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SIMULATION OF LOW FREQUENCY PERFORMANCE EHANCEMENT FOR PNEUMATIC VIBRATION ISOLATORS BY TIME DELAY CONTROL

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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 dual chamber pneumatic isolator. A 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 simulations. Comparison between passive and active pneumatic isolators is also presented.

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 piston, capillary tube, two chambers and diaphragm. The piston supports 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[4] to the active air-presure control. The time delay control(TDC) has been effectively applied to systems in the presence of nonlinearity, uncertainty and disturbance[4]. Thus, application of the TDC to the pneumatic isolators which show dynamic amplitude dependency significantly would be suitable.

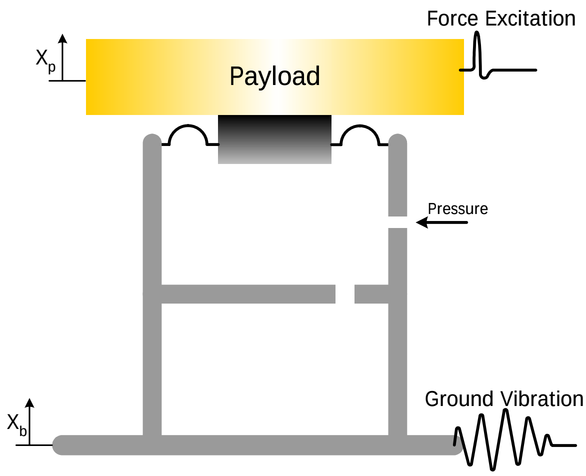


Figure 1. Dual-Chamber Pneumatic isolator and Payload

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 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, it is assumed that all state variables are measurable, which may not be the case in real situations. Considering that resonance frequencies of pneumatic isolatosr are in 1~10Hz, ground vibrations up to about 50Hz were used for simulations.

Time delay control[4]

Consider a nonlinear dynamic model given in Eq.(1) under the assumption that all state variables and their time derivativ s 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:.

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

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 $\hat{\mathbf{f}}(\mathbf{x},t\text{-}L) + \mathbf{d}(t\text{-}L)$ 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:

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

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:

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

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

State equation including control input

Eq.(6) is derived by applying Newton’s second law to Figure 1 which describes dynamic behaviour of the pneumatic vibration isolator.

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

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 piston. $k^*(x_p,\omega)$ is written as a function of frequency and dynamic amplitude due to its frequency and dynamic amplitude dependency as like Eq.(7) [5].

$$k^*(x_p,s)=k_s\frac{s+s_0(x_p,s)}{s+s_0(x_p,s)(N+1)} \tag{7}$$

where $k_s\equiv\frac{\kappa p_0A_p^2}{V_{t0}}$, $s_0(x_p,s)=\frac{\kappa RT_0}{V_bC(x_p,s)}$, $N\equiv\frac{V_{b0}}{V_{t0}}$. It enables the development of linear model under assumption that the amplitude of vibration on the piston is infinitesimal[1].

$$k^*(s)=k_s\frac{s+s_0}{s+s_0(N+1)} \tag{8}$$

where $s_0=\frac{\kappa RT_0}{V_bC_u}$, $C_u=\frac{128\mu L_c}{\pi\rho D_c^4}$ (flow restriction constant).

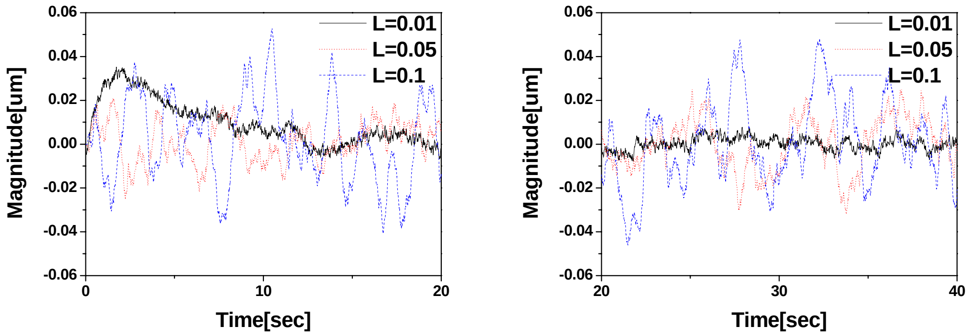
In this paper, complex stiffness of the pneumatic isolator is used by Eq.(8) because TDC which is less needed the information of the plant is applied to the pneumatic isolator. Substituting Eq.(8) into Eq.(6) and performing Laplace transform lead to

$$\begin{aligned}&[ms^3+ms_0(N+1)s^2+k_ss+k_ss_0]X(s)\\&=[s+s_0(N+1)]A_pP_{tc}+k_s(s+s_0)X_b(s)+[s+s_0(N+1)]F\end{aligned} \tag{9}$$

where the first term on right-hand side denotes control input, the second term ground vibration, the third term force excitation on the piston. Eq.(10) is the state equation form of Eq.(9).

$$\begin{bmatrix}\dot{x}_1\\\dot{x}_2\\\dot{x}_3\end{bmatrix}=\begin{bmatrix}0&1&0\\0&0&1\\-\frac{k_ss_0}{m}&-\frac{k_s}{m}&-s_0(N+1)\end{bmatrix}\begin{bmatrix}x_1\\x_2\\x_3\end{bmatrix}+\begin{bmatrix}0\\0\\\frac{A_p}{m}\end{bmatrix}\left[s_0(N+1)P_{tc}+\dot{P}_{tc}\right]+\begin{bmatrix}0\\0\\\frac{k_s}{m}\end{bmatrix}\left[s_0x_b+\dot{x}_b\right]+\begin{bmatrix}0\\0\\\frac{1}{m}\end{bmatrix}\left[s_0(N+1)F+\dot{F}\right] \tag{10}$$

SIMULATION RESULTS FROM TIME DELAY CONTROL APPLICATIONS TO PNEUMATIC ISOLATOR



(a) Time : 0~20s *(b) Time : 20~40s*

Figure 2. Vibration on the piston

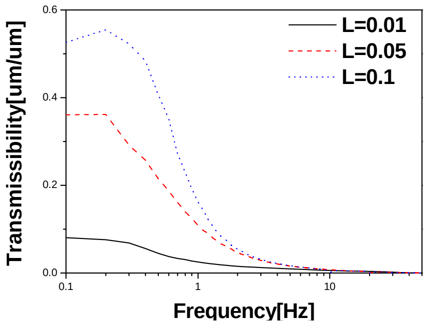


Figure 3. Transmissibility of the active isolator

Random signals of frequency 0~50Hz and amplitude 20μm in RMS value were used for the ground excitation. A single DOF system of 0.5Hz for natural frequency and critical damping was chosen for the reference model. Although performance of the vibration isolation can be more enhanced by choosing a natural frequency lower than 0.5Hz, it was done so for further comparative studies which will be done later based on actual experiments accelerations lower than 0.5Hz are not easy to measure. Vibration displacements of the piston in time domain obtained by applying three different time delays are shown in Figure 2. The sampling interval in data processing was 0.01sec, meaning that L=0.01 in Figure 2 represents the case where the state and the control input one sampling interval ago are used for estimation of the current control input. Although effects of different time delays in 0~10sec are not obvious, these become clear as time passes and it can be said that vibrations of the piston are smaller with shorter time delay.

Adjusting the time delay in the above simulations could be related to dynamic characteristics of the servo valve in a real active control system. That is, the faster the valve response, the smaller the time delay which can be realizable. In reality, the time delay in active control may be limited by the response time of the pressure sensor inside the servo valve. Transmissibilities for the three different time delays are shown in Figure 3. As can be expected from the time domain results in Figure 2, the smaller the time delay, the lower the transmissibility.

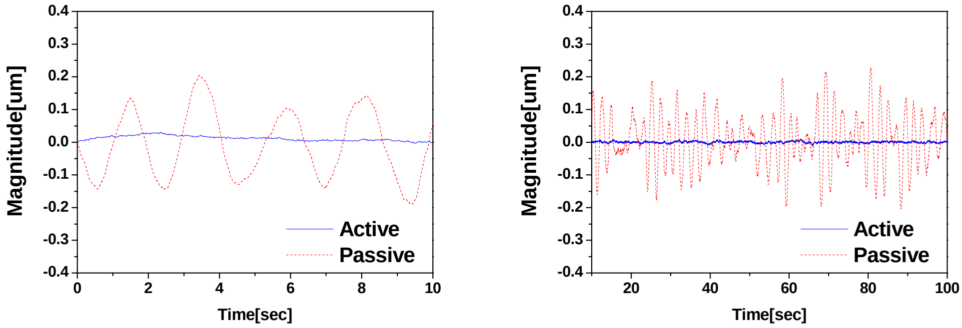


Figure 4. Vibrations of the piston of the pneumatic isolator

For comparison purpose, vibrations of the piston without control in time domain are as shown in Figure 4 and transmissibilities of the active and passive isolator are shown in Figure 5. While vibration of the ground is 20µm, that of the piston for passive type is 0.08µm and active type is 0.005µm in RMS value. Vibration of the piston for the active pneumatic isolator is reduced to 6% of the passive one. In simulation, settling time is about 10sec[6] because a single DOF system of 0.5Hz for natural frequency and critical damping was chosen for the reference model. If natural frequency is lower than 0.5Hz, vibration isolation performance is improved in spite of long settling time. However vibration of the piston for the active pneumatic isolator is much smaller than that of passive one in transient state. In general, damping ratio is chosen to 0.4~0.8 in order to protect from the large overshoot and proper settling time[6]. In this paper, critical damping was chosen for the reference model because the goal is to make transmissibility for the pneumatic isolator lower.

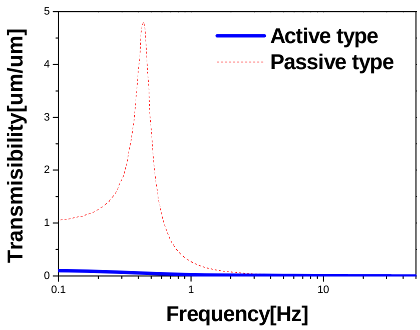


Figure 5. Transmissibility of the pneumatic isolator

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 20μm 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 disturbance and track the reference model.

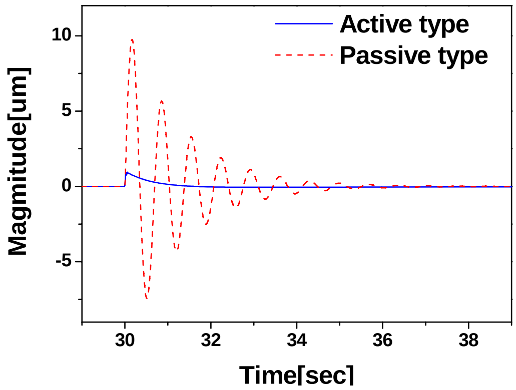


Figure 6. Force disturbance on the piston

Figure 6 shows the impulse response on the piston for the active and passive pneumatic isolator. The high level excitation is generated and short settling time is appeared on the piston for the passive isolator because damping ratio is about 0.1. However the low level excitation and short settling time is appeared in the active one. It means that the active isolator tracks the reference model well and not only excitation of ground but that of the piston also can be reduced when time delay control law is applied. Because, as mentioned before, control input is composed to eliminate the ground vibration by considering disturbance and track the reference model.

In real system, the ground excitation is about several tens μm, dynamic pressure in chamber is about several tens μPa. Thus the sensitivity of the pressure sensor inside the servo valve has to be chosen properly to control the pneumatic isolator. It is hard to measure physical quantities such as acceleration of low frequency range by sensors. It is regarded to have same performance as the passive pneumatic isolator below the lowest frequency range limit of sensor measurement. Thus high sensitivity accelerometer is required to performance improvement for the pneumatic isolator.

CONCLUSIONS

In this paper, performance improvement for the pneumatic isolator with active control law has been discussed based on simulation. TDC law which is easy to apply and robust against disturbance is proposed as a control law.

Although the passive pneumatic isolator can't be played a role of vibration isolation element with the ground excitation in near to and lower natural frequency range, the active one is possible to isolate not only low frequency excitation of the ground but also force excitation of the piston.

On going research, transmissibility of the active pneumatic isolator will be measured in actual experiments. Experimental results will be compared with simulation results.

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