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Broadband active control of noise and vibration in a fluid-filled pipeline using an array of non-intrusive structural actuators

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Fluid-filled pipelines occur in a wide variety of installations where they are excited by sources such as pumps and compressors through both the structure of the pipe wall and the fluid in the system. These vibrations may propagate downstream and result in unwanted radiated noise from the pipeline or connected structures. Although passive measures are available to control both structure and fluid borne vibration, their operating frequency range is generally limited by the volume and weight constraints that are imposed by the practical installation. Active control methods may overcome these volume and weight constraints and such systems have been investigated for the control of both fluid and structure borne transmission. In particular, previous work has highlighted the ability to control the internal acoustic pressure using non-intrusive fluid-wave actuators. However, although these systems are able to achieve control of the internal acoustic pressure they produce enhancements in the structural vibration and this limits the practical benefit of these systems. This paper investigates the behaviour of a non-intrusive structural controller consisting of an array of piezo stack actuators positioned circumferentially around the pipe and acting on its outside wall. The ability of the system to control the internal, fluid-borne, acoustic pressure and the externally radiated sound pressure is investigated through a series of experiments. These results are then used to highlight the trade-off between controlling the internal and external pressures using the non-intrusive array of structural actuators.

1 INTRODUCTION

Pipeline systems are used in a variety of engineering applications for the transfer of fluids, for example, in the petrochemical and process industries or in marine vessels. Although the primary function of these pipeline systems is the transmission of fluids, they also provide an

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efficient mechanism for the transmission of unwanted noise and vibration throughout the network and any connected structures. This noise and vibration can be induced via a variety of mechanisms such as pump and compressor noise, or flow induced noise, and due to the fluid-structure coupling these excitations may be transmitted through both fluid- and structure-borne paths [1]. The transmitted vibration may then either be directly radiated as noise by the vibration of the pipe wall, or transmitted into connected structures and subsequently radiated as noise. In many applications this noise radiation is undesirable and, therefore, a number of noise control techniques have been developed.

Passive control of noise and vibration in fluid-filled pipelines can be achieved using a variety of methods depending on the source of noise and its transmission path. For example, lowering the flow velocity can reduce flow noise, however, to maintain the same volumetric flow rate this requires a proportional increase in the cross-sectional area of the pipe. This increase in pipe volume is often not possible, however, due to space constraints in many practical installations. Pulsation dampers have also been used to attenuate fluid pressure pulses generated by pumps, for example; however, these devices result in a loss in the static pressure, which may decrease system performance [2]. In terms of the structural vibration it is possible to use standard vibration absorbers to attenuate the levels of pipe vibration and resilient mounts to reduce transmission into connected structures. However, the bandwidth of these passive systems is typically limited and it is particularly difficult to achieve high performance at lower frequencies.

To overcome the limitations of passive control measures a number of active control technologies have been proposed for pipeline systems. For example, a simple active vibration absorber has been proposed in [3] which is able to reduce the levels of vibration in a pipeline system corresponding to the fundamental and second harmonic of the pump. Compared to the alternative passive absorber this active system is able to adapt to the rotational speed of the pump and, therefore, attenuate the vibration over a range of operating conditions. In order to control the propagation of fluid-borne acoustic pressure, Kojima et al [4] proposed an active control system employing a servo actuator driving a piston in a pipe, and demonstrated single frequency feedforward control of the fluid-borne pressure pulsation. Although this control system achieved significant levels of attenuation of the pressure pulsation, it is intrusive to the pipeline and this may not be practical in many installations. Brennan et al proposed an alternative non-intrusive method of controlling the fluid-borne acoustic pressure [5]. This system employed a fluid-wave actuator, comprising of a magnetostrictive actuator coupled to the pipe via a constrained annular volume of water, and a fluid-wave sensor, implemented using PVDF film. This system was shown to achieve between 30 and 40 dB of attenuation in the fluid-borne acoustic pressure between 50 Hz and 1 kHz; however, it also resulted in an increase in the structural vibration of up to 35 dB, which would need to be controlled using a secondary control system. In addition to the significant enhancements in the structural vibration, the device proposed in [5] has been applied to a perspex pipe, which is relatively flexible compared to the steel pipes typical used in industrial applications. To overcome this limitation Maillard et al [2] have proposed an alternative non-intrusive fluid-wave actuator. This system comprised of 6 piezo-electric stack actuators positioned circumferentially around the pipe and driven in-phase in order to produce axisymmetric excitation. In order to achieve a high level of force input into the fluid, the thickness of the pipe wall was reduced at the location of the circumferential actuator array in order to increase the radial wall compliance. The control system is shown to achieve between 9 and 17 dB of attenuation in the first four harmonics of the pressure measured at an error hydrophone positioned downstream of the fluid-wave actuator. However, it is reported in [6] that this control system also causes significant increases in the structural vibration and, due to the fluid-structural coupling, this means that the level of control downstream of the error sensor is

also limited. To overcome this limitation it is suggested that it is necessary to control additional propagating waves, rather than just the axisymmetric wave.

This paper aims to extend the previous work in the control of noise and vibration in fluid-filled pipes by investigating the limits on the control of the internal fluid-borne acoustic pressure and the externally radiated noise when employing a non-intrusive array of structural actuators. In Section 2 the experimental test facility is described and the details of the non-intrusive array of structural actuators are presented. In section 3 the broadband multichannel feedforward active control formulation is presented, with the inclusion of an error signal weighting matrix, which allows the trade-off between controlling different responses of the pipeline system to be investigated. In Section 4 the performance of the non-intrusive actuator array is evaluated for a number of different cost functions and, in particular, the trade-off between control of fluid-borne and radiated noise is investigated. Finally, in Section conclusions are drawn.

2 EXPERIMENTAL FACILITY AND CONTROL SYSTEM

2.1 Fluid-Filled Pipeline

In order to investigate the control of noise and vibration in a fluid-filled pipeline, an experimental test facility has been constructed, as shown in Figure 1. The system consists of three, 1 m long, sections of steel pipe connected together to form a 3 m pipeline. The outer diameter of the pipe is 114 mm and has a wall thickness of 12 mm. The primary disturbance is provided by an inertial mass actuator attached to a flexible steel diaphragm which terminates the pipeline. The pipeline is also connected to a header tank via a port in the pipe wall, which provides a static pressure. This helps to ensure that trapped air inside the pipeline is limited and any excess air can be released via a bleed valve. A flexible steel diaphragm terminates the downstream end of the pipeline. Since the pipe is not infinite, or anechoically terminated, standing waves will be setup in the pipeline, however, this is not unlike practical installations where changes in the pipe diameter and bends in the pipeline will lead to impedance discontinuities that will also result in standing waves.

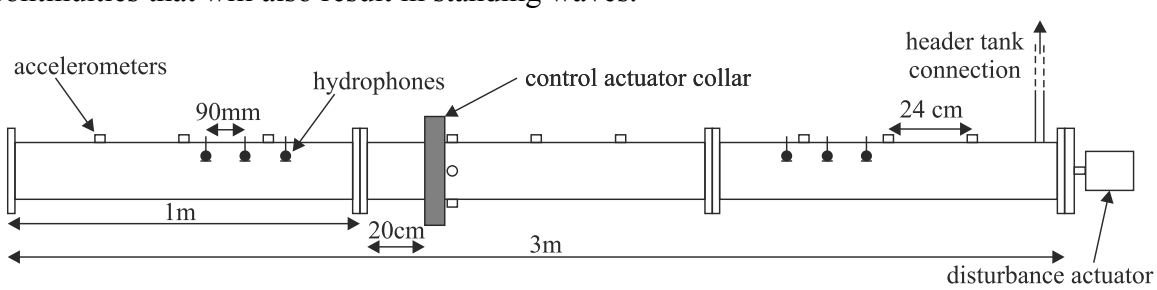


Figure 1 – Fluid-filled pipeline experimental setup.

2.2 Non-Intrusive Array of Structural Actuators

The non-intrusive array of structural actuators is similar to the system proposed in [2], however, 8 piezoelectric stack actuators are employed and the thickness of the pipe wall has not been reduced at the location of the collar. As shown in Figure 1 the collar of 8 actuators is positioned on the central section of the pipeline and a photograph of the collar is shown in Figure 2. The collar is secured by 8 locating bolts, which also provide a preloading force on the actuators. In comparison to the arrangement proposed in [2], where the 6 actuators are connected

in parallel and driven in-phase, the 8 actuators in the proposed arrangement are driven independently. This allows the array to potentially control higher-order modes than the arrangement employed in [2], and also avoids the sensitivity of the array to any differences in the preloading applied to the individual actuators. The employed actuators are Physik Instrumente P.810.1 and the specifications supplied by the manufacturer are detailed in Table 1.

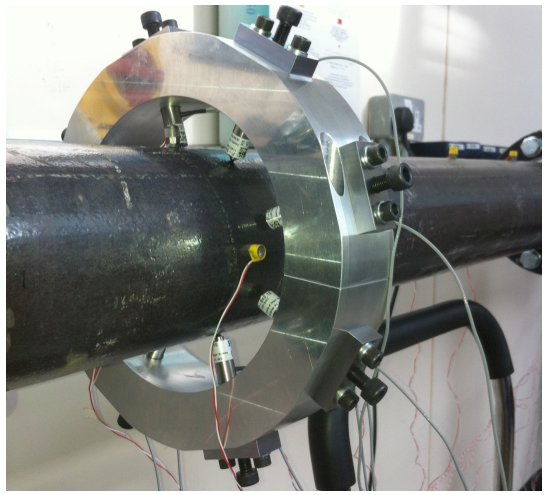


Figure 2 – The non-intrusive array of 8 piezoelectric stack actuators and the mounting collar.

Table 1 – Piezo-electric stack actuator – PI P.840.1 – manufacturer specifications.

Property	Value
Open-loop travel @ 0 to 100 V	15 µm
Static stiffness (Large-signal)	57 N/µm ±20%
Pushing force	1000 N
Pulling force	50 N
Max. torque on tip	0.35 N
Electrical capacitance	1.5 µF
Unloaded resonance frequency	18 kHz
Operating temperature	-20 to +80 °C
Mass	20 g
Length	32 mm

2.3 Structural and Acoustic Sensor Arrangement

To measure the response of the pipeline system and determine the effect of different control strategies, the pipeline has been instrumented with hydrophones, accelerometers, and microphones. As shown in Figure 1 six hydrophones have been installed in the pipeline; three upstream of the control collar (towards the disturbance actuator) and three downstream. Three accelerometers have been mounted on each pipe section, as shown in Figure 1 these accelerometers are distributed longitudinally along the length of the pipe. The radial vibration of the pipe at the location of the control collar has also been measured using 8 accelerometers circumferentially distributed around the pipe. In order to measure the radiated noise level from the pipeline, three microphones have been positioned in the far-field.

3 MULTICHANNEL BROADBAND ACTIVE CONTROL

In previous work employing fluid-wave actuators single frequency, or tonal active control has been investigated. However, in a number of applications it is important to control broadband noise induced in the pipeline by mechanisms such as fluid flow. Therefore, the broadband multichannel formulation of the filtered-*x* LMS feedforward active control algorithm will be detailed here. In order to facilitate the investigation of the trade-off between controlling fluid-borne acoustic pressure, structural vibration and radiated noise, the filtered-x LMS algorithm with an error signal weighting matrix will be presented.

For the multichannel feedforward controller employing *M* control signals and *K* reference signals, the vector of, *L*, error signals can be expressed as

$$\mathbf{e}(n) = \mathbf{d}(n) + \mathbf{R}(n)\mathbf{w} \tag{1}$$

where $\mathbf{d}(n)$ is the $L \times 1$ vector of disturbance signals, $\mathbf{w}$ is the *MKI* vector of filter coefficients of the *I*-th order control filters and $\mathbf{R}(n)$ is the $L \times MKI$ matrix reference signals filtered by the plant response. If the general cost function is defined as the weighted sum of the squared error signals, which is given using equation 1 by

$$J = E\left[\mathbf{e}^T(n)\mathbf{Q}\mathbf{e}(n)\right] = \mathbf{w}^T E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{R}(n)\right]\mathbf{w} + 2\mathbf{w}^T E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{d}(n)\right] + E\left[\mathbf{d}^T(n)\mathbf{Q}\mathbf{d}(n)\right] \tag{2}$$

where *E* is the expectation operator and $\mathbf{Q}$ is the $L \times L$ diagonal error signal weighting matrix, then the optimal, Wiener, set of filter coefficients can be calculated as [7]

$$\mathbf{w}_{opt} = -\left\{E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{R}(n)\right]\right\}^{-1} E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{d}(n)\right] \tag{3}$$

The matrix to be inverted in this calculation has dimensions of $MKI \times MKI$ and can therefore quickly become very large. For example, if we consider the system employing 8 control actuators, 3 error signals and 1024 filter coefficients this matrix has dimensions of 24576 × 24576. The inversion of such a large matrix is susceptible to conditioning problems. Therefore, it is often necessary in such cases to include some form of regularisation to improve the conditioning of the matrix to be inverted. For the feedforward active control problem presented here, regularisation of the matrix $\left\{E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{R}(n)\right]\right\}$ manifests itself as a constraint on the control effort, or the sum of the squared controller coefficients, $\mathbf{w}^T\mathbf{w}$. This regularisation term is consistent with the leakage parameter often employed in adaptive control algorithms. In general this regularisation term leads to the optimal solution being modified to

$$\mathbf{w}_{opt} = -\left\{E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{R}(n)\right] + \beta\right\}^{-1} E\left[\mathbf{R}^T(n)\mathbf{Q}\mathbf{d}(n)\right] \tag{4}$$

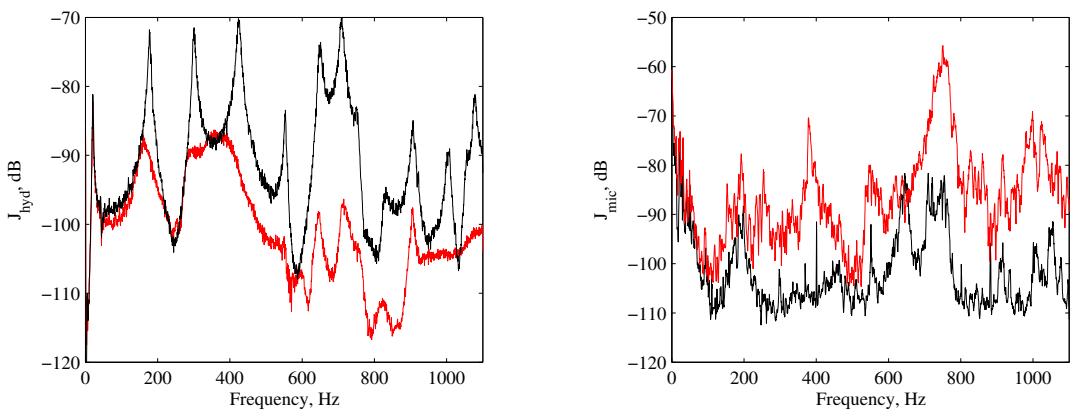
where β is an $MKI \times MKI$ diagonal matrix of regularisation, or leakage parameters. These parameters can be set to allow an accurate solution of the matrix inverse to be obtained, but can also be set for a practical application to constrain the filter coefficients such that the control actuators do not saturate. In the following section the leakage parameters have been set to ensure that the amplitude of the control signals are within the operating range of the actuators.

4 EXPERIMENTAL RESULTS

The performance of the non-intrusive array of structural actuators has been evaluated for a number of different control scenarios. In each case the primary disturbance has been generated by driving the disturbance actuator, shown in Figure 1, with a white noise signal. A sampling frequency of 2.2 kHz has been employed and the optimal control filter responses have been calculated according to equation 4. The reference signal has been provided by the signal driving the disturbance actuator and a control filter length of I = 512 has been employed.

4.1 Control of Fluid-Borne Vibration

Figure 3 shows the response of the pipeline system before and after control, when the non-intrusive array of actuators has been optimised to minimise the sum of the squared pressures measured at the 3 downstream hydrophone positions, $J_{hyd.}$. From Figure 3a it can be seen that significant reductions in the sum of the squared pressures measured in the fluid have been achieved. The peak attenuation is around 30 dB, which is comparable to that reported in [5] and significantly larger than that reported in [2]. The average attenuation over the full bandwidth of the system is 12.8 dB. However, as previously reported the level of structural vibration has been significantly increased this results in an increase in the radiated noise levels, as shown by the change in the sum of the squared pressures measured in the far-field of the pipeline shown in Figure 3b.



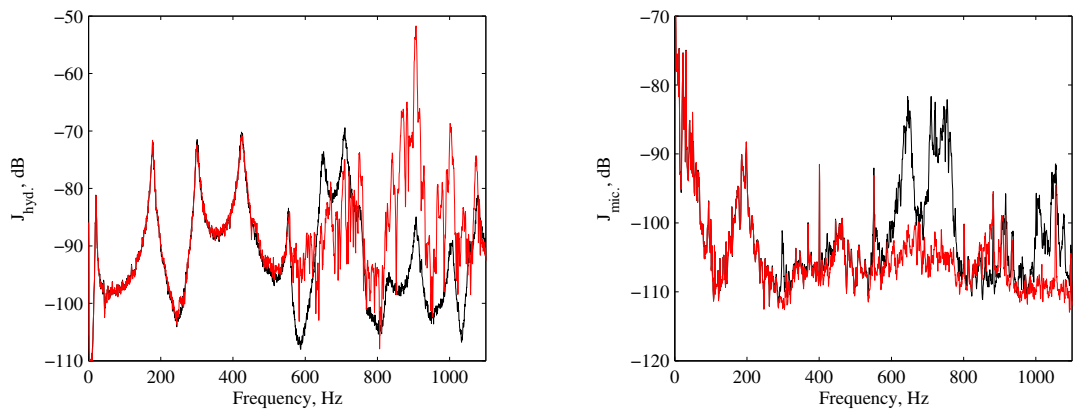
(a) Sum of squared downstream hydrophone pressures.

(b) Sum of squared far-field microphone pressures.

Figure 3 – The performance of the non-intrusive actuator array when optimised to minimise the sum of the squared pressures measured at the downstream hydrophones. The responses to the hydrophones, microphones and accelerometers are shown before control (black) and after control (red).

4.2 Control of Radiated Noise

In many practical applications the radiated noise is the primary quantity of interest and, therefore, active structural acoustic control systems have been developed to manipulate the vibration of a structure in order to minimise the radiated noise levels rather than the structural vibration [8, 9]. In the case of the pipeline system actuated by the non-intrusive array of structural actuators, Figure 4 shows the system response before and after control when the control filters have been optimised to minimise the sum of the squared pressures measured at the three far-field microphones, $J_{mic.}$. From Figure 4b it can be seen that the strongly radiating modes between 600 and 800 Hz have been attenuated by around 20 dB to the noise floor. However, the internal fluid-borne acoustic pressure measured by the hydrophones has also been significantly increased, although there is a small degree of attenuation between 650 and 750 Hz.



(a) Sum of squared downstream hydrophone pressures.

(b) Sum of squared far-field microphone pressures.

Figure 4 – The performance of the non-intrusive actuator array when optimised to minimise the sum of the squared pressures measured at the far-field microphones. The responses at the hydrophones, microphones and accelerometers are shown before control (black) and after control (red).

4.3 The trade-off between control of the fluid-borne, structure-borne and radiated noise

From the results presented in the previous sections, it is clear that when controlling either the internal fluid-borne acoustic pressure or the radiated noise alone, the uncontrolled responses of the pipeline system are generally enhanced. Therefore, it is interesting to consider the case when the actuator array is optimised to minimise some weighted combination of the internal fluid-borne acoustic pressure and the radiated noise level.

Figure 5 shows the trade-off between the broadband average attenuation in the sum of the squared fluid-borne acoustic pressures measured at the downstream hydrophones and the broadband average attenuation in the sum of the squared pressures measured at the far field microphone locations. From this plot it can be seen that a significant level of attenuation in the fluid-borne acoustic pressure can still be achieved when the enhancements in the radiated noise level are avoided. Specifically, the broadband average attenuation in the sum of the squared hydrophone signals is 12 dB when the broadband average change in the sum of the squared microphone signals is limited to 0 dB. In order to provide more insight into these results, the spectra of the two cost functions, J_{hyd} and J_{mic}, are shown in Figure 6. From these results it can be seen that significant reductions in the downstream hydrophone pressures are achieved throughout the bandwidth, however, it can be seen from the sum of the squared microphone signals shown in Figure 6b that although the dominant resonances between 600 and 800 Hz have been attenuated, there is an increase in the level between 200 and 500 Hz.

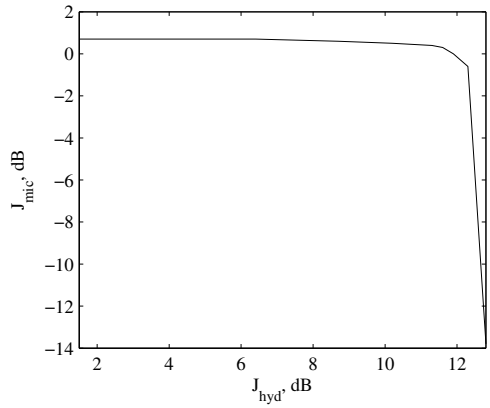
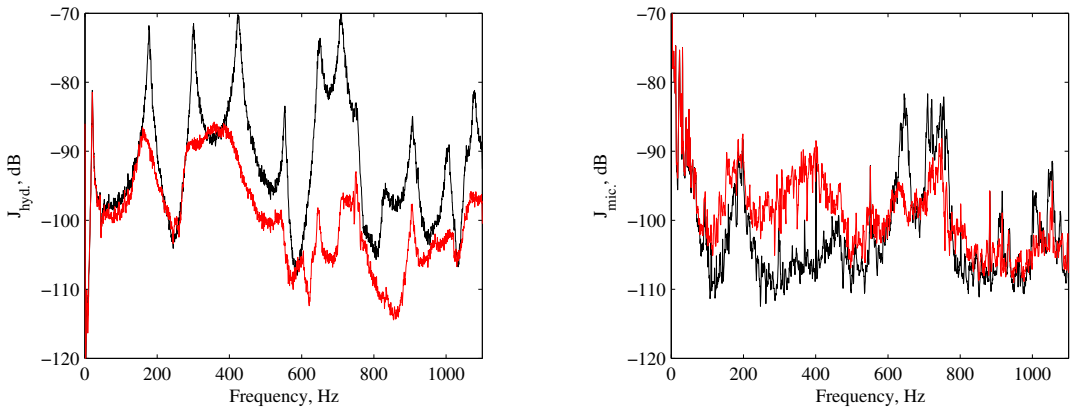


Figure 5 – The trade-off between the mean attenuation in the sum of the squared far-field microphone signals and the mean attenuation in the sum of the squared downstream hydrophone signals.



(a) Sum of squared downstream hydrophone pressures.

(b) Sum of squared far-field microphone pressures.

Figure 6 – The performance of the non-intrusive actuator array when optimised to minimise a weighted combination of the sum of the squared pressures measured at the downstream hydrophones and the far-field microphones. The responses at the hydrophones, microphones and accelerometers are shown before control (black) and after control (red).

5 CONCLUSIONS

Pipeline networks pose a particular problem for the transmission of noise and vibration around a variety of systems and environments. Therefore, there have been a variety of methodologies developed for their control. Although passive noise and vibration control measures have been successfully applied in practice, their performance is generally limited by space, weight and operational performance limitations. Active control methods have been developed to overcome some of the limitations of passive control measures, but in practice fluid-filled pipeline systems pose a complex problem due to the potential for the propagation of both fluid- and structure-borne noise. Previous work has developed non-intrusive fluid-wave actuators that are able to control the fluid-borne acoustic pressure, however, it has been shown that these control systems tend to significantly increase the levels of structural vibration. Previous work has

highlighted this problem for tonal noise control applications, however, it has not been fully investigated.

This paper has investigated the limitations on broadband active control in a fluid-filled pipeline of the fluid-borne acoustic pressure and the radiated acoustic noise using a non-intrusive array of structural actuators. In order to conduct this investigation a fluid-filled pipeline system has been constructed from standard steel piping and instrumented with 6 hydrophones, 3 far-field microphones and 16 accelerometers. The primary disturbance has been provided by a inertial actuator attached to a steel diaphragm which terminates one end of the pipeline. The non-intrusive structural actuator array is based on the system proposed in [2], however, it consists of 8 piezoelectric stack actuators distributed circumferentially around the pipe and the actuators are driven independently. The signals driving the control actuators have been calculated using the optimal time-domain Wiener solution, which has been derived for the case when an error sensor weighting matrix is included in the formulation.

The ability of the non-intrusive array of structural actuators to control the fluid-borne acoustic pressure measured at three downstream hydrophones has initially been evaluated. It has been shown that a peak attenuation of around 30 dB in the sum of the squared pressures measured at the down- stream hydrophones is achievable and the broadband average attenuation is 12.8 dB. However, it has also been shown that the levels of radiated noise are significantly increased by this control strategy. This increase in radiated noise is consistent with the increases in structural vibration that have been reported in previous work, however, the increase in radiated noise has not previously been explicitly reported. The performance of the non-intrusive array of structural actuators has also been investigated when the controller is optimised to minimise the radiated noise and in this case it is shown that the controller is able to achieve around 20 dB of attenuation in the dominantly radiating modes, but results in increases in the fluid-borne acoustic pressure.

The ability of the non-intrusive array of structural actuators to control both the fluid-borne acoustic pressure and the radiated noise level has been investigated, and it has been shown that a useful trade- off can be achieved between these two parameters. That is, it has been shown that the levels of enhancement in the radiated noise level can be significantly reduced with only a small reduction in the level of fluid-borne noise control. This may be useful in applications where the uncontrolled noise radiation from the pipeline is low, but must not be increased by the action of the active control system.

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