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An Electro-magnetic Air Spring for Vibration Control in Semiconductor Manufacturing

반도체 생산에서 진동 제어를 위한 전자기 에어 스프링

HyungTae Kim, CheolHo Kim, KangWon Lee, GyuSeop Lee and SungWan Son

김 형 태*·김 철 호†·이 강 원*·이 규 섭**·손 성 완**

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ABSTRACT

One of the typical problems in the precise vibration is resonance characteristics at low frequency disturbance due to a heavy mass. An electro-magnetic(EM) air spring is a kind of vibration control unit and active isolator. The EM air spring in this study aims at removing the low frequency resonance for semiconductor manufacturing. The mechanical and electronic parts in the active isolator are designed to operate under a weight of 2.5 tons. The EM spring is floated using air pressure in a pneumatic elastic chamber and actuated by EM levitation force. The actuator consists of a EM coil and a permanent magnetic plate which are installed inside of the chamber. An air mount was constructed for the experiment with a stone surface plate, 4 active air springs, 4 gap sensors, a DSP controller, and a multi-channel power amp. A PD control method and operating logic was applied to the DSP. Simulation using 1/4 model was carried out and compared with the experiments. The time duration and maximum peak at resonance frequency can be reduced sharply by the proposed system. The results show that the active system can avoid the resonance caused by the natural frequency of the passive system.

요 약

정밀 방진에서 전형적인 문제로 고하중으로 인한 저주파 공진 특성이 있다. 전자기 에어 스프징은 진동 제어 장치이자 능동형 방진 장치이다. 이 연구에서 전자기 에어 스프링은 반도체 생산을 위한 저주파 공진을 제거하는 것을 목적으로 한다. 능동형 방진 장치로 기계 및 전가 부분은 2.5톤의 하중에 작동되도록 설계하였다. 전자기 스프링은 탄성 공압 챔버 내에 공기압을 이용하여 띄우고, 전자기 된 시스템에 의하면 공진 주파수 영역에서 제어 시간 및 최고 피크가 상당히 줄어들었고, 그 결과 피동형 시스템 상의 고유 진동에 의해 발생되는 공진을 피할 수 있음을 보였다.

† 교신저자 ; 정회원, 한국생산기술연구원 생산시스템연구부
E-mail : cheolho@kitech.re.kr
Tel : (041)589-8313, Fax : (041)589-8460

* 정회원, 한국생산기술연구원

** 정회원, 알엠에스테크놀러지(주)

Nomenclature

κ : Adiabatic coefficient of air

ϕ : Magnetic flux

δ : Distance of a gap

μ : Permeability
A : Cross-sectional area
c : Velocity constant
D : Diameter
f_n : Resonance frequency
F : Force
H : Magnetic intensity
I : Current in a coil
k : Spring constant
K : P gain
l : Magnetic path
L : Inductance of a coil
m : Mass of a surface plate
N : No. of turns of a coil
P : Pressure of an air spring
p : Magnetic permeance
v : Volume of an air spring
V : Voltage of a coil
x : Displacement of an air spring

1. Introduction

Major environmental factors in semiconductor manufacturing environment are the temperature, humidity and cleanness, which are strictly controlled. Vibration is also one of the critical factors, but has not been considered seriously. The vibration in semiconductor manufacturing is hard to predict and insulate against. The environmental factors other than vibration can be controlled more easily because of the hermetic structures of fabs. Vibration can be emitted from a machine and transmitted through the floor, but is difficult to block. The vibration in fabs is frequently generated by workers, vehicle carriers and machines. Vibration caused by earthquakes is particularly serious and can have an effect on sensitive processes such as photo lithography. Sometimes, the manufacturing machines are shutdown due to these vibration, so the importance of industrial isolators is growing.

Passive isolators have been widely used in semiconductor manufacturing. A passive isolator is usually composed of air springs and a surface plate. Machines in fabs are mounted on the air springs, a weight of a couple of tones acts on the air spring. The problem with using this type is its slow response and resonance caused by low frequency vibration. Most of the vibration, higher than 100 Hz, is isolated by the mechanism, but vibration under 10 Hz brings about longer oscillations because of the natural frequency of the air spring. This resonance can be found in Kato's[1] and Shin's[2] studies. Therefore, the vibration control in semiconductor manufacturing requires decrease of the resonance effect.

Active control for air mounts has been studied using air control valves. Various formulations has been derived to control the air flow[3~8]. Erin designed an isolator using diaphragms and top-bottom chambers[9]. The problem of air control in fab lines is its the slow response to shock impact. There has been some attempts at vibration isolation using magnetics. Jones proposed the concept of a vibration isolator which functions by magnetic levitation[10]. Hoque developed a 3DOF isolator using magnetics. Pneumatics is not used in the isolator, and the capacity of the load weight is insufficient for industrial machines[11]. Magnetic isolators are found in some cases[12]. A piezo-electronic damper and a magneto-rheological damper were also developed. However, these springs and dampers cannot be applied to semiconductor manufacturing because of the load capacity[13].

Therefore, vibration in semiconductor manufacturing vibration control is related with the resonance problem. In this study, an electro-magnetic air spring was proposed for precise vibration control. An active actuator performs vibration control by electro-magnetic levitation and a real-time DSP. The control logic was based on position control using PD with micro resolution. The model of an EM air spring was constructed

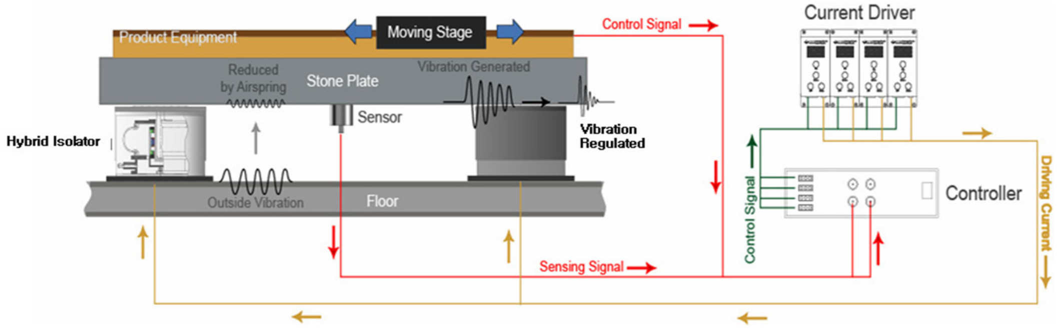

Fig. 1 Active vibration control for semiconductor manufacturing machines

and simulated. The performance of the active control is evaluated to check resonance response.

2. Active Pneumatic Spring

2.1 Concept of air mount

Air mounts for semiconductor manufacturing machines usually have 4 air springs which support a surface plate. These mounts reduce the vibration on the plate and isolate it from the floor. The passive air springs used for industrial loads do not function well in the case of low frequency vibration. The passive types have a slow response and resonance at 2~4 Hz due to their natural frequency. Therefore, active control is required to remove the oscillation caused by the resonance component in a disturbance. Figure 1 shows the concept of the active air mount for machines.

Active air springs are attached to the frame, the surface plate is placed on the springs, and then the machine is installed on the plate. Vibrations are transmitted through the surface of the floor or generated when a stage in the machine is moving. This vibration can be detected using sensors which are attached under the plate. 4 gap sensors are used in this study and detect the relative position from a base point. The controller is composed of a microprocessor and a real-time DSP. The microprocessor communicates with the user through a terminal. The DSP samples the analog signal from the sensors and calculates the analog outputs by control methods. The outputs from the DSP are too low in power to activate the actuator, so current drives are required. These current drives amplify the low power signal enough to regulate the vibration for semiconductor manufacturing. The active air springs are driven by the driving current. The feedback loops are composed of an active air spring, a position sensor, a DSP controller and a current driver. The control input is the initial position and the system output is the current position.

Figure 2 Concept of an electro-magnetic isolator Control algorithms should be derived to regulate the vibration as rapidly as possible. Control logic can be programmed using various methods and downloaded into the DSP. Vibration in a mount can have 6 DOF modes, but the most governing mode is Z axis. So, the air mount in this study controls vibration using Z axes. The performance of the vibration isolator should determined by a standard, as shown in Table 1. The table shows vibration conditions of acceleration, velocity and displacement in semiconductor manufacturing.

The granite surface plate floats on the upper plate, and a bottom plate contacts with floor. A permanent magnet plate is attached under the upper plate, placed inside of the rubber chamber, floated with the upper plate. A EM coil is also installed in the rubber chamber. The magnetic plate should be parallel with the EM coil and vertical to the direction of gravity. Concept of an EM air spring is shown in Fig. 2.

The force-deformation and damping of the rubber chamber by vertical movement is non-linear but vibration in the semiconductor is infinitesimal,

Table 1 Vibration criteria for semiconductor manufacturing

Class	Manufacturing system	Vibration criterita(rms)		
		4~8 Hz		8~80 Hz
		Acc. (gal)	Disp. (μm)	Vel. (μm/s)
A	x400 > microscope, measurement room, optical comparator, probe test, electric manuf. system	0.25	1	50
B	x400 < microscope, ophthalmology, neural operation, aligner, stepper < 1MDRAM	0.13	0.5	25
C	x30,000 > electric microscopy, MRM, semiconductor manuf. aligner, stepper > 1MDRAM	0.06	0.25	12
D	x30,000 < microscope, mass Spectrometer aligner, stepper 4MDRAM	0.03	0.12	6
E	Unsolated laser and optical research system aligner, stepper 64MDRAM	0.016	0.06	3

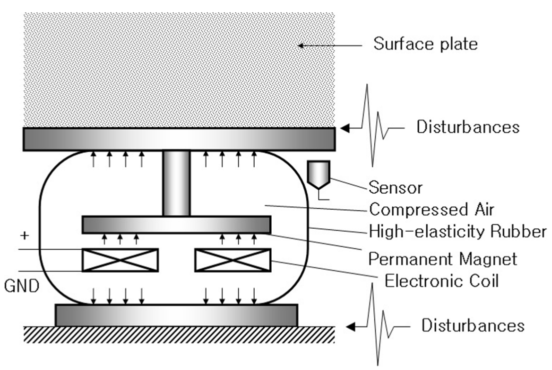

Fig. 2 Concept of an electro-magnetic isolator

so we can assume a linear system model of deformation and damping[14]. A coil and permanent magnet in the chamber generate the electro-magnetic force by driving current. The positive and negative direction of movement can be specified by adjusting the sign of the driving voltage. The permanent magnet is coupled with the upper plate, so the movement of the permanent magnet can be transferred to the surface plate. This air spring can isolate vibration from floor and absorb one on the mount. Some of the vibration is absorbed by the pneumatics and the remainder is removed by the active control unit. However, the power of pneumatics is much larger than that of EM, so EM is used to regulate vibration rapidly and avoid resonance. It is impossible to enlarge the EM force to thousands of newtons because the volume of EM parts is limited. The surface plate is so heavy that the actuators cannot lift up the load, so it is necessary to maintain the original level mechanically.

2.3 System analysis

The load mass of the machine and surface plate can be defined as a mass, m. The passive part of the air spring has a damping factor c and spring constant k. The k value can be varied by a big displacement. The value of the passive air spring shows non-linear and monotone increase in the full stroke range of the vertical movement. The estimated model of the non-linearity should include the air dynamics and rubber elasticity, as Eq. (1).

$$k = 4\pi f_n m - \frac{\kappa A_p^2 P_0}{v_0} \tag{1}$$

However, the movement range is narrow, so that k can be assumed to be a constant[14]. The natural frequency of the system is determined by the mass and the spring constant. The vertical movement of the mass is defined as x and that of the floor as z. The displacement, z, is slight in semiconductor manufacturing and has high frequency components, so it is not in the range of active control. The disturbance force resulting from the mass is defined as D_1 and the disturbance force resulting from the floor as D_2. These disturbances are commonly generated by a machine or operators. The control force F directly acts on the surface plate, as shown in Fig. 3. Therefore, the system model can be built as follows. The initial compression of the spring and air pressure form an equilibrium with the load weight.

$$m\ddot{x} + c\dot{x} + kx = -F + F_{d1} + F_{d2} \tag{2}$$

The driving current for the coil is related to the input voltage, V , inductance, L, and resistance, R.

$$L\frac{dI}{dt} + RI = V \tag{3}$$

The magnetic force between the coil and the core can be calculated by the permeance model as shown in Fig. 4. The dot line in the figure is a path of the magnetic flux in the permeance model. The permeance can be approximated to the shape of the magnetic path in the air gap. Equation (4) shows the relation between current, magnetic force and permeance. The magnetic force, F, can be calculated approximately as written the Eq. (4).

$$f = \frac{1}{2}(NI)^2\frac{dp}{dx} \tag{4}$$

where, N is the number of turns of the coil and p the permeance. Table 2 shows the parameters in the simulation.

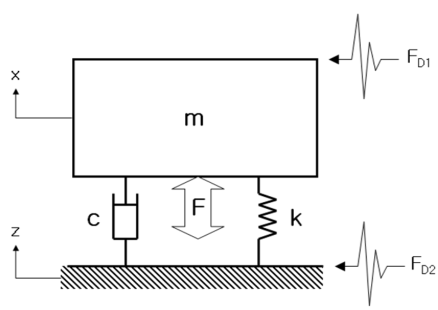

Fig. 3 Model of an active air spring

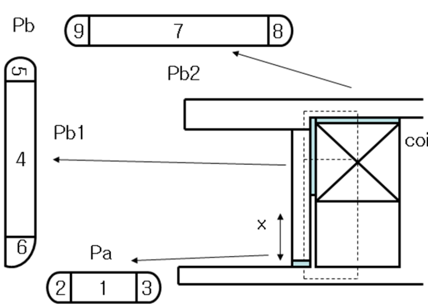

Fig. 4 Permeance model between the coil and the core

2.4 Control method

A feedback loop can be formed by any kind of sensor, including position, velocity and acceleration sensors. Velocity and acceleration sensors are good at detecting vibration, but they have a poor response to low frequency signals. Position sensors indirectly detect vibrations from the displacement of the mass, but good at detecting low frequency signals. Even an extreme-low frequency movement can cause the displacement from an initial position, so the position sensor is better to removing the low frequency vibrations. If a velocity sensor is used, but the final position will

Table 2 Simulation parameters

Variable	Value	Variable	Value
D_1	0.0150 m	L	0.69 H
D_2	0.084 m	R	3.0 Ω
D_3	0.084 m	N	500
D_4	0.070 m	m	625 kg
x_{01}	0.005 m	c	700 kg/s
x_{02}	0.2 m	k	1507 N/m
δ	0. 00025 m	F_{D1}	1000 N
δ_{b1}	0. 00025 m	F_{D2}	0 N
δ_{b2}	0. 00025 m		

be offset from the initial position. Acceleration sensors are sensitive to detect conventional vibrations, but have a poor response to low frequency signals. So, position control is suitable to control vibrations in semiconductor manufacturing.

The control method can be designed to reduce the error between the initial and current positions. The algorithm tries to recover the original point, even though positional errors can occur due to problems other than vibration. The algorithm helps the air mount maintain the correct yaw, roll and pitch angles. The positional error can be defined as shown in the following equation, but calibration is necessary to define the original point afterwards.

$$E(s) = X_r(s) - X(s) \tag{5}$$

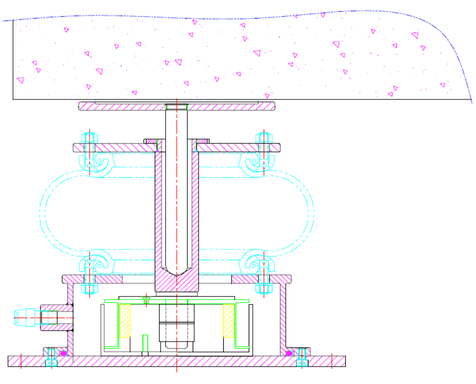

Fig. 5 Cross section of active air spring

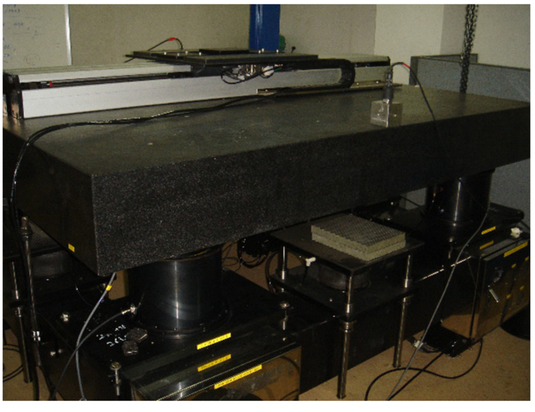

Fig. 6 Air mount for experiment

PD control was applied to increase the sensitivity, as shown in as Eq. (6). The $G(s)$ is the transfer function of the electro-magnetic actuator and air spring. The dynamic model of the surface plate is not considered in this study.

$$\frac{X(s)}{E(s)} = \left[K_p + K_d s\right] G(s) \tag{6}$$

3. Experiment

3.1 Active air mount

The cross section of the active air spring is shown in Fig. 5. The electro-magnetic actuator is placed at the bottom of the air chamber. Compressed air is supplied from the inlet at the side of the actuator and shared with the air chamber. The upper plate blocks the air at the top of the system, and has a cavity in the center. The shaft of the surface plate is inserted in this cavity to support the load weight. The permanent magnet is fixed at the end of the shaft, so, the permanent magnet can float in the air. The magnet makes a relative motion to the coil fixed on the base. The current which drives the actuator is supplied from a high pressure resistant connector. The plate under a surface plate has a pin and inserted to a hole in the shaft.

The space between the pin and hole make it

Table 3 System specification

Contents	Specification
Height	267 mm
Diameter	310 mm
Load weight	2000 kg/4ch
Natural frequency	2.8 Hz
Control force	1000 N
Frequency range	1.0~20.0 Hz
Air pressure	6.0 bar
Stroke	±5.0 mm
Accuracy	3.0 μm
AD bit	12 bit
DA bit	12 bit

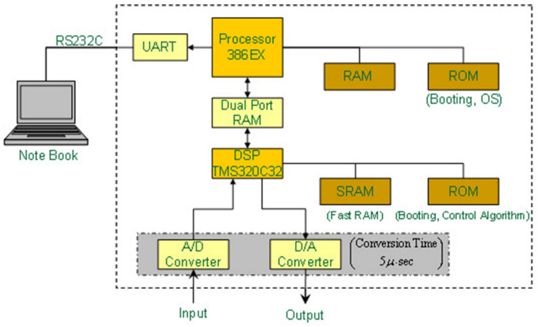

Fig. 7 Layout of a DSP controller

possible to form 6 DOF motion of surface plate. The surface plate is so heavy, it is impossible for the pin to escape out of the hole. The specification of the air spring is shown in Table 3. The air mount was constructed for the experiment, as shown in Figs. 6. 4 air springs were installed under the stone surface plate. 3 levelers were installed beneath the plate to keep it horizontal level of surface plate. The surface plate is so heavy that the actuators cannot lift up the load, so it is necessary to maintain the original level mechanically.

The magnetic force of the actuator is used to attenuate the vibration. A linear motor is installed on the plate to produce a dynamic disturbance in the horizontal direction. A position sensor is attached to the upper side of each air spring. The digital controller based on the TMS320C32 processor has 4 AD ports, 4 DA ports and 8 TTL level inputs. The source code of control program was written using ANSI-C. The source is compiled and downloaded through the RS-232 port. The signals from the sensors are transferred to the AD ports of the controller. The controller determines the outputs by PID logic and then sets the DA ports. Figure 7 shows the layout of the controller. The DA outputs were connected to each channel of the power AMP. The signal in each channel is amplified to 250W max. in proportion to the voltage used for driving the actuator.

3.2 Procedure

Air was supplied to each air spring to float the surface plate. A leveler was installed to adjust pneumatic pressure during the experiment. It controls the flow of air from the compressor when the surface plate is tilted. Then the levelers are fixed and the gap of each sensor is adjusted. The voltage of the IOs of the controller and AMP was checked for calibration. The center of the surface plate was impacted point to generate vibration. The central impact gives same forces to each air springs, so we can expect same motion of 4 air springs. This can be defined as 1/4 model. Full model will be presented in the future. An air spring is made of rubber and has unstable equilibrium under heavy mass. The spring collapses easily, so 3 springs at least should support one heavy mass. The result was monitored with velocity sensors separated from the feedback sensors. The velocity sensors are attached at the center, front, and right of the table on top of the surface. The impact response was measured and analyzed in the non-control and control modes using a B&K 3032A.

4. Results

The experimental results are compared with the simulation done with MATLAB simulink, as shown in Fig. 8. In the experiment, K_p was varied from 0.25 to 1.5 and K_d was 0.5 to 3.0 for watching the effect of the gains and finding optimal gains. Figure 9 shows the response variation of K_p in time domain when K_d is fixed to 1.0. The duration of impact response becomes shorter by applying higher gains. Figure 10 is the result of frequency domain using the same data as Fig. 9. The resonance peak decreased, which means the proposed system absorbs the low frequency components. But when the K_p is too high, the response was not converged. Fig. 11 shows the response variation of K_d in time domain when K_p

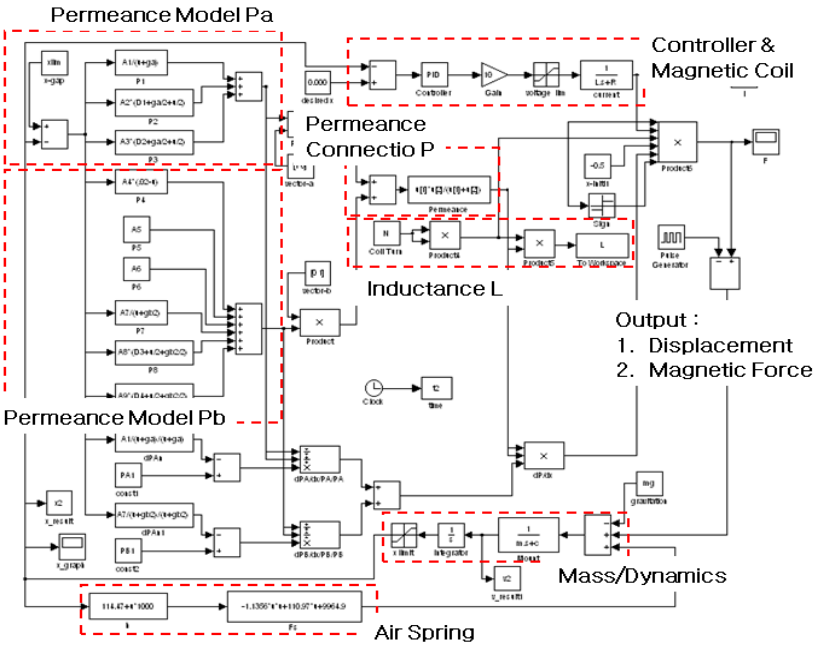

Fig. 8 Simulation diagram

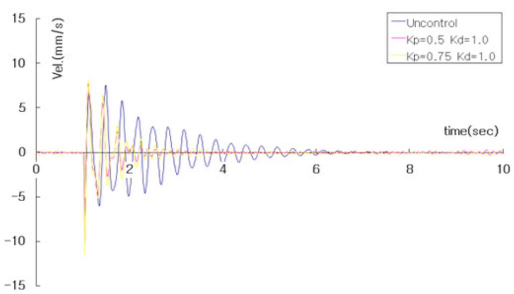

Fig. 9 Effect of K_p in time domain when K_d=1.0

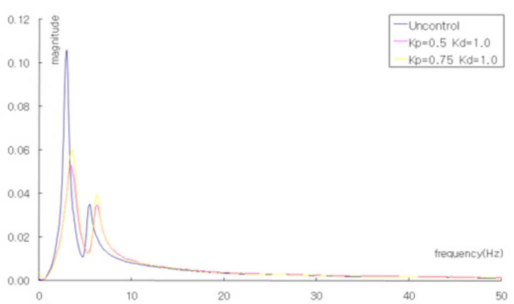

Fig. 10 Effect of K_p in frequency domain when K_d=1.0

is fixed to 0.5. Figure 12 is the result of frequency domain using the same data as Fig. 11. The trend was similar to that of the Fig. 9 and 10, but noise increased by higher K_d and made undesirable vibration near the neutral position.

The system of this study is non-linear, so K_p and K_d cannot be determined by various methods in a linear system. An optimal gain was found by varying K_p and K_d in simulation. The aim of this system is to minimize vibration, so the optimal gain is related with the largest magnitude in frequency domain. The maximum magnitude will decrease after control and the magnitude will be the smallest in the optimal condition. Table 4 shows the variation of the maximum magnitude from the frequency analysis by K_p and K_d in the simulated result. The magnitude decreases by increasing K_d and the minimum by K_p is when K_p=0.5. In the actual case, the system will be unstable due to noise when K_d increase. The optimal value is K_d=1.5 the experiment. The system became unstable when K_p>1.5 and K_d>2.5.

Figure 13 shows the simulated result with K_p =0.5, K_d=1.5 and an impulse force of 10 kN in the time domain. The horizontal axis is the time and

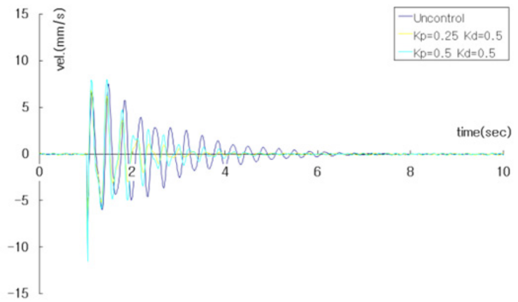

Fig. 11 Effect of K_p in time domain when K_d=0.5

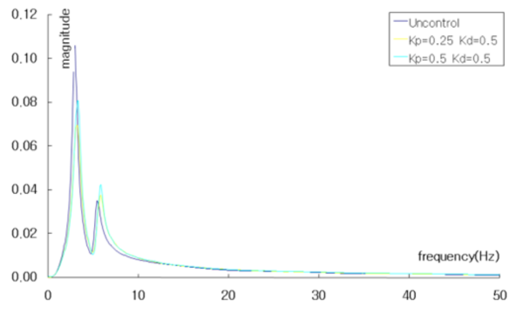

Fig. 12 Effect of K_p in frequency domain when K_d=0.5

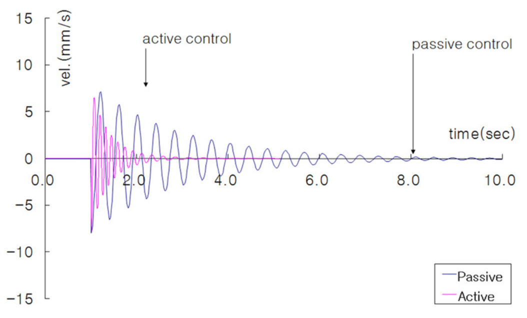

Fig. 13 Simulated results when K_p=0.5 and K_d=1.5

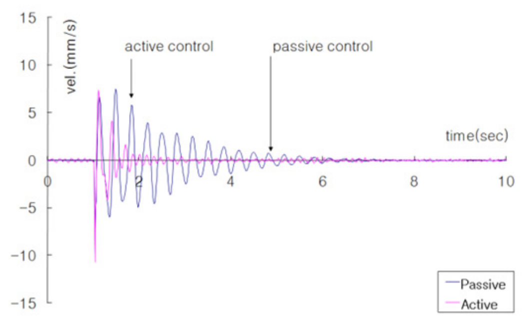

Fig. 14 Experimented results when K_p=0.5 and K_d=1.5

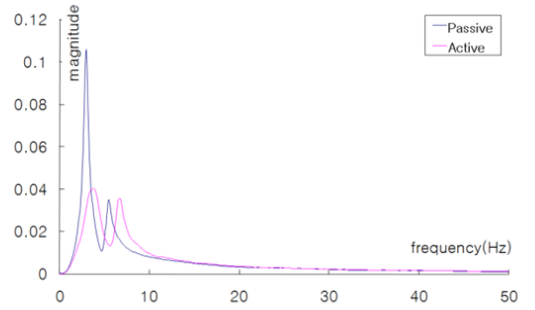

Fig. 15 Frequency analysis of the experimented results

Table 4 Magnitude of maxium peak in the simulation

$K_d \backslash K_p$	0.25	0.5	0.75	1.0	1.5
0.5	2.0088	1.6741	1.8104	1.9820	3.5697
1.0	1.5263	1.5825	1.6446	1.7246	2.1101
1.5	1.1776	1.1740	1.2167	1.2322	1.3447
2.0	0.8238	0.8278	0.8136	0.7965	0.7934
2.5	0.0003	0.5851	0.5962	0.6248	0.6910

the vertical axis is the magnitude of the velocity sensor. The duration of transient phase in the passive system was about 6.92 sec after impact, but that by the active system was only about 1.11 sec. The magnitudes of maximum peak in the both of the systems were almost similar. The experiments were performed under the same conditions and the results are shown as in Fig. 14. The impulse force was not measured, but the force was larger than 22 kN, due to the specification of the impact hammer(PCB 086D20). The impulse response was similar to the simulated result. The duration of transient phase by the passive system was 3.86 sec and that by the active system was 0.88 sec. Both of the experimental responses were faster than that of the simulation.

The short duration in the experiment can be explained by mechanical friction which can absorb the vibration energy. Final response of the graphs shows zero, which means the proposed system follows ground vibration and can satisfy class of semiconductor manufacturing environment[15]. A frequency analysis of the experimental result was conducted, as shown in Fig. 15. However, the duration can be reduced by 77 % by active control. The resonance frequency was 3.0 Hz with the passive control only. The magnitude became smaller at the resonance frequency and the frequency was shifted by 3.9 Hz. The magnitude of resonance frequency decreased into 62 %.

5. Conclusion

An active air spring was designed, simulated, constructed and experimented for semiconductor manufacturing in this study. The active air spring consisted of a passive pneumatic spring, an electro-magnetic actuator, a sensor, a power AMP and a controller. The actuator was attached beneath the air spring in the pneumatic chamber. The vibration was detected by a positional sensor and attenuated by a digital controller. The controller was based on a DSP processor and programmed using PD logic. The system was modeled and simulated using the permeance model. An air mount was constructed by means of 4 active air springs placed under the stone surface plate to support the heavy load. After the simulation, we found that the simulation and experimental results were similar. The duration was shortened by 77 % with the active system. The resonance frequency moved to the higher frequency region and the peak was reduced by 62 %. The active air spring is effective at removing vibration in semiconductor manufacturing.

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