Active damping in precision equipment using piezo

Bayan Babakhani, Theo de Vries
Control Engineering group, Faculty of EEMCS, University of Twente
Email: `b.babakhani@ewi.utwente.nl, t.j.a.devries@imotec.nl`

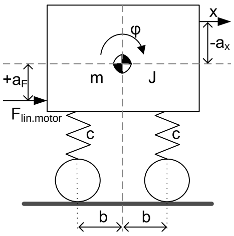
1 Introduction

In this paper, the rotational vibration in the linearly actuated precision machines with low damping is discussed. This so called Rocking mode is e.g. caused by the compliance in the guiding system of a linear actuator and leads to a long settling time of the end-effector. Another problem occurs when a feedback motion controller is applied to the plant. Complex poles present in the loop transfer that are close to the imaginary axis due to low damping, are destabilized by a relatively small gain. A possible solution is actively damping the resonance frequencies. By flattening the resonance peaks, the bandwidth of the system can increase without the danger of instability. In turn, this allows for higher integral gain in the motion control algorithm.

2 Active damping in simulation

Figure 1 shows a 1-Dimensional model of the Rocking mode and the corresponding transfer function. The actuator force F initiates a translational movement and at the same time, a rocking mode around the COM, due to the present compliance, c. This causes a ripple on the measured position, x.

The plant consists of three main parts; the actuator, the guiding system and a moving part. The damping should be applied between the moving part and the guiding system of the actuator, where the compliance causing the rocking mode is located. A platform on which the Active Vibration Control device (AVC) can be mounted, can be created by dividing the moving part into two parts: a lightweight carriage and the rest (containing the end-effector) called the head (see Figure 2). The resonance frequency of the carriage is relatively high due to its light weight and is thus negligible. The plant model now consists of a linear actuator, the carriage (translational mass), AVC and the head (mass and inertia).



$$P(s) = \frac{1}{ms^2} \cdot \frac{s^2+\omega_a^2}{s^2+\omega_r^2} \cdot \frac{\omega_r^2}{\omega_a^2}$$

$$k = 2cb^2$$

$$\omega_a = \sqrt{k/(J+ma_F a_x)}$$

$$\omega_r = \sqrt{k/J}$$

Figure 1: 1D model of a plant with rocking mode

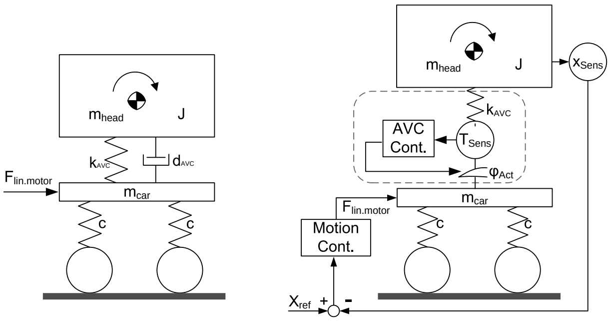


Figure 2: Active damping

AVC loop operates in parallel with the motion control loop. The implemented AVC algorithm is a Leaking Integral Force Feedback, as described by [2]. The transfer function of this intrinsically passive controller is $C_{AVC}(s) = \frac{K_{LIFF}}{s+p_{LIFF}}$, where $d_{AVC} = K_{LIFF}^{-1}$ and $k_{AVC} = p_{LIFF} \cdot d_{AVC}$. For motion control, a PID controller with high frequency roll-off is used, which is tuned based on the moving mass transfer function, according to [1]. Adding this controller to the plant without AVC results in an unstable system. The pole-zero plots (Fig. 3) show that by adding active damping, the closed-loop system remains stable over a wider gain range.

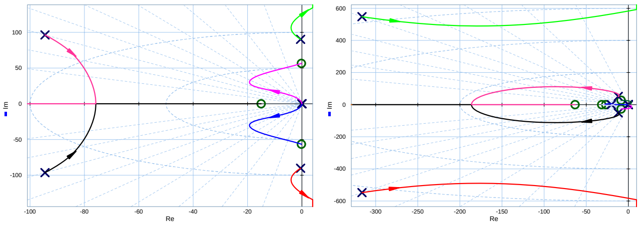


Figure 3: Plant with motion control; left:no AVC, right:with AVC

3 Conclusion

The effect of active damping om a plant with Rocking mode has been investigated in simulation. This results in a stable closed-loop system with high bandwidth, which allows for fast response, low settling-time and low steady-state error.

References

[1] H.J. Coelingh,“Design support for motion control systems”, University of Twente, The Netherlands, 2000

[2] J. Holterman,“Vibration control of high-precision machines with active structural elements”, University of Twente, The Netherlands, 2002