Optimal design of a control actuator for sound attenuation in a piping system excited by a positive displacement pump

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ABSTRACT

This paper investigates technology aimed at reducing noise propagated in a piping system. The piping system consists of a long pipe, with one end excited by a positive displacement pump and the other end connected to a water tank. Compared to a centrifugal pump, a positive displacement pump generally produces a larger pressure, so sound attenuation is more difficult. As a first step towards sound attenuation, the current study focuses on attenuating the acoustic wave in the pipe fluid. Before attempting to control the fluid motion in the pipe, a thorough understanding of the dynamic behavior of fluid-filled pipes is required. The first part of this paper reviews the work carried out by other investigators in this area, while the second part of the paper details the experimental program undertaken by the authors. A design of a fluid wave actuator for low-frequency plane fluid waves is provided. The actuator consists of a PZT (piezoelectric ceramic) cylinder which is mounted compliantly and concentrically inside a steel pipe. Since the pressure generating surface of the actuator is axisymmetric, only the radial axisymmetric mode is expected. The induced radial wall motion can in turn drive the fluid through structure-fluid coupling. Next, a practical design is proposed for active control of noise propagating from the piping system by using the actuator. Finally, the properties of the pipe, mounting systems and measurement setup are discussed in detail. The study will provide a guideline for setting up a piping system with active and passive control measures.

INTRODUCTION

Piping systems are a central part of many engineering installations including marine vessels. A number of sources such as pumps can generate vibrations which can propagate along the pipes and transmit to other mediums causing unwanted noise. This phenomenon is a direct consequence of the coupled nature of a fluid-filled pipe vibration, which means that vibration energy can propagate in both the pipe wall and in the fluid, making it difficult to attenuate.

de Jong (1994) has shown that in a pipe system connected to a rotary pump, the dominant fluid pulsations are at frequencies corresponding to the pump's blade passing rate and its harmonics. He further showed that a flexible bellows section fitted in an unpressurised pipework system was effective in suppressing the propagation of fluid pulsations. In practical pipework systems, however, the fluid contained in a pipe is generally pressurized, and this tends to stiffen the flexible sections making them less effective.

It has been recognized that passive methods are sufficient to attenuate high frequency noise. However, passive devices are not always practical at low frequencies. Thus, there is motivation for attempting to control fluid-borne vibrations using active control.

Work has been carried out into the active control of fluid waves in finite pipes. Brévart and Fuller (1993) and Fuller and Brévart (1995) have conducted theoretical and experimental studies into the active control of total power propagating along fluid-filled pipes using point forces as control actuators. Harper and Leung (1993) carried out an experimental study where helical PVDF (Polyvinylidene fluoride) cables were embedded in a rubber section of a pipe to control structural waves while a hydro-sounder was used to control the axisymmetric fluid wave. Brennan *et al.* (1996) reviewed the above work and developed an actuator for controlling the fluid wave propagating in a soft-walled, water-filled perspex pipe. Compared to a steel pipe, the perspex pipe is more amenable to deformation by the actuator. Thus, their approach is less effective for a steel pipe. Podlesak (1997) experimentally examined active control of noise transmission in a pipe excited by a tonal noise source, using a PVDF noise source as a control actuator at very low frequencies, where the structural waves could not be easily excited by applying the fluid-wave control source. Using a similar approach to Podlesak, Maillard (1998) investigated active control of noise radiation from the discharge line of an oil driven hydraulic engine. Baris Kiyar *et al.* (2002) tested an active control system using inertial actuators as controllers on a PVC pipe system and found reductions of 8 dB in structural waves and 6 dB in fluid waves could be achieved when using error signals from both accelerometers and hydrophones together.

A preliminary experimental investigation into the fluid-borne and structure-borne noise transmission along a water filled steel pipe due to the excitation of a centrifugal pump was carried out by Pan and Dickens (2001). The theoretical study of the work was conducted by Pan and Juniper (2003). The studies indicated that structural waves depend on experimental configurations. Fluid waves should be controlled at the same time as structural waves excited in the system.

This paper investigates a technology aimed at reducing noise propagated in a piping system excited by a positive displacement pump. A new design of a PZT fluid wave actuator for low-frequency plane fluid waves is discussed in detail and the actuator is manufactured. Next, a practical design is realized for active control of noise propagating from the piping system by using the actuator.

LOW-FREQUENCY DYNAMIC BEHAVIOUR OF A FLUID-FILLED PIPE

Propagating Waves in a Fluid-filled Thin Shell

This section recalls the basic mechanisms involved in the vibration of fluid-filled cylindrical thin shells. The analysis below is limited to low frequency, well below the ring frequency, defined as $\omega_{ring} = c_L / a$ where c_L is the compressional wave speed in a plate of the same material and thickness and a is the shell mid-plane radius. At the ring frequency, the shell vibrates in a breathing mode and the pipe circumference equals the associated compressional wavelength in a plate, $\lambda_L = 2\pi c_L / \omega_{ring}$.

Assuming a thin cylindrical shell, structural vibrations can be decomposed into a set of circumferential modes of order $n = 0, 1, 2, \cdots$ and associated waves propagating along the shell axis in the positive and negative directions.

The first mode $n = 0$ is referred to as the breathing mode due to its axisymmetric motion. In the case of the $n = 1$ mode (beam bending mode), the pipe cross section remains undeformed. Higher order modes feature lobar type mode shapes with $2n$ anti-nodes around the pipe circumference. Below the ring frequency, four wave types can propagate energy (Pavić 1992 and Pinnington and Briscoe 1994). The first three wave types represent axisymmetric waves of circumferential mode $n = 0$. The fourth wave type is associated with flexural waves of higher circumferential modes. The four wave types can be described in terms of their structural and fluid components. The first axisymmetric wave is predominantly a compressional wave in the shell along the axial direction with some associated radial wall motion influenced by Poisson's ratio and fluid loading. The second axisymmetric wave is predominantly fluid-based with some radial wall motion due to the shell compliance. It will be referred to as the acoustic or fluid wave. The third axisymmetric wave, referred to as the torsional wave, exhibits only motion in the tangential direction which is uncoupled from axial and radial motion. This wave does not have any fluid-based components. The $n = 1$ beam bending wave is characterized by lateral motion of the pipe, such that its cross-section remains virtually undeformed. In this mode, all three orthogonal motions are coupled. The higher-order flexural waves ($n \geq 2$) feature cut-on frequencies below which they cannot propagate (Fuller and Fahy 1982).

For a steel pipe of a common size, the ring frequency is usually much greater than the excitation frequency of potential sources such as a pump. Thus, a simplified model may be sufficient to estimate the system response.

Characteristic Equations of a Fluid-filled Pipe

At low frequencies, the wave-numbers of a fluid-filled pipe can be represented by the following simple expressions, derived from the approximate dispersion equations for longitudinal (wall-axial), acoustic, torsional and bending (wall-radial) wave-numbers given by Pavić (1992) and based on Flügge's model (Flügge1973)

$$\kappa_l \approx \varsigma_l \Omega, \qquad \kappa_a \approx \varsigma_a \Omega, \qquad \kappa_t \approx \varsigma_t \Omega,$$
$$\kappa_b \approx \varsigma_b \sqrt{\Omega} \tag{1}$$

where $\kappa_l = \kappa_x a$ is a non-dimensional representation of the axial wave-number κ_x and similarly for the other wave-numbers, Ω is the non-dimensional frequency defined as $\Omega = \omega / \omega_{ring} = \omega a / c_L$. The constants ζ are

$$\zeta_l = (1+\Delta)^{1/2}, \quad \zeta_a = [\psi + (2\eta + \nu^2)/(1-\nu^2)]^{1/2},$$
$$\zeta_t = [2/(1-\nu)^{1/2}], \quad \zeta_b = (2+\eta)^{1/4} \tag{2}$$

where

$$\psi = (c_s / c_f)^2, \quad \eta = (\rho_f / \rho_s)(a/h),$$
$$\Delta = \nu^2 (\psi - 1)/[(\psi - 1)(1-\nu^2) + 2\eta + \nu^2].$$

Subscripts *s* and *f* refer to the structure and fluid respectively. The wave-length associated with a given propagating wave and frequency can be estimated as $\lambda = 2\pi / k_{l,a,t,b}$.

Fahy (1987) derived approximate formulae for the cut-on frequencies of the first few circumferential modes in the case of an in-vacuum pipe. In particular, the cut-on frequencies of the $n = 2$ and $n = 3$ modes are approximated as

$$\begin{aligned} &\frac{h}{\sqrt{12}a} 2.68 f_{ring}, \ n = 2 \\ &\frac{h}{\sqrt{12}a} 7.65 f_{ring}, \ n = 3 \end{aligned} \tag{3}$$

where a is the pipe radius, h is the pipe wall thickness and $f_{ring} = \omega_{ring} / 2\pi$.

THE PIPING SYSTEM EXCITED BY A POSITIVE DISPLACEMENT PUMP

Positive Displacement Pump

A typical type of positive displacement pump called a screw pump, which transfers fluid along an axis, is considered here. Important specifications of a screw pump include maximum discharge pressure and flow rate. Unlike a centrifugal pump, the screw pump has a more or less constant flow regardless of the system pressure. Screw pumps generally give more pressure than centrifugal pumps, so sound attenuation is more difficult.

Characteristic Properties of the Piping System

Table 1 presents the characteristic properties of the system under consideration, where natural frequencies are derived from the equations defined in Equations (1)-(3). These properties are commonly used in the analysis of fluid-filled pipes. Their meaning will be presented in later sections.

Table 1. Characteristic properties of the piping system

Pipe inner radius (m)	0.050
Pipe thickness (m)	0.006
Pipe ring frequency (Hz)	15615
$n=1$ beam bending wave (Hz)	1100
$n=2$ cut-on frequency (Hz)	1367 (in vacuum)
$n=3$ cut-on frequency (Hz)	3903 (in vacuum)
Pump fluid rate	30m^3/h (1.06m/s mean velocity in pipe)
Pump rotation speed (RPM)	620
Pump pressure (bar)	Up to 3
Pump fundamental frequency (Hz)	10.5
Static pressure (bar)	0.5

Note the $n=2$ and $n=3$ cut-on frequencies are calculated based on the in-vacuum results of Equation (3). The effect of fluid loading will slightly reduce the above values by mass loading the pipe. The static pressure is based on 5m of water (see Figure 2: the water tank is 6 m deep and the pipe is attached 1m up from the bottom).

Propagating and Non-propagating Modes

Podlesak (1997) has discussed the behaviour of propagating and non-propagating modes for a water-filled pipe whose main results are recalled here. Before discussing the experiments and the results, the following questions had to be addressed: first, what types of modes are likely to be excited during the experiments; and second, what is the effect of non-propagating modes on the error hydrophone.

Since the screw pump transfers fluid along the longitudinal axis, the pressure generating surfaces of the noise would be approximately axisymmetric. Mainly, radial axisymmetric modes are expected to be generated. The ring frequency for the pipe is well outside the 0 to 1000 Hz measurement range. The first waterloaded radial-circumferential flexural mode (n=1 beam bending mode) was estimated at 1100 Hz. Clearly, the radial-circumferential flexural modes in the pipe would not be excited within the frequency range of the measurements. Therefore, acoustic and compressional modes will be the main propagating modes of interest for the current piping system. These two waves are inherently coupled, since the fluid and wall-axial waves interact with one another through a Poisson-type mechanism. The coupling increases as the frequency increases (Podlesak 1997).

The effect of non-propagating components generated by a PVDF fluid-wave actuator on an error hydrophone was examined by Podlesak (1997). The attenuation factors he calculated for the first six higher order, axisymmetrically excited fluid modes were over 200 dB. The current PZT fluid-wave actuator has similar specifications to those of the PVDF fluid-wave actuator. Therefore, the current PZT fluid-wave actuator would not be expected to degrade the error sensor performance.

Dispersion relation

To illustrate the dispersion relation given in Equation (1), the wave-numbers of the acoustic and longitudinal waves are plotted versus frequency in Figure 1. The wave-numbers of the bending and torsional waves are also shown in the figure for comparison

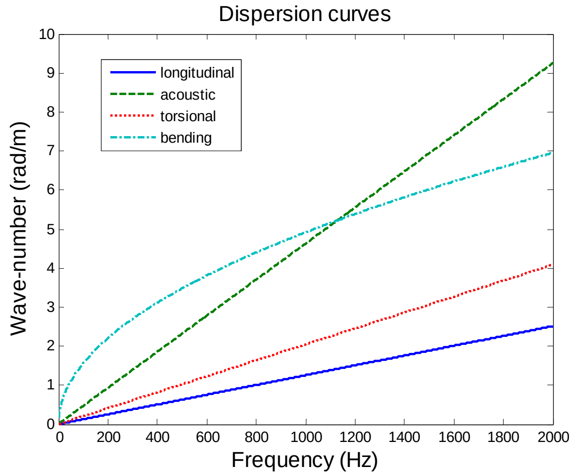

Figure 1. Dispersion curves of the water-filled steel pipe.

Note that 1150 Hz is the coincidence frequency, where the acoustic and bending wave-numbers are the same. This phenomenon indicates that sound radiation into the fluid is significant at this frequency. Below 1150 Hz, the fluid and structural waves are weakly coupled, and the fluid and structural response of the pipe is governed by the resonance of each wave type (Liu *et al.* 2002). Above 1150 Hz, the pipe stiffness in the radial direction decreases and fluid-structural coupling becomes strong. Well below the coincidence frequency, coupling between the fluid and pipe wall occurs through the axisymmetric longitudinal and acoustic waves as discussed above.

Determination of Fully Developed Flow

It was found that fluid velocity varies considerably in the inlet of the pipe which is in close proximity to the pump. In practice, actuators and sensors should be located where fluid flow is stable. Thus, it is necessary to classify the flow and determine the entry length of the pipe beyond which the flow is fully developed, i.e. the fluid velocity profile over the pipe cross-section reaches its steady state. The Reynolds number is $\mathrm{Re} = \rho UD/\mu = 1.06\times10^5$ with fluid density $\rho_f = 1000\ \mathrm{kg/m}^3$, mean velocity $U = 1.06$ m/s, pipe diameter $D = 0.100$ m and dynamic viscosity at 20ºC $\mu = 1.002\times10^{-3}$ Ns/m^2 for the current system. Pipe flow is usually considered to be turbulent for $\mathrm{Re} > 2000$ (Douglas 1985). Therefore, the flow can be assumed to be turbulent.

As described by Douglas *et al.* (1985), the entry length for establishment of fully developed turbulent flow is taken as 60 pipe diameters (6.00 m). Therefore, transducers along the piping should be located at least 6.5 m from the pump.

DESIGN OF AN ACTIVELY CONTROLLED PIPING SYSTEM

Design of a Test System

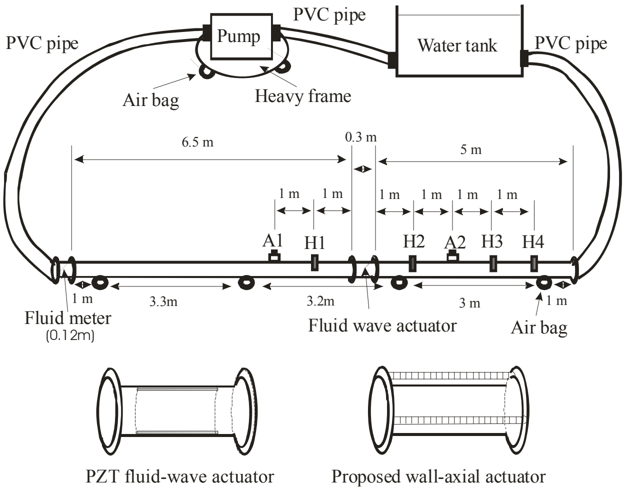

Figure 2. New design of pipe-pump system for active control of noise propagation (A: refers to accelerometer and H: refers to hydrophone).

A pipe-pump model is proposed and a schematic diagram is shown in Figure 2. The pump pumps water from a holding tank of dimensions $10\times10\times6$ m through a composite PVC pipe (6 m) into a fluid meter and then steel transmission pipe. The reason for introducing the PVC pipe is to damp vibration transmission to the steel pipe. The steel transmission pipe has a length of 6.5 m. The output of the steel transmission pipe is connected to the fluid wave actuator, attached to another steel transmission pipe of length 5 m, which is connected to a PVC pipe and then back to the water tank. All pipes have the same diameters of 100 mm. The sound pressure is monitored by four hydrophones, and structural vibration of the transmission pipe is monitored by two accelerometers. The hydrophones and accelerometers are located upstream and downstream of the control actuator. Details of the actuators, sensors and mounting systems will be described below.

Design of a Fluid Wave Actuator

The fluid wave actuator (refer to Figure 2) was manufactured from a 100 mm long by 3 mm thick PZT cylinder with 100 mm outer diameter. This was mounted compliantly and concentrically inside a short section of stainless steel pipe using a silicone rubber compound.

Driving the PZT cylinder via an applied voltage results in expansion and contraction of the cylinder. The induced radial wall motion of the PZT cylinder in turn drives the fluid, thus producing a controlled acoustic wave to cancel the primary acoustic wave at an error hydrophone and resulting in reduced noise downstream.

Pipes and Mounting

The properties of the pipes are considered based on cost and influence on the fluid. The piping used should also be similar to that used in realistic applications. Two galvanized steel pipes (6.5 m and 5 m long) will be used as the combined steel transmission pipe (refer to Figure 2). This avoids contaminating the circulated water with rust. The pipe will be mounted on four evenly spaced soft isolators. The system of the pipe with the isolators needs to have natural frequency less than half of the pump excitation frequency (e.g., 5 Hz) so that appropriate isolation of the pipe occurs. It has been calculated that the pipe (approximately 12 m long and 6 mm wall thickness) filled with water weighs approximately 280 kg. Thus, each isolator is required to have a stiffness of 69 kN/m. Suitable airbags can be selected to fulfill this requirement.

Pump Mounting

The positive displacement pump is proposed to be mounted on a heavy steel mass supported by three air bags with the natural frequency less than half of the pump excitation. The pump weight without a motor is 284 kg and the steel mass is 1000 kg. Thus, the total weight is more than 1300 kg including the motor. Air bags (for example Goodyear air bags, 1B6-536) will be used for this application. It has been calculated that using each air bag with a working height of 0.12 m and pressure of 345 kPa will provide a natural frequency of 2.15 Hz.

Measurement Setup

In practice, noise and vibration levels vary considerably along the pipe (see Section 2.1). Also, introducing the fluid wave control actuator may affect both noise and vibration levels upstream and downstream of the actuator. Thus, various noise and vibration sensor locations are proposed. Four hydrophones will be mounted inside the pipe vertically through the pipe wall. Each of the hydrophones will be housed in a cap which will be plugged through the pipe wall. The mounting hole will be machined and welded onto the pipe. The sensor locations are shown in Figure 2. One hydrophone is located upstream of the fluid wave actuator and three additional hydrophones are located on the downstream section. Two triaxial accelerometers will be mounted (using epoxy glue) on the piping, one upstream and one downstream, to estimate the pipe acceleration in the radial, axial and tangential directions.

Initial Pressure Response

It may be important to determine the frequency range where pump excitation dominates, as actuators and amplifiers are normally valid only for a certain frequency range. Furthermore, measurements can then be targeted at the dominant frequency range. Figure 3 shows the pressure level measured at the third hydrophone (H3) due to excitation from the pump running at 620 RPM. This measurement is from a previous experimental rig, with some similarities to the proposed system, but with different mounting arrangements. The fundamental pump harmonic is at 10.5 Hz. It can be seen that the pressure is significant at higher order harmonics as well. However, the pressure level decreases dramatically from 200 Hz onwards.

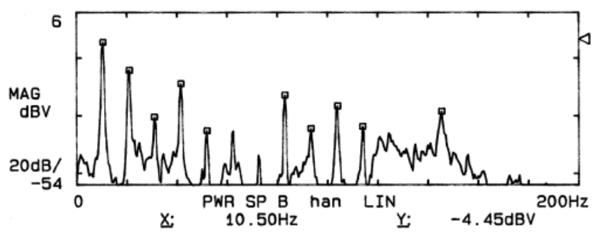

Figure 3. Initial measured pressure level downstream of terminating pipe.

CONCLUSIONS AND RECOMMENDATIONS

At low frequencies, four propagating wave types exist in a fluid-filled pipe. They are acoustic, longitudinal, torsional and bending waves. These waves are weakly coupled, and the fluid and structural response at the low frequencies considered is governed by the resonance of each wave type. When the pressure generating surface of the noise is axisymmetric and at frequencies below the beam bending mode (n=1), the acoustic and longitudinal waves dominate noise propagation.

A piping system excited by a positive displacement pump is investigated in this paper. The pump transfers fluid along the longitudinal axis of the piping system, so it is expected that the radial axisymmetric modes dominate. The initial testing results from an earlier, similar system showed that pressure levels in the pipe are significant at the pump fundamental and harmonics below 200 Hz and then decrease dramatically from 200 Hz onwards. The frequency range of 0 to 200 Hz is well below the n=1 beam bending mode. Therefore, the acoustic and longitudinal waves are the dominant waves propagated in the piping system and these two waves need to be controlled.

Three aspects were further investigated. Firstly, a fluid wave actuator was designed that could be excited to expand and contract in the radial direction to form a controlled fluid wave. The actuator was made from a PZT cylinder mounted compliantly and concentrically inside a steel pipe. Secondly, the properties of the pipes, mounting systems and transducer locations necessary for setting up the piping system were examined. Thirdly, a practical design of an actively controlled piping system using the fluid wave actuator was proposed.

As the longitudinal wave can also propagate and is inherently coupled with the acoustic wave through a Poisson-type mechanism as the frequency increases, a specific longitudinal wave actuator may be required to supplement the fluid wave actuator. The longitudinal wave actuator is envisaged as three PZT stacks mounted axially along the external wall of a pipe section and bolted at both flanges of the pipe section (refer to Figure 2). The PZT stacks would then be driven in-phase to produce longitudinal motion along the axis of the pipe wall. The longitudinal wave actuator could be placed in series with the fluid wave actuator to control zero order longitudinal and acoustic waves and would be expected to provide a reasonably independent control of the two wave regimes at low frequencies.

The experimental work to confirm the results of this study will be done in the future. The overall aim of this study is that, upon completion, the issues discussed and actuators designed in the study will provide a guideline for setting up a piping system with active and passive control measures.

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LIST OF SYMBOLS

a	Mean radius of the shell
c	Velocity of propagation
D	Pipe diameter
h	Thickness
n	Circumferential mode number
Re	Reynolds number
RPM	Rotation per minute

U	Mean velocity of fluid
ζ	Wavenumber constant
η	Non-dimensional constant
κ	Non-dimensional wavenumber
λ	Wave-length
μ	Dynamic viscosity
ν	Poisson's ratio
ρ	Density
ψ	Non-dimensional constant
ω	Angular frequency
ω_{ring}	$\omega_{ring} = c_L / a$, ring frequency
Ω	$= \omega / \omega_{ring}$, non-dimensional frequency

Subscripts

a	Acoustic
b	Bending
f	Fluid
l, L	Longitudinal
s	Solid
t	Torsional
x	Axial