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Realization of Hull Stability Control System for Continuous Track Vehicle with the Robot Arm

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Abstract. 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 of two independent axes. The stability control system in such a moving vehicle has to perform disturbance rejection well. . 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, and presents the reduced system modeling 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.

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

1 Introduction

The controlled system in a moving vehicle with disturbances has 6 dof motion in the pitching, yawing and rolling directions of two independent axes[1]~[9]. The controller in such a system should meet the requirements in disturbance rejection ratio, position accuracy, and velocity 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 servovalve flow has been performed by considering the kinetic, potential and dissipation energies. The 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 proposed controllers for 2 axes plants have been certified by the simulation and experiment results, which have a good disturbance rejection characteristics and indicate the improvement of the system performances in a moving vehicle.

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2 Control System of Continuous Track Vehicle

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.

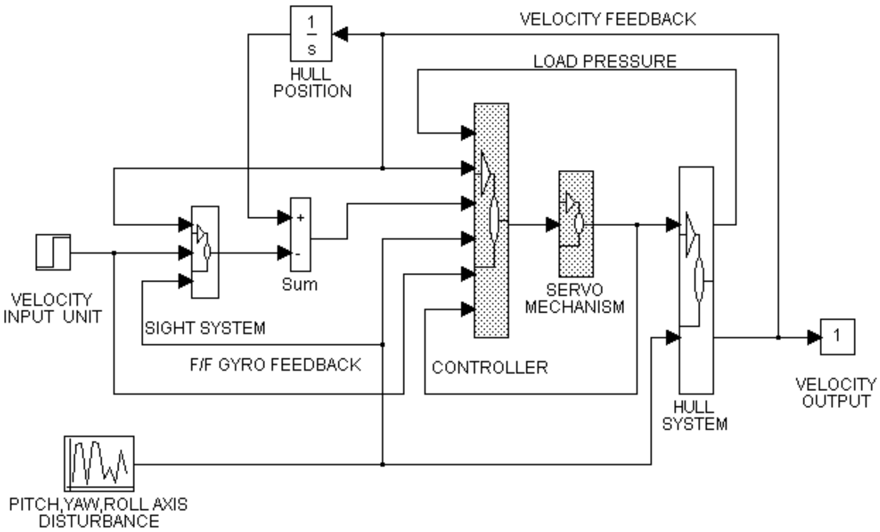

Fig. 1. Contril System Configuration

3 Mechanical Load and Servo-valve Flow Dynamics

The azimuth load, elevation load and electro-hydraulic servo-valve flow dynamics are derived as follows by considering kinetic, potential and dissipation energies[11].

3.1 Azimuth Axis 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 eq. (1).

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

3.2 Elevation Axis 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 eq (2).

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

In eq (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

3.3 Servo-valve Flow Dynamics [10]~[13]

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

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

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.

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

Where $\alpha_3 = \dfrac{V_{30}M_v}{4B_eA_{v3}}$

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

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

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

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

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

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 eq (5) is reduced into the 1 order function of eq (6) due to $\omega_{sv2} \ll \omega_{v3}$.

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

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

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

4 Controller Design

4.1 Elevation Axis Controller of Robot Arm

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].

$$(K_p + K_d s + \frac{K_i}{s})G(s) \tag{8}$$

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

$$\text{LPF \#1} = \frac{K_{cl}\omega_{cl}^2}{S^2 + 2\zeta_{cl}\omega_{cl}^2} \tag{9}$$

$$\text{BPF\#1} = \frac{K_{s1}\omega_{cb}^2 s}{s^2 + 2\zeta_{cb}\omega_{cb}s + \omega_{cb}^2} \tag{10}$$

$$\text{PID \#1(s)} = (K_{cp} + BPF\ \#1 + LPF\ \#1)K_{mgc} \tag{11}$$

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

$$\text{LPF \#1} = \frac{K_{sl}\omega_{sl}^2}{s^2 + 2\zeta_{sl}\omega_{sl}s + \omega_{sl}^2} \tag{12}$$

$$\text{BPF \#1} = \frac{K_{sb}\,\omega_{sb}^{2}\,s}{s^{2} + 2\zeta_{sb}\,\omega_{sb}\,s + \omega_{sb}^{2}} \tag{13}$$

$$\text{PID \#2(s)} = (K_{sp} + BPF\ \#2 + LPF\ \#2)\cdot K_{mgs} + \frac{K_{si}}{s} \tag{14}$$

The PID#1 and #2 controller are realized by the analog circuits of Sallen-Key filters as show in fig. 2, which have the characteristics of low sensitivity and non-inverting gain [13].

4.2 Azimuth Axis Controller of Hull vehicles Simulation and Experiment Result

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 [14][15][16].

5 Conclusion

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 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 characteristics and indicate the improvement of the system performances in a moving vehicle.

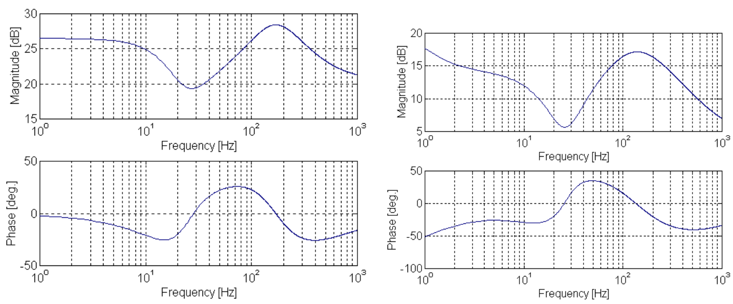

Fig. 2. Bode Plots (a) PID#1, (b) PID#2

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