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BROADBAND ACTIVE CONTROL OF FLUID-BORNE SOUND IN A PIPELINE USING AN ACTUATED ORIFICE PLATE

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Abstract

Pipeline systems are used for the transportation of fluids in a wide range of engineering applications such as water distribution, oil pipelines and marine vessels. Noise from pumps and compressors can propagate efficiently through such systems, often resulting in sound being radiated far from the original source. Existing passive treatments are widely used in industry to minimise the effect this has on radiated noise, but the effective frequency range is limited by mass and size constraints. In contrast, active control systems, that use a secondary source to cancel the sound or vibration from a primary source, generally exhibit high performance at low frequencies. The inclusion of a secondary actuator can, however, adversely affect the fluid flow in the system or require complex modifications to the pipeline. This paper proposes a potential solution where the secondary actuator is integrated into an orifice plate, which are commonly used to reduce pressure or restrict flow in pipeline systems. The actuated orifice plate is used in an optimal feedforward control system to minimise the downstream broadband sound pressure. This work serves as a proof of concept that such an active orifice plate has the potential for controlling fluid-borne noise in pipeline systems.

Keywords: *active control, pipeline, fluid-borne noise, orifice plate*

1. Introduction

Pipeline systems are used for the transportation and distribution of fluids with widespread industrial applications, including marine vessels, oil pipelines and water distribution. Many of the components used in pipeline systems, such as pumps and compressors, generate significant levels of noise that cannot be easily attenuated without compromising performance. At low frequencies relative to the cross-sectional dimensions of a pipe, sound propagates in plane waves, meaning that propagation occurs entirely in one dimension and geometrical spreading does not occur. This results in very little attenuation due to the distance the wave has travelled from the sound source. Therefore, sound can efficiently propagate over large distances in pipeline systems and can then be radiated from the end of the system or via structural interaction at some position downstream from the sound source. The radiation of noise is often undesirable, since it can result in annoyance, discomfort and disruption for both humans and local wildlife. It is therefore of significant interest to develop methods for the reduction of sound transmission through pipeline systems.

Traditionally, passive methods for the reduction of pipeline noise are employed, such as the use of pulsation dampers [1], which can be used to reduce the level of sound transmitted from a periodic source such as a pump or compressor. This is achieved by coupling the fluid flow with a flexible membrane, bellow or piston which deforms to absorb the pressure fluctuations. The drawback of this approach is that it impedes the flow through the pipe and results in a loss of static pressure. Another strategy, which has no impact on the flow in the pipe, is to use viscoelastic damping treatment to attenuate structural vibration, since pipe vibration and fluid-borne sound are closely coupled [2]. However, this approach is only effective over a limited frequency range and often adds significant mass and volume to the structure and may be completely ineffective when the sound is transmitted via the fluid in the pipe and then radiated from the pipe termination.

Active control offers an alternative approach to controlling sound in pipelines, where a secondary acoustic source is used to produce sound that destructively interferes with the unwanted sound, resulting in significant attenuation. This concept has been applied to fluid-borne pipeline noise by Kojima et al. [3], who used a servo-actuated piston installed in a side-branch of the main pipe as a secondary actuator. The feed-

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forward control system they implemented achieved significant attenuation of tonal disturbances, but the actuator affects the flow within the pipeline system which may not be desirable. Later work [4] implemented a fluid-wave actuator that excited sound waves within the fluid inside a perspex pipe by applying a constrictive force to the pipe itself, thus achieving control without impeding the flow within the pipe. One drawback of this approach, as well as later approaches applied to stiffer, steel pipe sections [1], is that there exists a trade-off between controlling the fluid-borne sound and enhancing the structural vibration. The vibrations in the structure could then radiate both into the fluid inside the pipe, and also into the external environment, potentially creating a new problem. In an attempt to address this, a multi-channel control strategy has been implemented that accounts for both the internal fluid-borne sound and the externally radiated sound pressure [5]. This system demonstrated that the level of enhancement in the externally radiated sound can be limited at the expense of a degradation in the performance of the downstream fluid-borne sound control. More recently, several different actuator designs have been applied to the problem of actively controlling fluid-borne sound in a pipe, including a fast-response valve [6] and a hybrid active-passive actuator that combines a stacked piezoelectric fluid actuator with a perforated plate [7]. However, none of the approaches listed here achieve control of fluid-borne sound without affecting the fluid flow or enhancing the structural vibration of the pipe. This paper proposes an Actuated Orifice Plate (AOP) as an acoustic control source in a feedforward, broadband active control strategy and presents an initial feasibility study into the control potential of this device. Orifice plates are commonly used to control the pressure and flow rate within pipeline systems so, although the actuator is intrusive, it can be built into existing components in a pipe system without increasing the overall footprint. The experimental set-up used to assess the performance of the AOP is detailed in Section 2, along with the AOP design. Section 3 outlines the calculation of the optimal control filter, before Section 4 presents the results of control simulations using two different reference signals. Finally, Section 5 draws conclusions from the paper.

2. Experimental set-up

2.1. Pipeline set-up

In order to assess the suitability of the AOP design for active noise control applications, the actuator has been installed within a water-filled circulatory pipe flow rig as shown in Fig. 1. The system consists of a straight pipe section with hydrophones placed both upstream and downstream of the orifice plate, which are used as reference and error sensors respectively. The upstream hydrophones are denoted H_1, H_2 and H_3, with H_3 closest to the orifice plate and H_1 furthest away. Likewise, the downstream hydrophones are denoted H_4, H_5, and H_6, with H_4 closest to the orifice plate and H_6 furthest away. For both the upstream and downstream sets of hydrophones, the spacing between individual sensors is 90 mm, as shown in Fig. 1. In this proof-of-concept study, the effects of flow in the system will not be considered. Therefore in the absence of a pump-generated acoustic disturbance, a DataPhysics IV-40 inertial actuator is attached externally to the pipe wall upstream of the orifice plate to act as the primary acoustic disturbance.

2.2. Actuated orifice plate

The AOP consists of four Visaton EX 30 S electrodynamic inertial actuators housed within a sealed aluminium enclosure, as shown in Figure 2. The design has filleted edges on the upstream side to minimise the turbulent flow generated around the housing. The actuator housing is attached to the perimeter of the circular orifice in the plate, in order to maximise the coupling of the inertial forces to the desired diaphragmatic motion of the orifice plate. This will in turn maximise the electro-acoustic efficiency and acoustic power output of the AOP. The AOP is installed between two flanged pipe sections with rubber gaskets used to both ensure water tightness and to mechanically isolate the AOP from the pipe structure.

3. Broadband optimal control

In pipe systems where the noise from a pump or compressor is dominated by discrete tonal components, a tonal controller can be employed [3]. However, this paper considers the case of a broadband disturbance where a broadband controller is required. Here, a leaky FxLMS feedforward control algorithm will be used due to its ease of implementation, relative robustness to challenges associated with practical implementation and its widespread use as a benchmark algorithm. In this study, a single channel FxLMS algorithm is utilised with the optimal control filter calculated based on measurements of the system. The single channel system uses a single error sensor and single control source, with the error signal provided by the downstream hydrophone H_4, and all four actuators in the AOP driven in-phase to provide a single effective control source.

The error signal at the n-th sample, $e(n)$, can be expressed as

$$e(n) = d(n) + \boldsymbol{w}^{\mathrm{T}}\boldsymbol{r}(n), \tag{1}$$

where $d(n)$ is the disturbance signal, $\boldsymbol{w}$ is the vector of control filter coefficients with length I and $\boldsymbol{r}(n)$ is the vector of current and $I-1$ past values of the reference signal, $\boldsymbol{x}(n)$, filtered through the plant model, $\boldsymbol{g}$. A block diagram representation of the control strategy is shown in Fig. 3. The cost function of the leaky FxLMS controller is defined as

$$J = E\left[e^2(n)\right] = \boldsymbol{w}^{\mathrm{T}}\boldsymbol{A}\boldsymbol{w} + 2\boldsymbol{w}^{\mathrm{T}}\boldsymbol{b} + c + \beta\boldsymbol{w}^{\mathrm{T}}\boldsymbol{w} \tag{2}$$

where E denotes the expectation operator,

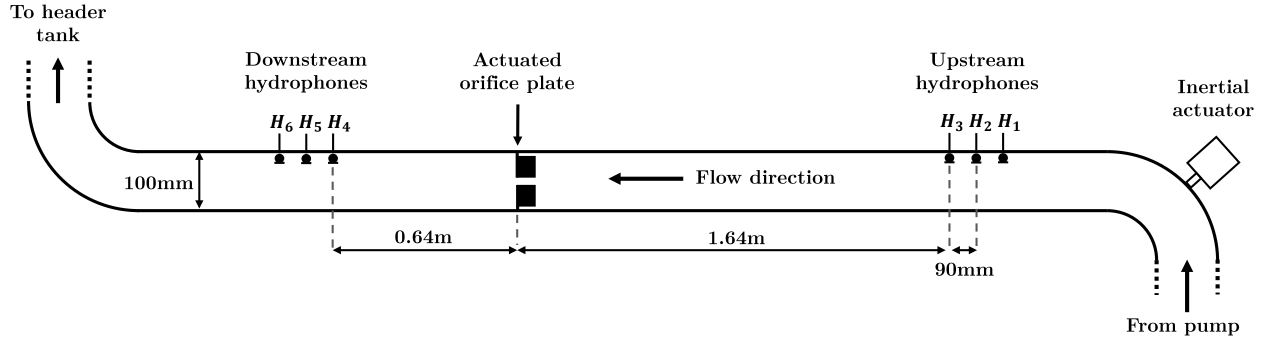

Figure 1: Diagram of the pipeline set-up, showing the position of the AOP, the six hydrophones used to monitor the fluid-borne sound and the inertial actuator used to provide a primary acoustic disturbance.

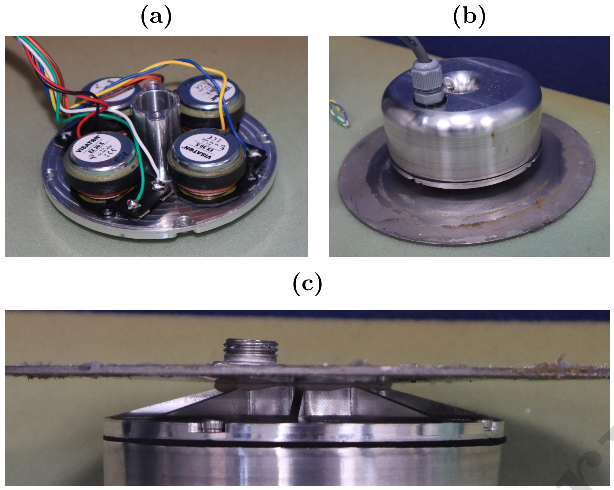

Figure 2: Pictures of the AOP showing (a) the arrangement of the four actuators within the enclosure, (b) the sealed enclosure on the orifice plate and (c) the minimised connection area between the housing and the orifice plate.

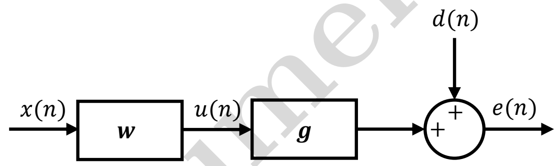

Figure 3: Block diagram of the single channel feedforward control system, showing the reference signal, $x(n)$, disturbance signal, $d(n)$, error signal, $e(n)$, control signal, $u(n)$, control filter, $\boldsymbol{w}$ and the system plant, $\boldsymbol{g}$.

$\boldsymbol{A} = E\left[\boldsymbol{r}(n)\boldsymbol{r}^{\mathrm{T}}(n)\right]$, $\boldsymbol{b} = E\left[\boldsymbol{r}(n)d(n)\right]$, $c = E\left[d^2(n)\right]$ and β is a regularisation parameter used to constrain the control effort, $\boldsymbol{w}^{\mathrm{T}}\boldsymbol{w}$. The optimal control filter coefficients that minimise this cost function can then be calculated as [8]

$$\boldsymbol{w}_{\mathrm{opt}} = -\left\{E\left[\boldsymbol{r}(n)\boldsymbol{r}^{\mathrm{T}}(n)\right] + \beta\mathbf{I}\right\}^{-1} E\left[\boldsymbol{r}(n)d(n)\right]. \quad (3)$$

where $\mathbf{I}$ is the $I \times I$ identity matrix.

4. Experimental results

In order to investigation the control performance for the system described in Section 2 via offline simulations, the acoustic response of the system to the primary acoustic disturbance and to the AOP have been measured experimentally. A sampling frequency of 51.2 kHz has been used to minimise delays in the system and to ensure that the relative time advance of the reference signal is sufficiently large. The signals from the hydrophones have been filtered through anti-aliasing low-pass filters at half the sampling frequency. From the measured responses, the MATLAB function *invfreqz* is used to calculate FIR filters that approximate the response of the system. With H_4 defined as the error sensor, these filters are then used to calculate the optimal control filters according to the leaky FxLMS algorithm outlined in Section 3. The regularisation term, β, is set to constrain the control force to ensure the control actuators are operating within their linear dynamic range and protected from overheating. The resulting control performance is then evaluated by calculating the error signal using Eq. 1. Although only H_4 is used as an error sensor in the control strategy, it is important to also consider the response at H_5 and H_6 to ensure that attenuation is not limited to the location of the error sensor. It is also worth noting that in a practical setting, a multichannel or wave-based controller could be implemented to achieve a more global control of downstream noise.

4.1. Actuator input reference

In the first instance, the input signal to the inertial actuator used to generate the acoustic disturbance is used as the reference signal in the control simulations, giving the maximum possible time advance, as well as good coherence with the error signal. The resulting power spectral density (PSD) of the downstream hydrophone responses without control (solid) and with control (dashed) are shown in Fig. 4. It can be seen that the acoustic pressure at the error sensor, H_4,

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is significantly attenuated across the majority of the frequency range, achieving up to 30 dB of attenuation at higher frequencies. The performance of the controller is, however, limited at lower frequencies, where the level of attenuation is lower and some frequencies experience enhancement. Below around 500 Hz, the attenuation seen at the other two hydrophones, H_5 and H_6, is comparable to that at H_4. However, as frequency increases above 500 Hz the attenuation at H_5 and H_6 decreases, meaning that the controller is only achieving local attenuation.

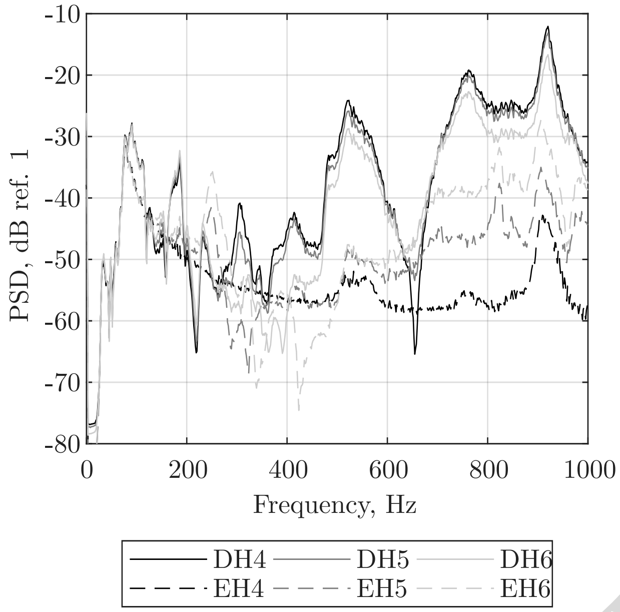

Figure 4: PSDs of the downstream hydrophone responses without control (solid) and with control (dashed) for the controller using the disturbance actuator input signal as a reference.

4.2. Upstream hydrophone reference

Although using the acoustic disturbance actuator input as the reference signal has demonstrated good performance, in most practical broadband noise control applications such a signal will not be available. Therefore, the use of a practically realisable reference signal in the form of one of the upstream hydrophones is explored here. Fig. 5 shows the PSDs of the downstream hydrophone responses without control (solid) and with control (dashed) from the implementation of the controller using H_3 as a reference signal. H_3 has been selected because it provided the highest coherence with the error signal compared to the three upstream hydrophones. It can be seen from these results that the performance of the controller has been significantly degraded compared to the idealised reference signal case, particularly below 500 Hz, where very little attenuation is achieved. As frequency increases, the controller exhibits up to 18 dB of attenuation at the error sensor when the disturbance level is high, but also shows significant enhancement when the disturbance level is low. However, observation of the acoustic response at H_5 and H_6 suggests that this controller achieves more uniform downstream attenuation compared to the actuator input case.

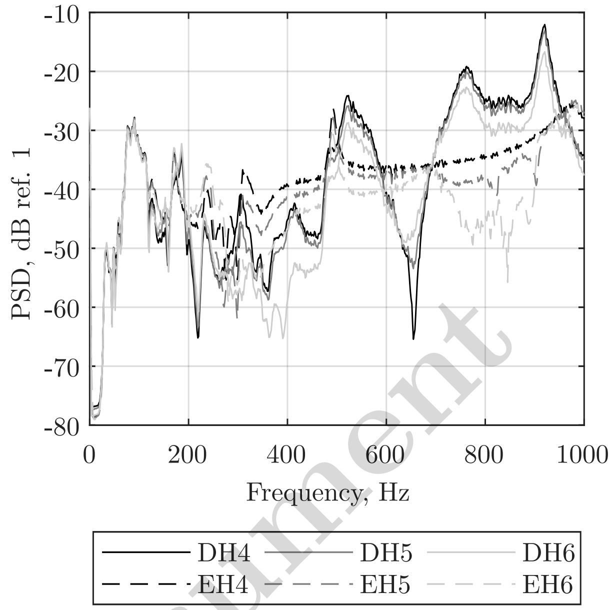

Figure 5: PSDs of the downstream hydrophone responses without control (solid) and with control (dashed) for the controller using H_3 as a reference signal.

4.3. Reference signal coherence and cross-correlation

The coherence between the reference and error signals in a feedforward control system is known to limit control performance [8], so it is useful to compare the coherence for the two reference signals considered in the previous sections. Figure 6 shows the coherence between the error signal and each of the considered reference signals. From these results it can be seen that the coherence is relatively consistent for the two reference signals, with a slight degradation at low frequencies for the H_3 hydrophone reference signal. These results suggest that the performance degradation when using the H_3 hydrophone signal as the reference is not due to poor coherence. Instead, it is likely that the shorter time advance provided when using the H_3 reference is responsible for the reduction in attenuation performance.

The difference in time advance can be demonstrated using the cross-correlation between each of the reference signals and the error signal. For the H_3 reference case the time advance is 2.2 ms between the reference signal and the error signal. In the case of the actuator input reference the time advance is 4.8 ms, meaning that this reference signal has approximately double the time advance of the H_3 signal. This accounts for the significant drop-off in performance observed when the upstream hydrophone, H_3, is used as a reference signal. This also means that, for a longer system where an upstream hydrophone could be placed further from the secondary source in order to increase

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the time advance of the reference signal, the control performance could be significantly increased using a practical reference signal.

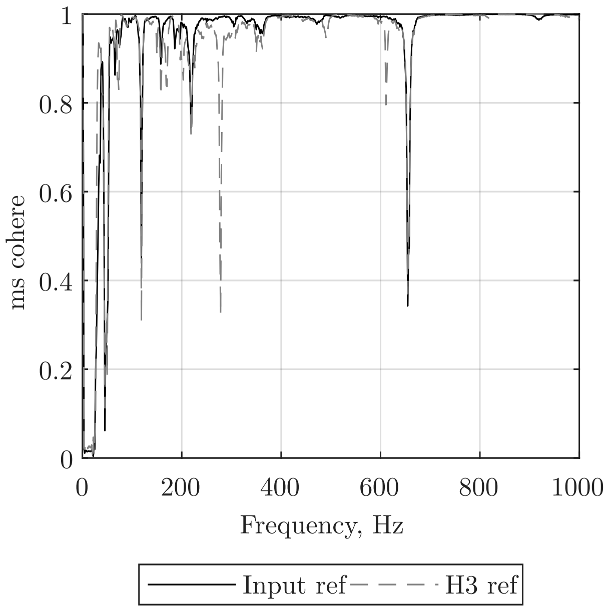

Figure 6: Coherence between the reference and error signals for reference signal considered: the actuator input and the signal from H_3.

5. Conclusions

The work reported in this paper has demonstrated the potential of an Actuated Orifice Plate (AOP) for the active control of fluid-borne sound in a pipeline. Although the objective of the proposed AOP is to control flow-induced noise, in this preliminary study a broadband disturbance is provided by an inertial actuator fixed externally to the pipe wall. Measurements of the experimental set-up have been taken and used to calculate optimal filter coefficients for a leaky FxLMS controller in a single-channel feedforward configuration. With the disturbance actuator input used as the reference signal, significant attenuation of the primary disturbance is achieved at the error sensor. The sound pressure at the two hydrophones further downstream is also attenuated, although to a lesser degree, demonstrating that the control performance is not limited to a single discrete position in the pipeline. Implementation of the control strategy using the more practical case of a hydrophone positioned upstream of the AOP to provide the reference signal achieves some control, but it is notably more limited than the case with an idealised reference signal. This has been shown to be due to the limited time advance available in this case. However, the controller still achieves up to 18 dB of attenuation at the frequency where the level of the disturbance is highest. The control configuration with a hydrophone as the reference signal also shows more uniform control across the three downstream hydrophones at higher frequencies.

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References

[1] J. Maillard, "Active control of pressure pulsations in piping systems," University of Karlskrona, Tech. Rep., 1998.

[2] G. Pavic, "Vibroacoustic energy flow through straight pipes," *Journal of Sound and Vibration*, vol. 154, no. 3, pp. 411–429, 1992.

[3] E. Kojima, M. Shinada, and T. Yamaoka, "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*, vol. 36, no. 2, pp. 230–237, 1993.

[4] M. J. Brennan, S. J. Elliott, and R. J. Pinnington, "A non-intrusive fluid-wave actuator and sensor pair for the active control of fluid-borne vibrations in a pipe," *Smart Materials and Structures*, vol. 5, pp. 281–296, 1996.

[5] J. Cheer and S. Daley, "Broadband active control of noise and vibration in a fluid-filled pipeline using an array of non-intrusive structural actuators," *Proc. of 44th Int. Con. and Exp. on Noise Control Engineering*, 2015.

[6] C. Wu, Z. Jiao, Y. Xu, C. Li, Q. Liu, and S. Wu, "Active control method for fluid borne noise in aerospace fluid systems of variable operation states," *Mechanical Systems and Signal Processing*, vol. 214, 2024, ISSN: 111375.

[7] G. Cheng, Q. Liu, and W. Yu, "Development and active noise control of a piezo-actuated attenuator for underwater vehicle pipelines," *Smart Materials and Structures*, vol. 34, 2025, ISSN: 125027.

[8] S. Elliott, *Signal Processing for Active Control*. Academic Press, 2001.

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