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Real-time and simultaneous control of air mounts for suppressing pneumatic resonance caused by high speed lateral motion

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Abstract
Pneumatic isolators are widely used to block vibrations from the floor in semiconductor and LCD manufacturing machines. However, pneumatic isolators can cause resonance due to low stiffness under heavy weight. The resonance adversely affects the products being manufactured and also requires long time for stabilizing the manufacturing machines. We have previously shown an air mount which was a combination of a magneto-rheological (MR) damper and a conventional isolator. The MR damper was designed for and installed in an empty space of isolators, which comprises an air mount. In this study, the MR damper was controlled using a real-time digital controller which can be coupled to the high speed motion of an XY stage. The motion and the vibration control were achieved simultaneously in real time. The vibration was detected using digital laser sensors under four corners of a motion stage. The laser sensors measured the vertical positions and the controller transformed these into displacement, velocity, acceleration and 3 DOF postures. These values were feed back to the drive power of pneumatic servo values and MR coils. The pneumatic parts maintained reference positions, and the MR dampers removed the disturbances. In the experiments, high speed and lateral motion of XY directions were used for internal vibration source. The major axes of conventional isolators are in the direction of gravity (Z), so isolation systems are critical at these lateral motions (XY). The resonance was induced using pneumatic control from the XY motion and was effectively suppressed by the MR control.

Resonance control, MR fluid damper, Vibration absorption, Pneumatic spring, Hybrid air mount

1. Introduction

Vibration is one of the most critical environmental factors in sub-micro manufacturing processes such as photolithography and inspection. Vibration criteria have been proposed for these processes and BBN is commonly applied to semiconductor manufacturing [1]. According to the BBN, class E is required for manufacturing, however it is difficult to achieve because there are various vibration sources in the manufacturing line. Isolators are installed under these manufacturing machines, which minimizes contact with the floor of the manufacturing line. The isolators can be constructed with power devices such as pneumatics, magnetic levitation, piezo-actuators, the Stewart platform and MR fluid. Air spring is the most common device used for the isolation. The air spring can generate high power to support tons of weight and is simple to handle. However, long oscillations are inevitable because of the much lower stiffness of the air spring as compared with the mass of the machine and the isolator is the only device for dissipating the energy of the internal vibration. Hybrid isolators, proposed recently, use multiple power devices based on pneumatics to mitigate such oscillations [2].

A conventional vibration control system is separated from the motion controller, which causes delay. As stage motion becomes faster, it is necessary to integrate the vibration and the motion to improve response of the vibration control [3]. We propose an integrated controller which shares both the control states and performs the coordinate conversion in real time. Mechanical resonance of isolators is caused by the high speed motion of the stage, and we show that the controller can restrict the resonance.

2. Lateral motion and resonance

Isolators commonly support a heavy weight under a machine using pneumatic pressure and control. The weight is heavy but stiffness and damping are relatively low, which causes long oscillations due to the induced vibration. Therefore, our previous study showed a combined mechanism using an MR fluid to reduce the settling time by increasing damping [2]. The mechanism was designed considering motion in the gravitational direction, Z, and the stiffness and the damping could be varied using electric inputs. However, high speed motion of the machine is along the lateral directions, xy, so, lateral stiffness and the damping of the isolators affects lateral vibration, and they are fixed parameters in most cases. The XYZ vibration of a machine is usually infinitesimal compared with the xyz motion of a moving mass. The moving mass m is also smaller than the mass of the machine. The vibration coordinate can be estimated using the sensor coordinates as follows:

$$(X,Y,Z) = \left(\frac{1}{n}\sum_{i=1}^{n} X_i, \frac{1}{n}\sum_{i=1}^{n} Y_i, \frac{1}{n}\sum_{i=1}^{n} Z_i\right) \quad (1)$$

As shown in the figure 1, the other coordinates of machine vibration are the roll and the pitch which can be written as Z_i, they can be controlled with the pneumatic pressure and the damping forces by varying electric current at actuation points.

$$(\psi,\varphi) \approx \left(\frac{Z_1 - Z_2 - Z_3 + Z_4}{4l_1}, \frac{-Z_1 - Z_2 + Z_3 + Z_4}{4l_2}\right) \quad (2)$$

The measurement and actuator points, l and d, are usually inside the surface plate because of mechanical interference. The equation of the gravitational motion is written as follows, and the equations of XY planar motion have same form.

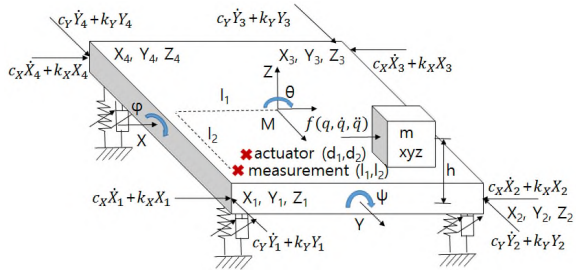

Figure 1. 6 DOF model of typical motion stage

$$M\ddot{Z}+\sum_{i=1}^{n}c_{zi}\dot{Z}_i+\sum_{i=1}^{n}k_{zi}Z_i\approx M\ddot{Z}+nc_z\dot{Z}+nk_zZ=m\ddot{z} \quad (3)$$

The resonance frequency is $\sqrt{nk_z/M}$, and the motion of the stage is the vibration source. Control inputs change C, the damping coefficient of the MR fluid, and K, the stiffness of air spring. The equations of roll and pitch imply that the lateral motion can be controlled using forces in the Z direction. The payload, m, is 5% of the M, so it is neglected.

$$J_{YY}\ddot{\psi}+d_1\sum_{i=1}^{n}sgn_1(i)c_{zi}\dot{Z}_i+d_1\sum_{i=1}^{n}sgn_1(i)k_{zi}Z_i=m\ddot{y}h \quad (4)$$

$$J_{XX}\ddot{\varphi}+d_2\sum_{i=1}^{n}sgn_2(i)c_{zi}\dot{Z}_i+d_2\sum_{i=1}^{n}sgn_2(i)k_{zi}Z_i=m\ddot{x}h \quad (5)$$

The equations can be approximated into simple equations of motion in the same form as equation (3) by replacing the summation in the equation (2). Then the resonance will be induced at $2\sqrt{l_1d_1\,k_z/J_{YY}}$ and $2\sqrt{l_2d_2\,k_z/J_{XX}}$. The shock energy of motion is indirectly absorbed in the Z direction.

3. Experiment and results

The target system with MR-air isolators and a simultaneous controller was constructed in our previous research [3]. The xy planar motion was driven at high speed (400 mm/s) and acceleration (0.8G) in lateral directions of (400 mm, 400 mm) strokes, and vibration control was performed simultaneously [3]. Zψφ vibration was converted from four laser position sensors, which was decoded by the controller in real time.

The resonance of the pneumatic control was induced by the lateral motion. After activating the MR control, the resonance was removed and the settling time was found to be 3.9 sec, as shown in figure 2. Figure 3 shows FFTs of the Zψφ vibration before and after MR control. The theoretical resonance frequencies were approximately 3.2 Hz for ψ and φ. The peak frequencies in the pneumatic control were 3.2 Hz, respectively, but the resonance was removed after applying MR control, as shown in figure 3. Therefore, the proposed control devices are effective in removing unstable cases of the resonance induced by lateral motion.

4. Conclusion

Simultaneous control of motion and vibration was applied to a lateral motion which can induce resonance. The controller interfaced with linear motors, vibration sensors and air mounts as well as performed feedback control. The air mount was composed of pneumatic spring and an MR damper. After applying MR control, the amplitude of the resonance frequency was removed. The proposed control system will be effective when applied to machines which use high speed motion.

Acknowledgement

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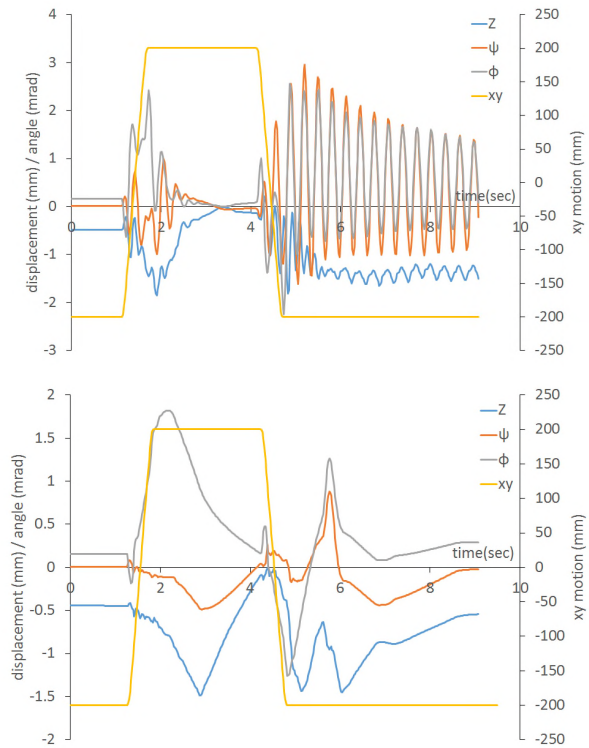

Figure 2. Time domain responses before and after MR control

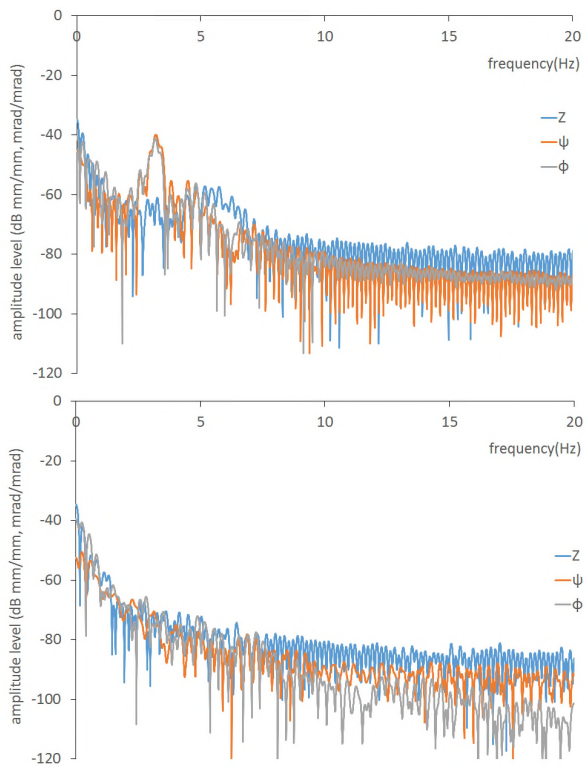

Figure 3. Frequency domain responses before and after MR control (FFT with zero-padding m=32)

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