THE VIBRATION ACTIVE CONTROL ON THE FLUID BORNE NOISE

Jiao Zongxia, Chen Ping, Hua Qing, Wang Shaoping
College of Auomation Science and Electrical Egineering
Beijing University of Aeronautics and Astronautics, 100083, Beijing, P.R.China
zxjiao@buaa.edu.cn

ABSTRACT

Vibration is the vital problem in the dynamic design of the aircraft power supply system, which is correlation with safty and performance of the hydraulic system.The conventional passive absorber can not adapt the variable work conditions, whearas the hydraulic systems of aircraft are developing toward high pressure, large power and becoming more complex. This paper presented a vibration active control method to reduce the vibrations of fluid power supply and pipeline systems, which is of adaptive and robust aganist disturbance, and can keep the vibration minmum under variable speed and load. This paper illustrated the theory and application of vibration active control of fluid power supply and pipeline systems in detail. The multi-layer PZT is taken as the driver which is of characteristics of large force output, high frequency bandwith and small volume. The auto optimum control method is adopted, which can adjust control parameters at any time to aganist the disturbance.

KEYWORDS: PZT Fluid Borne Noise

INTRODUTION

The device of vibration active control can be traced back to 20 Century, the magnetic valve controlled buffer was used first then. The more complex and practical vibration active control systems were put into application till sixties, mainly focused on field of aeronautics and aerospace engineering [1]. Now it is widely applied in aeronautics and astronautics, motor mobile and mechanical engineering. The fluid power technology is usually used as the method for vibration active control, but the hydraulic system itself is seldom to reduce vibration by active control technology. The Japanese scholar Yokota has ever carried out the similar work, the active accumulator was designed which adjusts the flow peak through piston movement controlling, and result in a large volume.
The paper proposes a new kind of vibration active control method. An active control orifice valve is designed, which is used as active absorber. It is controlled by optimum control algorithm adaptively.

THE SCHEMA OF VIBRATION ACTIVE CONTROL

Fig.1 is a fluid power supply pipeline system. In generally there is fluid borne noise due to structure of the pump, which will induce the vibration of the fluid pipeline network connected. If the fluid pulsation frequency is near to the harmony frequency of the pipeline network, it will result in resonance, furthermore, induce fluid-structure interaction, which is closely related to the fluid borne noise frequency and model of pipe networks.
In order to reduce the system vibration, an orifice valve is designed and connected into the system bypass, which can adjust flow rate pulsation dynamically through a adaptively optimum control method. The orifice valve is droved by PZT. Normally, relieving flow rate through orifice valve is proportional to jaw opening of the valve. If it can be adjusted according to the flow rate pulsation dynamically, it is available to control the vibration. The frequency spectrum is got through FFT analysis of the pressure pulsation, which is taken as the reference vibration control frequencies. The other key problem is how to determine the controlling amplification and phase shift of the active absorber. This paper shows that an efficient, fast and valid optimum is needed to adjust above control parameters.

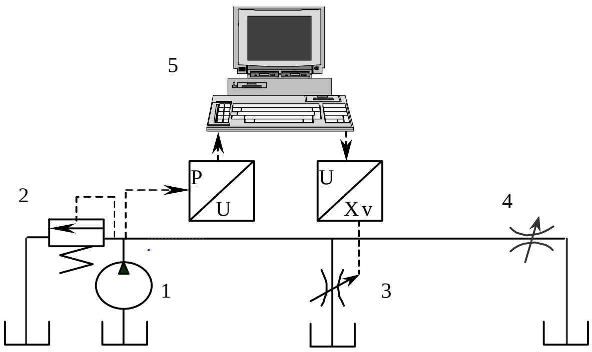

Fig. 1 The schema of vibration active control

1-pump, 2-relief valve, 3-active relief valve, 4-Load, 5-Industry PC

This Paper is financed by Aeronautics Science Foundation and Youth backbone teacher foundation of Education Ministry

DESIGN OF ACTIVE ORIFICE VALVE

The active orifice valve is droved by multi-layer PZT, see Fig.2 is the schema of it. The jaw opening of the valve can be controlled by input voltage.

(1) The Mathematics of Piezoelectric Ceramic

Normally, piezoelectric ceramic can produce electric charge under external force. That is the direct piezoelectric effect. On the other hand, it will flex under external electric field. That is converse piezoelectric effect. Mathematics model is as follows:

Direct piezoelectric effect:

$$D = d^t T + e^T E \tag{1}$$

Converse piezoelectric effect:

$$S = s^e T + dE \tag{2}$$

D -- Electric displacement(C/m^2)

D=[D_1 D_2 D_3]

S -- Strain

E -- Electric Field V/m

T -- Stress Pa

e^T -- Dielectric constant matrix

$$e^T_{i,j} = [\partial D_i / \partial E_j] \text{ C/(V*m)}$$

s^e -- Flexible Matrix (Pa^{-1})

d -- Piezoelectric strain constant matrix(m/V)

d^t -- Strain charge constant C/N

$$d_{i,j} = [\partial S_j / \partial E_i]_T = [\partial D_i / \partial T_j]_E$$

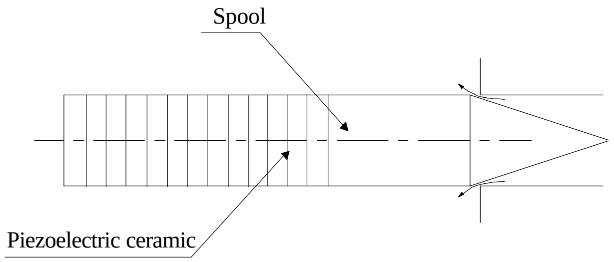

Fig.2 The schema of active orifice valve

The relation between strain and electric field intensity regardless load influence is as follows:

$$S = \begin{bmatrix} 0 & 0 & 0 & 0 & d_{15} & 0 \\ 0 & 0 & 0 & d_{14} & 0 & 0 \\ d_{31} & d_{32} & d_{33} & 0 & 0 & 0 \end{bmatrix}^t \begin{bmatrix} E_1 \\ E_2 \\ E_3 \end{bmatrix} \tag{3}$$

Because the vibration frequency is not greater than 1000Hz, and the force is small in the application field this paper involved, the relation between displacement and voltage independent of the frequency can be simplified as:

$$\Delta l = d_{33} V \tag{4}$$

V— Voltage (V)

Fig.2 is the driver part of valve spool. It will flex proportional to external electric field intensity.

(2) Mathematics of Orifice Valve

The flow rate equation of orifice valve is

$$Q_l = C_V W X_V \sqrt{\frac{2(P_s - P_o)}{r}}$$

Above equation can be linearized on zero zone as:

$$Q_l = K_Q X_V + K_C P_s \tag{5}$$

Combined (4) and (5)

$$Q_l = K_Q d_{33} V + K_C P_s \tag{6}$$

where

K_Q is the flow rate amplifier:

$$K_Q = \partial Q_l / \partial X_V = C_V W \sqrt{\frac{2(P_s - P_o)}{r}}$$

K_C is the pressure amplifier.

$$K_C = \partial Q_l / \partial P_s = \frac{C_V W X_V \sqrt{\frac{(P_s - P_o)}{2r}}}{P_s - P_o}$$

C_V -- Flow rate coefficient

W -- Window width of the orifice m

X_V -- Spool displacement m

P_s -- Power supply pressure Pa

P_0 -- Return Pressure (Pa)

r -- Oil density (kg/m^3)

V -- Voltage exerted on the piezoelectric ceramic (V)

MODELING ON FLUID POWER SUPPLY SYSTEM

The model of the hydraulic system is divided into two types [2]: One is bump parameter model, normally it is the form of general differential equations. Another is distributed parameter model, normally is the form of the partial differential equations. The pump, and valve is the typical bump parameter model and fluid pipeline is the typical distributed parameter model, which is related to pipe position [2][4].

(1) Mathematics of Axial Piston Pump

The pump flow rate model and its iterative algorithm is as follows [4]:

$$Q_{out} = Q_m - \Delta Q_{leak} - \Delta Q_{comp} - \Delta Q_{ps} \qquad (7)$$

$$\Delta Q_{ps} = C_d A_{il} \sqrt{\frac{2}{r}(P_p - P_{vs})} \qquad (8)$$

$$p_p = \frac{1}{2}(-FB \sqrt{|FB^2 - 4FC|}) \qquad (9)$$

$$FB = 2[Q_{out} - (Q_m + P_c K_{leak} + \frac{P_{pold} V}{b\Delta t})] \frac{1}{K_{leak} + \frac{V}{b\Delta t}} - \frac{\left[C_d A_{il} /(K_{leak} + \frac{V}{b\Delta t})\right]^2}{r} \qquad (10)$$

$$FC = [Q_{out} - (Q_m + P_c K_{leak} + \frac{P_{pold} V}{b\Delta t})]^2 \frac{1}{K_{leak} + \frac{V}{b\Delta t}} - \frac{2P_{vs}\left[C_d A_{il} /(K_{leak} + \frac{V}{b\Delta t})\right]^2}{r} \qquad (11)$$

$Q_m = \frac{A_p \Delta x}{\Delta t}\sin(y - wt)$, Q_m is the transient flow rate of the pump(m^3/s) A_p is the piston cross area(m^2) w is the angular velocity(rad/s).

Q_{leak} Q_{comp} is the flow rate increment due to leakage and oil compress (m^3/s).

P_{vs} -- Pressure of inlet (Pa)

A_{il} -- Cross-area of inlet (m^2)

K_{leak} -- Leakage coefficient (m^3/(s*Pa))

b -- Oil bulk module (Pa)

P_{Pold} -- The previous outlet pressure (Pa)

(2) Mathematics of Fluid Pipeline

The model of one dimensional pipeline

$$\frac{\partial P}{\partial x} + \frac{r}{A}\frac{\partial Q}{\partial t} + f(Q) = 0 \qquad (12)$$

$$\frac{\partial Q}{\partial x} + \frac{A}{rc}\frac{\partial P}{\partial t} = 0 \qquad (13)$$

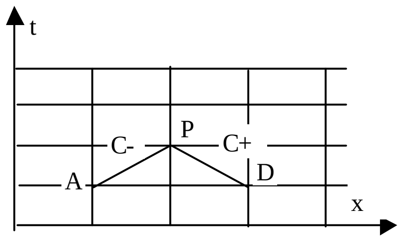

Fig. 3 Schema of characteristic line algorithm

In order to get the solutions of the above partial equations, the characteristic line method are adopted.

$$\begin{aligned} Pp &= (CL + CR)/2 \\ Qp &= (CL - CR)/2 \end{aligned} \qquad (14)$$

Where

$$CL = P_A + ZcQ_A - 0.5c\Delta t[f(Qp) + f(Q_A)]$$

$$CR = P_A - ZcQ_D + 0.5c\Delta t[f(Qp) + f(Q_D)]$$

P--pressure; Q—flow rate; Z_C--Characteristic Impedance $(Pa * s / m^3)$; r --density (kg/m^3)

c--sonic velocity(m/s); x--long coordinate(m); t--time(s); A- Cross area;

Pp ¡¢ Qp ¡¢ P_A ¡¢ Q_A ¡¢ P_D ¡¢ Q_D represents the state of P, A, D points respectively.

$f(Q)$ - The viscous friction related to flow rate:

$$F(Q) = Fs(Q) + Fd(Q) \qquad (15)$$

$Fs(Q)$ -- Steady Friction

$Fd(Q)$ -- Dynamic Friction

Regard as to the steady friction

Laminar flow:

$$Fs(Q) = 128\, rvQ / pd^4$$

turbulent flow

$$Fs(Q) = 0.213\, rv^{0.25} Q^{1.75} / d^{4.75}$$

v -- Viscosity cSt d -- Diameter

Regard as to the dynamic friction according to Trikha model

$$Fd(Q) = 16\, rv / d^4 \sum_{j=1}^{3} y_j(t + \Delta t)$$

Where

$$y_i(t + \Delta t) = y_i(t)\exp(-n_i v / a^2) + m_i[Q(t + \Delta t)] - Q(t)]$$

The values of m_i ¡¢ n_i [10] are as follows:

$m_1 = 8000$ $m_2 = 200$ $m_3 = 24.4$

$n_1 = 160$ $n_2 = 32.4$ $n_3 = 4$

In the solution, the coupling problem between fluid pipeline and pump flow rate models must be considered, see reference [2].

THE CONTROL PRINCIPLE

Fig.4 is the vibration active control function diagram of the whole fluid power supply and pipeline systems. It reduces vibration by connecting an active orifice valve bypass, which can adjust flow ripple through controlling jaw opening of the valve. Normally pump flow ripple includes some sinuous products. It is difficult to measure for its high frequency characteristics, whereas the measure of pressure is no problem. It just is the standard for judging vibration. The frequency of pressure pulsation is same as pump flow ripple, so that we can get the control frequency through FFT analysis of the pressure. The follow key problem is to determine the vibration amplifier and phase shift. The adaptive-optimum control method is adopted in this paper. It can adjust control amplifier and phase shift in real time to keep minimum output of the pressure fluctuation.

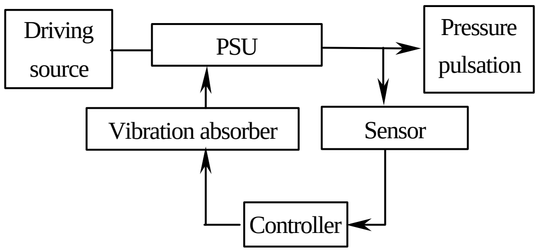

Fig. 4 Diagram of vibration active control

The so-called adaptive-optimum control method is that it can adjust control parameters at any moment according to a optimum ruler in order to keep minimum output of pressure pulsation. It should include follow factors:

- The goal function

The root-mean-square value (RMS value) of pressure pulsation is taken as optimum goal in a period of sample:

$$\bar{p} = \frac{1}{n}\sum_{i=0}^{n-1} p(i) \tag{16}$$

$$E[p^2(i)] = \frac{1}{n}\sum_{i=0}^{n-1}[p(i) - \bar{p}]^2 \tag{17}$$

- Design variables (control parameters)

Fig.5 is the schema of control method. The objective controlled is power supply system. The frequency spectrum can be got through FFT analysis of p(i). The most strong frequency point is chosen as the control frequency

$$x_v = A\sin(wt + j) \tag{18}$$

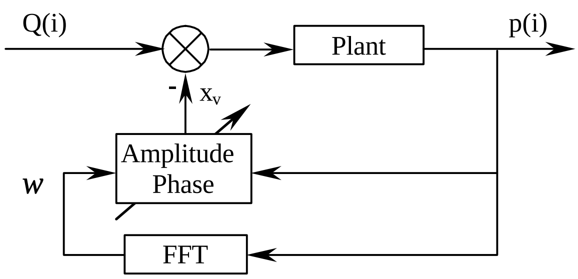

Fig. 5 The structure diagram of control algorithm

Where A is the amplitude, j is the phase shift are the control signals. They are the design variables in adaptive-optimum algorithm.

- Optimum method

The vibration frequency is slowly varying signal in the most of the applications of fluid power supply systems. This paper chooses optimum method in reference [9], the rotate vector method. Its characteristic is free of the calculation of differential quotient, easy to get maximum value, and algorithm is steady. The control parameters are modified and effective on the system in every step during optimum.

SIMULATION FOR A FLUID POWER SUPPLY PIPELINE SYSTEM

The simulation for a fluid power supply pipeline systems shown in Fig.1 is carried out. The simulation parameters are shown in Table 1.

Table 1 Simulation parameters

pump displacement (ml/r)	speed (r/min)	pipe diameter (mm)	pipe length (m)
5.5	1000	10	1
steady pressure (MPa)	maximum displacement of valve(m)		
1.3	$5*10^{-6}$		

The simulation results are shown in Table 2.

Table 2 Simulation results

Flow spulsation (m^3/s) without VAC	Flow spulsation (m^3/s) with VAC	Pressure spulsation (Pa) without VAC
$5.5*10^{-5}$	$0.22*10^{-5}$	$1*10^{5}$
Pressure spulsation (Pa) without VAC	Ratio for vibration reduced	
$0.31*10^{5}$	69%	

The leakage for VAC is only $0.07*10^{-5}$ M^3/s, the power lost is 0.38%. Fig.6 is the simulation curves indicate procedure of vibration active control.

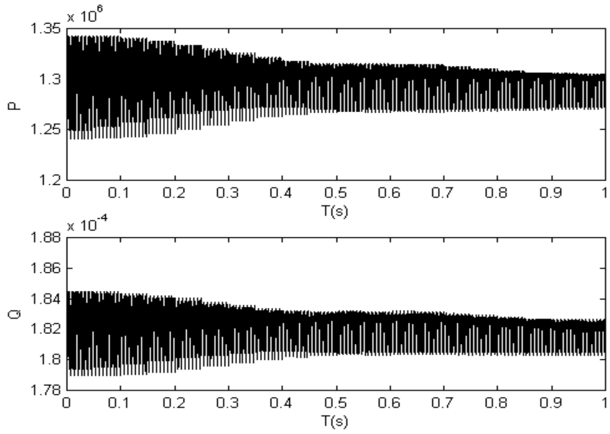

Fig.6 The procedure of the VAC for pressure and flow rate pulsation

CONCLUSION

This paper put forward a new idea for designing active orifice valve droved by PZT technology, which is taken as active absorber; Furthermore, the principle of vibration active control for fluid power supply pipeline systems is presented, subsequently is modeled in detail; The adaptive-optimum control method is adopted to adjust control parameters dynamically to keep pressure pulsation minimum to against external disturbance.

The active absorber is of follow advantages compared with passive one: strong adaptability; robust; small volume. Furthermore is to put it into the practice and widen its application areas.

REFERENCE

[1] Gu Zhongquan etc. Vibration active control, Defense industry publisher, April.1997

[2] Jiao Zongxia, “The research and development of the pump CAD system”, Ph.D. dissertation, Zhejiang University, 1991.7

[3] Zhu Xiaojin, Tao Baoqi, “Structre vibration active control based on auto-optimum control”, Vibration, Measurement and Diagnosis, Vol.19, No.2, March, 1999

[4] Lin Song, “The CAD of hydraulic system”, The West-North Industry publisher, June, 1992

[5] Zhang Yanzhong, “Quick FFT and wahl transformation, Aeronautics Industry Publisher, August, 1989

[6] Zhang Azhou, Yao Qihang etc. “Vibration Control Engineering”, Aeronautic Industry Publisher, 1989

[7] Shinichi YOKOTA, Hisashi SOMADA and Hirotugu YAMAGUCHI Study on an Active Accumulator JSME Series B, Vol.39, NO.1, 1996

[8] Dubowsky S. Active control of mechanical systems: the state-of-the-art for robotic manipulators. AIAA In: Proc. AIAA/ASME/ASCE/AHS26th SSDMC. 1985. 258~261

[9] Jiao Zongxia, Lu Yongxiang, A new engineering optimum method—the rotation method, Journal of Mechanical Engineering, June, 1996

[10] Lei Tianjue, Hydraulic engineering manual, Mechanical industry publisher, 1991