Hybrid Fluid-borne Noise Control in Fluid-filled Pipelines

M Pan, N Johnston, A Plummer

Centre for Power Transmission and Motion Control, Department of Mechanical Engineering, University of Bath, BA2 7AY

m.pan@bath.ac.uk

Abstract. This article reports on an initial investigation of a hybrid fluid-borne noise control system in hydraulic pipelines. The hybrid system is built by integrating an active feedforward noise controller with passive tuned flexible hoses. The active attenuator is designed to cancel the dominant harmonic pressure pulsations in the fluid line, while the passive hose is tuned to attenuate the residual high frequency pulsations. The active attenuator can effectively decrease the fluid-borne noise by superimposing a secondary anti-phase control signal. Adaptive notch filters with the filtered-X least mean square algorithm were applied for the controller and a frequency-domain least mean square filter was used for the secondary path on-line identification. The transmission line model was used to model the pipeline, and a time-domain hose model which includes coupling of longitudinal wall and fluid waves was used to model the flexible hose. Simulation results show that very good noise cancellation was achieved using the proposed approach, which has several advantages over existing fluid-borne noise control systems, being effective for a wide range of frequencies without impairing the system dynamic response much. While the flexible hoses may be less effective than purpose-built passive silencers, they can form an inexpensive and practical solution in combination with active control.

1. Introduction

Fluid-borne noise (FBN) is a major contributor to air-borne noise (ABN) and structure-borne noise (SBN) in fluid power systems as well as leading to increased fatigue in system components. A number of different approaches, including passive and active techniques, can be used to reduce the FBN in hydraulic systems. Normally, these can be categorized as follows [1]:

- Reduction of pump or motor flow ripple;
- Tuning of the circuit in order to avoid resonant conditions;
- Use of a silencer or pulsation damper;
- Use of flexible hoses and accumulators.
- Active FBN control device;

Passive systems or components, such as silencers, pulsation dampers, accumulators and flexible hoses, have been shown to be effective to reduce FBN. They are generally reliable and simple. However, they require tuning to specific systems and their attenuation frequency range is limited and they may be bulky for some applications. Furthermore, attenuation devices based on expansion chambers, accumulators or hoses are likely to be unsuitable for high dynamic response systems as they add compliance to the system and would impair the dynamic response.

Active devices can potentially be effective at a much wider range of frequencies and system designs without significantly affecting the system dynamic response. They are successfully applied in the area

of ABN and SBN cancellation [2-5]. The principle is to use the intentional superposition of waves to generate a destructive interference pattern in order to attenuate the unwanted noise. Some applications have been proposed and applied for the purpose of FBN cancellation using active noise control (ANC) methods [6-15]. Klees filed for a patent in 1967 to introduce ANC principle for attenuating the periodic motion in a fluid conduit [7]. The experimental results were promising but with some limitations due to the restriction of the hardware in 1960s. Kojima *et al*. proposed an active noise attenuator for FBN in filled-fluid piping systems, and good cancellation results were achieved [8]. The method used an upstream real-time progressive wave signal to generate an anti-phase control signal to attenuate the FBN using the principle of wave superposition. This approach was highly reliant on the accuracy of the real-time progressive wave, and the inaccuracy of the estimation of the progressive wave could result in poor noise attenuation and system instability. A novel method of using the unsteady flowrate to estimate the progressive wave was proposed by Johnston and Pan [9]. The unsteady flowrate measurement was based on wave propagation analysis by using the measured pressures as boundary conditions [10]. This can be used as an enhancement to Kojima's method [8]. Maillard designed a fluid wave actuator which was arranged in series with the noise source to attenuate FBN in a piping system where the filtered-X least mean square (FXLMS) control algorithm was applied. The actuator consisted of a circumferential ring of Plumbum (lead) Zirconate Titanate (PZT) stacks acting on the pipe to generate an axisymmetric plane wave in the fluid through radial motion coupling. Experimental results showed good FBN cancellation [11]. Wang and Johnston proposed an active FBN attenuator using a servo-valve as an anti-noise generator [12]. It proved that the FXLMS algorithm with an estimated secondary path $S(z)$ could attenuate the FBN in a conventional hydraulic system. However, the experimental validation was based on a servo valve which limits the cancellation to the low frequency range. Also the servo valve was sensitive to contamination. The active FBN control principle is firstly implemented on a switched inertance hydraulic system in [13, 14]. A rectangular-wave reference signal was used in [15] and a sinusoidal-wave reference signal was used in [14]. Results show that very good cancellations were achieved, and the controller is capable of performing robustly with changing conditions.

2. Hybrid fluid-borne noise control system

A promising ANC method for FBN attenuation is to sense the system pressure ripple and to generate a secondary anti-pressure source to superimpose a compensating pressure wave to attenuate the pulsation. In the condition of periodic narrowband noise, such as that produced by pumps, electric motors and digital switching valves, an adaptive method can predict the noise harmonics and signal bandwidth, based on the characteristics of the source, and has the benefit of requiring only one sensor measurement. With a clever system design based on the combination of passive control approaches and active attenuators, the FBN can be reduced while the system still maintains good dynamic response, as shown in figure 1.
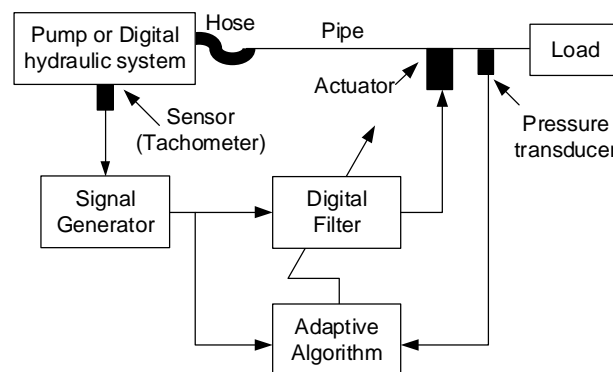

Figure 1. Hybrid (active and passive) fluid-borne noise control

2.1. Active feedforward noise controller

FBN generated from hydraulic systems can generally be divided into broadband and narrowband according to the energy distribution. Broadband noise can be introduced by the instability and cavitation of valves, while narrowband noise is caused primarily by unsteady flow from pumps and motors. The mechanism of pumps and motors generates the periodic noise in the system. Modern fluid power techniques, such as switched hydraulic systems, electrohydraulic actuators and digital hydraulics, are also facing the noise problem because of their natural design or control techniques.

The periodic fluid-borne noise is modelled by using a series of sinusoidal waves with the frequency of $200n$ (n=1 to 5). The amplitude of each frequency of the FBN is 3bar, 0.8bar, 0.7bar, 0.6bar and 0.5bar. For a conventional pumping system, the fundamental frequency is determined by the number of pumping section and motor driving speed. For a switched hydraulic system, the fundamental frequency is determined by the switching frequency of the high-speed switching component.

Figure 2 shows the implementation of a narrowband feed-forward FXLMS algorithm with a piezoelectric valve on a fluid-filled pipeline. The FXLMS algorithm is applied on the fluid-filled pipeline to attenuate the periodic noise. The details of the FXLMS algorithm can be found in [16, 17] and it has been successfully applied for pressure pulsation cancellation in a switched hydraulic system recently [14, 15]. A piezoelectrically actuated hydraulic valve (described in the next section) is used to generate the anti-noise control signal. The Fast Block Least-mean Square (FBLMS) algorithm operating in the frequency domain is used for the secondary path $S(z)$ identification. The secondary path $S(z)$ considers the dynamics of the piezoelectric valve and the couplings connected to the primary fluid path. In practice, the measured pressure signal $e(n)$ usually contains some components uncorrelated with the reference noise $v(n)$, which can cause problems with the on-line identification when using a time domain filter [17]. The corrupted estimated $S(z)$ can increase the convergence time of the cancellation filter and result in identification instability. The FBLMS in the frequency domain can substantially improve the robustness of the identification process. The details of the FBLMS algorithm can be found in [16, 17].
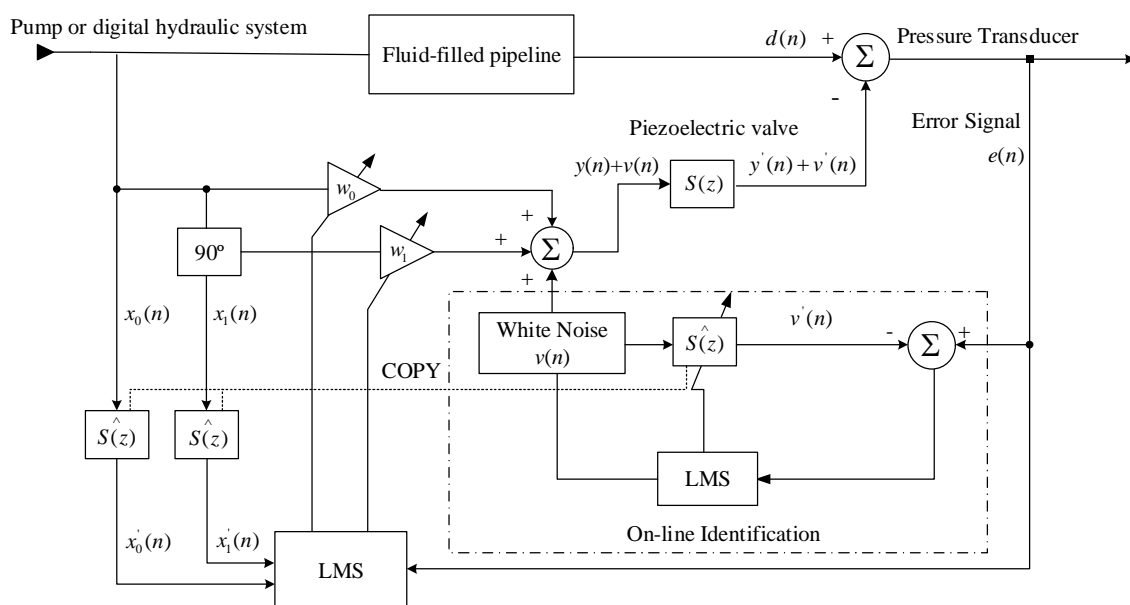
Figure 2. Implementation of the narrowband feed-forward FXLMS algorithm on a fluid-filled pipeline

2.1.1 Piezoelectrically actuated hydraulic valve

The principle and design of this piezoelectric actuated valve were described in [18]. The Horbiger plate method, which is based on the use of annular grooves in two mating plates, was applied to form multiple metering edges for allowing large flow path areas at relatively small plate separation distances. Figure 6 shows the flow-pressure characteristics of the valve with an applied voltage of 400V.

The pressure-flow rate relationship for the piezoelectric valve can be described as:

$$q = 0.00634\,p^5 - 0.15412\,p^4 + 1.45355\,p^3 - 6.76936\,p^2 + 16.53784\,p - 7.96590 \qquad (1)$$

where p is the pressure difference and q is the flow rate. The detailed valve model (including the amplifier) can be found in [19]. The high bandwidth valve (~1000Hz) is modelled using a second-order transfer function S(s) based on the previous experimental study [19, 20].

$$S(s) = \frac{\omega^2}{s^2 + 2\zeta\omega s + \omega^2} \qquad (2)$$

where the natural frequency $\omega = 2000\pi$ rad/s and the damping ratio $\zeta = 0.8$.

Figure 3 shows that simulated and experimental results agree very well. This valve model is used in the simulation to represent the secondary path anti-noise generator.
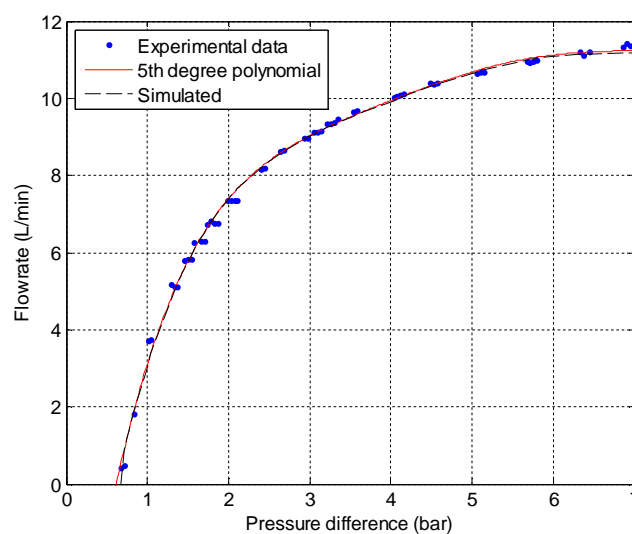
Figure 3. Steady state characteristics for the piezoelectric valve with an applied voltage of 400V
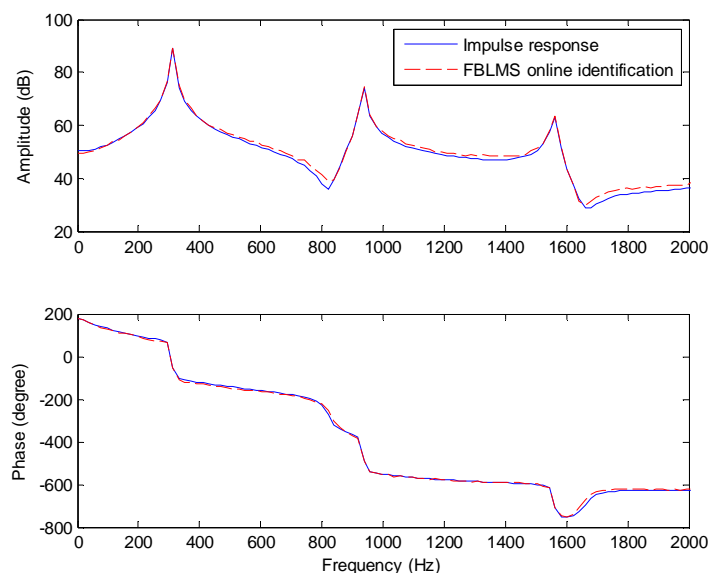
2.1.2 Secondary path identification and noise cancellation

The schematic of the piping system is shown in figure 4, where the secondary path is considered from position p_2 to p_1. The loading valve controls the downstream boundary condition of the pipe. The pipe is modelled using the Transmission Line Method (TLM), which makes use of the inherent delay in transmission of pressure and flow from one end of the line to the other. The length of the pipe is 1m, which is composed of two short tubes, 0.75m and 0.25m, in order to fit in the by-pass control branch. The pressure transducer is arranged 0.12m downstream from the by-pass junction. A high downstream impedance 8.25×10^{13} kg/m^{4}s is applied to represent the almost-closed end condition.
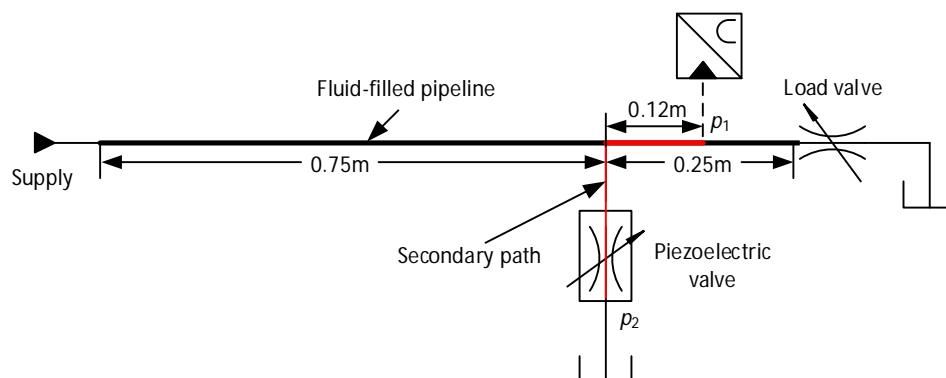
Figure 4. Schematic of the fluid-filled pipeline

Figure 5 shows the secondary path identification result using the online identification technique based on the FBLMS algorithm. The length of the identification filter was 512 and the sample frequency was 10kHz. The result was compared with this using the Fourier transform of the impulse response. Good agreement is obtained. A shorter length filter can be used to achieve faster computation speed, but it may result in some inaccuracy.


**Figure 5.** Secondary path identification

Figure 6 shows the FBN cancellation results with the active controller in the time domain and frequency domain. Table 1 lists convergence rates applied in the adaptive notch filters for different frequency components and the amplitudes of the original pressure and the total cancellation. The maximum cancellation of 58.3 dB was achieved at the frequency of 600 Hz, and the average cancellation was over 40dB. It can be noted that the noise at the frequencies of 310, 930, 1550, 2170 and 2790 Hz increased after the cancellation. This is because of the white noise which is amplified at the system's resonant frequencies which occur at $f = nc/4L$ (n=1, 3, 5…). A passive tuned flexible hose will be applied to attenuate the noise at these harmonics.

Table 1. Filter convergence rates and cancellation results for different harmonics

Frequency (Hz)	200	400	600	800	1000
Original pressure pulsation (dB)	109.5	98.0	96.9	95.5	93.8
After cancellation (dB)	54.3	56.3	38.6	38.4	39.7
Total cancellation (dB)	55.2	41.7	58.3	57.1	54.1
Convergence rate	1×10^{-21}	1×10^{-21}	1×10^{-21}	1×10^{-21}	1×10^{-21}
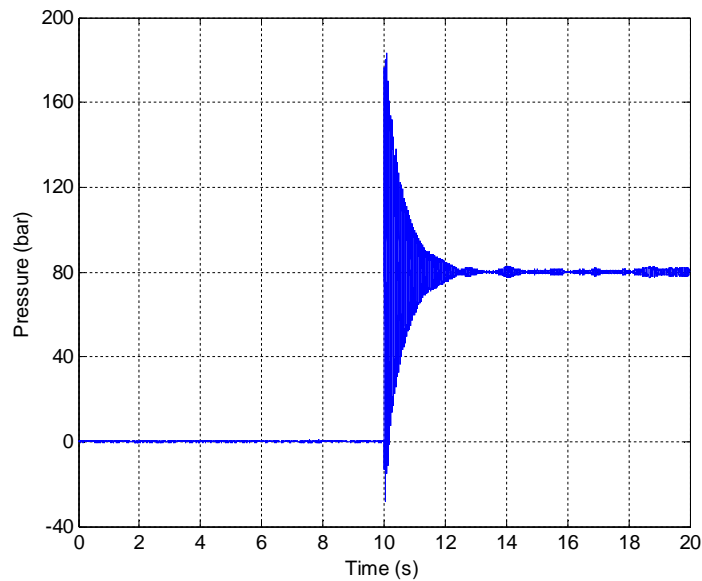
(a) Time domain result
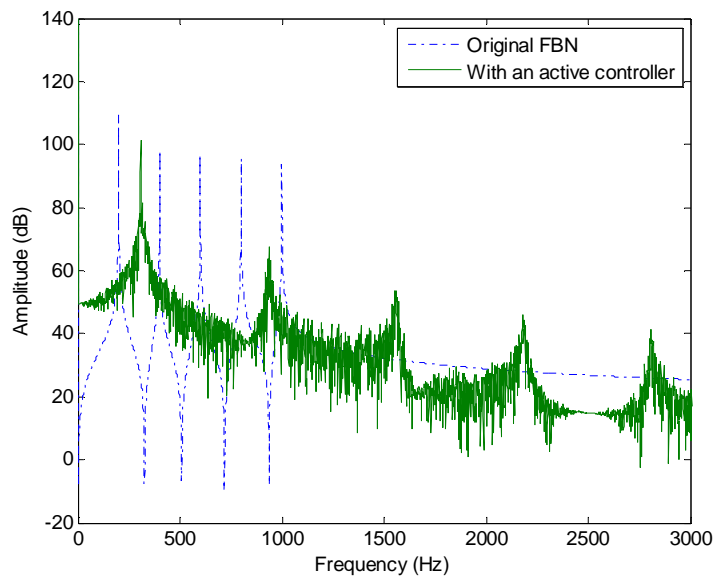
(b) Frequency domain result

Figure 6. FBN cancellation with an active noise controller

2.2. Passive tuned flexible hoses

Flexible hoses have an important effect on the dynamics and noise of many hydraulic fluid power systems. Flexible hoses are known to exhibit complex wave propagation characteristics. Because of coupling between the longitudinal motion in the fluid and in the hose wall, two sets of waves occur, travelling at different speeds, both in the fluid and in the wall [21]. The effective bulk modulus, stiffness and Poisson's ratio have an important effect on the dynamic behavior, and these depend in a complex way on the construction of the hose. In addition, the viscoelastic materials in the hose wall introduce a high level of damping, which can benefit the FBN reduction.

Traditionally hoses have been modelled as simple pipes with a reduced bulk modulus. This is adequate in many situations but is only accurate for a very limited bandwidth and does not capture the true behavior of waves in the hose. Taylor developed a comprehensive model based on a one-dimensional finite element method (FEM), and includes pressure-dependent properties and viscoelastic damping [22]. Johnston developed a flexible hose model based on the transmission line method (TLM). The two modes of wave propagation are modelled in detail, and corrections are included for viscoelastic relaxation and non-linearities [21]. This time-domain hose model is applied here to capture wide bandwidth dynamics.

The dynamic and static properties of a wide range of flexible hoses were presented in [23]. A double layer steel braid hose is selected and used in this work, as described in table 2. The hose properties and wave properties are shown in table 3, which are used to estimate the wave propagation constants of the hose models. Neglecting viscous friction [21, 24], the propagation constants γ_1 and γ_2 are given as:

$$\gamma_i = \omega \overline{\gamma}_i \tag{3}$$

$$\overline{\gamma}_i = \frac{j}{c_i} + \alpha_i \tag{4}$$

where $i = 1, 2$; c_i is the speed of sound and α_i is the damping constant.

Therefore, the amplitude and phase of $\overline{\gamma}_i$ can be calculated. The amplitude of $\overline{\gamma}_1$ and $\overline{\gamma}_2$ are 0.0017 s/m and 9.39×10^{-4} s/m and the phase of $\overline{\gamma}_1$ and $\overline{\gamma}_2$ are 88.8733 degree and 89.5424 degree, respectively. Firstly the hose is modelled in the frequency domain using '*Prasp*' [25] to investigate the tuned parameters, which are applied in the time domain model [21] to complete the passive control structure.

Table 2. Hose descriptions [23]

Description	Internal diameter $2r_b$ (mm)	Reinforcement diameter $2r_r$ (mm)	Rated pressure (bar)	Mass per unit length m (kg/m)
Double layer steel braid	12.9	17.0	293	0.53

Table 3. Wave properties and dynamic hose properties [23]

Wave properties	
c_1	596 m/s
α_1	3.3×10^{-5} s/m
c_2	1065 m/s
α_2	7.5×10^{-5} s/m
n_{21} (magnitude, phase)	0.432, -178.5 degree
Dynamic hose properties	
Bulk modulus B_H	17670 bar, 2.2 degree
Axial stiffness Ex	4.3
Poisson's ratio v_x	0.463, -0.8 degree
Fluid properties (from manufacturer's data)	
Density ρ	870 kg/m^3
Viscosity v	16 cSt
Bulk modulus B_F	13360 bar

Figure 7 shows the amplitude ratio of the system with and without hoses. The amplitude ratio is defined as the quotient of the pressure at the outlet and inlet points of the system. It shows the resonant frequencies of the tube, which decrease in frequency and magnitude when the hose is applied. Different lengths of the hose were implemented at the upstream of the system (figure 4). As can be seen, the

optimal hose length is about 0.8m, which has a low amplitude ratio in high frequency range and good performance at low frequency range. A longer hose may be used to achieve better attenuation at high frequencies, but it may cause higher pressure loss and add extra compliance. The FBN cancellation result with the hybrid control method is shown in figure 8, where the amplitudes at the resonant frequencies were effectively attenuated by the passive hose.
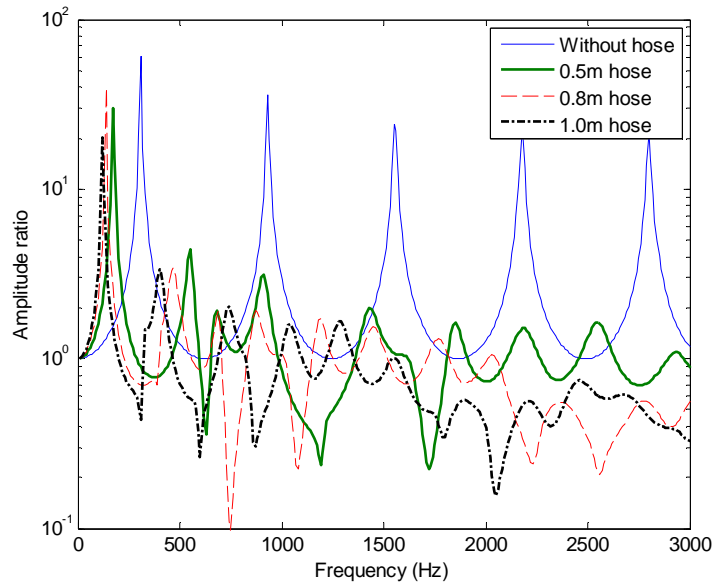

Figure 7. Amplitude ratio
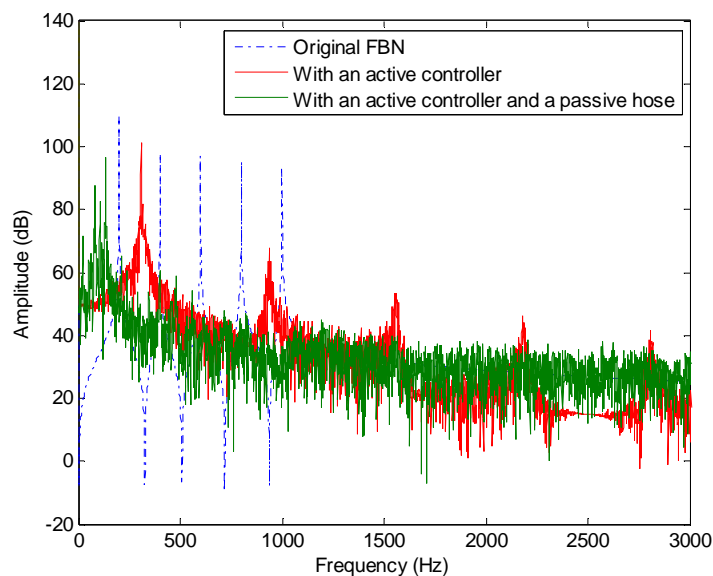
Figure 8. FBN cancellation with an active noise controller and a passive hose

It is crucial to keep a low overall amplitude ratio at high frequencies when a flexible hose is used and tuned. This can ensure the attenuation at a wide range of high frequencies. The strongest resonances should be considered and tuned in order to avoid excitation frequencies. This can reduce the power requirements of the active attenuator and avoid high noise levels when the attenuator is inactive. It can also improve controller stability because the system dynamics are less resonant at those frequencies. The tuning can be achieved using frequency domain analysis software such as *Prasp*, but it needs knowledge of the properties of the hose [24].

3. Simulation

The varying loading condition is modelled to test system stability and robustness. The downstream pipe impedance was varied from 8.25×10^{13} kg/m^{4}s to 8.25×10^{9} kg/m^{4}s (characteristic impedance, no reflection, lower pulsation expected) in 10s, which represents the loading flow rate gradually increased. The FBN cancellation results in the time domain and frequency domain are shown in figure 9, where the active attenuator adapts and performs robustly for the varying boundary condition. The adaptive weights of the filter for the fundamental frequency cancellation are plotted in figure 10, where w_1 remained at 3.97×10^{-5} and w_2 remained at -1.57×10^{-5} with very small oscillations. The total cancellation at all harmonics was over 50dB. The tuned hose effectively attenuated the high frequency FBN, as seen in figure 9 (b). Table 4 lists the cancellation results at different harmonics.
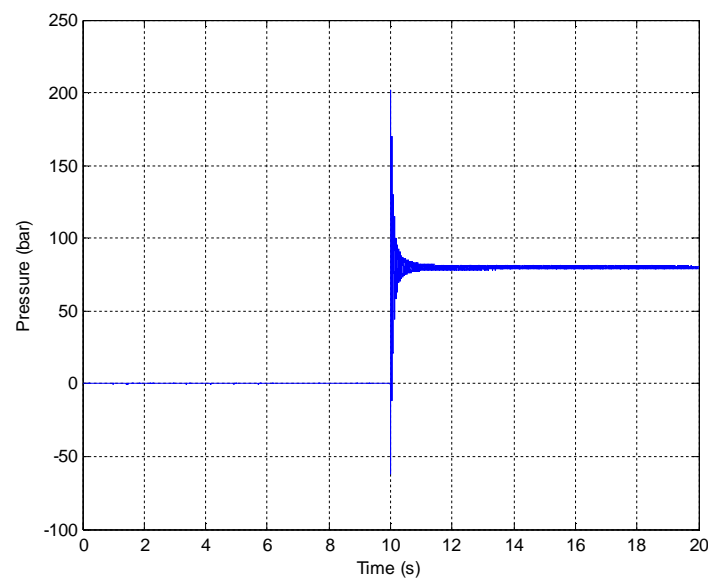

(a) Time domain result
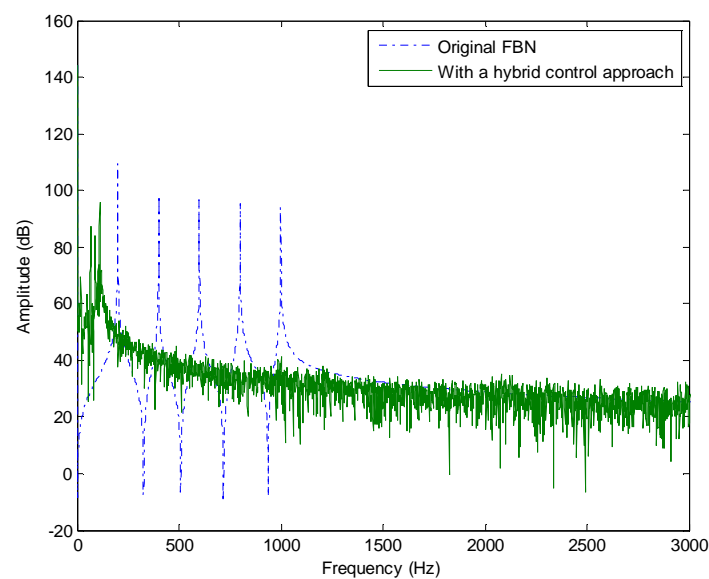
(b) Frequency domain result

Figure 9. FBN cancellation with an active noise

controller

9

This hybrid approach provides an effective FBN cancellation solution for a wide range of frequencies without impairing the system dynamic response.

Table 4. Results of cancellation at different harmonics

Frequency (Hz)	200	400	600	800	1000
Original pressure pulsation (dB)	109.5	98.0	96.9	95.5	93.8
After cancellation (dB)	48.2	42.7	34.3	35.7	41.3
Total cancellation (dB)	61.3	55.3	62.6	59.8	52.5
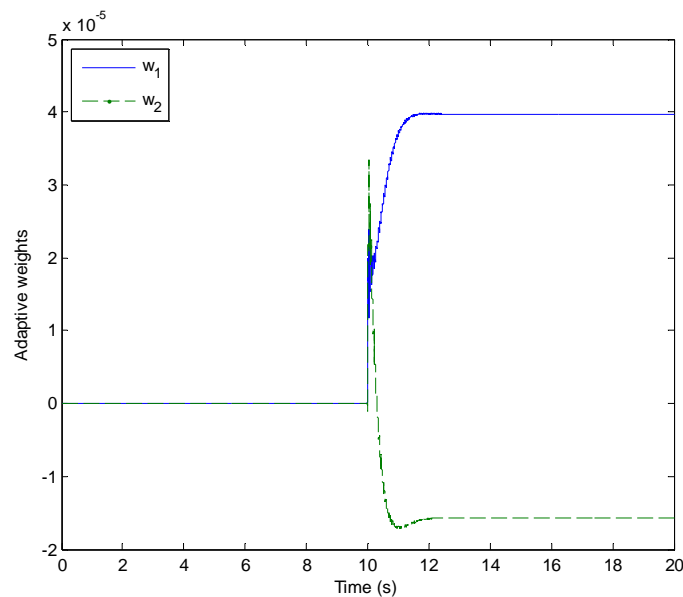
Figure 10. Adaptive weights of the filter for the fundamental frequency cancellation with a varying loading condition

4. Discussion

The active attenuator is proposed with the passive hose for FBN cancellation in the fluid-filled pipe. The secondary path actuator is effective at generating an accurate control signal to the pipe, and very good FBN cancellation was obtained at the downstream end. The performance of the active attenuator is robust with varying loading conditions. While the flexible hoses may be less effective than purpose-built passive silencers, they can provide an inexpensive and practical solution in combination with active control. Also, the hose is very essential to most hydraulic systems, and currently the hose length is selected or picked arbitrarily in most cases. The tuned hose can benefit the system noise reduction without any extra cost, and precise tuning is not needed. The following still needs to be considered:

- The power of the white noise for the secondary path identification needs to be considered in a reasonable range as extra broadband noise is introduced to the pipeline system. The secondary path actuator/valve may suffer from the broadband noise, which may cause component instability or damage due to noise interference.

- There is a significant increase in FBN due to the white noise at the system resonance of about 100Hz; however this could be reduced by adaptive filtering of the white noise to reduce its power at this frequency.

- The piezoelectric valve has been tested and applied as the secondary path valve in active FBN control study [14]. It has a limited maximum pressure and a constant noticeable leakage. A high bandwidth and force actuator/valve will be designed for further experimental study.

- The xPC system is considered to use as the real-time platform. The sampling frequency may be limited to below 10kHz because of the hardware computation ability. This can affect the accuracy the secondary path identification and the harmonics attenuated.

5. Conclusion

This paper proposed a hybrid control system for fluid-borne noise cancellation in a fluid-filled pipe. The hybrid system is composed of an active feedforward noise controller and a passive tuned flexible hose. The active attenuator is designed to cancel the dominant harmonic pressure pulsations in the fluid line, while the passive hose is tuned to attenuate the residual high frequency pulsations. Simulation results show that very good FBN cancellation was achieved using the hybrid control approach. The active attenuator performs and adapts effectively and robustly for low frequency noise cancellation and the tuned passive hose is able to attenuate high frequency noise. Discussion about the implementation of the proposed system in real time are presented. It can be concluded that the proposed hybrid FBN control system can be a very promising solution for FBN control in fluid-filled pipes. The hose can form a practical and inexpensive solution in combination with the active device.

References

[1]   Henderson, R. 1981 Quieter Fluid Power Handbook: Silencing Fluid-Borne Noise. *BHRA Fluid Engineering*. pp. 33-44.

[2]   Kuo, S and Morgan D. 1995 Active noise control systems: algorithms and DSP implementations. *John Wiley & Sons*, Inc. New York, NY, USA.

[3]   Bao C, Sas P and Van Brussel H. 1993 Adaptive active control of noise in 3-D reverberant enclosures. *Journal of Sound and Vibration*, vol. 161, pp. 501-514.

[4]   Zhang, M., H. Lan., and W. Ser. 2001 Cross-updated active noise control system with online secondary path modelling. *Speech and Audio Processing, IEEE Transactions* on Vol. 9(5), pp. 598-602.

[5]   Hillis, AJ, Harrison, AJL and Stoten, DP. 2005 A comparison of two adaptive algorithms for the control of active engine mounts. *Journal of Sound and Vibration*, vol. 286, pp. 37-54.

[6]   Wang, L. 2008 Active control of fluid-borne noise. PhD thesis, *University of Bath*, UK.

[7]   Klees, G. 1967. Attenuating device. US Patents, No. 3323305.

[8]   Kojima, E., Shinada, M. and Yamaoka, T. 1993 Development of an active attenuator for pressure pulsation in liquid piping systems (trial construction of the system and fundamental experiments on attenuation characteristics). *JSME international journal. Series B, fluids and thermal engineering*, 36, pp. 230-237.

[9]   Johnston, DN. and Pan, M. 2010 Use of Pipeline Wave Propagation Model for Measuring Unsteady Flowrate. *In: Proc. Fluid Power and Motion Control*, September, Bath, pp.307-320.

[10]   Johnston, DN., Pan, M, Kudzma, S. and Wang, P. 2014 Use of Pipeline Wave Propagation Model for Measuring Unsteady Flow Rate. *Journal of Fluids Engineering*, 136, no. 3 (2014): 031203.

[11]   Maillard, J, Lago, T. and Fuller, C. 1999 Fluid wave actuator for the active control of hydraulic pulsations in piping systems. *Proceedings of the International Modal Analysis Conference IMAC*, pp. 1806-1812.

[12]   Wang L and Johnston DN. 2008 Adaptive attenuation of narrow band fluid borne noise in a simple hydraulic system. *In: Bath/ASME Symposium on Fluid Power and Motion Control*. Bath. University of Bath, pp. 357-368.

[13]   Pan, M., Johnston, DN. and Hillis, AJ. 2012 Active control of pressure pulsation in a switched inertance hydraulic system using a rectangular-wave reference signal. *In: Bath/ASME Symposium on Fluid Power and Motion Control*, 2012-09-12 - 2012-09-14, Bath. UK.

[14]   Pan, M., Johnston, DN., and Hillis, AJ. 2013 Active control of pressure pulsation in a switched inertance hydraulic system. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering* vol. 227(7), pp. 610-620.

[15]   Pan, M., Johnston, DN. and Hillis, AJ. 2012. Active control of pressure pulsation in a switched

inertance hydraulic system using a rectangular-wave reference signal. *In: Bath/ASME Symposium on Fluid Power and Motion Control*, 2012-09-12 to 2012-09-14, Bath. UK.

[16] Haykin, S. 1996 Adaptive filter theory. Eaglewood Cliffs, *NJ: Prentice Hall*, Inc.

[17] Kuo, S. and Morgan, D. 1995 Active noise control systems: algorithms and DSP implementations, *John Wiley & Sons, Inc*. New York, USA.

[18] Branson DT, Johnston DN, Tilley D, *et al*. 2010 Piezoelectric actuation in a high bandwidth valve. *Ferroelectrics* Vol 408: pp. 32–40.

[19] Branson DT, Wang FC, Johnston DN*, et al*. 2011 Piezoelectrically actuated hydraulic valve design for high bandwidth and flow performance. *Proc IMechE, Part I: J Systems and Control Engineering* Vol 225: pp.345–358.

[20] Pan, M. 2012 Active control of pressure pulsation in a switched inertance hydraulic system. PhD thesis, *University of Bath*, UK.

[21] Johnston, D.N., 2006 A time-domain model of axial wave propagation in liquid-filled flexible hoses. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 220(7), pp.517-530.

[22] Taylor, SEM. 1998 Development of numerical models for hydraulic pipelines and flexible hoses. PhDThesis, *University of Bath*, UK.

[23] Johnston, D.N., Way, T.M. and Cone, K.M., 2010 Measured dynamic properties of flexible hoses. *Journal of Vibration and Acoustics*, 132(2), p.021011.

[24] Drew, J.E., Longmore, D.K. and Johnston, D.N., 1998 Theoretical analysis of pressure and flow ripple in flexible hoses containing tuners. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 212(6), pp.405-422.

[25] Johnston, D.N., 2007 Pressure ripple analysis software Package PRASP: user guide and reference manual. *Centre for Power Transmission and Motion Control*, University of Bath. UK.