A Study on Intelligent Active Roll Angle Controller Design Analysis and Modeling Algorithm

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Abstract

An Intelligent active roll angle controller design algorithm is discussed. The detailed mathematical formulation and analysis are discussed, and then modeling and design method for active roll angle controller are presented. This paper proposes a design method based upon intelligent robust controller design algorithm to control actively roll angle for improving cornering performance problems. The intelligent robust controller is designed for steady speed driving vehicle system model with representation of steering angle and yaw angular velocity parameters for cornering stability. And the detailed formulation and analysis for the objective vehicle system are investigated.

Keywords : roll angle, intelligent control, robust control, yaw angular velocity, steer angle

I. Introduction

This paper proposes analysis, modeling and design algorithm in vehicle system to control roll angle when cornering for driving stability. The equations of motion including vehicle rolling angular motion are formulated. The objective vehicle system is analyzed and optimal design modeling is investigated and then, intelligent robust active controller is designed for cornering stability to control roll angle with yaw angular velocity and steer angle as control input.

II. Dynamic System Formulation

In this paper, I deal with an analysis object system based upon fixed roll axis notion theory in Fig. 1 [1-2]. In Fig. 1, X-Y plane is a driving ground surface plane and X-Y-Z are absolute coordinates. CG is the static center of gravity point of total vehicle, and P is a crossing point of roll axis and vertical line across the CG point. As point P a starting point, x axis is front-to-rear axis parallel with driving ground surface plane, y axis is lateral direction axis at right angles with x axis, and z axis is a vertical directi on axis to x axis with upper-to-lower direction; $x-y-z$ is the relative coordinates on sprung mass part. Also the point P as a standard point, it is considered as relative coordinates $x'-y'-z'$ on unsprung mass part.

The sprung mass is distributed symmetrically to the $x-z$ plane, and S is its center of gravity point on $x-z$ plane. The unsprung mass is distributed on the $x'-y'$ plane, and its center of gravity point is U on x' axis. In this paper, m denotes the total vehicle mass; m_S and m_U denote the sprung mass and the unsprung mass; h_S denotes length of S to x axis; c and e denote the length of the sprung part S and unsprung part U to z axis, respectively. θ and β denote the yaw and side slip angle to the center of gravity of vehicle. r and p denote yaw and roll angular velocity; ϕ denote the roll angle. And throughout in this paper, it is assumed that the vehicle was driven at the steady speed V.

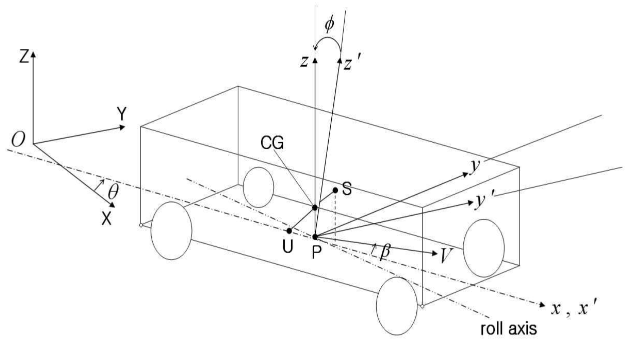

Fig. 1. Objective Vehicle System Coordinates

The analysis model of an objective system is considered as dynamic motions of translation and rotation of its center of gravity and around that point.

In translational motion, the accelerations to the lateral directions of sprung and unsprung mass parts, the acceleration of y direction α_S and the acceleration of y' direction α_U can be defined as

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접수 일자 : 2009. 2. 16 수정 완료 : 2009. 4. 25
게재확정일자 : 2009. 4. 29

$$\alpha_S = \dot{v} + ur - h_S\dot{p} + c\dot{r} \quad (1)$$
$$\alpha_U = \dot{v} + ur - e\dot{r}$$

where u and v denote the velocity elements of x and y (or y') directions, respectively [3]. Assuming that side slip angle $|\beta| \ll 1$ at the point P and steady speed V conditions, from the relations of $u \approx V$ and $v \approx V\beta$, follows can be obtained [4-6].

$$\alpha_S = V\dot{\beta} + Vr - h_S\dot{p} + c\dot{r} \quad (2)$$
$$\alpha_U = V\dot{\beta} + Vr - e\dot{r}$$

The inertia forces of sprung mass part and unsprung mass part, Y_S and Y_U is obtained as follows [2].

$$Y_S = m_S\alpha_S = m_S V(\dot{\beta} + r) - m_S h_S\dot{p} + m_S c\dot{r} \quad (3)$$
$$Y_U = m_U\alpha_U = m_U V(\dot{\beta} + r) - m_U e\dot{r}$$

The inertia force of total vehicle system Y can be obtained as follows.

$$Y = Y_S + Y_U \quad (4)$$
$$= (m_S + m_U)V(\dot{\beta} + r) - m_S h_S\dot{p} + (m_S c - m_U e)\dot{r}$$

Here, $m_S + m_U = m$, and because the CG is the center of gravity of total vehicle system, in translational motion the inertia force can be obtained as

$$Y = mV(\dot{\beta} + r) - m_S h_S\dot{p} \quad (5)$$

where $m_S c - m_U e = 0$.

In rotational motions of sprung mass part, the yawing moment N_S around the z axis to S point and rolling moment L_S around the parallel axis to the x axis can be obtained as

$$N_S = -I_{zxS}\dot{p} + I_{zzS}\dot{r} \quad (6)$$
$$L_S = I_{xxS}\dot{p} - I_{xzS}\dot{r}$$

where $I_{xxS}, I_{xzS}, \cdots$ are moment of inertia and product of inertia around the axis parallel to $x-y-z$ axis across the S point. And the yawing moment N_U around the parallel axis to the z' axis across U point on the unsprung part obtained as follow.

$$N_U = I_{zzU}\dot{r} \quad (7)$$

From above, the yawing moment N around z(or z') axis of total vehicle system and the rolling moment L around x(or x') axis can be obtained as follows

$$N = N_S + N_U + cY_S - eY_U$$
$$= (I_{zzS} + I_{zzU} + m_S c^2 m_U e^2)\dot{r} - (I_{zxS} + m_S h_S c)\dot{p}$$
$$+ (m_S c - m_U e)V(\dot{\beta} + r)$$
$$= I_z\dot{r} - I_{zx}\dot{p}$$

$$L = L_S - h_S Y_S \quad (8)$$
$$= (I_{xxS} + m_S h_S^2)\dot{p} - (I_{xzS} + m_S h_S c)\dot{r} - m_S h_S V(\dot{\beta} + r)$$
$$= I_x\dot{p} - I_{xz}\dot{r} - m_S h_S V(\dot{\beta} + r)$$

because the each inertia forces of Y_U and Y_S should be adopted to U and S points. Here, under the assumption of $|\phi| \ll 1$, I_z is the yaw moment of inertia around the vertical axis across the center of gravity point of total vehicle system, and I_x is the rolling moment of inertia around the x axis of sprung mass part, and follows are also given.

$$I_z = I_{zzS} + I_{zzU} + m_S c^2 + m_U e^2 \quad (9)$$
$$I_{zx} = I_{xz} = I_{xzS} + m_S h_S c$$
$$I_x = I_{xxS} + m_S h_S^2$$

In external forces worked into the total vehicle system, its forces are considered to be lateral forces to the tires. The roll steers α_f and α_r of front and rear wheels can be defined as follows [2].

$$\alpha_f = \frac{\partial\alpha_f}{\partial\phi}\phi,\ \alpha_r = \frac{\partial\alpha_r}{\partial\phi}\phi \quad (10)$$

Because of adding steer angle α_f of front wheel and α_r of rear wheel about the real steer angle δ, the side slip angles of front-rear-wheels are,

$$\beta_f = \beta + \frac{l_f}{V}r - \delta - \alpha_f = \beta + \frac{l_f}{V}r - \delta - \frac{\partial\alpha_f}{\partial\phi}\phi \quad (11)$$
$$\beta_r = \beta + \frac{l_r}{V}r - \alpha_r = \beta + \frac{l_r}{V}r - \frac{\partial\alpha_r}{\partial\phi}\phi$$

and cornering forces worked into front and rear wheels $2Y_f$ and $2Y_r$ are obtained as follows

$$2Y_f = -2K_f\beta_f = 2K_f(\delta + \frac{\partial\alpha_f}{\partial\phi}\phi - \beta - \frac{l_f}{V}r) \quad (12)$$
$$2Y_r = -2K_r\beta_r = 2K_r(\frac{\partial\alpha_r}{\partial\phi}\phi - \beta - \frac{l_r}{V}r)$$

where K_f and K_r are tire cornering powers of each one front and rear wheel. The camber thrusts $2Y_{Cf}$ and $2Y_{Cr}$ of front-rear-wheels can be defined as follows. Here, K_{Cf} and K_{Cr} denote the camber thrust coefficients of front and rear wheels, respectively.

$$2Y_{Cf} = -2K_{Cf}\frac{\partial\phi_f}{\partial\phi}\phi,\ 2Y_{Cr} = -2K_{Cr}\frac{\partial\phi_r}{\partial\phi}\phi \quad (13)$$

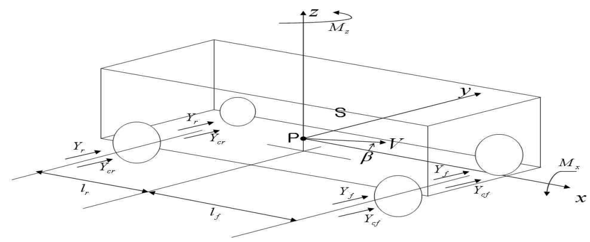

Fig. 2. External Forces into Objective System

In Fig. 2, the total external forces to the lateral direction can be obtained as follows.

$$
\begin{aligned}
F_y &= 2Y_f + 2Y_r + 2Y_{Cf} + 2Y_{Cr} \\
&= 2K_f(\delta + \frac{\partial \alpha_f}{\partial \phi}\phi - \beta - \frac{l_f}{V}r) - 2K_{Cf}\frac{\partial \phi_f}{\partial \phi}\phi \\
&+ 2K_r(\frac{\partial \alpha_r}{\partial \phi}\phi - \beta - \frac{l_r}{V}r) - 2K_{Cr}\frac{\partial \phi_r}{\partial \phi}\phi
\end{aligned} \tag{14}
$$

Also the yawing moment M_z around z axis and yawing moment M_x around x axis about to the total external forces are defined as follows.

$$
\begin{aligned}
M_z &= 2l_f Y_f - 2l_r Y_r + 2l_f Y_{Cf} - 2l_r Y_{Cr} \\
&= 2l_f K_f(\delta + \frac{\partial \alpha_f}{\partial \phi}\phi - \beta - \frac{l_f}{V}r) - 2l_f K_{Cf}\frac{\partial \phi_f}{\partial \phi}\phi \\
&+ 2l_f K_r(\frac{\partial \alpha_r}{\partial \phi}\phi - \beta - \frac{l_r}{V}r) + 2l_r K_{Cr}\frac{\partial \phi_r}{\partial \phi}\phi
\end{aligned} \tag{15}
$$

$$
M_x = (-K_\phi + m_S g h_S)\phi - C_\phi p \tag{16}
$$

In Eq. (16), K_ϕ and C_ϕ denote the total roll stiffness of front-rear suspensions and equivalent viscous friction coefficient of rolling motion, respectively.

III. Controller Design Algorithm

Based on the objective system analysis, the relations of lateral forces, yawing and rolling moments can be considered as follows.

$$
\begin{aligned}
Y - F_y &= 0 \\
N - M_z &= 0 \\
L - M_x &= 0
\end{aligned} \tag{17}
$$

From Eq. (5), Eq. (8) and Eqs. (14)-(16), the equations of the vehicle motion can be obtained as follows.

$$
\begin{aligned}
mV(\frac{d\beta}{dt} + r) - m_S h_S \frac{d^2\phi}{dt^2} &= 2K_f(\delta + \frac{\partial \alpha_f}{\partial \phi}\phi - \beta - \frac{l_f}{V}r) \\
&+ 2K_r(\frac{\partial \alpha_r}{\partial \phi}\phi - \beta - \frac{l_r}{V}r) \\
&- 2(K_{Cf}\frac{\partial \phi_f}{\partial \phi} + K_{Cr}\frac{\partial \phi_r}{\partial \phi})\phi
\end{aligned} \tag{18}
$$

$$
\begin{aligned}
I\frac{dr}{dt} - I_{xz}\frac{d^2\phi}{dt^2} &= 2K_f(\delta + \frac{\partial \alpha_f}{\partial \phi}\phi - \beta - \frac{l_f}{V}r)l_f \\
&- 2K_r(\frac{\partial \alpha_r}{\partial \phi}\phi - \beta - \frac{l_r}{V}r)l_r \\
&- 2(l_f K_{Cf}\frac{\partial \phi_f}{\partial \phi} + l_r K_{Cr}\frac{\partial \phi_r}{\partial \phi})\phi
\end{aligned} \tag{19}
$$

$$
\begin{aligned}
&I_\phi \frac{d^2\phi}{dt^2} - I_{xz}\frac{dr}{dt} - m_S h_S V(\frac{d\beta}{dt} + r) \\
&= (-K_\phi + m_S g h_S)\phi - C_\phi \frac{d\phi}{dt}
\end{aligned} \tag{20}
$$

Those are considered that $I_z = I$ is yawing moment of inertia of total vehicle and $I_{xx} = I_\phi$ is rolling moment of inertia around roll axis of the vehicle where x axis and roll axis can be considered as same location. Finally, the following dynamic equations are derived

$$
\begin{aligned}
&mV\frac{d\beta}{dt} + 2(K_f + K_r)\beta + [mV + \frac{2(l_f K_f - l_r K_r)}{V}]r \\
&\quad - m_S h_S \frac{d^2\phi}{dt^2} - 2Y_\phi \phi = 2K_f \delta
\end{aligned} \tag{21}
$$

$$
\begin{aligned}
&2(l_f K_f - l_r K_r)\beta + I\frac{dr}{dt} + \frac{2(l_f^2 K_f + l_r^2 K_r)}{V}r \\
&\quad - I_{xz}\frac{d^2\phi}{dt^2} - 2N_\phi \phi = 2l_f K_f \delta
\end{aligned} \tag{22}
$$

$$
\begin{aligned}
&-m_S h_S V\frac{d\beta}{dt} - I_{xz}\frac{dr}{dt} - m_S h_S V r + I_\phi \frac{d^2\phi}{dt^2} \\
&\quad + C_\phi \frac{d\phi}{dt} + (K_\phi - m_S g h_S)\phi = 0
\end{aligned} \tag{23}
$$

where

$$
\begin{aligned}
Y_\phi &= (\frac{\partial \alpha_f}{\partial \phi}K_f + \frac{\partial \alpha_r}{\partial \phi}K_r) - (\frac{\partial \phi_f}{\partial \phi}K_{Cf} + \frac{\partial \phi_r}{\partial \phi}K_{Cr}) \\
N_\phi &= (\frac{\partial \alpha_f}{\partial \phi}l_f K_f + \frac{\partial \alpha_r}{\partial \phi}l_r K_r) \\
&- (\frac{\partial \phi_f}{\partial \phi}l_f K_{Cf} + \frac{\partial \phi_r}{\partial \phi}l_r K_{Cr})
\end{aligned} \tag{24}
$$

To design intelligent robust yaw angle control system, Eqs. (21)-(23) are represented by

$$
\begin{aligned}
&m_S h_S \ddot{\phi} + 2Y_\phi \phi = F_1 + \overline{F_1}r - 2K_f \delta \\
&I_{xz}\ddot{\phi} + 2N_\phi \phi = F_2 + \overline{F_2}r - 2l_f K_f \delta \\
&I_\phi \ddot{\phi} + C_\phi \dot{\phi} + (K_\phi - m_S g h_S)\phi = F_3 + H_1 r + H_2 \delta
\end{aligned} \tag{25}
$$

where

$$
\begin{aligned}
&F_1 = mV\dot{\beta} + 2(K_f + K_r)\beta,\ \overline{F_1} = mV + \frac{2(l_f K_f - l_r K_r)}{V} \\
&F_2 = 2(l_f K_f - l_r K_r)\beta + I\dot{r},\ \overline{F_2} = mV + \frac{2(l_f^2 K_f + l_r^2 K_r)}{V} \\
&F_3 = m_S h_S V\dot{\beta} + I_{xz}\dot{r} + m_S h_S V r \quad .
\end{aligned} \tag{26}
$$

Here, H_1 and H_2 are control input coefficients. The matrix representations of Eq. (24) become as follows.

$$
\begin{aligned}
&\frac{d}{dt}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} = \begin{bmatrix}0 & 1 \\ -2(m_S h_S)^{-1}Y_\phi & 0\end{bmatrix}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} + \begin{bmatrix}0 \\ (m_S h_S)^{-1}\end{bmatrix}F_1 \\
&\quad + \begin{bmatrix}0 \\ (m_S h_S)^{-1}\overline{F_1}\end{bmatrix}r + \begin{bmatrix}0 \\ -2(m_S h_S)^{-1}K_f\end{bmatrix}\delta \\
&\frac{d}{dt}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} = \begin{bmatrix}0 & 1 \\ -2I_{xz}^{-1}N_\phi & 0\end{bmatrix}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} + \begin{bmatrix}0 \\ I_{xz}^{-1}\end{bmatrix}F_2 \\
&\quad + \begin{bmatrix}0 \\ I_{xz}^{-1}\overline{F_2}\end{bmatrix}r + \begin{bmatrix}0 \\ -2I_{xz}^{-1}l_f K_f\end{bmatrix}\delta \\
&\frac{d}{dt}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} = \begin{bmatrix}0 & 1 \\ -I_\phi^{-1}(K_\phi - m_S g h_S) & -I_\phi^{-1}C_\phi\end{bmatrix}\begin{bmatrix}\phi \\ \dot{\phi}\end{bmatrix} \\
&\quad + \begin{bmatrix}0 \\ I_\phi^{-1}\end{bmatrix}F_3 + \begin{bmatrix}0 \\ I_\phi^{-1}H_1\end{bmatrix}r + \begin{bmatrix}0 \\ I_\phi^{-1}H_2\end{bmatrix}\delta
\end{aligned} \tag{27}
$$

That is; $\Updownarrow$

$$
\begin{aligned}
\dot{x} &= A_1 x + B_{11}F_1 + B_{12}r + B_{13}\delta \\
\dot{x} &= A_2 x + B_{21}F_2 + B_{22}r + B_{23}\delta \\
\dot{x} &= A_3 x + B_{31}F_3 + B_{32}r + B_{33}\delta
\end{aligned} \tag{28}
$$

In this paper, the control system is designed with the intelligent robust h^{∞} control to improve the cornering stability. The state space equation form of Eq. (26) can be modeled and expressed as

$$\begin{aligned} \dot{\tilde{x}} &= A\tilde{x}+B_1 w+B_2 u \\ z &= C_1\tilde{x}+D_{12}u \\ y &= C_2\tilde{x}+D_{21}w \end{aligned} \tag{29}$$

where $\tilde{x}$ and u denote roll angle and roll angular velocity as the state system variables, and yaw angular velocity and steering angle as control input; y and z denote measured output and controlled output of the control system. w denotes external forces elements as disturbances input. System design variables and matrix parameters become as follows.

$$A=\begin{bmatrix} A_1 & 0 & 0 \\ 0 & A_2 & 0 \\ 0 & 0 & A_3 \end{bmatrix},\ B_1=\begin{bmatrix} B_{11} & 0 & 0 \\ 0 & B_{21} & 0 \\ 0 & 0 & B_{31} \end{bmatrix},$$

$$B_2=\begin{bmatrix} B_{12} & B_{13} \\ B_{22} & B_{23} \\ B_{32} & B_{33} \end{bmatrix},\ \tilde{x}=\begin{bmatrix} x \\ x \\ x \end{bmatrix},\ w=\begin{bmatrix} F_1 \\ F_2 \\ F_3 \end{bmatrix},\ u=\begin{bmatrix} r \\ \delta \end{bmatrix}$$

Intelligent robust h^{∞} control problem is to find a controller such that the closed-loop system is internally stable and the following h^{∞} norm condition, $\| T_{zw}(s) \|_{\infty} < \gamma$ is satisfied. $T_{zw}(s)$ is the transfer function from the disturbance input w to the controlled output z in the closed-loop system, γ is a prescribed positive number [7-8]. In order to design robust h^{∞} controller for controlled objective system, it is assumed that the following two Riccati equations about to γ and positive-definite matrices V_1, V_2

$$\begin{aligned} &A^TX+XA+\gamma^{-2}XB_1B_1^TX \\ &\quad -XB_2B_2^TX+C_1^TC_1+V_1=0 \end{aligned} \tag{30}$$

$$\begin{aligned} &AY+YA^T+\gamma^{-2}YC_1^TC_1Y \\ &\quad -YC_2^TC_2Y+B_1B_1^T+V_2=0 \end{aligned} \tag{31}$$

have two solutions of positive X, Y and $\gamma^2Y^{-1}>X$. Then there exists controller such that the closed-loop system of objective system is internally stable and the above h^{∞} norm condition is satisfied [9-10]. In this paper, one of the intelligent robust controllers can be defined as follows

$$\begin{aligned} \dot{\hat{x}} &= (A+\gamma^{-2}B_1B_1^TX-B_2B_2^TX \\ &\quad -ZC_2^TC_2-\gamma^{-2}ZV_1)\hat{x}+ZC_2^Ty \\ u &= -B_2^TX\hat{x} \end{aligned} \tag{32}$$

where

$$Z=\gamma^2(\gamma^2Y^{-1}-X)^{-1}$$

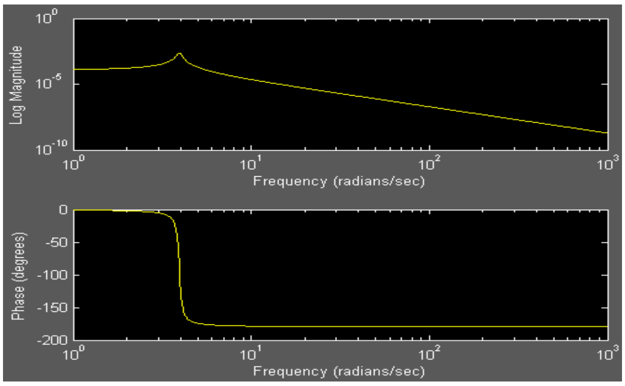

Fig. 3. Bode Frequency Response of Roll Angle ϕ

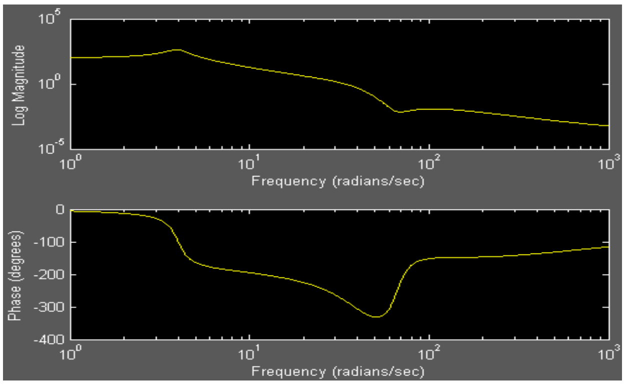

Fig. 4. Bode Response of Controlled plant

The bode frequency responses of design objective plant yaw angle ϕ and h^{∞}controller are shown in Fig. 3-4. In detail numerical simulation specifications, those were set that the vehicle mass m=100 slugs, m_S=90 slugs, m_U=10 slugs, I=2500 slug·ft^2, l=10 ft, l_f=5.5 ft, K_f, K_r=9000, 8900 lb/rad, I_{ϕ}=520 slug·ft^2, h_s=1.2 ft, V=100 ft/s, and $(K_{\phi}-m_Sgh_S)$ = 8100 lb·ft/rad. Also it is considered that $d\beta/dt=dr/dt=d^2\phi/dt^2=d\phi/dt=0$. The result of γ-iteration [11] calculation to solve the robust h^{∞} control problem in this paper is shown as Table 1.

Table 1. Result of γ-iteration

Test bounds: 0.0000 < gamma <= 10.0000

gamma	hamx_eig	xinf_eig	hamy_eig	yinf_eig	nrho_xy	p/f
10.000	1.7e-001	9.7e-013	2.6e-001	0.0e+000	0.0003	p
5.000	1.7e-001	9.7e-013	2.6e-001	0.0e+000	0.0011	p
2.500	1.6e-001	9.7e-013	2.6e-001	0.0e+000	0.0045	p
1.250	1.6e-001	9.7e-013	2.6e-001	0.0e+000	0.0185	p
0.625	1.5e-001	9.7e-013	2.5e-001	0.0e+000	0.0801	p
0.313	5.7e-002	9.7e-013	2.1e-001	0.0e+000	0.5555	p
0.156	1.2e-014#	*******	4.4e-016#	*******	******	f
0.234	2.3e-014#	*******	1.6e-001	0.0e+000	******	f

Gamma value achieved: 0.3125

Ⅳ. Conclusions

In this paper, I dealt with a design method based upon intelligent robust h^{∞} control system design algorithm to control actively roll angle for improving cornering performance problems. The intelligent robust controller was designed for steady speed driving vehicle system model with representation of steering angle and yaw angular velocity parameters for cornering stability. The detail formulation and analysis for the objective vehicle system were investigated and expected small γ value was achieved.

References

[1] L.Segal "Theoretical prediction and experimental substantiation of the response of the automobile to steering control," *Proc. of I. Mech. E. (A.D.)*, 1957.

[2] M.Abe, *Automotive dynamics and control,* TDU-Press, 2008

[3] G. W. Housner and D. Hudson, *Applied mechanics dynamics*, D. Van Nostrand Company, 1959.

[4] E. I. Ono, "A Study on the Integrated Control of Automotive Dynamics", *Journal of Systems, Control and Information*, Vol. 49, No. 6, pp. 205-210, 2005.

[5] *The Vehicle System Dynamics and Control*. JSME, Yokendo, 1999.

[6] A. V. Zanten and R. Erhart, "The Vehicle Dynamics Control System of Bosch", *SAE PT,* vol. 57, pp.497~514, 1996.

[7] J. H. Park, "A study on adopting intelligent control system in active suspension equipment," *Journal of The Korea Society of Computer and Information*, vol. 12, No. 3, pp. 287 - 293, 2007.

[8] J. H. Park and W. S. Ahn, "h^{∞} Yaw Moment Control with Blake for Improving Driving Performance and Stability" *Proc. of IEEE/ASME Conference on Advanced Intelligent Mechatronics September 19 23*, Atlanta, USA, 1999.

[9] J. H. Park, "A study on active suspension control system in vehicle bouncing and pitching vibration for improving ride comfort," *Journal of The Korea Society of Computer and Information*, vol. 12, No. 2, pp. 325 - 331, 2007.

[10] J. H. Park, "Combined Optimal Design with Minimum Phase System", *Journal of Control, Automation, and Systems Engineering,* Vol. 10, No 2, pp. 192-196, 2004.

[11] J. Mita, h^{∞} *Control*, Shokodo Press, 1994.

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