공압능동제어를 이용한 저주파 영역에서의 공압제진대 제진성능 개선에 대한 연구

Performance Enhancement of Pneumatic Vibration Isolation Tables in Low Frequency by Active Control

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Key Words : Pneumatic vibration isolation table, Active control, Air spring, Time delay control

ABSTRACT

As environmental vibration requirements on precision equipment become more stringent, use of pneumatic isolators has become more popular and their performance is subsequently required to be further improved. Dynamic performance of passive pneumatic isolators is related to various design parameters in a complicated manner and, hence, is very limited especially in low frequency range by volume of chambers. In this study, an active control technique, so called time delay control which is considered to be adequate for a low frequency or nonlinear system, is applied to a single chamber pneumatic isolator. The procedure of applying the time delay control law to the pneumatic isolator is presented and its effectiveness in enhancement of transmissibility performance is shown based on simulation and experiment. Comparison between passive and active pneumatic isolators is also presented.

1. Introduction

Precision instruments such as steppers for semiconductor production, electron-beam microscopes and laser systems are highly sensitive to environmental vibrations. As more precision is required, requirement on ground vibration level for such instruments becomes accordingly more stringent as in VC and NIST[1]. Thus, pneumatic isolators are often used to isolate the ground vibration or reduce transmission of the force excitation onto the floor.

As shown in Figure 1, a dual-chamber pneumatic vibration isolator consists typically of a piston, a capillary tube, two chambers and a diaphragm. The piston supports the payload and the diaphragm, a rubber membrane of complicated shape installed for prevention of air leakage, works as an additional spring. The air in the pneumatic chambers is the main stiffness element. The capillary tube works as a damping element [2]. Yet, some problems in the capillary tube, e.g., dynamic amplitude dependency of the diaphragm, haven't been completely solved out[3].

In this paper, a methodology to improve dynamic performance of the pneumatic isolators is proposed, which is to apply so called time delay control law(TDC)[4] to the active air-pressure control. Also, the experimental results which are applied TDC theory are shown and examined compared with passive one. The time delay control has been effectively applied to systems in the presence of nonlinearity, uncertainty and disturbance[4]. Thus, application of TDC to the pneumatic isolators which show dynamic amplitude dependency significantly would be suitable.

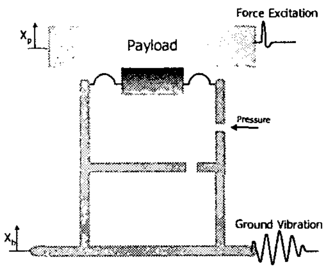

Figure 1. Dual-Chamber Pneumatic Isolator and Payload

2. Design of Time Delay Controller

A mathematical model of the pneumatic isolator derived in [2] disregards dynamics of the diaphragm such as vibration amplitude dependency observed in actual measurements[5]. Time delay control technique is applicable to such a deficient mathematical model

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because it employs an additional reference model. However, time delay control requires all state variables such as displacement, velocity and acceleration. This technique is useful for systems of slow response because it replaces current status with the one before a given amount of, e.g., sampling time.

In this paper, state equation for one chamber type isolator and payload was derived because time delay of actuator response is larger when dual-chamber and capillary tube are used. Considering that resonance frequencies of pneumatic isolator are in 1~5Hz, ground vibrations up to about 50Hz were used for simulations.

2.1 Time Delay Control

Consider a nonlinear dynamic model given in Eq.(1) under the assumption that all state variables and their time derivative are available.

$$\dot{\mathbf{x}} = \mathbf{f}(\mathbf{x},t) + \mathbf{B}(\mathbf{x},t)\mathbf{u} + \mathbf{d}(t) \tag{1}$$

In the above equation, $\mathbf{x}$ represents the state vector, $\mathbf{u}$ the control input, $\mathbf{f}(\mathbf{x},t)$ the unknown plant dynamics, $\mathbf{d}(t)$ the unknown disturbance, $\mathbf{B}(\mathbf{x},t)$ the control distribution matrix.

In the time delay control, exact information of the plant is not necessarily required because the plant is controlled to track dynamic characteristics of a reference model. The reference model, consisting of a desired natural frequency and a damping ratio in this study, is linear and time invariant as given in Eq. (2),

$$\dot{\mathbf{x}}_m = \mathbf{A}_m\mathbf{x}_m + \mathbf{B}_m\mathbf{r} \tag{2}$$

where $\mathbf{x}_m$ denotes the state vector, $\mathbf{A}_m$ the system matrix, $\mathbf{B}_m$ the command distribution matrix, $\mathbf{r}$ the command vector. Using Eqs.(1) and (2), the control input is derived such that the plant in Eq. (1) should track the reference model as follows:.

$$\mathbf{u} = \hat{\mathbf{B}}^{+}[-\hat{\mathbf{f}}(\mathbf{x},t) - \mathbf{d}(t) + \mathbf{A}_m\mathbf{x} + \mathbf{B}_m\mathbf{r} - \mathbf{K}\mathbf{e}] \tag{3}$$

where $\hat{\mathbf{B}}^{+}$ denotes the pseudo-inverse of $\hat{\mathbf{B}}$, $\hat{\mathbf{B}}$ a constant matrix representing the known range of $\mathbf{B}(\mathbf{x},t)$, $\hat{\mathbf{f}}(\mathbf{x},t)$ the unknown parts in the plant, $\mathbf{K}$ the feedback constant. All of the input terms in Eq.(3) are already known except $\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t)$. It is clear that the control input $\mathbf{u}$ can be decided as soon as $\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t)$ is available. Based on the fact that $\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t)$ is a continuous function, it is assumed that $\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t)$ and $\hat{\mathbf{f}}(\mathbf{x},t-L)+\mathbf{d}(t-L)$ are almost the same when the time delay L is small. That is, $\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t)$ is approximated as shown below:

$$\begin{aligned}\hat{\mathbf{f}}(\mathbf{x},t)+\mathbf{d}(t) &= \dot{\mathbf{x}} - \hat{\mathbf{B}}\mathbf{u} \\ &\cong \dot{\mathbf{x}}(t-L) - \hat{\mathbf{B}}\mathbf{u}(t-L)\end{aligned} \tag{4}$$

where time delay L corresponds to integer multiples of sampling time in discrete time control. By substituting Eq.(4) into Eq.(3), the control input is derived as follows:

$$\mathbf{u} = \hat{\mathbf{B}}^{+}\left[-\dot{\mathbf{x}}(t-L) + \hat{\mathbf{B}}\mathbf{u}(t-L) + \mathbf{A}_m\mathbf{x} + \mathbf{B}_m\mathbf{r} - \mathbf{K}\mathbf{e}\right] \tag{5}$$

As mentioned before, values of the state and its derivative must be provided somehow.

2.2 Derivation of State Equation

Eq.(6) is derived by applying Newton's second law to pneumatic vibration isolator including payload.

$$m\ddot{x} + k^{*}(x_p,\omega)\otimes(x - x_b) = A_pP_{tc} + F \tag{6}$$

where $\otimes$ represents convolution integral, $k^{*}(x_p,\omega)$ complex stiffness of the pneumatic vibration isolator except the diaphragm, P_{tc} control pressure and F force excitation on the payload. $k^{*}(x_p,\omega)$ is written as a function of frequency and dynamic amplitude due to its frequency and dynamic amplitude dependency as like Eq.(7) [5].

$$k^{*}(x_p,\omega)=k_s+k_d^{*}(x_p,\omega) \tag{7}$$

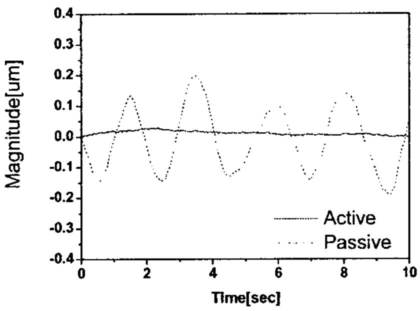
where k_s denotes stiffness of single chamber, $\frac{\kappa p_0 A_p^2}{V_{t0}}$ [2], k_d^{*} complex stiffness of diaphragm. In this paper, complex stiffness of diaphragm regards as nonlinear dynamics since it shows the dynamic amplitude dependency and is hard to be estimated the behavior. The mean value of complex stiffness of diaphragm is used for simulation and realization of pneumatic vibration isolators. The state equation is derived by Eq.(8)

$$\begin{aligned}\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\end{bmatrix} &= \begin{bmatrix}0 & 1\\ \frac{k_s+k_d}{m} & \frac{c_d}{m}\end{bmatrix}\begin{bmatrix}x_1\\ x_2\end{bmatrix} + \begin{bmatrix}0\\ \frac{A_p}{m}\end{bmatrix}P_{tc} \\ &+ \begin{bmatrix}0\\ \frac{1}{m}\end{bmatrix}[(k_s+k_d)x_b + c_d\dot{x}_b] + \begin{bmatrix}0\\ \frac{1}{m}\end{bmatrix}F\end{aligned} \tag{8}$$

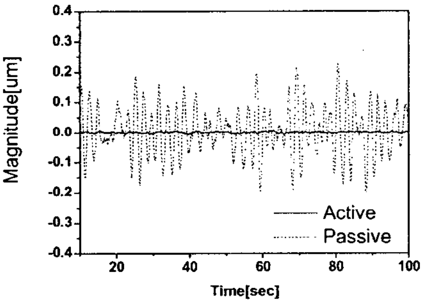
where the second term on right-hand side denotes control input, the third term ground vibration, the fourth term force excitation on the payload, k_d real part of complex stiffness of diaphragm and c_d equivalent viscous damping of diaphragm.

3. Simulation Results from Time Delay Control Applications to Pneumatic Isolator

3.1 Simulation Results for Ideal Case



(a) Time 0~10 sec



(a) Time 10~100 sec

Figure 2. Vibration on the Payload

For comparison purpose, vibrations of the payload with and without control in time domain are shown in Figure 2 and transmissibilities of the active and passive isolator are shown in Figure 3. While vibration of the ground is 20㎛, that of the payload for passive type is 0.08㎛ and for active type is 0.005㎛ in RMS value. Vibration of the payload for the active pneumatic isolator is reduced to 6% of the passive one. The settling time doesn't need to be discussed because the overall transient time signals of active type between 0 and 10 sec are much smaller than passive one. In this paper, a single DOF system of 0.5Hz for natural frequency and critical damping was chosen for the reference model. The reason why the natural frequency and critical damping were chosen for the reference model is that the goal is to make transmissibility for the pneumatic isolator lower and frequency range of accelerometer has limitation to measure the vibration.

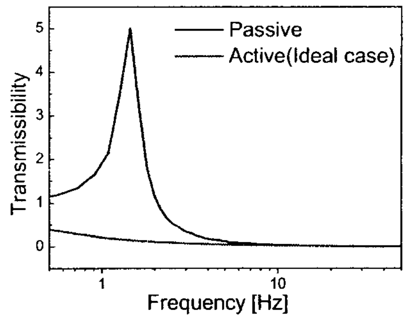


Figure 3. Transmissibility between Displacement of Payload and Ground

Transmissibility for the active pneumatic isolator is calculated from steady state data. Total time was 50sec, the sampling interval in data processing was 0.01sec and random signals of frequency 0~50Hz and amplitude 20㎛ in RMS value were used for the ground excitation. In this simulation, it is shown that transmissibility for the active pneumatic isolator is much lower than that of the passive one. Comparing with performance for the passive pneumatic isolator, the active one isolates the ground excitation in near to and lower natural frequency range because control input is composed to eliminate the ground excitation by considering it as a disturbance and to track the reference model.

3.2 Simulation Results Including Actuator Dynamics and Modified Integration Method

A pneumatic servo-valve used as an actuator has some problems in aspect of response accuracy and speed. Figure 4 shows the transfer function of actuator without any action, output pressure for incoming signal voltage to control volume, 2.98×10^{-4} m^3. In aspect of the magnitude and phase, the servo-valve needs to improve the response accuracy and speed. PD controller is applied to servo-valve to improve it. The determination of controller gain is discussed later in this paper and only the results applied PD controller is dealt with for the simulation. The transfer function of actuator applied PD controller is Eq.(9).

$$\text{T.F.}=\frac{\text{Output Pressure}}{\text{Input Voltage}}=\frac{1068.2s+554.5}{s^2-347.3s-178.6} \tag{9}$$

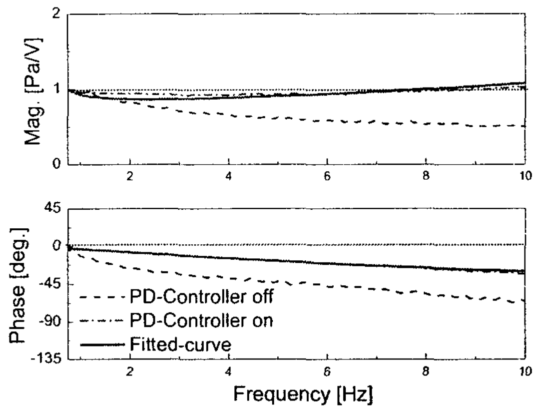


Figure 4. Transfer Function of Pneumatic Servo-valve

In aspect of cost and difficulty to install the sensors like velocity or displacement sensor, accelerometer is used to estimate the states, velocity and displacement, which are needed to compute the control input to apply the time delay control theory. However the DC components are accumulated infinitely as integrating the acceleration signal. Some reference[7] suggested the modified integration method to estimate them, Eq.(10).

$$\frac{1}{s} \to \frac{1}{s+a}, \tag{10}$$

Whereas ' a ' denotes the modified integration coefficient and it is determined by a tenth of the lowest frequency of interested frequency range, 0.5Hz. The modified integration method can prevent the estimation signal from accumulating DC infinitely.

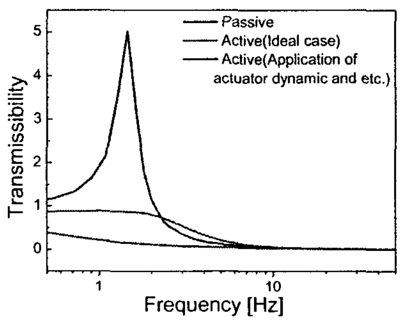


Figure 5. Transmissibility between Displacement of Payload and Ground

Figure 5 shows the simulation results of the transmissibility of passive and active type pneumatic vibration isolator including the one which is applied the actuator dynamics and modified integration method. The isolation performance is naturally decreased as applying them but still better than passive one, especially near and below the natural frequency. The RMS values of transmissibility for passive one and active one applied the actuator dynamics are 0.49 and 0.11 respectively.

4. Configuration of Experimental Equipment for TDC applications to Pneumatic Isolator

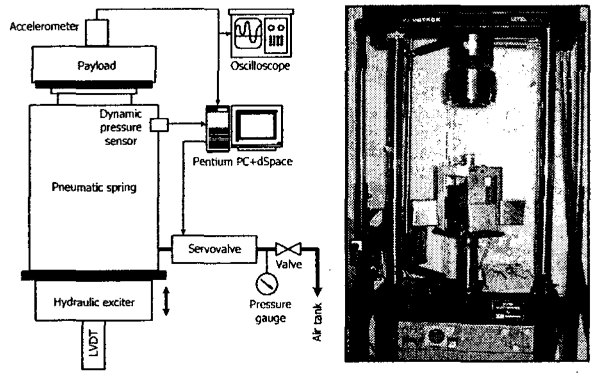


Figure 6. Configuration of Experimental Equipment

Figure 6 shows the configuration of experimental equipment for TDC applications to pneumatic isolator. The dSpace and Matlab are used to compute the control input voltage to produce the pressure. The pneumatic servo-valve is used to keep the static pressure, 3.51bar, and supply the dynamic pressure. Because the dynamic behavior of payload is only concerned, a dynamic pressure sensor, PCB 106B, that has high pressure amplitude resolution is installed in the control volume and a high sensitive seismic accelerometer, PCB 393B05, is installed on the payload. The mass of payload is 87kg, about a fourth of total payload of isolation table. From simulation results, the dynamic amplitude of pressure as control input is about 50 Pa when the ground vibration is about 20 ㎛, common laboratory vibration, but the servo-valve, Bosch Rexroth ED12, used in this research can't afford it. To show the effectiveness of control theory and isolation performance, the hydraulic exciter is used to produce the higher ground vibration, 500 ㎛. In the near future, we will examine the effectiveness under the common ground vibration as changing or redesigning the pneumatic servo-valve.

5. Experimental Results

5.1 Application of PD Controller to Improve Actuator Response

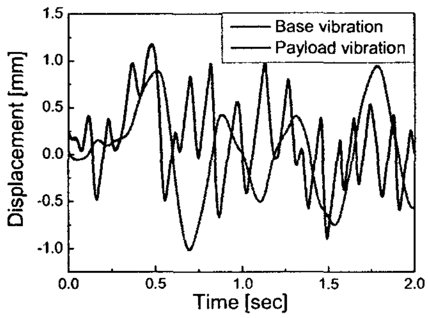
In previous chapter, the necessity of PD controller to the pneumatic servo-valve was discussed. For fast response, the lager P and D gains are more effective but the actuator system easily reaches the unstable region and has saturation point of pressure supply. Therefore the gains of P and D are determined as high as possible in the stable and unsaturated range, Table 1.

	P-gain	D-gain
Controller Gain	4	0.03

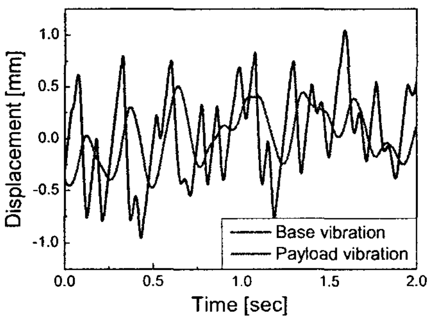
Table 1. Gain of PD Controller Applied to Servo-valve

Figure 4 shows the transfer function of actuator before and after applications of PD controller. Magnitude and phase of transfer function are improved as applying PD controller to pneumatic servo-valve.

5.2 Experimental Results



(a) Passive type



(b) Active type

Figure 6. The Vibrations of Payload and Base

Figure 6 shows the time signals of base excitation and payload vibration for passive and active ones respectively. The payload vibration of active one is lower than base vibration while the passive one is higher. Also, the necessity of isolation near natural frequency, about 2 Hz, can be found as shown it. The isolation efficiency can be described by the ratio of RMS values between payload and base, 1.20 and 0.56 on base excitation, 500 ㎛ and 0~12.5Hz, respectively.

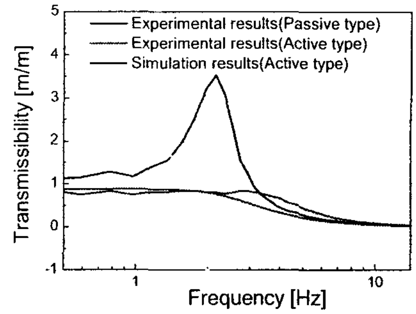


Figure 7. Transmissibility between The Vibrations of Payload and Base

Figure 7 shows the experimental results of transmissibility of passive and active type pneumatic vibration isolator with the simulation results. Total time was 100sec, the sampling interval in data processing was 0.01sec and random signals of frequency 0~12.5Hz and amplitude 500 ㎛ in RMS were used for the base excitation. The performance of isolation is improved, especially near and below natural frequency. The RMS values of transmissibility for passive and active ones are 1.02 and 0.48 respectively which are similar results with time domain ones and the isolation performance of 53 %, the difference between passive and active ones based on passive one, is improved. The experimental results are very similar to simulation one and the main factor caused the deterioration of isolation performance is explained by simulation results as response accuracy and speed of actuator compared with ideal case. The isolation performance can be advanced by changing or redesigning the actuator.

6. Conclusion

In this paper, a methodology to improve dynamic performance of the pneumatic isolators was discussed, which was application of so called time delay control law to the active air-pressure control. Also the experimental results applied TDC theory were shown and examined compared with passive one. The results showed the effectiveness of TDC to pneumatic vibration isolator with respect to improvement of isolation performance. Lastly, the necessity of changing or redesigning the actuator was discussed.

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