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Disturbance Rejection and Performance Improvement in a Moving Vehicle

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Abstracts - The moving vehicle with disturbances has the 6 dof motion in the pitching, yawing and rolling directions of two independent axes. The control system in such a moving vehicle has to perform disturbance rejection well. 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.

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. The controller in such a system should meet the requirements in disturbance rejection ratio, position accuracy, 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 controller has been implemented by analog circuits. The proposed 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.

2. 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, servovalve, controller, sight system, velocity input handle and sensors as shown figure 1.

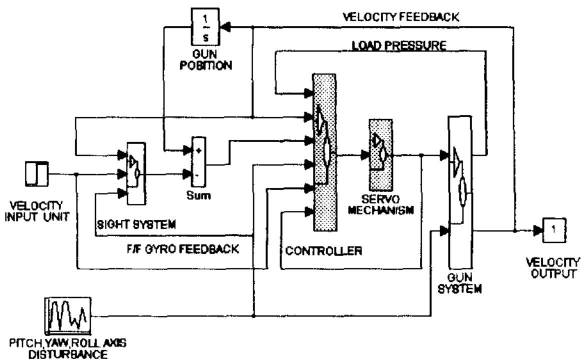

Fig. 1 Control System Configuration

3. Mechnical Load and Servovalve Flow Dynamics

The azimuth load, elevation load and electro-hydraulic sevovalve flow dynamics are derived as follows by considering kinetic, potential and dissipation energies[1].

3.1 Azimuth Load Dynamics

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 deadzone. The equation describing the azimuth dynamics is presented in eq (1).

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

3.2 Elevation Load Dynamics

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** piesents 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 Servovalve Flow Dynamics[2]~[4]

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

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

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

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

$$x_{v3} = \frac{K_{q2} \cdot x_{sv2}}{a_3 s^3 + a_2 s^2 + a_1 s + a_0} \quad (4)$$

$$\text{where, } a_3 = \frac{V_{30}M_v}{4\beta_e A_{v3}}, \qquad a_2 = \frac{V_{30}B_v}{4\beta_e A_{v3}} + \frac{K_{c2}M_v}{A_{v3}}$$

$$a_1 = A_{v3} + \frac{V_{30}K_v}{4\beta_e A_{v3}} + \frac{K_{c2}B_v}{A_{v3}} \qquad a_0 = \frac{K_{c2}K_v}{A_{v3}}$$

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

$$x_{v3} = G_2(s) \cdot K_{q2} \cdot x_{sv2} \quad (5)$$

$$\text{where, } G_2(s) = \frac{\omega_{v3}^2}{A_{v3}s(s^2 + 2\zeta_{v3}\omega_{v3} + \omega_{v3}^2)}$$

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

$$x_{v3} \approx \frac{K_{q2}}{A_{v3} \cdot s} x_{sv2} \quad (6)$$

Consequentlly 2-3 stage servovalve is shown as the reduced model of eq (7).

$$x_{v3} = G_2(s) \cdot K_{q2}K_{sv} \cdot V_{in} \quad (7)$$

4. Controller Design

Fig. 2 is the detailed representation of the azimuth and elevation control and stabilization system.

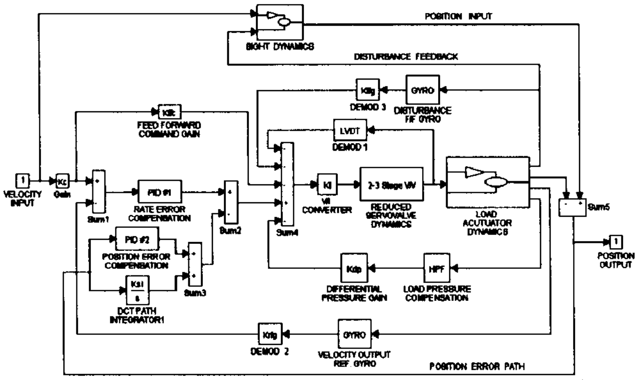

Fig. 2 The Proposed Control and Stabilization System

4.1 Elevation Axis Controller

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

① Magnitude reduction about 20 dB in about 25Hz
② System bandwidth extention by phase lead in freq.>25Hz
③ Minimize Phase delay in freq.<25Hz
④ Improvement of Disturbance rejection characteristics using notch filters

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

① Magnitude reduction about 5 dB in freq = 25Hz
② System bandwidth extention 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 eq (8) and (9). The cut-off frequency is selected by trade-off method and total controller is shown as eq (10).

$$LPF\ \#1 = \frac{K_{cl}\omega_{cl}^2}{s^2 + 2\zeta_{cl}\omega_{cl}\, s + \omega_{cl}^2} \quad (8)$$

$$BPF\ \#1 = \frac{K_{cb}\omega_{cb}^2\, s}{s^2+2\zeta_{cb}\omega_{cb}\, s+\omega_{cb}^2} \tag{9}$$

$$PID\ \#1(s) = (K_{cp}+BPF\ \#1+LPF\ \#1)K_{mgc} \tag{10}$$

The PID#2 controller is realized using the eq (11), (12) and (13). The cut-off frequency is selected by the above procedure.

$$LPF\ \#2 = \frac{K_{sl}\omega_{sl}^2}{s^2+2\zeta_{sl}\omega_{sl}\, s+\omega_{sl}^2} \tag{11}$$

$$BPF\ \#2 = \frac{K_{sb}\omega_{sb}^2\, s}{s^2+2\zeta_{sb}\omega_{sb}\, s+\omega_{sb}^2} \tag{12}$$

$$PID\ \#2(s) = (K_{sp}+BPF\ \#2+LPF\ \#2)\cdot K_{mgs}+\frac{K_{si}}{s} \tag{13}$$

The PID#1 and #2 controllers are realized by the analog circuits of Sallen-Key filters as shown in fig. 3 and fig. 4, which have the characteristics of low sensitivity and noninverting gain[5].

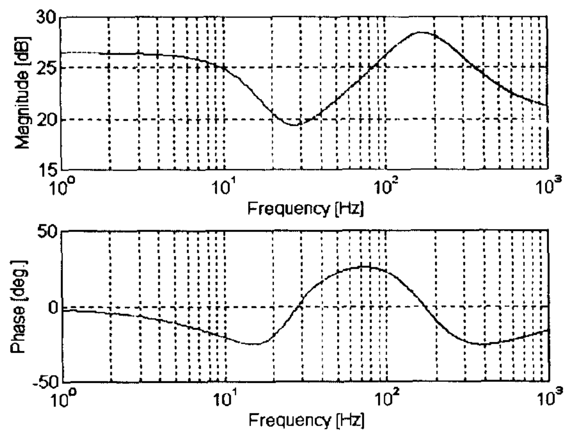

Fig. 3 Bode Plots of PID#1

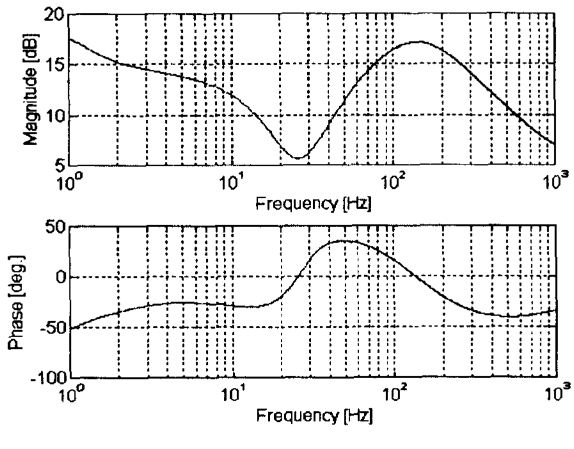

Fig. 4 Bode Plots of PID#2

4.2 Azimuth Axis Controller

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[6][7].

4.3 Disturbance Rejection and Stabilizaion Capability

In the moving vehicle, the disturbance of pitch, yaw and roll axes which affect the plant, are measured by the velocity gyro. The measured disturbance signals is used to the DRR(Disturbance Rejection Ratio) analysis of simulation and experiment. In the referance data, the pitch data varys from -50 to 60 deg/sec, yaw data varys from -5 to 7 deg/sec and roll data varys from -30 to 25 deg/.sec, which is shown in fig. 5.

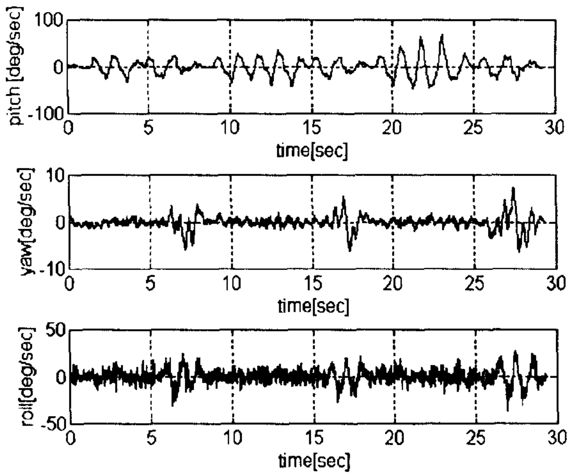

Fig. 5 Disturbance Waveforms

5. Simulation and Experiment Result

The time response to a 5.0 deg/sec step command input of the azimuth and elevation plants are shown in fig. 6 and fig. 7. The RMS position errors are shown in fig.8 and the RMS velocity errors are shown in fig.9. The aboved experimental results is summarized in table 1.

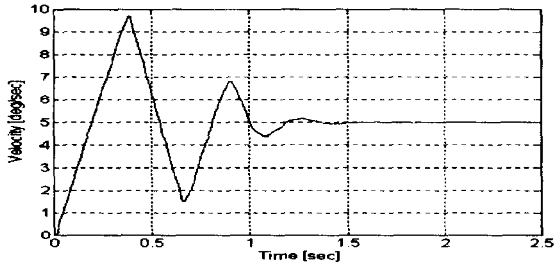

Fig. 6 Step Response of Azimuth Axis

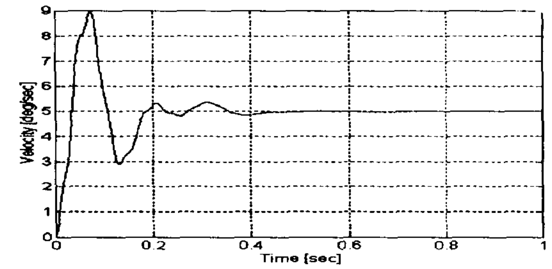

Fig. 7 Step Response of Elevation Axis

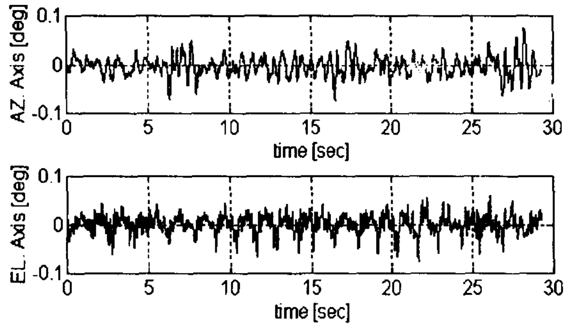

Fig. 8 Experiment Results of Position Error

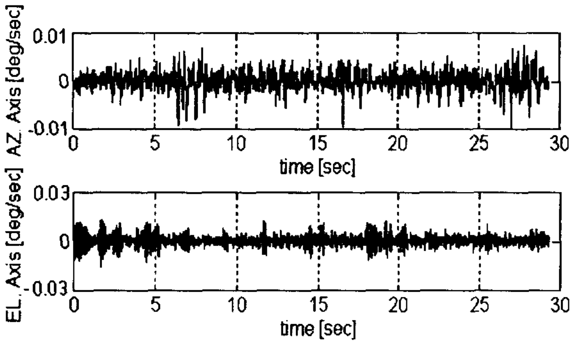

Fig. 9 Experiment Results of Velocity Error

6. 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 stabiliztion, the paper performs the load dynamics by considering a flexible body, unbalance moments, stiction and coulomb frictions, and presents the reduced system modelling 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 disturbanmce rejection charateristics and indicate the improvement of the system performances in a moving vehicle.

Table 1. Experimental Results

Items	Azimuth		Elevation	
	Spec.	Results	Spec.	Results
Overshoot [%]	90.0	88.3	85.0	80.2
Peak Time [sec]	0.4	0.38	0.083	0.079
Settling Time [sec]	1.5	1.2	0.4	0.3
Bandwidth [Hz]	10	14.3	8	10.5
Gain Margin Frequency [Hz]	6.5	7.1	8.0	10.1
Position Error [rms-mrad]	0.6	0.34	0.5	0.28
Input Disturbance Magnitude [mrad/sec]	13.42	13.92	132.9	138.6
DRR Mag.[dB]	Max -15	Max -20	Max -20	Max -25

Referance

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