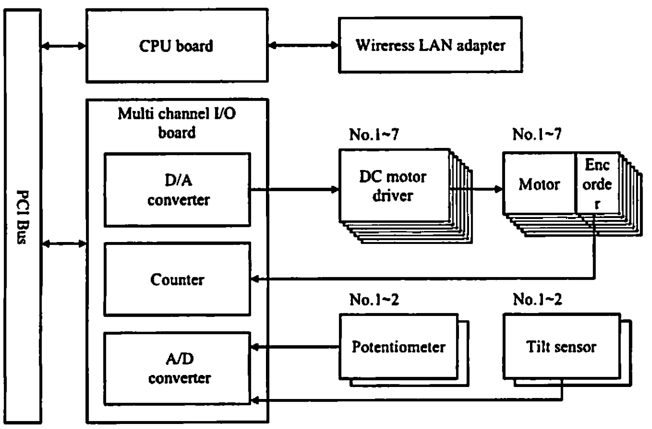
3. Control system

3.1 Passing over the step

In general, the wheeled vehicle for rough terrain drives all wheels and has redundant actuations. Therefore, the vehicle system calculates from its reference speed to the actuator velocity commands based on its kinematic model (Carlisle, 1979). However, the passive suspension mechanism changes the body shape and it is required to modify its kinematic model.

If the system does not modify the vehicle's kinematic model, the actuator velocity commands are unsuitable to an actual situation because of the difference between an actual body configuration and assumed configuration of the model used for calculation. The unstable command leads the wheel slips or blocks during passing over the step and these actions cause that the vehicle loses the balance of the body posture as shown in Figure 6. Thus, we proposed a wheel control method based on the body configuration information.

Figure 5 Overview of the vehicle's control system



3.2 Sensor setup

Our vehicle has two potentiometers on each passive joint and tilt sensors which are attached on the rear part of the vehicle body as shown in Figure 7.

We can measure the following angles using these sensors:

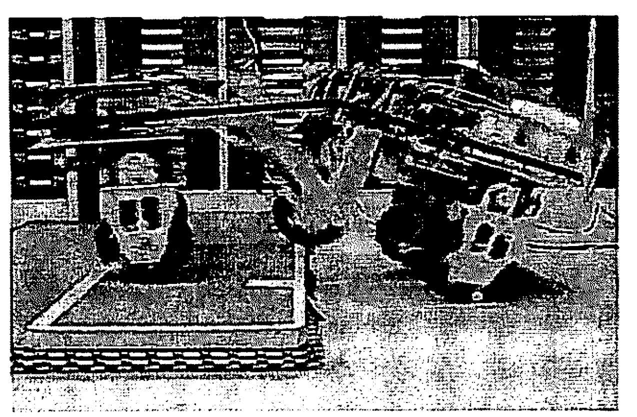
- The roll angle θ_a and pitch angle γ_a of the body shape from potentiometers.
- The roll angle θ_b and pitch angle γ_b of the body inclination from tilt sensors.

Our developed vehicle is designed on the assumption that it passes over non-flat ground in only forward or backward direction. The body shape angle changes to only the advance direction, therefore, we only consider the roll angle (θ_a and θ_b).

3.3 Kinematics modification

Figure 8 shows the arrangement of the wheels (we display as wheel i: $i = 1\ldots7$) and the definition of the coordinates, the length of each links, and the rotate speed of each wheel,

Figure 6 The off-balance situation on the irregular terrain



Note: Middle wheel is not grounded and it causes a reducing of step-overcoming performance

Figure 7 The angle meters

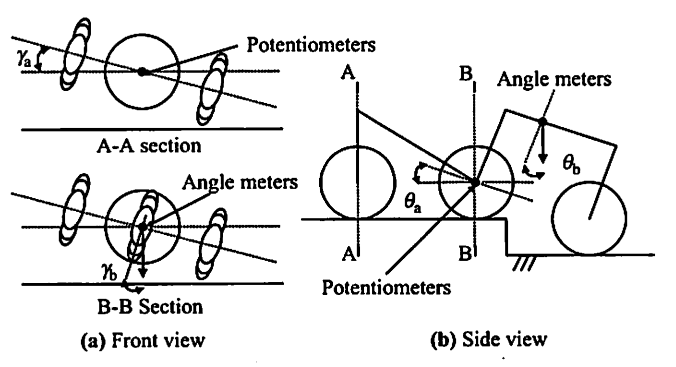
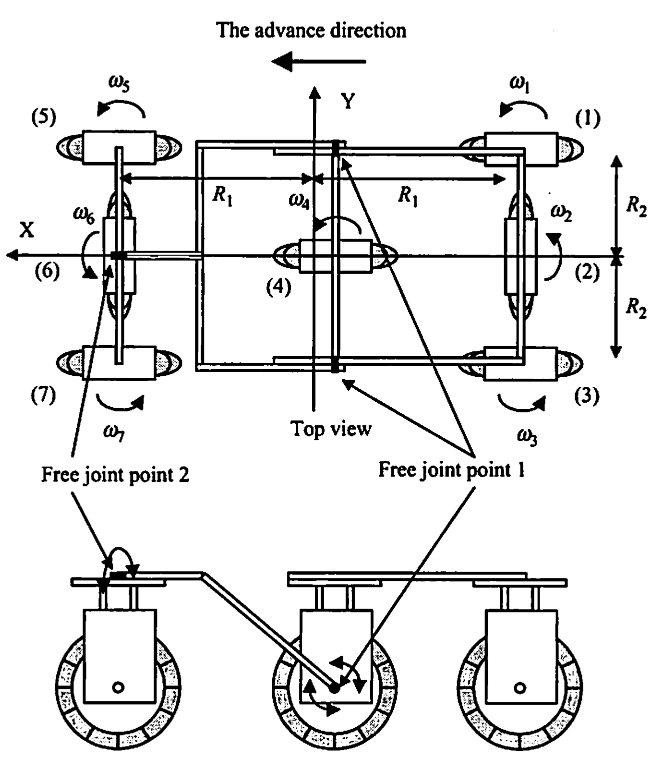


Figure 8 Coordination and parameters



respectively. R_1 and R_2 indicate the length of each links and $\omega_1, \ldots, \omega_7$ are the rotation velocity of each wheel. The relations between the rotation velocity of each wheel ω_i and the rotation velocity of each actuator v_i are calculated in equation (1):

$$\omega_i = kv_i \tag{1}$$

where i is the number of wheel ($i = 1\ldots7$) and k is gear ratio on the actuator.

The wheels 1 and 3 are located on the same straight line and symmetrically on X-axis, therefore the rotation velocity of the wheel 1 is equal to one of the wheel 3 when the vehicle moves forward (Carlisle, 1979). The same relation on Y-axis is realized also about wheels 5 and 7:

$$\omega_1 = -\omega_3 \tag{2}$$

$$\omega_5 = -\omega_7 \tag{3}$$

As the same, when the vehicle moves beside, the rotational velocity of wheel 2 must be equal to the rotation velocity of wheel 6:

$$\omega_2 = -\omega_6 \tag{4}$$

Now, we consider the geometric relationships among the velocity vector of the front wheel (wheels 5 and 7), the middle wheel (wheel 4) and the rear wheel (wheels 1 and 3) when the vehicle passes over the step.

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When the vehicle passes over the slope at an incline of α deg, the position relationship and the velocity vector relationship of wheels 1, 4 and 5 are shown in Figure 8. Figure 9 shows the frame-kinematic model of the vehicle body and we can consider this model as the slider link mechanism. In this model, the slider is the slope surface and the pin joint is the axle point of the wheel. The pin joints are attached on the slider and connected by the link. The degree of the potentiometer is $-\theta_a$ and the degree of the tilt sensor is θ_b.

The body's instantaneous center point of the wheels 1 and 4 is P_{41} in Figure 8 and the center point of the wheels 4 and 5 is P_{54}. Now, we discuss the relationship between wheels 1 and 4 using P_{41}. Equation (5) is derived from geometry in Figure 8:

$$h = a\cos(\alpha - \theta_b) = b\cos\theta_b \tag{5}$$

a and b are the distance between P_{41} and the wheels 4 and 5, respectively. h is the length of the perpendicular vector to link 2. Therefore, equation (6) shows the relationship of the velocity of wheels 4 and 5. Moreover, from equation (2), equation (7) is derived:

$$v_1 = \frac{\cos(\alpha - \theta_b)}{\cos\theta_b} v_4 \tag{6}$$

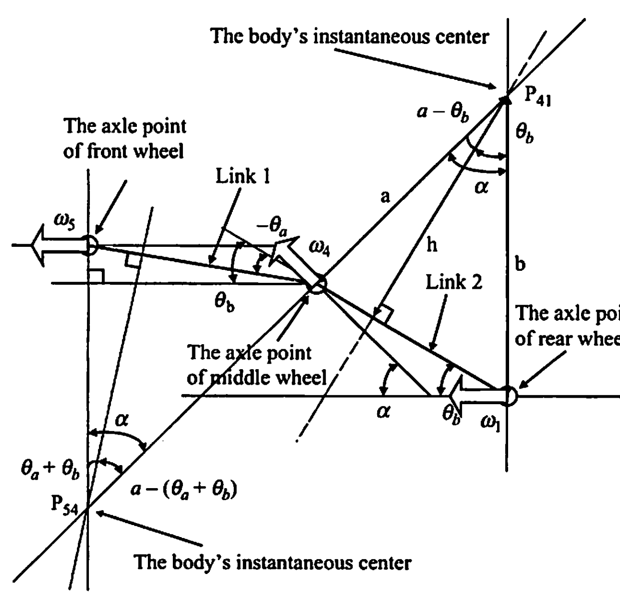
$$v_3 = -\frac{\cos(\alpha - \theta_b)}{\cos\theta_b} v_4 \tag{7}$$

Also, from the view of P_{54}, the relationship of the velocity of wheels 4 and 5 are expressed in equations (8) and (9) is calculated by using equation (3):

$$v_5 = \frac{\cos(\theta_a + \theta_b)}{\cos\{\alpha - (\theta_a + \theta_b)\}} v_4 \tag{8}$$

$$v_7 = -\frac{\cos(\theta_a + \theta_b)}{\cos\{\alpha - (\theta_a + \theta_b)\}} v_4 \tag{9}$$

Figure 9 The relation of the body shape and steps



After all, utilizing these equations, following equations are derived:

$$\dot{x}=\frac{1}{5}\left(\frac{\cos\theta_2}{\cos(\alpha-\theta_2)}v_1-\frac{\cos\theta_2}{\cos(\alpha-\theta_2)}v_3+v_4+\frac{\cos\{\alpha-(\theta_1+\theta_2)\}}{\cos(\theta_1+\theta_2)}v_5-\frac{\cos\{\alpha-(\theta_1+\theta_2)\}}{\cos(\theta_1+\theta_2)}v_7\right) \quad (10)$$

$$\dot{y}=\frac{1}{2}(-v_2+v_6) \quad (11)$$

$$\dot{\theta}=\frac{1}{6}\left(\frac{v_1}{R_2}+\frac{v_2}{R_1}+\frac{v_3}{R_2}+\frac{v_4}{R_2}+\frac{v_6}{R_1}+\frac{v_7}{R_2}\right) \quad (12)$$

Thus, equation (13) is derived as follows:

$$\dot{\mathbf{X}}=\mathbf{J}(\theta_a,\theta_b)\cdot\mathbf{V} \quad (13)$$

$\mathbf{V}=[v_1 \ldots v_7]^{\mathrm{T}}$ is rotation velocity vector of each actuator, $\dot{\mathbf{X}}=[\dot{x}\ \ \dot{y}\ \ \dot{\theta}]^{\mathrm{T}}$ is the velocity vector of vehicle motion and $\mathbf{J}(\theta_a,\theta_b)$ is the Jacobian matrix based on body configuration:

$$\mathbf{J}(\theta_a,\theta_b)=kr\cdot\begin{bmatrix}\frac{\cos(\alpha-\theta_b)}{5\cos\theta_b} & 0 & -\frac{\cos(\alpha-\theta_b)}{5\cos\theta_b} & \frac{1}{5} & \frac{\cos(\theta_a+\theta_b)}{5\cos\{\alpha-(\theta_a+\theta_b)\}} & 0 & -\frac{\cos(\theta_a+\theta_b)}{5\cos\{\alpha-(\theta_a+\theta_b)\}}\\ 0 & -\frac{1}{2} & 0 & 0 & 0 & \frac{\cos\gamma}{2} & 0\\ \frac{1}{6R_2} & \frac{1}{6R_1} & \frac{1}{6R_2} & 0 & \frac{1}{6R_2} & \frac{1}{6R_1} & \frac{1}{6R_2}\end{bmatrix} \quad (14)$$

Therefore, the control reference value of each actuator can be calculated by equation (15):

$$\mathbf{V}=\mathbf{J}^{+}(\theta_a,\theta_b)\cdot\dot{\mathbf{X}} \quad (15)$$

where $\mathbf{J}^{+}(\theta_a,\theta_b)$ is pseudo-inverse of Jacobian matrix:

$$\mathbf{J}^{+}(\theta_a,\theta_b)=(\mathbf{J}^{\mathrm{T}}\mathbf{J})^{-1}\mathbf{J}^{\mathrm{T}}=\frac{1}{kr}\cdot\begin{bmatrix}\cos\theta_b/\cos(\alpha-\theta_b) & 0 & R_2\\ 0 & -1 & R_1\\ -\cos\theta_b/\cos(\alpha-\theta_b) & 0 & R_2\\ 1 & 0 & 0\\ \cos\{\alpha-(\theta_a+\theta_b)\}/\cos(\theta_a+\theta_b) & 0 & R_2\\ 0 & 1 & R_1\\ -\cos\{\alpha-(\theta_a+\theta_b)\}/\cos(\theta_a+\theta_b) & 0 & R_2\end{bmatrix} \quad (16)$$

Thus, the vehicle system must synchronize the rotation velocity of the wheels according to equation (15).

3.4 Traction control

In our control system, we utilize PID-based torque control method (Chugo *et al.*, 2005a, b). The torque reference of each wheel is calculated by equation (17):

$$\tau_i=k_{\mathrm{p}}e+k_{\mathrm{i}}\int e\,\mathrm{d}t+k_{\mathrm{d}}\frac{\mathrm{d}e}{\mathrm{d}t} \quad (17)$$

where e is error value and $k_{\mathrm{p}}, k_{\mathrm{i}}, k_{\mathrm{d}}$ are proportional, integral and derivative gain of PID controller, respectively.

This control method adjusts the control reference may not exceed the marginal torque. The controller calculates the coefficient c_i as follows:

$$c_i=\begin{cases}(\tau_{\max}/\tau_i) & \text{if } \tau_i>\tau_{\max},\\ 1 & \text{if } \tau_i\le\tau_{\max},\end{cases} \quad (18)$$

where $\tau_{\max}$ is marginal torque of the actuator.

The torque output is calculated by equation (19):

$$\tau_i^{\mathrm{out}}=c\cdot\tau_i \quad \text{where } c=\min\{c_1,\ldots,c_7\} \quad (19)$$

Comparing from c_1 to c_7, the controller can maintain each wheel's rotation velocity.

4. Experiment

4.1 Simulation setup

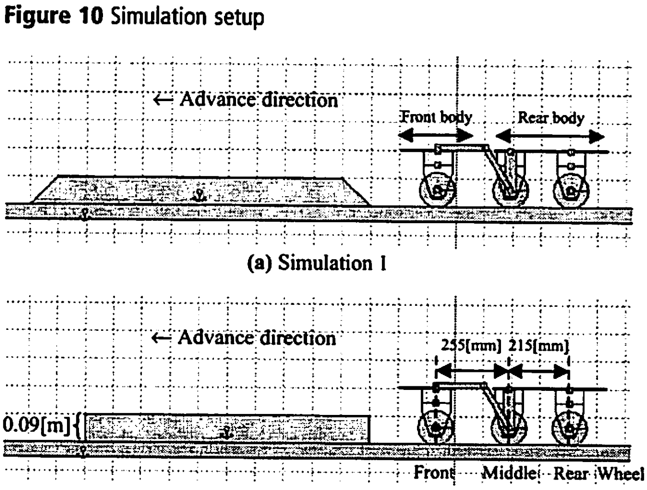
We verify the performance of our proposed controller which uses our proposed control reference based on its body shape by computer simulations. Here, we discuss the vehicle motion in the direction of the X-axis in Figure 8, because our vehicle passes over the step in its forward and backward direction. As initial conditions, simulation parameters are chosen from the prototype model. The parameters are shown in Table I.

In order to verify that proposed system is effective, we test two situations. In simulation (1), the vehicle runs on 45° slope as shown in Figure 10(a). The height of slope is 0.09 m and

Table I Vehicle parameters

Length	Front 195 mm Rear 400 mm
Body weight	Front 7.8 kg Rear 13.8 kg
Wheel diameter	132 mm
Distance between wheels	Front 255 mm Rear 215 mm
Friction coefficient	Static 0.4 Dynamic 0.3
Running speed	0.25 m/sec

Figure 10 Simulation setup



(b) Simulation 2

the length is 1 m. In simulation 2, the vehicle passes over the step as shown in Figure 10(b). The height of the step is 0.09 m and the length is 1 m. We compare the result by our proposed controller with the result utilizing standard controller, which uses fixed control reference and does not consider the body shape. Both controllers utilized our PID-based control scheme as shown in Section 3.4.

We use the Working Model 2D as a physical simulator and MATLAB. Both applications are linked by dynamic data exchange function, which is equipped on Windows OS.

4.2 Simulation result

Tables II and III show these simulation results. The slip ratio (Yoshida and Hamano, 2001) of the wheel and the disturbance ratio (Chugo *et al.*, 2005a, b) of the wheel are calculated by equations (20) and (21), respectively. The disturbance ratio means the error ratio of the rotation velocity:

$$\hat{s} = \frac{r\omega - v_\omega}{r\omega} \tag{20}$$

$$\hat{d} = \frac{\omega_{ref} - \omega}{\omega} \tag{21}$$

ω is the rotation speed of the actuator and ω_{ref} is the reference value of wheel rotation velocity. r and v_ω indicate the radius of the wheel and the vehicle speed (we measure the axle of middle wheel) in world coordination, respectively.

By these simulations, we verified the proposed controller improves the performance of the step overcoming. The slip ratio of the wheel decreases 44 percent in the simulation (1) and 45 percent in the simulation (2). And the disturbance ratio of the wheel decreases 41 percent in the simulation (1) and 55 percent in the simulation (2). The performance of our proposed controller is better than one of the standard controller with fix references regardless of the angle of the step.

Furthermore, the load which is applied to one wheel is distributing. For example, the difference between disturbance ratio of the middle and rear wheel decreases to 50 percent in the simulation (1) and 33 percent in the simulation (2). This means wheels can transmit its traction force to the ground more effectively using control references according to the body shape.

Table II The error result of simuration (1) (percent)

	Reference	Front wheel	Middle wheel	Rear wheel	Average
Slip	Fixed	7.5	6.7	7.0	7.1
	Proposed	4.0	4.0	4.0	4.0
Disturbance	Fixed	2.6	1.8	2.2	2.2
	Proposed	0.7	1.1	0.9	0.9

Table III The error result of simulation (2) (percent)

	Reference	Front wheel	Middle wheel	Rear wheel	Average
Slip	Fixed	17.7	16.4	15.6	16.6
	Proposed	8.3	9.6	9.3	9.1
Disturbance	Fixed	2.7	1.9	2.5	2.0
	Proposed	0.8	1.3	1.1	1.1

4.3 Experiment

Here, we verify the performance of our control method by the experiment. In this experiment, the prototype vehicle passes over the step and we verify about two topics. One topic is the height of the step which the vehicle can climb up. The other topic is the slippage and the rotation error of the wheels when the vehicle passes over 0.09 m height step. We measure the moving velocity of the axle of the middle wheel using the motion capture system as the vehicle speed. As the same as simulation, we compare the result by our proposed controller with the result utilizing standard controller, which uses fixed control reference and does not consider the change of body shape during step climbing.

As the result of the experiment, the vehicle can pass over the 0.128 m height step with our proposed controller as shown in Figure 11. With standard controller, the vehicle can pass over the only 0.096 m height step.

When the vehicle passes over the 0.09 m height step, the slip ratio of the wheel decreases to 60 percent and the disturbance ratio of the wheel decreases to 51 percent by our proposed control scheme as shown in Table IV. As the results, the vehicle can passes over the 0.09 m height step more smoothly with our proposed controller as Figure 12.

Figure 13 shows the control reference of our proposed controller during passing over 0.9 m height step. Usually, the control reference of the wheel which contacts the step is larger than the references of other wheels which are on the flat ground. From Figure 12, when the front wheel contacts the

Figure 11 Passing over the 128 mm height step

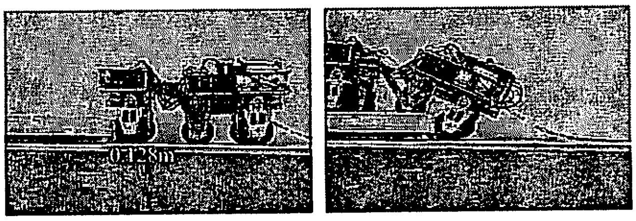
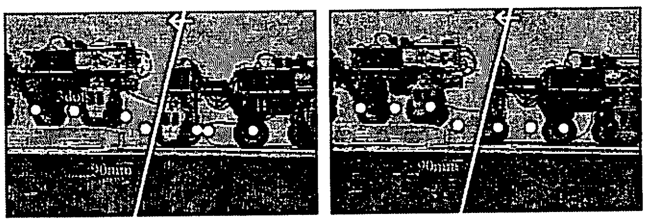


Table IV The error result of experiment (percent)

	References	Front wheel	Middle wheel	Rear wheel	Average
Slip	Fixed	18.8	17.9	18.4	18.4
	Proposed	10.8	12.0	10.4	11.1
Disturbance	Fixed	12.6	11.1	11.9	11.9
	Proposed	6.2	5.1	7.0	6.1

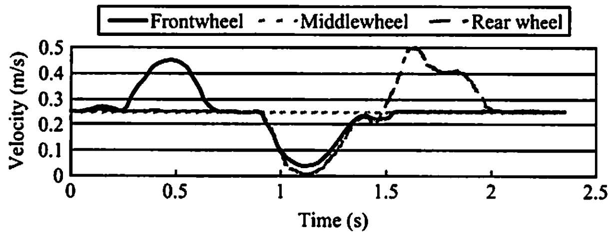
Figure 12 Passing over the 90 mm height step



(a) With fixed reference (b) With proposed reference

Note: White points are plotted at every 0.3 [sec]

Figure 13 Control references during step climbing

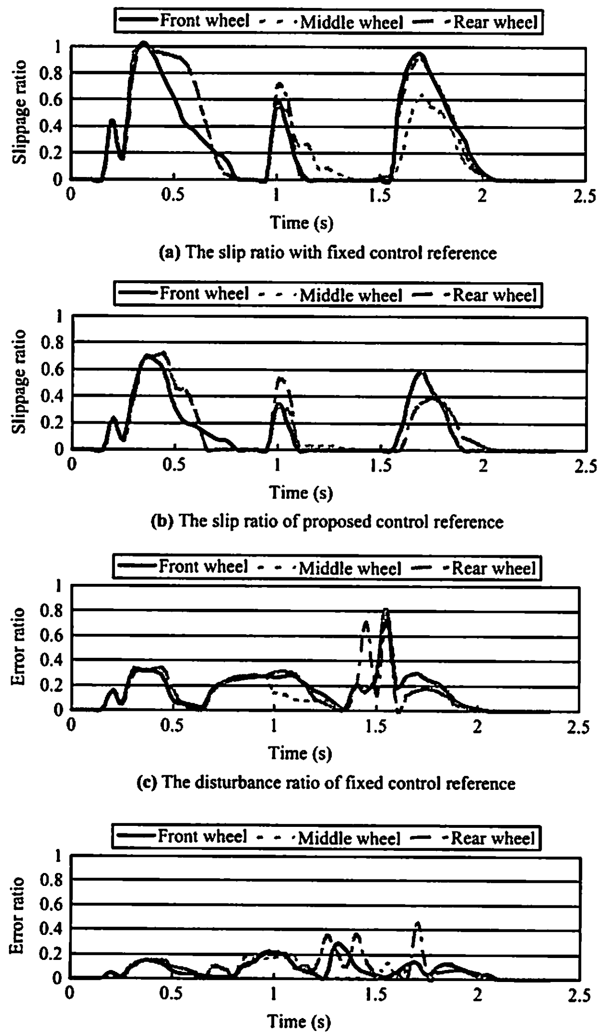


step, its reference is larger than one of other wheels. As the same, when the middle wheel contacts the step and when the rear wheel contacts it, its reference is larger than one of other wheels. Therefore, we can verify our proposed controller derives the reference of each wheel coordinated according to the body configuration.

Figure 14 (a) and (b) show the slip ratio by standard controller with fixed control reference and proposed controller, respectively. As the same, Figure 14 (c) and (d) show each disturbance ratio. In Figure 14 (a) and (b), there are three peaks in both cases. They show the contact between the step and the front, middle and rear wheel and at that time, the change of body shape is large. Using proposed control references, three peaks are reduced and this means the modification of wheel control reference is useful especially when the vehicle body changes largely. In Figure 14 (c) and (d), the disturbance ratio also reduced during step-overcoming evenly. This means that using control reference according to the body shape, the load is distributed to each wheel equally and each wheel can rotate according to their reference.

From these results, our proposed controller derives suitable control reference according to the change of body configuration. As the result, our controller reduces the slippage and the rotation error of the wheels. Furthermore, our method improves the vehicle's performance of the step-overcoming. Therefore, our proposed control method is effective for passing over the step for the vehicle with passive linkages.

Figure 14 The result of the experiment



(a) The slip ratio with fixed control reference

(b) The slip ratio of proposed control reference

(c) The disturbance ratio of fixed control reference

(d) The disturbance ratio of proposed control reference

5. Conclusions

In this paper, we propose the wheel control method according to change of vehicle's body shape. We discuss the kinematic model referring to the body configuration. Using this method, the controller can adjust the suitable wheel control references when the vehicle passes over the obstacles changing its body shape.

We verified the effectiveness of our proposed method by the computer simulations and experiments using our prototype. Utilizing our proposed control scheme, the slip ratio and the disturbance ratio of the wheel reduces when the vehicle passes over the step and its step-overcoming performance is improved. It is effective not only the step but also other obstacles, for example, the slopes.

As our future works, we will consider the motion planning method based on the environment information.

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