International Journal of Control and Automation
Vol.8, No.5 (2015), pp.221-230
http://dx.doi.org/10.14257/ijca.2015.8.5.21

Design and Implementation of Dynamic Attitude Control of Continuous Track System Using Gyro Sensors

Kyoojae Shin[1]* and Uyeol Jeon[2]

Department of Digital Media Eng., Busan University of Foreign Studies, Korea
kyoojae@bufs.ac.kr[1], jeon4600@gmail.com[2]

Abstract

The continuous track system with the hull disturbances has 6 degrees of freedom (dof) motion in the pitching, yawing and rolling directions of two independent axes. The controller in such a system must satisfy the requirements in disturbance rejection ratio, position accuracy, speed and acceleration magnitude. The paper presents PID controller with disturbance rejection function, low sensitivity and notch filter against the bending frequency by the disturbances. The dynamic analysis of mechanical load and servo-valve flow has been performed by considering the kinetic, potential and dissipation energies. The control scheme has been certified by the MATLAB simulation using practical disturbances. The performance of a designed system has been certified by the simulation and experiment and experiment results, which indicates the improvement of the system performance in case of the existence of external disturbances.

Keywords: *Disturbance Rejection, Hull stabilization control, 6 dof motion, PID with Filter*

1. Introduction

In this paper, the continuous track vehicle with disturbances in the driving condition has the 6 dof motion in the pitching, yawing and rolling directions for two independent axes. The stability control system in such a moving vehicle has to perform disturbance rejection well [1-3, 11-17]. In order to improve the performance of DRR(Disturbance Rejection Ratio) and stabilization, the paper performs the load dynamics by considering a flexible body, unbalance moments, stiction and coulomb frictions, which presents the reduced modeling of the dynamic continuous track system and PID controller with disturbance rejection function, low sensitivity and the improved bandwidth frequency.

The paper presents PID controller with disturbance rejection function, low sensitivity filter and notch filter for the bending frequency rejection. The performance of a designed system has been certified by the simulation and experiment results. The proposed controllers for 2 axes plants have been certified by the simulation and experiment results, which have a good disturbance rejection characteristics and the improvement of the system performances in a moving vehicle. The performance of a designed system has been certified by the simulation and experiment and experiment results.

2. Design of Dynamic Attitude Control System

The paper investigates the 2 axes independent drive planar model of which the azimuth and elevation axes are uncoupled axes. One axis of motion does not affect the other and each axis is independent. The control system is composed of plant, servo valve, controller, sight system, velocity input handle and sensors as shown Figure 1 [11].

ISSN: 2005-4297 IJCA
Copyright ⓒ 2015 SERSC

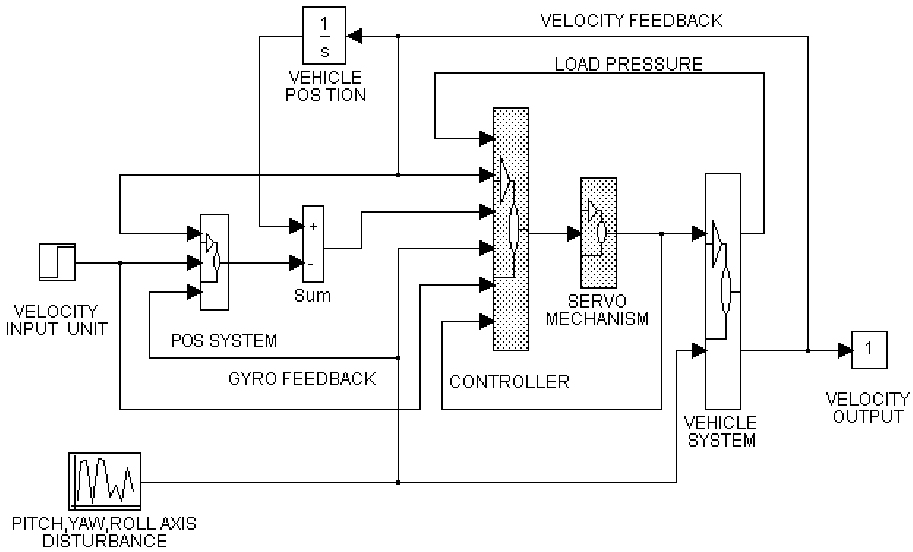


Figure 1. The Configuration of Dynamic Attitude Control System

2.1. Mechanical Load and Servo valve Flow Dynamics

Considering others research results, we had designed the more efficient, safe, and attractive of the hydraulic model [11]. The azimuth load, elevation load and electro-hydraulic servo-valve flow dynamics are derived as follows by considering kinetic, potential and dissipation energy [4].

2.2. Dynamics of Hull

The azimuth load dynamics represent the rotational motion in the lateral axis. The nonlinear elements described in the azimuth load dynamics include coulomb frictions and dead zone. The equation describing the azimuth dynamics is presented in equation. (1).

$$J(h,t,n,b,gr)\theta''+D(h,t,m,b,gr)\theta'+G(g,t,m,b,gr)\theta=U(t,m,um) \qquad (1).$$

2.3. Dynamics of Robot Arm

The elevation load dynamics represent the rotational motion in the pitch axis. The nonlinear elements described in the elevation load dynamics include coulomb frictions. The equation describing the elevation load dynamics is presented in equation (2).

$$J(h,g,um)\theta''+D(h,g,um)\theta'+G(h,g,um)\theta=U(g,p,tc) \qquad (2)$$

In equation (1) and (2), J is a inertia matrix, D presents viscous damping matrix, G is the vector of gravitational torques, and U is the vector of applied torques. The dynamics is also characterized by the unbalance moment, stiction and coulomb frictions

2.4. Servo-valve Flow Dynamics [11]

The second stage servo-valve has been modeled with a second order transfer function as shown in equation (3).

$$\chi_{sv2}=G_1(s)\cdot K_{sv}\cdot V_{in} \qquad (3)$$

 Copyright ⓒ 2015 SERSC

where, $G_1(s) = \dfrac{\omega_{sv2}^2}{s^2 + 2\zeta_{sv2}\omega_{sv2}s + \omega_{sv2}^2}$

The third stage spool has been modeled as integrator with its spool area.

$$\chi_{v3} = \frac{K_{\alpha 2} \cdot \chi_{sv2}}{\alpha_3 s^3 + \alpha_2 s^2 \alpha_1 s + \alpha_0} \tag{4}$$

where, $\alpha_3 = \dfrac{V_{30}M_v}{4B_e A_{v3}}$

$$\alpha_2 = \frac{V_{30}B_v}{4B_e A_{v3}} + \frac{k_{c2}M_v}{A_{v3}}$$

$$\alpha_1 = A_{v3} + \frac{V_{30}K_v}{4B_e A_{v3}} + \frac{K_{c2}B_v}{A_{v3}}$$

$$\alpha_0 = \frac{K_{c2}M_v}{A_{v3}}$$

The linear model of 2-3 stage servo valve is totally 5 order system in equation (3) and (4). The order of the servo-valve is reduced by the following conditions; $\dfrac{V_{30}B_v}{4B_e A_{v3}} << \dfrac{K_{c2}M_v}{A_{v3}}$ in equation (4) and the ignorance of valve stiffness K_{v° equation (4) is reduced to equation (5).

$$\chi_{v3} = G_2(s) \cdot K_{\alpha 2} \cdot \chi_{sv2} \tag{5}$$

Where, $G_2(s) = \dfrac{\omega_{v3}^2}{A_{v3}s(s^2 + 2\zeta_{v3}\omega_{v3} + \omega_{v3}^2)}$

The 3 order function of equation (5) is reduced into the 1 order function of equation (6) due to $\omega_{sv2} << \omega_{v3}$.

$$\chi_{v3} \approx \frac{K_{\alpha 2}}{A_{v3} \cdot s}\chi_{sv2} \tag{6}$$

Consequently 2-3 stage servo valve is shown as the reduced model of eqation (7).

$$\chi_{v3} = G_2(s) \cdot K_{\alpha 2}K_{sv} \cdot V_{in} \tag{7}$$

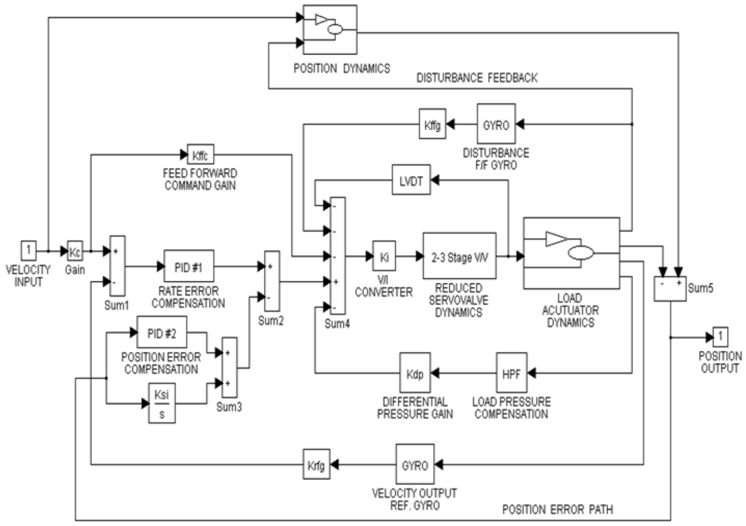


Figure 2. The Proposed Conrol and Stalilization System

3. Controller Design

3.1 Design Theory of Controller

Classical control system design is usually performed using transfer function description; the two most popular techniques are root locus and bode plot design. Closed loop specifications are most often given in terms of steady state error and such desired system step response parameters as rise time, peak time, settling time, peak overshoot, etc.

Figure 2 is the detailed representation of the azimuth and elevation control and stabilization system. The proportional-integral-derivative (PID) controller can take several forms: proportional only, proportional plus derivative (PD), proportional plus integral(PI) and full PID. It allows the designer, however, to satisfy only one closed loop specification; GM, PM, steady state error, etc. The addition of derivative control increases the damping in the closed loop system while integral control increases the system type and decreases steady state error. PID is usually effective in meeting most specifications. It is by far the most widely used controller in the process industry. It is also widely available in various forms.

Because three parameters must be adjusted in the design of PID controllers, root locus and bode design. The Ziegler and Nichols had developed a method for tuning a PID controller, which is based on a simple stability analysis. The proportional gain is then multiplied by 0.6, and the other two gains are calculated as

$$K_p = 0.6k_m \qquad K_d = \frac{k_p \pi}{4\beta\omega_m} \qquad K_i = \frac{K_p \omega_m}{\pi} \tag{8}$$

where K_m is gain at which the proportional system oscillates, and ω_m is the oscillation frequency. Note that this technique does not design to any specifications. Rather, Ziegler and Nichols found that this design procedure provided “good” behavior for process controllers. Years of experience by process control engineers have indicated that it is indeed a good technique.

Either the root locus or the bode plots can be used to determine K_m and ω_m . For example, a root locus is obtained for the given plant transfer function. The gain at which the root locus crosses the $j\omega$ axis is K_m, and the frequency on the $j\omega$ axis defines ω_m. Alternatively, bode plots are plotted for the given plant transfer function. The GM is determined at the frequency ω_{pc}, $K_m = 10^{(GM/20)}$ and $\omega_m = \omega_{pc}$. Be aware that the Bode technique gives approximate answers only.

The Ziegler-Nichols technique does not allow us to design a PID controller to achieve specific closed loop behavior. An analytical technique can be developed to determine the PID parameters given steady state error and performance specifications. The loop gain of a PID controller system is given by [10].

$$(K_p + K_d s + \frac{K_i}{s})G(s) \tag{9}$$

The velocity controller (PID#1) is designed according to the following conditions;

① Magnitude reduction about 20 dB in about 25Hz

② System bandwidth extension by phase lead in freq. >25Hz

③ Minimize Phase delay in freq.>25Hz

④ Improvement of Disturbance rejection characteristics using notch filters

 Copyright ⓒ 2015 SERSC

The position controller (PID#2) is designed according to the following conditions;

① Magnitude reduction about 5 dB in freq = 25Hz

② System bandwidth extension by phase lead in freq. >12Hz

③ Minimize Phase delay in freq.>12Hz

④ Improvement of Disturbance rejection characteristics using notch filters such a position controller

The PID#1 controller is realized using the equation (10) and (11). The cut-off frequency is selected by trade-off method and total controller is shown as equation (12).

$$\text{LPF \#1} = \frac{K_{cl}\omega_{cl}^2}{S^2 + 2\zeta_{cl}\omega_{cl}^2} \tag{10}$$

$$\text{BPF\#1} = \frac{K_{s1}\omega_{cb}^2 s}{s^2 + 2\zeta_{cb}\omega_{cb}s + \omega_{cb}^2} \tag{11}$$

$$\text{PID \#1(s)} = (K_{cp} + BPF\#1 + LPF\#1)K_{mgc} \tag{12}$$

The PID#2 controller is realized using the equation (13), (14) and (15). The cut-off frequency is selected by the above procedure.

$$\text{LPF \#1} = \frac{K_{sl}\omega_{sl}^2}{s^2 + 2\zeta_{sl}\omega_{sl}s + \omega_{sl}^2} \tag{13}$$

$$\text{BPF \#1} = \frac{K_{sb}\omega_{sb}^2 s}{s^2 + 2\zeta_{sb}\omega_{sb}s + \omega_{sb}^2} \tag{14}$$

$$\text{PID \#2(s)} = (K_{sp} + BPF\#2 + LPF\#2)\cdot K_{mgs} + \frac{K_{si}}{s} \tag{15}$$

The PID#1 and #2 controller are realized by the analog circuits of Sallen-Key filters as show in figure 3 and figure 4, which have the characteristics of low sensitivity and non-inverting gain [6].

3.2 Azimuth Axis Controller of Hull Vehicle

The notch filters are set to 22[Hz] and 67[Hz]. The notch filter is used to eliminate the structural resonance associated with gear box and the bending frequency in a moving vehicle. The feed forward command is used to improve the system's ability to overcome coulomb friction. The feed forward command enables the control system to build up torque more quickly. The pressure feedback and 3rd stage feedback help to shape the servo valve response. The pressure feedback compensates the differential load pressure [7-8, 10, 18-19].

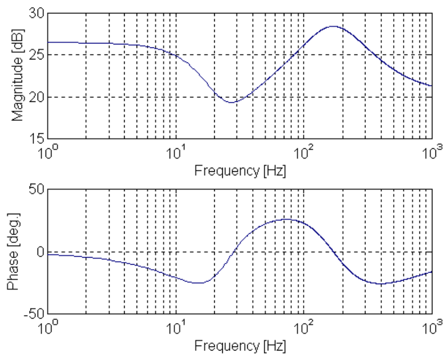


Figure 3. Bode Plots of PID#1

Copyright ⓒ 2015 SERSC

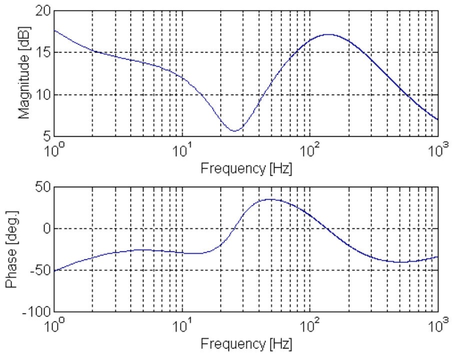


Figure 4. Bode Plots of PID#2

3.3 Disturbance Rejection and Stabilization Capability

The continuous track vehicle with disturbances in the driving condition has the 6 dof motion in the pitching, yawing and rolling directions for two independent axes. In the moving vehicle, the disturbance of pitch, yaw and roll axes which affect the plant, are measured by the DRR (Disturbance Rejection Ratio) analysis of simulation and experiment. In the reference data, the pitch data is variable from -50 to 60 degree/sec, yaw data is variable from -5 to -7 degree/sec and roll data is variable from -30 to 25 degree/sec, which is shown in Figure 5.

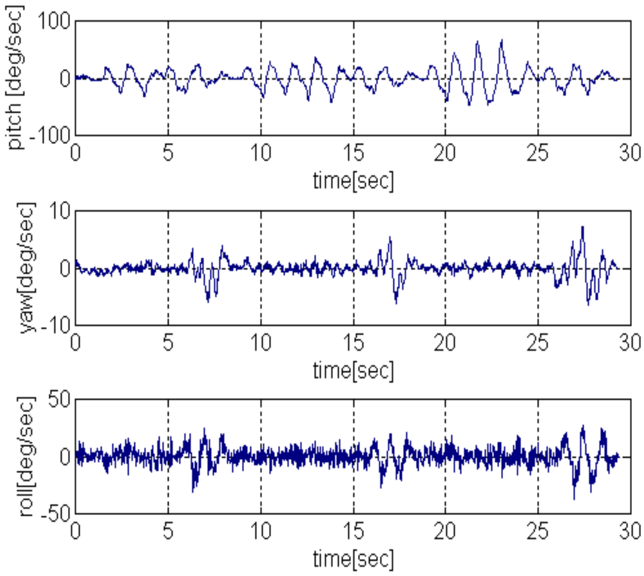


Figure 5. Disturbance Waveforms

4. Simulation and Experiment Result

The above system has appeared in several papers [2, 13-16] and is well known for being difficult to the stability control. It is an example of a system that requires a disturbance rejection. None of the traditional classical techniques apply to this continuous track vehicle, but the PID#1 and PID#2 with disturbance rejection ratio methods can easily handle this plant. The dynamic attitude control scheme has been certified by the simulation using practical disturbances and experiment results which indicates the improvement of the system performance in case of the existence of external disturbances.

The first step is finding an appropriate desired closed loop transfer function. The time response to a 5.0 degree/sec step command input of the azimuth and elevation pants are

 Copyright ⓒ 2015 SERSC

shown in Figure 6 and Figure 7. The RMS position errors are shown in Figure 8 and the RMS velocity errors are shown in figure 9. The above experimental results are summarized in Table 1.

The following specifications are given in the hull driving axis that overshoot = 90.0[%], peak time = 0.4[sec] and settling time = 1.5[sec]. We can determine that the designed closed loop system that overshoot = 88.3[%], peak time = 0.38[sec] and settling time = 1.2[sec]. The following specifications are given in the arm driving axis that overshoot is 85.0[%], peak time = 0.083[sec] and settling time = 0.4[sec]. We can determine that the designed closed loop system that overshoot = 80.2[%], peak time = 0.079[sec], settling time = 0.3[sec] in the no driving condition of vehicle.

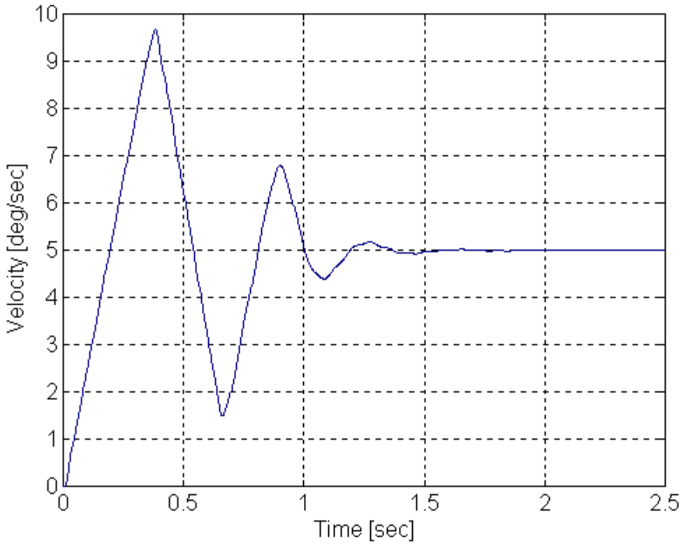


Figure 6. Step Response of Azimuth Axis

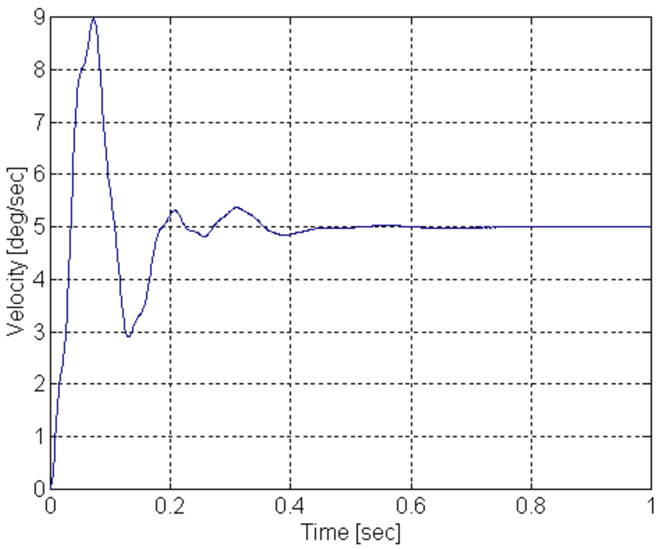


Figure 7. Step Response of Elevation Axis

The following specifications are given in azimuth axis that position error = 0.6[mil rad] in the disturbance condition 13.42[mil rad/sec]. We can determine that the designed closed loop system that position error = 0.34[mil rad], DRR magnitude < Max -20[dB]. Also, the following specifications are given in elevation axis that position error = 0.5[mil rad] in the disturbance condition 132.9[mil rad/sec]. We can determine that the designed closed loop system that position error = 0.28[mil rad], DRR magnitude < Max -25[dB].

The paper presents PID controller with disturbance rejection function, low sensitivity filter and notch filter for the bending frequency rejection. The performance of a designed system has been certified by the simulation and experiment results. The proposed controllers for 2 axes plants have been certified by the simulation and experiment results, which have a good disturbance rejection characteristic and indicate the improvement of the system performances in a moving vehicle.

The dynamic attitude control system in such a moving vehicle has to perform disturbance rejection well. The moving vehicle with disturbances has the 6 dof motion in the pitching, yawing, and rolling directions of two independent axes such as Figure 5. In order to improve the dynamic attitude performance, the paper presents PID controller with disturbance rejection function, low sensitivity filter and notch filter for bending frequency rejection such as the equations (10)~(15), Figure 3 and Figure 4. The performance of a designed dynamic attitude control system has been certified by the simulation and experiment and experiment results for Figure 2. The performance of a designed system has been certified by the simulation and experiment and experiment results of Figure 8 and Figure 9.

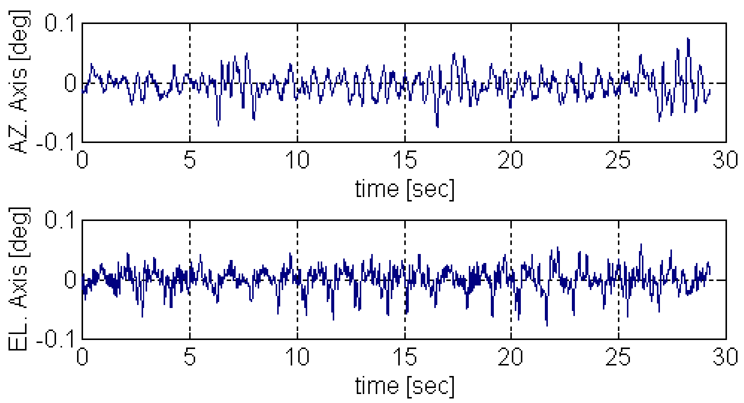


Figure 8. Experiment Results of Attitude Error

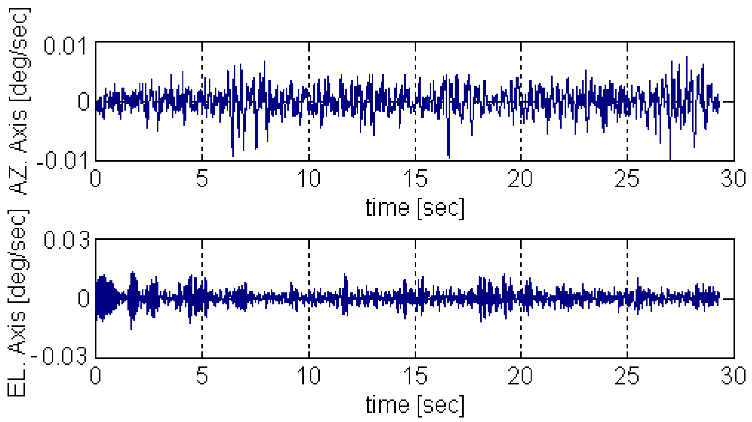


Figure 9. Experiment Results of Velocity Error

5. Conclusions

The paper designs and realizes the controller with disturbance rejection function in a moving vehicle. In order to improve the performance of DRR and stabilization, the paper

 Copyright ⓒ 2015 SERSC

performs the load dynamics by considering a flexible body, unbalance moments, stiction and coulomb frictions, and presents the reduced system modeling and PID controller with disturbance rejection function, low sensitivity and the improved bandwidth frequency . The proposed controllers for 2 axes plants have been certified by the simulation and experiment results, which have a good disturbance rejection characteristic and indicate the improvement of the system performances in a moving vehicle.

Table 1. Experimental Results

Items	Hull		Arm	
	Spec.	Results	Spec.	Results
Overshoot [%]	90.0	88.3	85.0	80.2
Peak Time[sec]	0.4	0.38	0.083	0.079
Setting Time [sec]	1.5	1.2	0.4	0.3
RMS Attitude Error [mil rad]	0.6	0.34	0.5	0.28
DRR Mag.[dB]	Max 15	Max 20	Max 20	Max 25

6. Acknowledgements

This work was supported by the research grant of the Busan University of Foreign Studies in 2015.

7. References

7.1. Journal Article

[1] Y. Hori, "Future Vehicle driven by Electricity and Control-Research on Four Wheel Motored UOT Electric March II", IEEE Transaction on Industrial Electronics, vol. 51, no. 5, **(2004),** pp. 954-962.
[2] E. K. Liebemann, "Safety and Performance Enhancement: The Bosch Electronic Stability Control (ESP)", Robert Bosch GmbH, Germany, Paper Number 05-0471
[3] A. Shaoout, "Real Time System in automotive Applications: Vehicle Stability Control", Electrical Engineering Research, vol. 1, no. 4, **(2013).**
[4] R. M. Krupka, "Mathematical Simulation of the Dynamics of a Tank", SAE Technical Paper, Series Paper No.850416 **(1985).**
[5] D. J. Martin and C.R.Burrows "The Dynamic Characteristics of an Electro-Hydraulic Servo valve, Measurement, and Control", Journal of Dynamic Systems, **(1976),** pp. 395-406.
[6] A. De Pennington, J. J. Mannnetje and R. Bell, "The Modellings of Electro-Hydraulic Control Valves and Its Influence On The design Of Electro-Hydraulic Drives", Journal Mechanical Engineering Science, vol.16, **(1974),** pp. 972-979.
[7] T. R. Welch, "The Use of Derivative Pressure Feedback in High Performance Hydraulic Servomechanism", Journal of Engineering for Industry, **(1962),** pp. 8-14**.**
[8] R. Bell and A de Pennington, "Active Compensation of Lightly Dampe Electro-hydraulic Cylinder Drives Using Derivative Signals", vol. 184, no. 4, **(1969),** pp. 83-98.

7.2. Book

[9] B. Shahian and M. Hassul, "Control System Design Using Matlab", Prentice Hall **(1993).**
[10] H. K. Kim, "Circuit Analysis and Synthesis", KIEE, **(1990),** pp. 219-350.

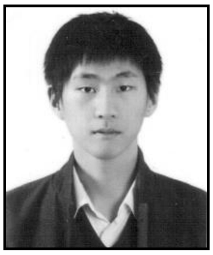
7.3. Conference Proceedings

[11] K. Shin and U. Jeon, "Realization of Hull Stability Control System for Continuous Track Vehicle with the Robot Arm", Advanced Science and Technology Letters Vol.86, Ubiquitous Science and Engineering, pp. 96-101 **(2015**) , April 15, Jeju, Korea.

[12] K. Kawashima, T. Uchida and Y. Hori, "Rolling Stability Control Based on Electronic Stability Program for In-wheel-motor Electric Vehicle, Evs24, **(2009),** Stavanger, Norway.
[13] H. Fujimoto, A. Tsumasaka and Toshihiko Noguchi, "Vehicle Stability Control of Small Electric Vehicle on Snowy Road", JSAE Review of Automotive Engineers, vol. 27, no. 2, **(2006),** pp. 279-286.
[14] S. Satou and H. Fujimoto, "Proposal of Pitching Control for Electric Vehicle with In Wheel Motor", IIC-07- 81 IEE Japan, **(2007),** pp. 65-70.
[15] P. He, Y. Hori, "Improvement of EV Maneuverability and Safety by Dynamic Force Distribution with Disturbance Observer", WEVA-Journal, vol.1, **(2007),** pp. 258-263.
[16] NTRCI, Doug pape, "U31: Vehicle Stability and Dynamics Electronic Stability Control Final Report", DTRT-06-G-0043 **(2011).**
[17] M. Paine, "Vehicle design and Research Pty Limited for Road and Traffic Authority of NSW" **(2005).**
[18] IJAST, Nawal Azoui and Lamir Saidi, "Passivity Based Adaptive Control of Robotic Manipulators Electrically Controlled", **(2011),** pp. 45-54.
[19] IJAST, Salim Aissi, Lamir Saidi, Rachid Adbessemd and Fakhr Eddine Ababsa, "Passivity Based Control of Doubly Fed Induction Machine Using a Fuzzy Controller", **(2011),** pp. 51-64.

Authors

Kyoo-jae Shin
1985~BS degree in Electronics Engineering, Wonkwang University.
1988~MS degree in Electrical Engineering, Cheonbuk National University.
2009 ~PhD degree in Electrical Engineering, Busan National University.
1988.4~1991.7 Technical Professor of Navy Technology School
1991.8~1997.2 Research Manager Engineer, Doosan Ind.
1997.3~2014.8 Professor in Dept. of Electrical Control Engineering, Suncheon First University
2014.9~Present Professor in Dept. of Digital Media Eng, Busan University of Foreign Studies



Uyeol Jeon
2015~ BS degree in Computer Engineering, Hoseo University.
2015~Present MS degree in Dept. of ICT Creative Design, University of Foreign Studies

 Copyright ⓒ 2015 SERSC