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CONTROL OF A VIBRATION ISOLATION SYSTEM USING PRESSURE DIFFERENTIATORS AND SPOOL TYPE SERVO VALVES

Tomonori KATO*, Yosuke YAMAZAKI** , Koichi SAWAMOTO***, Kenji KAWASHIMA***
and Toshiharu KAGAWA***

* Department of Mechanical Engineering, Tokyo Metropolitan College of Technology
1-10-40 Higashi-oi, Shinagawa, Tokyo, 140-0011 Japan
(E-mail: kato@tokyo-tmct.ac.jp)
**Sumitomo Metals, 1-8 Fuso-cho, Amagasaki City, Osaka,
***Tokyo Institute of Technology
4259 Nagatsuta-cho, Midori-ku, Yokohama City, Kanagawa 226-8503 Japan

ABSTRACT

Active control type pneumatic isolation systems are widely used in semiconductor production processes. Most of them are actively controlled by nozzle flapper type servo valves, since they have fine characteristics as imperfect pressure integrators. But nozzle flapper type servo valves exhaust a lot of compressed air, for the sake of controllability. In this research, we firstly develop a pressure differentiator using an isothermal chamber, a cylindrical type laminar flow element and a differential pressure sensor. Then, by using the developed pressure differentiator and a spool type servo valve, an imperfect pressure integration system with much lower exhausted air flow rate compared to the system with nozzle flapper type servo valve, is constituted. Finally, the proposed pressure integration system is applied to the control of active vibration isolation system. The experimental results indicate superior performance of the developed system, especially in much lower energy consumption.

KEY WORDS

Pneumatics, Vibration Isolation System, Energy Saving, Pressure Differentiator

NOMENCLATURE

d : nominal length [m]
f : frequency [Hz]
f_c : cut-off frequency [Hz]
G : mass flow rate [kg/s]
K : gain coefficient [Pa・s]
L : length of slit [m]
L_i : inlet length [m]
P_a : atmospheric pressure [Pa]
P_c : pressure in chamber [Pa]
P_{do} : output signal of pressure differentiator [Pa]

P_j : differential pressure[Pa]

P_s : measured pressure [Pa]

Q : volume flow rate [m^3/s]

R : gas constant [J/(kg・K)]

r_1 : Outer diameter [m]

r_2 : Inner diameter [m]

s : laplace operator [1/s]

st : sampling time [s]

t : time [s]

T : time constant of the sensor [s]

v : velocity in laminar flow element [m/s]

V : volume of chamber [m^3]

W : air mass in chamber[kg]

x : displacement [m]

θ : temperature in a chamber [K]

θ_a : room temperature [K]

μ : viscosity [Pa・s]

ρ_a : density of air at atmospheric pressure [kg/m^3]

ρ_c : density of air when pressurized [kg/m^3]

INTRODUCTION

Since air is an ideal gas for design of vibration isolation systems, air spring systems are widely used in industry, as seen in machine tools or transportation systems[1]. Air spring systems are required such performances as steady-state stability, quick self-motion control, and good floor vibration isolation. In order to satisfy these demands, nozzle flapper type servo valves with linear and precise pressure controllability are widely used. But if pressure control with high gain is needed, nozzle flapper type servo-valve requires large amount of steady-state exhausted air flow rate, i.e. energy efficiency is sacrificed to controllability. Though there is a over lap type spool type servo valve which has much smaller amount of steady-state exhausted air flow rate and high flow rate gain, since it has strong non-linearity between its control signal to output flow rate, spool type servo valves have so far been difficult to use for precise pressure control like air spring systems.

In this paper, in order to reduce the affection by the nonlinearity and dead-point shift which spool type servo valves have, a pressure differentiator using an isothermal chamber[2][3], which we developed is applied to the pressure control in a pressure tank, as constituting a cascade pressure control system. Then, the cascade pressure control system is applied to the isolation table.

DEVELOPMENT OF PRESSURE DIFFERNETIATOR

Principle of the proposed pressure differentiator

As shown in Fig.1, the proposed pressure differentiator is composed of an isothermal chamber, cylindrical laminar flow element, a diaphragm type differential pressure gauge and a pressure sensor. The cross sectional structure of a slit is shown in Fig.1.When the measured pressure P_s changes, air flows through the narrow slit between two plates, and pressure in chamber P_c changes slightly behind of P_s. By measuring the differential pressure $P_j=P_s-P_c$ with a diaphragm type differential pressure gauge, the differentiated value of P_s , dP_s/dt can be calculated.

Assuming the flow in the slit as a laminar flow, and taking equation of energy and Poiseuille law into consideration, the measurement principle of the sensor is explained as follows. The state equation for compressive fluids in a chamber can be written as Eq.(1).

$$P_c V = WR\theta \tag{1}$$

The following equation is derived by totally differentiating Eq.(1) and assuming that the chamber volume is constant:

$$\frac{dP_c}{dt}V = GR\,\theta + WR\,\frac{d\theta_a}{dt} \tag{2}$$

If the state of air in the chamber during charge or discharge remains isothermal, the following equation can be obtained from Eq.(2):

$$G = \frac{V}{R\theta_a}\frac{dP_c}{dt} \tag{3}$$

In Eq.(3), the average temperature in the chamber is equal to the room temperature. It is clear from Eq.(3) that if the volume of the chamber V and the room temperature θ_a are known, the mass flow rate G is proportional to the differentiated value of pressure in chamber P_s.

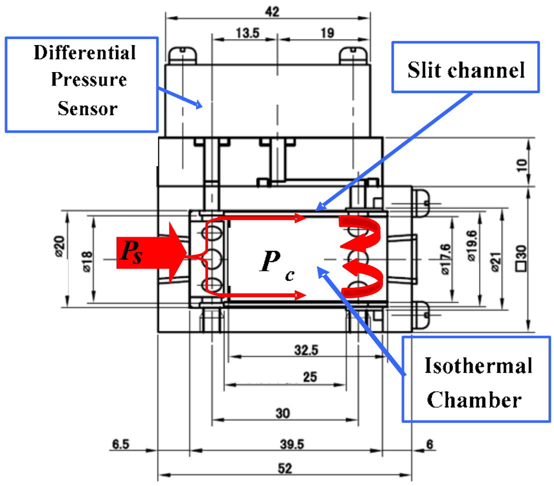

Fig.1 Structure of the proposed sensor

Suppose the width, outer and inner diameters and length of the slit are respectively *r1, r2*, *h*, *L*, and if flow in the slit is laminar, the relationship between the volume flow rate flowing through the slit *Q* and the differential pressure of both edges of slits can be expressed by Poiseuille law as

$$Q = \frac{\pi r_2 (r_1 - r_2)^3}{6\mu L}(P_s - P_c) \tag{4}$$

Taking the density of air at atmospheric pressure ρ_a and the pressure in a chamber P_c into consideration, the mass flow rate can be expressed by the equation (5).

$$G = \rho_a \frac{P_c}{P_a} Q = \frac{\rho_a P_c \pi r_2 (r_1 - r_2)^3}{6 P_a \mu L}(P_s - P_c) \tag{5}$$

Defying the resistance coefficient of flow rate as $r_s = 6 P_a \mu L / \{\rho_a \pi r_2 (r_1 - r_2)^2\}$, equation (5) can be rewritten as

$$G = \frac{P_c}{r_s}(P_s - P_c) \tag{6}$$

The output signal of the differential pressure gauge can be written as

$$P_j = P_s - P_c \tag{7}$$

Therefore, from by Eq.(6) and Eq.(7), G can be written as follows.

$$G = \frac{P_c}{r_s} P_j \tag{8}$$

From Eq.(3) and (8), defining $K = \frac{r_s V}{R\theta}$ as the gain coefficient,,

$$\frac{dP_c}{dt} = \frac{P_c P_j}{r_s}\frac{R\theta}{V} = \frac{1}{K} P_c P_j \tag{9}$$

By laplace transform of Eq.(9), assuming a tiny pressure change from the equilibrium point of no differential pressure, $P_j(0)=0$, i.e., $P_s(0) \fallingdotseq P_c(0)$, Eq.(10) can be derived.

$$\begin{aligned} & sP_c - P_c(0) \\ & \cong sP_c - P_s(0) = \frac{1}{K} P_j P_c(0) \end{aligned} \tag{10}$$

Since $sP_s - P_s(0) = \mathcal{L}\left(\frac{dP_s}{dt}\right)$, and defining T as

$$T = \frac{K}{P_c} = \frac{6 P_a \mu L V}{P_c \rho_a \pi r_2 (r_1 - r_2)^3 R\theta} \tag{11}$$

the output of the differential pressure sensor P_j can be written as Eq.(12).

$$P_j = \frac{T}{1 + Ts}\mathcal{L}\left(\frac{dP_s}{dt}\right) \tag{12}$$

So, it is clear that the output value of the differential pressure sensor P_j is T times value of dP_s/dt, with 1st-order lag filter.

Now, suppose the divided value of P_j by T as the output of the proposed differentiator P_{do}, as Eq.(13).

$$P_{do} = \frac{1}{T} P_j = \frac{1}{1 + \left(\frac{K}{P_c}\right)s}\mathcal{L}\left(\frac{dP_s}{dt}\right) \tag{13}$$

When the desired frequency band of the differentiated value of P_s is much lower than the cut off frequency of the proposed sensor,

$$f_c = \frac{1}{2\pi T} = \frac{P_c \rho_a r_2 (r_1 - r_2)^3 R\theta}{12 P_a \mu L V} \tag{14}$$

the proposed apparatus is effective as a pressure differentiator. The sensor in this research is fabricated with the specifications shown in Fig.1. The theoretical cut off frequency of the developed sensor at Pc=180kPa abs is f_c=317[Hz]. For more information about the developed sensor, see the reference paper [2].

APPLICATION OF PRESSURE DIFFERNETIATOR TO ISOLATION TABLE

Cascade pressure control system

The pressure control system in vibration isolation system is required to satisfy the performance as an imperfect pressure integrator. For that reason, ordinarily, nozzle flapper type servo valves are widely used. But, in order to reduce the exhausted air flow rate from the servo valves, a cascade pressure control system using a spool type servo valve and the developed pressure differentiator is proposed. In Fig.1, suppose the affect by the volume change, PA/V*dx/dt, is negligibly small, measuring the differentiated value of pressure, dP/dt, is equivalent to measuring the flow rate. In order to realize an imperfect pressure integrator, a cascade pressure control system, with dP/dt as the minor loop, is constituted. The control cycle of the minor loop is designed as about 900 times of that of the pressure control loop. Therefore, the non-linearity and dead-point shift of the spool type servo valve are compensated, and the existence of the minor loop is almost negligible.

Procedure of experiment

Experiment was conducted with the apparatus shown in Fig.2. The volume of the isolator's chamber is V=1.1×10^{-3}[m^3], the mass of the table is 45.1[kg], and the surface area of the air spring is 72.4×10-4[m^2]. The spool type servo valve used in the experiment is FESTO M5-SA. The gains are as follows : Kx:500[v/m], KxI:250[v/m], Ka:10[V/(m/s^2)], K:2.53×10^4, Kp:1.55

$\times 10^{-4}$[1/s], Kpd:6.24$\times 10^{-4}$[V/(Nl/min)], Kdp:20[V/Pa]. Fig.3 indicates the block diagram of the vibration isolation control system.

In the experiment, at 0s time, the control is started, and after settled at the set displacement, the steady state vibration of acceleration is compared to the value before the isolation table is controlled to float.

Experiment results

The experimental results are shown in Fig.4. The proposed control method can float the table without steady state error and unwanted vibration. achieves the vibration isolation performance within the amplitude of 2$\times$10-3[m/s^2]. The result is no way inferior to that of nozzle flapper type servo valve.

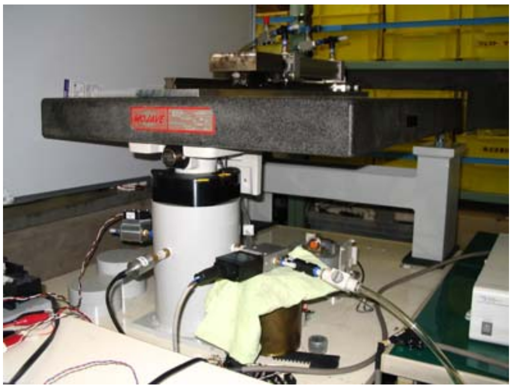

Fig.2 Picture of vibration isolation system

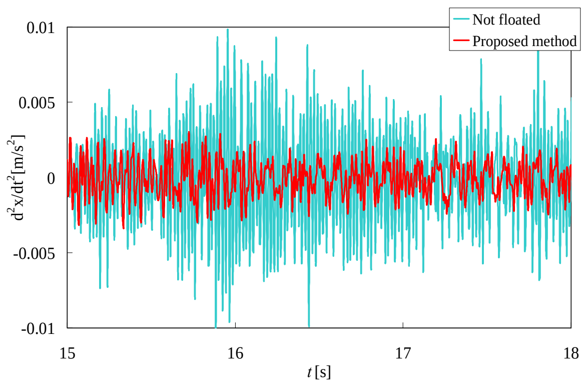

Fig.4 Experimental results of acceleration

Table 1 Comparison of exhausted air flow rate

Type of valve	Nozzle flapper	Spool
Exhausted flow rate[Nl/min]	15	0.4

Table 1 shows the comparison of the exhausted air flow rates on types of the servo valves. The proposed method can reduce the energy loss by exhausted air flow rate.

CONCLUSION

In this research, a pressure differentiator using an isothermal chamber and a cylindrical type laminar flow element is designed and fabricated.

Then, the developed pressure differentiator is applied to the pressure control of the tank of the isolation table, as constituting the cascade pressure control system. An imperfect pressure integrator is developed with a spool type servo valve and a developed sensor.

It is clarified that the proposed method can successfully control the vibration isolation table.

It is also clarified that there is a possibility of realizing the air spring system with much lower exhausted flow rate and higher maximum control flow rate.

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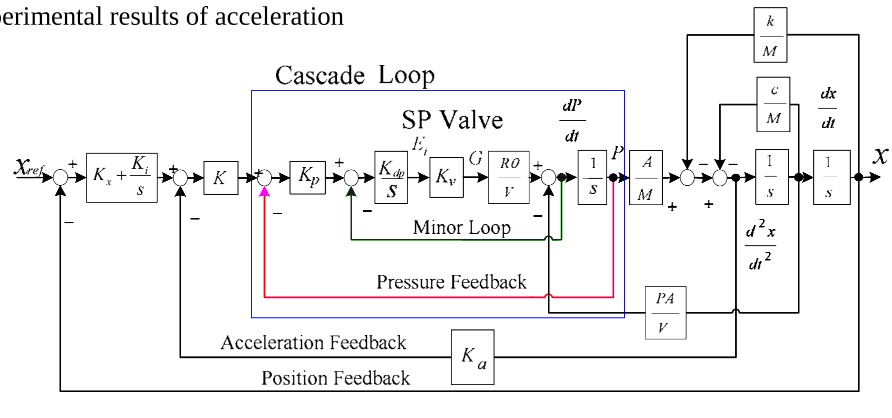

Fig.3 Block diagram of the proposed control method