電気自動車運動制御における車体速度の推定手法の実験検証

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Experimental Verification of Velocity Estimation Methods Used for EV Motion Control

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Abstract

In this paper, we discuss several methods for the estimation of EV body velocity, which will be used for the motion control of EV. We especially focus on a model based nonlinear velocity observer. In this study, the longitudinal and the lateral velocity of the EV chassis will be estimated in real time. Experiments are performed by using an EV named "UOT March II", which was built by the University of Tokyo. In the experiments, the EV drives in several driving cases under the normal road condition. In order to verify the estimation method, an optical sensor for detecting the velocity is used. It can be shown that results estimated by the proposed observer are quiet consistent with those obtained by the expensive optical sensor.

キーワード：電気自動車, 車体速度, 推定方法, 運動制御
(Electric vehicles, vehicle velocity, estimation methods, motion control)

1. Introduction

Since the utilization of the electrical motors for EVs, some advanced control methods which are impossible for inner combustion vehicles (ICVs) can be realized for EVs based on those motors [1] [2]. On the other hand, those control algorithms can much more greatly benefit if several parameters, such as body velocity, tire-road friction coefficient, wheel slip and body side slip angle are available on line [3].

Although, those vehicle states or parameters can be directly measured by some sensors such as optical sensors, there are practical problems such as cost, reliability and noise for the applications on the EVs. However, observers are often designed to estimate signals which are difficult or expensive to measure for motion control. Therefore, we try to use an observer to estimate the vehicle velocity which is important for the active motion control for EVs. This is one of the motivations of this study for the estimation of the vehicle velocity.

Recently, many analytical and experimental studies on the estimation of the vehicle velocity have been performed. Senger [4] estimated the lateral speed in the linear tire region. In his method, the cornering stiffness of the tire is assumed to be known. Based on this method, many researches accommodate the estimation of the cornering stiffness of the tire. Satisfactory results are obtained for different purpose. Farrelly [5], Chunting Mi [6] applied the estimation methods which are based on the physical modeling of the vehicles. They are kinds of kinematics methods, in which the observers can be tuned according to the kinematics and designed to be robust to the unknown of cornering stiffness. Lars Imsland [7], Cherouatc [8] proposed Lyapunov based nonlinear observer techniques for velocity estimation. Lars Imsland used a modular way to estimate the longitudinal and lateral vehicle velocity. In this method, the stability of the observer is proved by the Lyapunov method. In the experiments, this method shows a satisfactory performance. Fangjun Jiang [9] gave an adaptive nonlinear velocity observer. The design of this observer is performed according to the working conditions of one wheel and used for the ABS control. However the robustness of this estimation did not be verified.

In summary, those estimation methods are all model based methods. The dynamic model, especially the linear or nonlinear tire model, is often used. Many literatures about this topic, for example [10] [11], are typically published for the tire models. In this paper, these kinds of methods are applied. Next, in those methods, the estimations of the tire road friction coefficient are mentioned [12] [13]. We in this paper only use the results but do not focus on this topic. Although as mentioned above, the researches on the estimation of the vehicle velocity has been performed, there have few comparison among and summary about these methods. In order to apply these methods for the motion control of EVs, we here will discuss, compare, and verify several representative ones of those mentioned methodologies. This is another motivation of this study.

The paper is organized as follows. In section 2, a simplified model of EV dynamics is presented. In section 3, the design methods for velocity estimation are discussed. In section 4, these estimation methods, especially the nonlinear modular methods, are investigated using experiments. Finally, in section 5, the preliminary conclusions are given.

2. Vehicle Dynamics

EV dynamics is the certain performance accomplished during accelerating, braking, cornering and ride. In this study, dynamics of EV in planar motion will be concentrated.

〈2·1〉 Chassis Modeling Assume that the free form of the EV is shown as Fig.1. $X_{COG}OY_{COG}$ is the body fixed coordinate system, which is defined with the origin at the center of the gravity of the EV. X_{COG} axis points forward and Y_{COG} points to the left. V_{COG} is the vehicle velocity which is defined in this coordinate. $V_i,\ i=\{1,2,3,4\}$ is velocity of wheel center in the coordinate of $X_{COG}OY_{COG}$. δ_f is the front steering angle. β and γ are side slip angle and yaw rate of the body. $\beta_i,\ i=\{1,2,3,4\}$ is slip angle of each tire which is also illustrated in this coordinate system.

Another coordinate system, which is illustrated as *ton* in the Fig.1, is fixed with the origin at the center of the wheel. V_t and V_n are velocities of the wheel in this coordinate. F_{ni} and F_{ti}, which are also illustrated in this coordinate system, are traction forces acting on the ith tire.

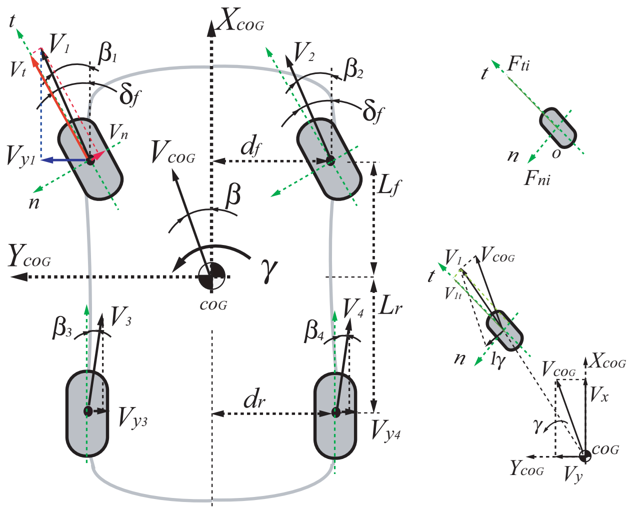

図 1 EV の平面運動力学モデル

Fig. 1. Dynamic planar motion model of the EV

In this study we consider the EV chassis as a rigid body. The effect of suspension is neglected. The longitudinal dynamics can be expressed as

$$Ma_x = \sum_{i=1}^{4} F_{xi} = \sum_{i=1}^{4} (F_{ti}cos\delta_i - F_{ni}sin\delta_i) \quad \cdots\cdots (1)$$

The lateral dynamics can be expressed as

$$Ma_y = \sum_{i=1}^{4} F_{yi} = \sum_{i=1}^{4} (F_{ti}sin\delta_i + F_{ni}cos\delta_i) \quad \cdots\cdots (2)$$

The yaw motion can be expressed as

$$\begin{aligned} J\dot{\gamma} = &\ \tfrac{d_f}{2}(F_{x2} - F_{x1}) + \tfrac{d_r}{2}(F_{x4} - F_{x3}) \quad \cdots\cdots\cdots (3) \\ &+ L_r(F_{n3} + F_{n4}) - L_f(F_{n1} + F_{n2}) \end{aligned}$$

where M is the mass of the EV. a_x and a_y are longitudinal and lateral acceleration. J is the inertial moment of the EV at the origin of the COG. L_f is the distance from the COG to the front axis of the EV. L_r is the distance from the COG to the rear axis of the EV.

In addition, the chassis velocity of the wheel center in the $X_{COG}OY_{COG}$ coordinate system can be expressed as

$$V_{1,x} = V_x - \gamma \cdot \frac{d_f}{2}; \quad V_{1,y} = V_y + \gamma \cdot L_f \cdots\cdots\cdots (4)$$

$$V_{2,x} = V_x + \gamma \cdot \frac{d_f}{2}; \quad V_{2,y} = V_y + \gamma \cdot L_f \cdots\cdots\cdots (5)$$

$$V_{3,x} = V_x - \gamma \cdot \frac{d_r}{2}; \quad V_{3,y} = V_y - \gamma \cdot L_r \cdots\cdots\cdots (6)$$

$$V_{4,x} = V_x + \gamma \cdot \frac{d_r}{2}; \quad V_{4,y} = V_y - \gamma \cdot L_r \cdots\cdots\cdots (7)$$

where V_x and V_y are longitudinal and lateral velocity of the COG in the $X_{COG}OY_{COG}$ coordinate system. $V_{i,x}$ and $V_{i,y}$ are longitudinal and lateral velocity of the wheel center in the $X_{COG}OY_{COG}$ coordinate system.

According to the relationship between the coordinate system $X_{COG}OY_{COG}$ and the coordinate system *ton* the velocity of the wheel center can also be written as

$$V_{i,t} = V_{i,x}cos\delta_i + V_{i,y}sin\delta_i \cdots\cdots\cdots\cdots\cdots\cdots\cdots (8)$$

$$V_{i,n} = V_{i,y}cos\delta_i - V_{i,x}sin\delta_i \cdots\cdots\cdots\cdots\cdots\cdots\cdots (9)$$

The side slip angle of each tire is expressed as

$$\beta_1 = arctan\frac{V_{1,y}}{V_{1,x}} - \delta_1; \quad \beta_2 = arctan\frac{V_{2,y}}{V_{2,x}} - \delta_1 \cdot \ (10)$$

$$\beta_3 = arctan\frac{V_{3,y}}{V_{3,x}}; \quad \beta_4 = arctan\frac{V_{4,y}}{V_{4,x}} \ \cdots\cdots\cdots\cdots (11)$$

〈2·2〉 Nonlinear Tire Modeling It is well known that the friction force between the tire and the road is a nonlinear function of the wheel slip. Fig.2(b) illustrates that function, which is also known as the $\mu - s$ curve. Note that those $\mu - s$ curves change for different vehicle speeds and on different road surfaces. The optimum wheel slip value for maximum friction also varies. Fig.2(a) shows dynamics, friction forces and their relationships.

The wheel slip can be calculated in the *ton* coordinate system.

$$\lambda_{i,t} = \frac{V_{i,\omega} - V_{i,t}}{V_{i,t}} \ \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (12)$$

$$\lambda_{i,n} = sin\beta_i \ \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (13)$$

where $\lambda_{i,t}$ and $\lambda_{i,n}$ are longitudinal and lateral slip ratio of the ith wheel respectively. $V_{i,\omega} = r\omega$. r is the radius of the

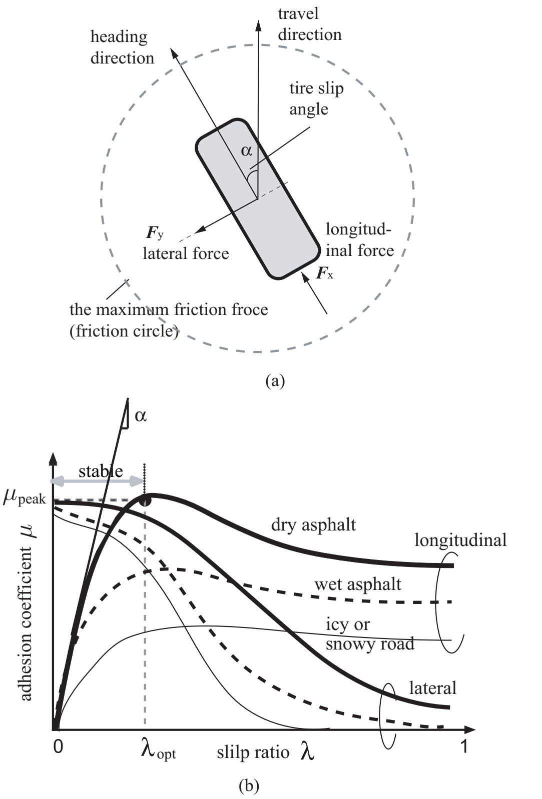

図 2 (a) タイヤに発生した摩擦力. (b) タイヤと路面違いでの $\mu - s$ 曲線

Fig. 2. (a) Working conditions and friction forces of the tire. (b) Different $\mu - s$ curves according to different tire-road conditions

ith wheel. ω is the rotational velocity of that corresponding wheel.

In normal drive condition, the vehicle velocity is almost the same as the wheel velocity. However, when a wheel becomes locked or slips, the vehicle velocity and the wheel velocity are quite different. The estimation of the tire friction forces based on the wheel slip is important for the velocity detection. As mentioned above, there are many literatures which are published about the tire friction estimation (10) (11) (12) (13) (14), we do not focus on this topic here. We only prefer the method proposed by H. B. Pacejka (14). The method calculates the friction forces according to the wheel slip as

$$F = D \cdot sin(C \cdot arctan(B\phi)) \cdots\cdots\cdots\cdots\cdots\cdots (14)$$

where B is the stiffness factor. C is the shape factor. D is the peak factor. E is the curvature factor. Those factors are previously defined. $\phi = (1 - E)\lambda + (E/B)arctan(B\lambda)$. F can be longitudinal and lateral forces according to the wheel slip $\lambda_{i,t}$ or $\lambda_{i,n}$ respectively.

This empirical method for tire force calculation is often used in almost all nonlinear observer designs for the velocity estimation.

3. Observer Design

We here summarize three main methods for velocity estimation which are published in the literatures recent years.

- Linear estimation method

  In this method, the vehicle dynamics is linearized and illustrated by a bicycle model. The transfer function from the steering angle to the yaw rate and lateral velocity can be obtained. One method uses the transfer function $H_{\delta_f \to \gamma}$ to fit the unknown parameters, such as the cornering stiffness. And then uses the estimated parameters to detect the lateral velocity, which can be expressed by the transfer function $H_{\delta_f \to V_y}$.

  Another method uses the linear model of the vehicle dynamics and calculates the velocity directly. Fig.3 shows this kind of method.

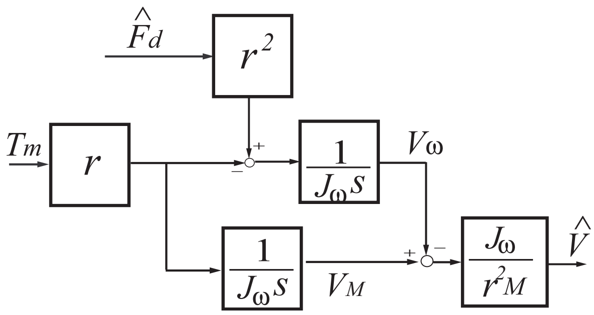

図 3 線形モデルに基づく車体速度推定の手法 (6)

Fig. 3. EV velocity estimation method based on the linear model

  where T_m is the driving torque. $\hat{F}_d$ is the estimated traction force. J_ω is the inertia moment of the wheel.

- Adaptive tuning

  This method only uses two or four measured wheel velocities. The acceleration or yaw rate signals are not needed. It works just as an adaptive nonlinear filter. It is often used in a special way, which is expressed as (9)

$$\dot{y}(t) = -R_g \cdot sign(y(t) - x(t)) \cdots\cdots\cdots\cdots (15)$$

$$y(t = 0) = y_0$$

  where $y(t)$ represents the actual vehicle velocity, the change of $y(t)$ reflects the road condition (9). $x(t)$ is the input of this filter. R_g is a tuning parameter. $sign(*)$ is a sign function.

  In a special case, this kind of method may give a satisfying result. However, the reliability of this method did not be proved in those kinds of literatures.

- Nonlinear estimation method

  This method often considers the kinematics of vehicle and is promised for the stability and convergence.

Lyapunov based nonlinear observer design is the typical example of this kind of method. In this paper, we mainly focus on one of this method.

As for this method, we consider the one proposed by Lars Imsland [7]. Further, based on this method, we re-design it and verify it by the experiments. The mentioned nonlinear observer is constructed in a modular way. The wheel speeds, yaw rate, and longitudinal accelerations are used for estimating the longitudinal velocity. Thereafter, lateral velocity will be estimated by using the information of estimated longitudinal velocity.

In our design, the original idea, in the estimation of lateral velocity, in order to avoid using complex nonlinear tire model, a simply linear full order side slip observer is used to estimate the slip angle, and then the tire forces are calculated by using the estimated side slip. The yaw rate, lateral accelerations are also used for the lateral estimations. The mentioned velocity observer is shown in Fig.4.

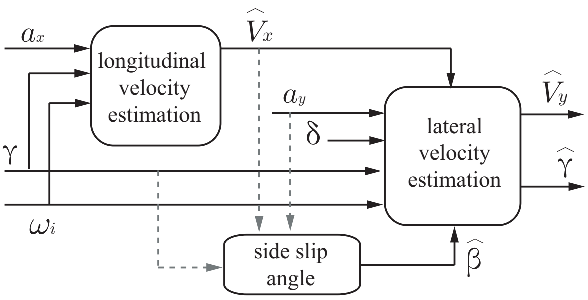

図 4　提案した車体速度推定オブザーバー
Fig. 4.　The proposed observer for EV velocity estimation

The longitudinal kinematics of EV can be written as

$$\dot{V}_x = a_x \quad \cdots\cdots (16)$$

Thus the longitudinal velocity can be estimated by [7]

$$\dot{\hat{V}}_x = a_x + \sum_{i=1}^{4} K_i(a_x)(V_{x,i} - \hat{V}_x) \quad \cdots\cdots (17)$$

where the observer gain $K_i(a_x)$ depends on the acceleration measurements.

The lateral kinematics of EV can be written as

$$\dot{V}_y = -V_x\gamma + a_y \quad \cdots\cdots (18)$$

$$\begin{aligned} J\dot{\gamma} = {} & \tfrac{d_f}{2}(F_{x2} - F_{x1}) + \tfrac{d_r}{2}(F_{x4} - F_{x3}) \quad \cdots\cdots (19) \\ & + L_r(F_{n3} + F_{n4}) - L_f(F_{n1} + F_{n2}) \\ & = \tfrac{1}{J} f(\lambda_t, \lambda_n) \end{aligned}$$

The lateral velocity is designed together with the yaw rate. The method can be summarized as

$$\dot{\hat{V}}_y = -V_x\hat{\gamma} + K_{v_y}(Ma_y - \sum \hat{F}_{y,i}) \quad \cdots\cdots (20)$$

$$\dot{\hat{\gamma}} = \frac{1}{J} f(\lambda_t, \lambda_n) + K_\gamma(\gamma - \hat{\gamma}) \quad \cdots\cdots (21)$$

where the observer gain K_{v_y} and K_γ also depend on the lateral acceleration and yaw rate measurements. The estimation of the total side forces are calculated [14] based on the estimation of side slip of the EV [15].

4. Verification by the Experiments

Experiments are performed to verify these estimation methods. The EV, which is illustrated in [3], is driven on a planar road at different initial velocities and accelerations. Driver gives different steering inputs to change the driving pattern. The experiments are only performed in a simple road conditions. Before the experiments, the maximum friction coefficient of the road is assumed to be known.

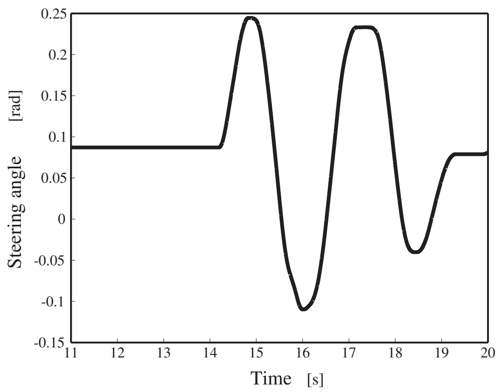

図 5　ドライバーのステアリング角度 (スラローム操舵)
Fig. 5.　Input steering angle by the driver(slalom)

Fig.5 shows the input steering angle of the slalom test case. In this case, the initial velocity is about 5m/s. The longitudinal and lateral accelerations are shown in Fig.6 and Fig.7. The estimated longitudinal vehicle velocity is shown in Fig.8. It can be shown that the estimated result is almost consistent with the measured result. Fig.9 shows the velocities of four wheels. Considering the Fig.8 and Fig.9, the normal calculation method which uses the tested wheel velocities to get the body velocity can not obtain the same accuracy result as what is shown in Fig.8.

The "J-turn" experiments are also performed. The input steering angle is shown in Fig.10. In this case, the initial velocity is about 11m/s. The longitudinal and lateral accelerations are shown in Fig.11 and Fig.12.

The estimated longitudinal and the lateral vehicle velocity is shown in Fig.13 and Fig.14 respectively. It can be shown that the estimated results are consistent with the measured results. Fig.15 shows the velocities of four wheels.

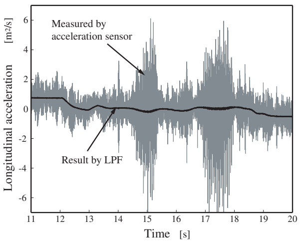

図 6　前後加速度 (slalom)
Fig. 6.　Longitudinal acceleration

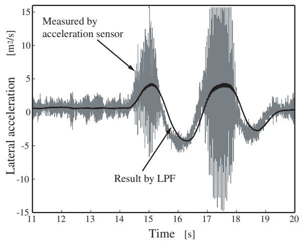

図 7　横加速度 (slalom)
Fig. 7.　Lateral acceleration

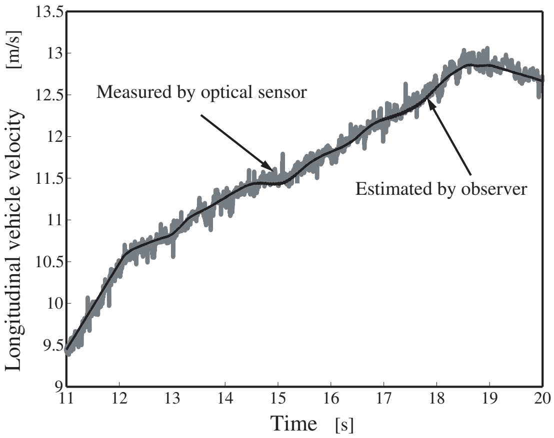

図 8　推定した車体前後速度、と計測した速度
Fig. 8.　Extimated Vs. measured longitudinal velocity(slalom)

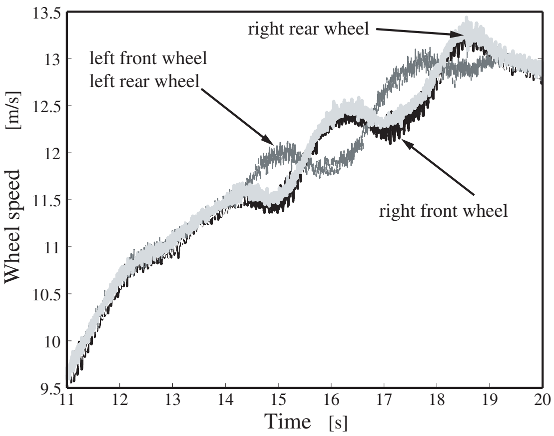

図 9　各輪の速度
Fig. 9.　Wheel velocities(slalom)

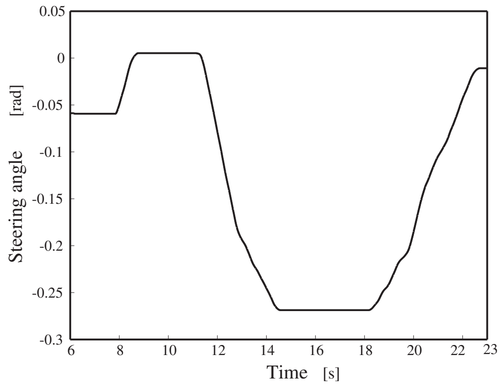

図 10　ドライバーのステアリング角度 (J-turn 操舵)
Fig. 10.　Input steering angles by the driver(J-turn)

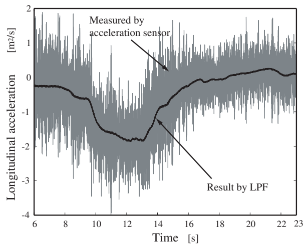

図 11　前後加速度 (J-turn)
Fig. 11.　Longitudinal acceleration

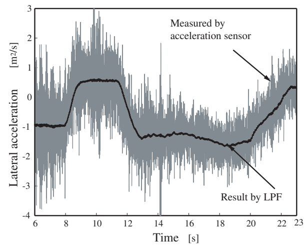

図 12　横加速度 (J-turn)
Fig. 12.　Lateral acceleration

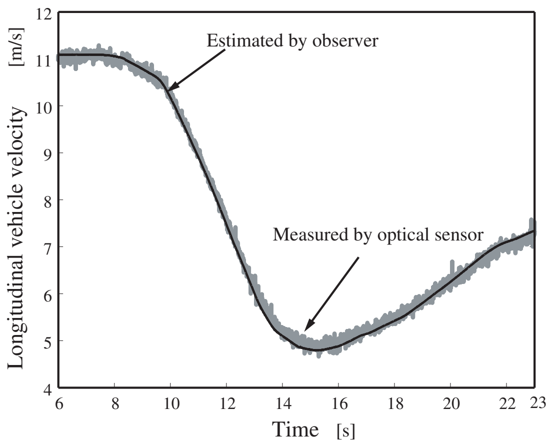

図 13　推定した車体前後速度、と計測した速度
Fig. 13.　Extimated Vs. measured longitudinal velocity(J-turn)

5.　Conclusions

We review main works on the estimation of the vehicle velocity. Based on that, we re-design and propose the nonlinear observer for velocity estimation. It is possible to design a modular observer such that both the estimated states and parameter estimation converge to system state and true parameter values. Observer-based methods for the tire-road friction estimation are very important for the work of the detection of the velocity. The proposed methods for the velocity estimation have been implemented and verified by a four-wheel driven EV.

The tire road friction force calculation algorithm has been modified such that the good estimated values of the road friction are provided. The proposed methods were shown

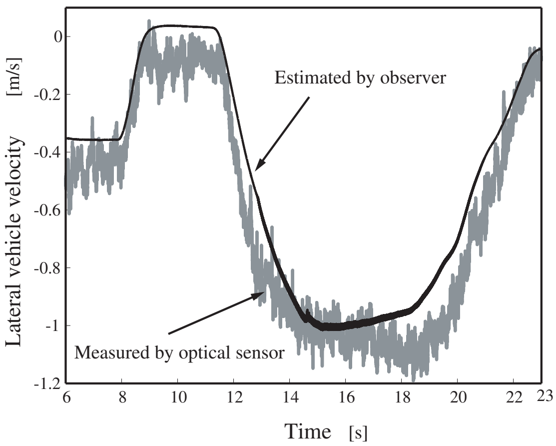

図 14 推定した車体横速度、と計測した速度
Fig. 14. Extimated Vs. measured lateral velocity(J-turn)

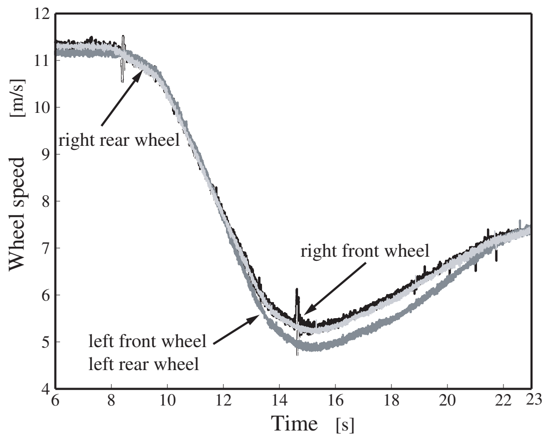

図 15 各輪の速度
Fig. 15. Wheel velocities(J-turn)

through the tests to have good properties in quickly reducing initial estimation error and have been shown to be robust to modeling errors and parameter uncertainties.

However the road friction coefficient is assumed to known in this study. Actually, in the real application, this knowledge is difficult to be known. The estimation of the tire-road friction coefficient will be studied in the future. Further, based on that research, the proposed method should be verified on different road conditions, such as the ice or snowy road.

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