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A DESIGN METHOD FOR CONTROL SYSTEM FOR STEER-BY-WIRE USING FAULT DETECTOR

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Abstract. *A drive-by-wire system has been of much interest as a next-generation method for research and manufacturing in automotive industries due to numerous advantages such as fuel efficiency, flexible designs, preventive maintenance, and better driving comfort. Nonetheless, safety-related issues are still challenging for an effective failure detection method. This paper therefore presents a design method of the steer-by-wire control using fault tolerant control technique for a 4-wheeled vehicle in which the steering can control the left and right rear wheels independently through an in-wheel motor. The proposed fault tolerant control technique comprises feedback fault detectors in order to compensate and estimate a system failure, and offers a rapid self-recovery to stabilize the overall vehicle system. Mathematical models and numeration analysis examples are included. The results show that the system is stabilized even if any single wheel fails, and the transient response error is less than 0.03. The proposed technique offers not only a simple control block diagram design for real application in microcontroller, but also a potentially full stabilization under system failure.*
Keywords: Steer-by-wire, Disturbance observer, Fault tolerant control system

1. **Introduction.** A drive-by-wire system has continuously been of much interest in the next-generation automotive manufacturing industry. Historically, vehicle steering systems have been introduced by various technologies, including (i) pure mechanical system integration, (ii) hydraulic power-assisted system, (iii) electro-hydraulic power-assisted steering, and (iv) electric power-assisted steering, prior to recent advances in drive-by-wire system. In the technical aspects of the drive-by-wire system, the mechanical transmission mechanism is eliminated and driver's intention will be converted to an electric signal, and hence the controller subsequently drives the actuator. As a result of using the steer-by-wire system, explicit advantages are the improvement of fuel efficiency, realization of preventive safety technology, improvement of collision safety, and also the improvement of mounting position and design freedom. In addition, further advantages are vehicle weight reduction, an expansion of the vehicle interior space, and driving comfort feelings [1].

In previous studies, the by-wire system has been considered in various mechanisms such as steer-by-wire, shift-by-wire and brake-by-wire systems [2]. In particular, this paper

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focuses mainly on the steer-by-wire system that uses a wire harness instead of the steering shaft.

Nonetheless, a significant shortcoming of the steer-by-wire system is in the case where the steering operation cannot be performed under failure or malfunction conditions, and therefore safety is required to be ensured [4]. As a solution to this critical issue, an effective control technique for failure compensation should be considered intensively. In particular for automotive application, a vehicle can independently control the driving force of the rear wheels with an in-wheel motor [5].

In this paper, a design method of the steer-by-wire control using fault tolerant control technique will be proposed for a 4-wheeled vehicle in which the steering can control the left and right rear wheels independently through an in-wheel motor. The proposed fault tolerant control technique comprises a fault detector, which is similar to a disturbance observer, in order to estimate a system failure and operate as a rapid self-recovery to stabilize the overall system. Mathematical models and numeration analysis examples are included. This paper is organized as follows. Section 2 describes a steer-by-wire system and problem formulations. Section 3 proposes a new design method for fault tolerant controls. In Section 4, a particular case of numerical examples will illustrate the effectiveness of the proposed method. Concluding remarks will be drawn in Section 5.

2. **Problem Formulation.** Consider a vehicle that has steer-by-wire and can control the left and right rear wheels independently shown in Figure 1. The equivalent three-wheel model to Figure 1 and steering system model considered in this paper are shown in Figure 2 and Figure 3, repectively. The meaning of symbols in Figure 2 and Figure 3 are summarized in Table 1. Assume that we do not consider that transient phenomena in case that the vehicle is suddenly accelerated or decelerated. In addition, we do not consider the case such that suddenly large steering operations occur. From these assumptions, the running speed of the vehicle can be regarded as constant [5].

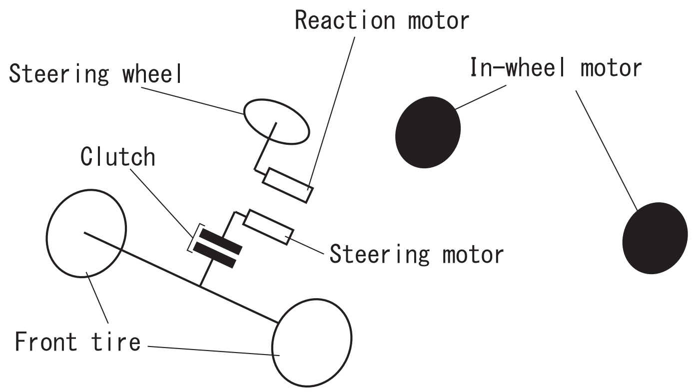

Figure 1. Vehicle structure

Under these assumptions, from Figure 2 and Figure 3, equations of motions are written by

$$m\dot{V} = 2X_f + X_{rr} + X_{rl}, \tag{1}$$

$$mV\left(\dot{\beta} + \gamma\right) = 2Y_f + Y_{rr} + Y_{rl}, \tag{2}$$

$$J\dot{\gamma} = 2Y_f l_f - (Y_{rr} + Y_{rl})l_r + \frac{d_r}{2}X_{diff}(s), \tag{3}$$

and

$$J_s\ddot{\delta} + C_s\dot{\delta} = T(s) - 2\xi Y_f, \tag{4}$$

where

$$Y_f = -K_f\left(\beta + \frac{l_f}{V}\gamma - \delta\right), \tag{5}$$

$$Y_{rr} = Y_{rl} = -K_r \left(\beta - \frac{l_r}{V} \gamma \right), \tag{6}$$

and

$$X_{diff} = X_{rr} - X_{rl}. \tag{7}$$

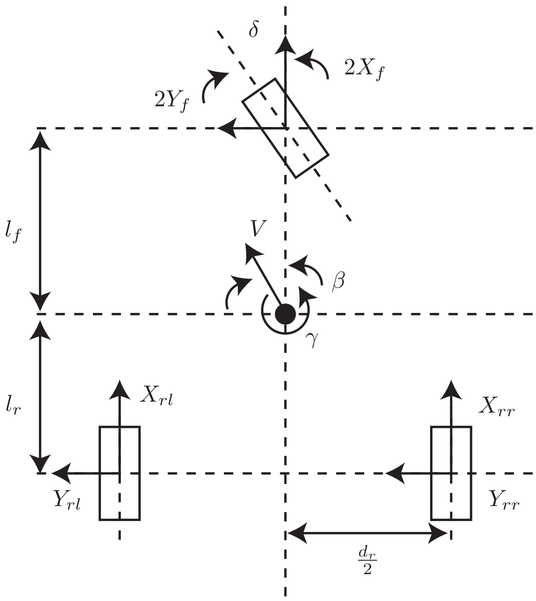

Figure 2. Vehicle model

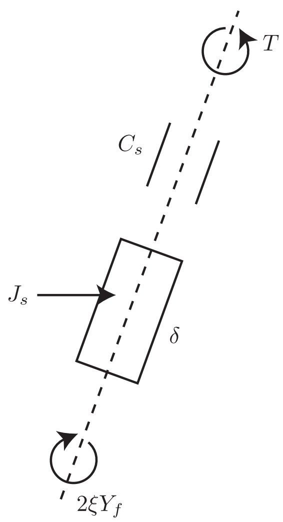

Figure 3. Steer model

The behavior of the vehicle when acceleration is generated is examined. There are two types of acceleration generated by steering operation. One is a lateral acceleration in the case of emergency avoidance. The other is a yaw rate in case that the vehicle turns. When we consider steering response, we need to examine the vehicle behavior of both lateral acceleration and yaw rate. A new physical quantity denoted by $D^*(t)$ that linearly combines the response of yaw rate and lateral acceleration is defined as

$$D^*(t) = \left(\dot{\beta} + \gamma \right) dV + \gamma(1 - d)V, \tag{8}$$

where d $(0 \leq d \leq 1)$ is an association constant [6]. We regard $T(s)$ and $X_{diff}(s)$ as input and $D^*(t)$ as an output of the plant.

Table 1. The meaning of symbols

Symbol	The meaning of symbol	Unit
m	Vehicle mass	kg
V	Vehicle velocity	m/s
X_f	Front tire driving/breaking force	N
X_{rr}	Rear right tire driving/breaking force	N
X_{rl}	Rear left tire driving/breaking force	N
β	Vehicle slip angle	rad
γ	Yaw rate	rad/s
Y_f	Front tire cornering force	N
Y_{rr}	Rear right tire cornering force	N
Y_{rl}	Rear left tire cornering force	N
J	Moment of vehicle inertia	kgm^2
l_f	Distance between front tire and center	m
l_r	Distance between rear tire and center	m
d_r	Rear tread	m
X_{diff}	Rear tire driving force difference	N
δ	Vehicle-wheel steering angle	rad
J_s	Moment of steering inertia	kgm^2
C_s	Damping coefficient of steering	kgm^2/s
T	Steering motor torque	Nm
ξ	Trail	m
K_f	Front tire cornering stiffness	N/rad
K_r	Rear tire cornering stiffness	N/rad

From the assumption that vehicle speed is constant, (1)-(8) are expressed by the state space expression written by

$$\begin{cases} \dot{x}(t) = Ax(t) + Bu(t) \\ y(t) = Cx(t) \end{cases}, \tag{9}$$

where

$$x(t) = \begin{bmatrix} \beta \\ \gamma \\ \dot{\delta} \\ \delta \end{bmatrix}, \tag{10}$$

$$u(t) = \begin{bmatrix} T(t) \\ X_{diff}(t) \end{bmatrix}, \tag{11}$$

$$y(t) = D^*(t), \tag{12}$$

$$A = \begin{bmatrix} -\frac{2(K_f+K_r)}{mV} & -\left\{1+\frac{2(l_fK_f-l_rK_r)}{mV^2}\right\} & 0 & \frac{2K_f}{mV} \\ -\frac{2(l_fK_f-l_rK_r)}{J} & -\frac{2\left(l_f^2K_f+l_r^2K_r\right)}{JV} & 0 & \frac{2l_fK_f}{J} \\ \frac{2\xi K_f}{J_s} & \frac{2\xi K_f l_f}{J_sV} & -\frac{C_s}{J_s} & -\frac{2\xi K_f}{J_s} \\ 0 & 0 & 1 & 0 \end{bmatrix}, \tag{13}$$

$$B = \begin{bmatrix} B_1 & B_2 \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ 0 & \frac{d_r}{2J} \\ \frac{1}{J_s} & 0 \\ 0 & 0 \end{bmatrix}, \tag{14}$$

and

$$C = \begin{bmatrix} -\frac{2d(K_f+K_r)}{m} & -\frac{2d(l_fK_f-l_rK_r)}{mV}+(1-d)V & 0 & \frac{2dK_f}{m} \end{bmatrix}. \tag{15}$$

The transfer function from u to y in (9) is written by

$$y(s) = G(s)u(s) \in R(s), \tag{16}$$

where

$$G(s) = \begin{bmatrix} G_1(s) & G_2(s) \end{bmatrix} \in RH_\infty^{1\times 2}, \tag{17}$$

$$G_1(s) = C(sI-A)^{-1}B_1, \tag{18}$$

$$G_2(s) = C(sI-A)^{-1}B_2, \tag{19}$$

and

$$u(s) = \begin{bmatrix} u_1(s) \\ u_2(s) \end{bmatrix} \in R^2(s). \tag{20}$$

In the case that the handle motor torque breaks, that is the system has a failure, the plant $G(s) = [\ G_1(s) \quad G_2(s)\ ]$ is changed to $G(s) = [\ 0 \quad G_2(s)\ ]$.

The problem considered in this paper is to propose a design method for control system that makes the output $y(s)$ follows the reference input $r(s)$ even when the system is failure or not, where $r(s) \in R(s)$. Therefore, we design a control system that satisfies the following equation

$$\lim_{t\to\infty}\{r(t)-y(t)\} = 0. \tag{21}$$

3. **Controller Design.** In this section, we propose a design method for control system that makes the output $y(s)$ follow the reference input $r(s)$ even when the system is failure or not. In order to solve this problem, the control system in Figure 4 is considered. Here, $C(s) \in R^2(s)$ is a controller to stabilize the control system in Figure 4, $G(s) \in RH_\infty^{1\times 2}$ is the plant such that when the system is normal

$$G(s) = \begin{bmatrix} G_1(s) & G_2(s) \end{bmatrix}, \tag{22}$$

and when the system is failure

$$G(s) = \begin{bmatrix} 0 & G_2(s) \end{bmatrix}. \tag{23}$$

$\hat{d}(s) \in R(s)$ works as a fault detector, $F_1(s) \in RH_\infty^{1\times 2}$ and $F_2(s) \in RH_\infty$ are controllers for fault detector and $\hat{F}(s) \in RH_\infty^{2\times 1}$ is a controller to compensate the influence of the failure.

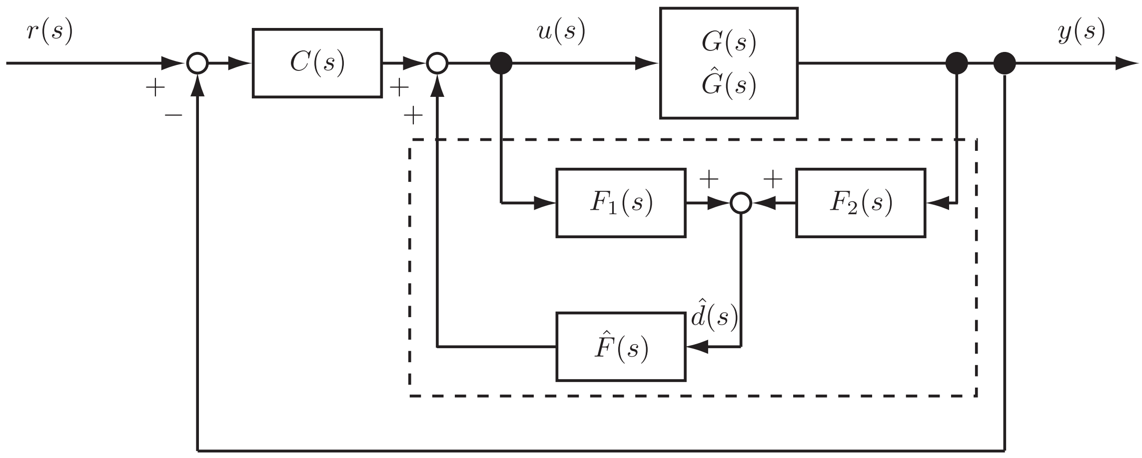

FIGURE 4. Fault tolerant control system

Next we explain the controller in Figure 4. $\hat{d}(s)$ works as a fault detector, that is following expressions hold.

1) When the system is normal, that is $G(s)$ is written by (22),

$$\hat{d}(s) = 0 \tag{24}$$

is satisfied.

2) When the system is failure, that is $G(s)$ is written by (23),

$$\hat{d}(s) = \left(G(s) - \hat{G}(s)\right) u(s) = G_1(s)u_1(s) \tag{25}$$

is satisfied.

From simple manipulations, when $F_1(s)$ and $F_2(s)$ are settled by

$$F_1(s) = \left[\begin{array}{cc} G_1(s) & G_2(s) \end{array}\right], \tag{26}$$

and

$$F_2(s) = -I, \tag{27}$$

then $\hat{d}(s)$ works as a fault detector.

It is necessary to design $\hat{F}(s)$ so that the output difference between normal state and failure state of the system and the output difference when $\hat{F}(s)\hat{d}(s)$ input to $G(s)$ when the system is failure are equal. Therefore, $\hat{F}(s)$ is designed to satisfy the following equation

$$G_2(s)\hat{F}(s)\hat{d}(s) = G_1(s)u_1(s). \tag{28}$$

There is no $\hat{F}(s)$ satisfying (28) for any $u_1(s)$ and any $\hat{d}(s)$. In order to satisfy a condition in (28) for the low-frequence range

$$\hat{F}(s) = \left[\begin{array}{c} 0 \\ \dfrac{1}{G_{2o}(s)}q(s) \end{array}\right], \tag{29}$$

where $q(s)$ is a low pass filter written by

$$q(s) = \frac{1}{(1+\tau s)^{\alpha}}, \tag{30}$$

τ is a small positive number, α is a positive integer to make $q(s)$ in (30) proper and $G_{2o}(s)$ is an outer function of $G_2(s)$ satisfying

$$G_2(s) = G_{2i}(s)G_{2o}(s), \tag{31}$$

and $G_{2i}(s)$ is an inner function satisfying $G_{2i}(0) = 1$. Note that in the low frequency range ω satisfying $q(j\omega) \simeq 1$, (28) is satisfied. Therefore, in order to satisfy $q(j\omega)$ in the wide frequency range, τ in (30) is settled small.

Next, a design method for $C(s)$ in Figure 4 is explained. Since $G(s)$ in (17) is stable and the parameterization of all stabilizing controllers by [7], $C(s)$ written by

$$C(s) = \begin{bmatrix} \dfrac{Q(s)}{1 - Q(s)G_1(s)} \\ 0 \end{bmatrix} \tag{32}$$

stabilizes control system in Figure 4 under the assumption that $Q(s) \in RH_\infty$. In order to make the output $y(s)$ follow the step reference input $r(s)$ without a steady state error, $Q(s)$ is settled by

$$Q(s) = \frac{1}{G_1(s)}\hat{q}(s), \tag{33}$$

where

$$\hat{q}(s) = \frac{1}{(1 + \tau_q)^{\alpha_q}}, \tag{34}$$

τ_q is a positive number and α_q is a positive integer to make $\hat{q}(s)$ in (34) proper.

4. **Numerical Example.** In this section, we show a numerical example to illustrate the effectiveness of the proposed method. Table 2 shows the related parameters for vehicle model [1].

TABLE 2. Parameters for simulation

Symbol	Value	Unit
m	1400	kg
J	2457	kgm^2
l_f	1.02	m
l_r	1.58	m
d_r	1.48	m
J_s	11.98	kgm^2
C_s	9	kgm^2/s
K_f	33700	N/rad
K_r	56200	N/rad
ξ	0.05	m
V	10	m/s
d	0.5	–

From Table 2 and (17), $G_1(s)$ and $G_2(s)$ are given by

$$G_1(s) = \frac{2.0093s^2 + 46.2900s + 413.5481}{s^4 + 27.8684s^3 + 494.8547s^2 + 5.6161 \cdot 10^3 s + 2.0333 \cdot 10^4}, \tag{35}$$

and

$$G_2(s) = \frac{0.0023s^3 + 0.00352s^2 + 0.8719s + 5.8851}{s^4 + 27.8684s^3 + 494.8547s^2 + 5.6161 \cdot 10^3 s + 2.0333 \cdot 10^4}. \tag{36}$$

For the plant $G(s)$ in (17), we design a control system in Figure 4. $F_1(s)$ and $F_2(s)$ in Figure 4 are settled by (26) and (27), respectively. $\hat{F}(s)$ in Figure 4 is designed by (29), where low pass filter $q(s)$ is chosen as

$$q(s) = \frac{1}{0.001s + 1}. \tag{37}$$

Then $\hat{F}(s)$ is written by

$$\hat{F}(s) = \begin{bmatrix} 0 \\ \dfrac{s^4 + 27.8684s^3 + 494.8547s^2 + 5.6161 \cdot 10^3 s + 2.0333 \cdot 10^4}{(0.001s+1)(0.023s^3 + 0.00352s^2 + 0.8719s + 5.8851)} \end{bmatrix}. \tag{38}$$

$C(s)$ in Figure 4 is designed by (32), where $Q(s)$ is given by (33) and

$$\hat{q}(s) = \frac{1}{(0.001s + 1)^2}. \tag{39}$$

Then we have

$$C(s) = \begin{bmatrix} \dfrac{s^4 + 27.8684s^3 + 494.8547s^2 + 5.6161 \cdot 10^3 s + 2.0333 \cdot 10^4}{0.00000201s^4 + 0.04023229s^3 + 0.09299355s^2 + 0.8270962s} \\ 0 \end{bmatrix}. \tag{40}$$

Using above parameters, we show the response of Figure 4. When $r(t) = 1$ and the failure occurs at $t = 10$ [sec], that is after $t = 10$ [sec], $G_1(s) = 0$, the response of the error

$$e(t) = r(t) - y(t) \tag{41}$$

is shown in Figure 5, where the dotted line shows the response of error $e(t)$. Figure 5 shows that

1) when the system is normal, the control system in Figure 4 is stable;
2) when the system is normal, the output $y(t)$ follows the step reference input without steady state error;
3) even if the system is failure, the control system in Figure 4 is stable;
4) when the system is failure, the output $y(t)$ follows the reference input without steady state error.

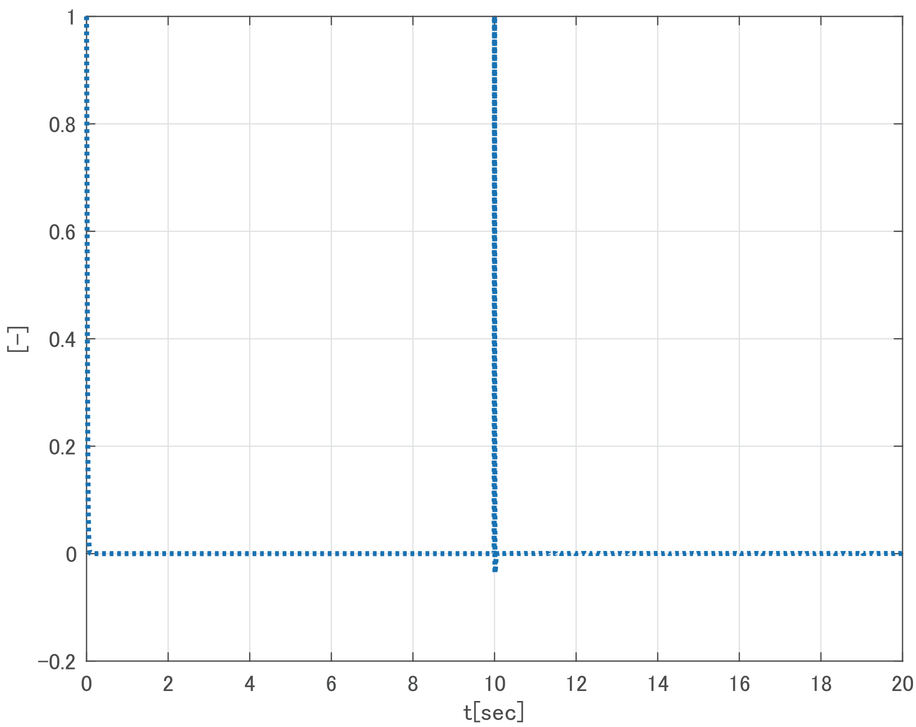

FIGURE 5. Response of the error $e(t) = r(t) - y(t)$

When $r(s) = \sin(5t)$ and the failure occurs at $t = 10$ [s], that is after $t = 10$ [s], $G_1(s) = 0$, the response of the error

$$e(t) = r(t) - y(t) \tag{42}$$

is shown in Figure 6, where the dotted line shows the response of error $e(t)$. Figure 6 shows that

1) when the system is normal, the control system in Figure 4 is stable;
2) when the system is normal, the output $y(t)$ follows the reference input with small steady state error. From the discussion in Section 3, in order to make the steady state error smaller, τ_q in (34) is set smaller;
3) even if the system is failure, the control system in Figure 4 is stable;
4) when the system is failure, the output $y(t)$ follows the reference input with small steady state error. From the discussion in Section 3, in order to make the steady state error smaller, τ_q in (34) is set smaller.

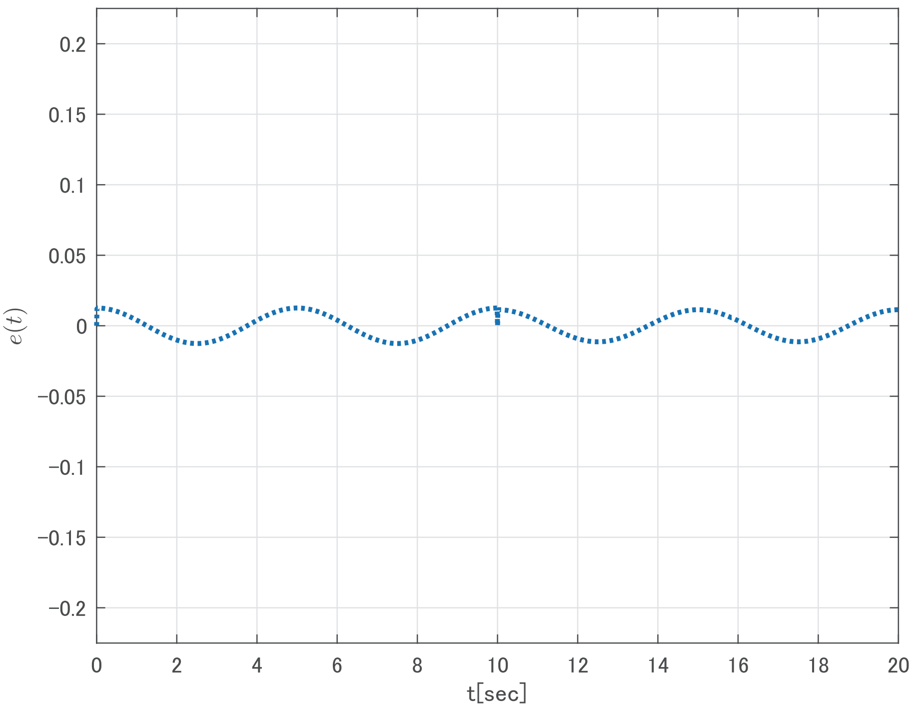

FIGURE 6. Response of the error $e(t) = r(t) - y(t)$

5. **Conclusions.** In this paper, we have proposed a design method for control system for steer-by-wire using the fault tolerant control system. Numerical examples are illustrated to show the effectiveness of the proposed method.

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