Active Control of Pressure Pulsations in Piping Systems

by

Julien Maillard
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Research Report

Active control of pressure pulsations
in piping systems

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Abstract

Fluid-borne vibrations in piping systems remains a serious problem in applications such as marine vessels where mechanical fatigue and radiated noise are critical factors. In the case of pumps or hydraulic engines, the main source of vibrational energy is in the fluid axisymmetric plane wave associated with the system pressure pulsations. Due to fluid/structure coupling, this wave propagates in both the pipe wall and fluid. For high levels of pressure pulsations, the resulting radial and axial wall motion can then cause mechanical fatigue and unwanted radiated noise. Passive pulsations dampers have been used traditionally to reduce the fluid pressure pulses. The use of such passive devices is limited however in critical applications due to the resulting static pressure loss which decreases the system performance. This report describes the design and testing of a non-intrusive fluid wave actuator for the active control of pressure pulses. The actuator consists of a circumferential ring of PZT stacks acting on the pipe outside wall to generate an axisymmetric plane wave in the fluid through radial motion coupling. After briefly describing a simplified model of the actuator along with predicted performances, experimental results will show the control performance of the actuator applied to the discharge line of an oil driven hydraulic engine.
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Chapter 1

Introduction

Piping systems are a central part of numerous engineering installations including marine vessels. A number of sources such as pumps generate vibrations which can propagate along the pipes and excite other structures. These vibrations can cause two main problems, namely mechanical fatigue and radiated noise. This report describes a method aimed at reducing radiated noise in piping systems.

Noise in piping systems is generated through pipe wall motion which can directly radiate sound as well as excite other potential acoustic radiators through pipe attachments. Two types of pipe noise are commonly defined. Structure-borne noise comes from direct mechanical excitation of the pipe through its attachments and connections to vibrating structures such as pumps. On the other hand, fluid-borne noise is associated with fluid excitation of the pipe wall. The present study focuses on low frequency fluid-borne noise. At low frequencies, fluid excitation is mainly associated with pressure fluctuations generated by pumps and hydraulic engines. As an example, de Jong has shown that the dominant fluid pulsations measured in a pipe system connected to a rotary pump occur at frequencies corresponding to the pump’s blade passing rate and its harmonics [1]. Note that higher frequency fluid-borne noise can be generated from flow perturbations associated with elbows, valves, or cross section changes in the pipe.

While passive methods are sufficient to attenuate high frequency noise, in the low frequency region passive devices are not always practical. In particular, effective passive rubber sections installed in a pressurized pipe-work system are not always available due to the resulting safety problem. Passive pulsation dampers have been used with success to reduce the fluid pressure ripples. However, their use can become limited in critical applications due to the resulting static pressure loss which decreases the system performance. Thus there is motivation for attempting to control fluid-borne vibrations using active control. As the nature of the disturbance is deterministic, it is well suited to the applications of active control techniques such as feedforward control using a reference signal derived from the rotation speed of the pump or hydraulic engine [2,3].

One of the particularity of piping systems is that vibrational energy can propagate throughout the entire system, thus causing unwanted noise in areas removed from the source of excitation, e.g., pumps. This property is a direct consequence of the coupled nature of fluid-filled pipe vibrations, which means that vibrational energy can propagate in both the pipe wall and in the fluid, making it difficult to attenuate. Due to the structure-fluid coupling, the two propagation paths are not independent. For instance, attenuating structural motion only may not be sufficient as energy can propagate through the fluid and excite the structure downstream. In terms of waves, vibrational energy is transmitted along the pipe through propagating waves of different types. Each wave has structural and fluid components [4] whose relative contribution will determine the type of actuator and sensors to use in an active control approach.

For the target system considered in this report, the pressure pulsations resulting from the pump and
hydraulic engine constitute the main source of vibrations in the system. The excitation being contained in the fluid, it will be shown that the acoustic wave will dominate the pipe wall motion. As a first step towards sound attenuation, the current study will thus focus on attenuating the acoustic wave, also referred to as fluid wave as most of its energy is contained within the fluid.

Only a limited amount of work has been carried out into the active control of fluid waves in pipes. Brévant and Fuller have conducted theoretical and experimental studies into the active control of total power propagating along fluid-filled pipes using point forces as control actuators [5,6]. Harper and Leung [7] carried out an experimental study where helical PVDF cables were embedded in a rubber section of pipe to control structural waves while a hydro-sounder was used to control the axisymmetric fluid wave. Finally, Brennan et al. developed an actuator for controlling the fluid wave propagating in a soft water-filled perspex pipe [8]. In contrast with the work mentioned above, the present study is aimed towards the implementation of an active control system on a real hydraulic system including pressurized pipes and high levels of fluid pulsations.

In a first chapter, the target system is briefly introduced by giving its main characteristics in terms of dimensions and properties. The next chapter focuses on the dynamic behavior of fluid-filled pipes within the low frequency region. Based on this analysis and the characteristics of the target system, the third chapter discusses the design and characteristics of the actuator used to achieve control of the fluid waves. In a fourth chapter, the experimental setup used in testing the control system is presented. Finally, the measured performances of the system are given and discussed in the last chapter, which is followed by the conclusions of the study and recommendations for future work.
Chapter 2

Characteristics of the target system

This chapter introduces the target system for the control approach developed in this project. The discussion will help understand the different requirements for the control system as well as provide a basis for the numerical simulations and experimental testing presented in the next chapters.

2.1 General presentation

The target system for this work is a hydraulic pipe-work on-board a marine vessel. The system is composed of a main pump located in the machine room at the rear of the ship and a hydraulic engine located at the front of the ship. The hydraulic engine is used to drive auxiliary propellers located on the front of the ship for fast maneuvering. The two pipe lines on the pressure side and return side of the pump will be referred to as the incoming line and outgoing line, respectively. A simplified schematic of the system is shown in Figure 2.1. The arrows indicate the direction of flow.

![Simplified schematic of the on-board hydraulic system](image)

As mentioned above, the hydraulic engine is located towards the front of the ship near a sleeping compartment for the crew. Initially, noise levels up to 94 dBA were measured in this compartment when running the system at full speed. After installing a new low noise level engine based on a direct drive rather than gear transmission and a set of passive pulsation dampers, a 20 dB attenuation in overall noise level was measured in the cabin bringing the maximum noise level down to 74 dBA [9]. At this point, further noise reduction is required in order to improve comfort in the sleeping compartment. Moreover, while effectively reducing the pressure pulsations by factors as high as one order of magnitude, the passive pulsation dampers also result in
a static pressure drop which in turns penalizes the performances of the hydraulic system. This motivates the use of alternative control methods such as the one investigated in this work.

2.2 Dimensions and estimated properties

Both incoming and outgoing lines are standard steel pipes. The incoming pipe has a 38 mm outside diameter and a 5 mm wall thickness. The outgoing pipe has a 60.3 mm outside diameter and a 2 mm wall thickness. The contained fluid is hydraulic oil. The above pipe dimensions are recalled in Table 2.1 along with estimated values for the material and fluid properties of the system. These values will be used in the next chapter for modeling the system. Note that the fluid density and bulk modulus estimates slightly differ from those found in standard tables. They correspond to more realistic values commonly measured in oil piping systems as discussed in Reference [10].

<table>
<thead>
<tr>
<th></th>
<th>incoming line</th>
<th>outgoing line</th>
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<tbody>
<tr>
<td>pipe mid-surface radius (mm)</td>
<td>16.5</td>
<td>29.15</td>
</tr>
<tr>
<td>pipe wall thickness (mm)</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>pipe Young’s modulus (GPa)</td>
<td>192</td>
<td></td>
</tr>
<tr>
<td>pipe mass density (kg/m³)</td>
<td>7700</td>
<td></td>
</tr>
<tr>
<td>pipe Poisson ratio</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>pipe loss factor</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>fluid bulk modulus (Gpa)</td>
<td>1.24</td>
<td></td>
</tr>
<tr>
<td>fluid mass density (kg/m³)</td>
<td>871</td>
<td></td>
</tr>
<tr>
<td>fluid loss factor</td>
<td>0</td>
<td></td>
</tr>
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Table 2.1: Dimensions and estimated properties of the piping system

Table 2.2 presents additional properties of the system derived from the values found in Table 2.1. These properties are commonly used in the analysis of fluid-filled pipes. Their meaning will be presented Chapter 3. It should also be mentioned that the fluid velocity is approximately 4.5 m/s in both incoming and outgoing lines when running the system at full speed. This corresponds to a very small Mach number and the effects of non-zero fluid velocity on the waves propagating in the system can therefore be neglected [11]. Finally, the temperature of the hydraulic oil is approximately 50 °C.

<table>
<thead>
<tr>
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<th>incoming line</th>
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<tbody>
<tr>
<td>pipe thickness to radius ratio (h/a)</td>
<td>0.303</td>
<td>0.069</td>
</tr>
<tr>
<td>pipe ring frequency (Hz)</td>
<td>50492</td>
<td>28850</td>
</tr>
<tr>
<td>pipe longitudinal wave speed (m/s)</td>
<td>5235</td>
<td></td>
</tr>
<tr>
<td>fluid wave speed (m/s)</td>
<td>1193</td>
<td></td>
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Table 2.2: Characteristic properties of the piping system
2.3 Measured system response

This section briefly presents some of the measurements performed on the target system by Karlskronavarvet, AB. The following figures were taken out of a recent measurement report [9]. All data presented in this section was measured at full pump speed, i.e., with the engine running around 550 rpm.

Radiated sound

Figure 2.2 shows the auto-spectrum of the sound pressure level in dB measured at one location inside the sleeping compartment between 0 and 1000 Hz. The various tones present in the noise spectrum are indicated on the plot by specifying from top to bottom their frequency and amplitude in dB. The table on the right side of the plot presents the frequencies of the first 15 harmonics of the hydraulic engine. As seen on the plot, the sound radiated in the cabin is directly correlated to the harmonics of the hydraulic engine as well as to some harmonics of the main pump, thus clearly indicating the propagation of pressure pulses and associated radial wall motion throughout the system.

Pressure pulsations

The following figures present the pressure pulsations measured at various locations along the incoming and outgoing lines. Each figure shows the pressure signal in the frequency domain (auto-spectrum over 0–1000 Hz) as well as in the time domain over 0.1 s. Also shown are the frequencies of the first few harmonics of the pump and/or engine in order to facilitate the identification of the various tones. The reader should refer to Figure 2.1 to locate the measurement points associated with each figure shown below.
On the pump pressure side, before the pulsation damper, the pressure pulses feature two main tones around 250 Hz and 380 Hz with peak-to-peak amplitudes reaching 1.4 MPa (see Figure 2.3). Both tones correspond to the second and third harmonics of the pump rotation speed. At location 2, i.e., on the engine pressure side before the pulsation damper, the two above tones decrease in amplitude as seen in Figure 2.4. The decrease is of about one order of magnitude for the first tone with a peak-to-peak amplitude around 0.15 MPa while the amplitude of the second tone decreases even further. This attenuation is mainly due to the pulsation damper installed after the pump, as well as some damping of the waves traveling over the incoming line. It should be pointed out that the damper is more effective at reducing the higher tone above 200 Hz. On the other side of the pulsation damper (location 3), the pulses peak-to-peak amplitude increases with high values around 0.15 MPa. The auto-spectrum shown in Figure 2.5 presents more tones with frequencies related to harmonics from both pump and engine. The pump first and second harmonics are further attenuated, but other harmonics at higher frequencies cause the overall level of pulsations to increase. This measurement shows that pulsations induced by the engine can propagate backwards, i.e., against the direction of flow. Now looking at measured pressure fluctuations on the engine return side, high levels of pulsations are present before the pulsation damper with peak-to-peak amplitudes reaching 0.1 MPa as seen in Figure 2.6. Note that all tones are now solely associated with the harmonics of the engine rotation speed. Downstream the pulsation damper, all the above tones are significantly reduced except for the first one whose frequency, 65 Hz, is too low for effective passive attenuation (see Figure 2.7). The overall peak-to-peak amplitude is around 0.03 MPa at this location. The pressure measurements shown above also indicate the static pressure levels in the incoming and outgoing line, reaching about 20 MPa and 2 MPa, respectively. The static pressure drop across the pulsation dampers can also be seen on the plots.

To summarize, the pressure pulses in between the two pulsation dampers have maximum peak-to-peak amplitudes around 0.15 MPa in the incoming line and 0.1 MPa in the outgoing line. The above levels are sufficient to induce radial wall motion and radiating noise into the surrounding air, as well as excite other structural radiators through the pipe attachments (see Figure 2.2).

**Pulsation damper performance**

Figure 2.8 presents the performance of the first pulsation damper located on the pump pressure side. The plot shows the pressure attenuation in dB within third octave bands between 0 and 1100 Hz. Passive attenuation levels reach 25 dB in some bands above 250 Hz. However, the device become much less efficient below this frequency, where the attenuation drops to about 5 dB. Therefore, an active attenuation device is needed in order to further improve attenuation at lower frequencies without inducing further static pressure drop which would result from adding passive dampers in series.
Figure 2.3: Pressure pulsations before damper on pump pressure side (location 1)
Figure 2.4: Pressure pulsations before damper on engine pressure side (location 2)
Figure 2.5: Pressure pulsations after damper on engine pressure side (location 3)
Figure 2.6: Pressure pulsations before damper on engine return side (location 4)
Figure 2.7: Pressure pulsations after damper on engine return side (location 5)
Figure 2.8: Pressure attenuation across pulsation damper (pump pressure side)
Chapter 3

Low frequency vibrations of fluid-filled pipes

This section recalls the basic mechanisms involved in the vibrations of fluid-filled cylindrical thin shells. The understanding of these mechanisms is critical in order to successfully apply active control to hydraulic piping systems. In particular, the different wave types and their relative contribution to the overall vibrations need to be assessed. The following discussion is based on the assumptions that the dynamic behavior of the pipe under consideration can be described using thin cylindrical shell theory. Also, the discussion of the characteristics of the system vibrational response is limited to the low frequency region. As mentioned in Chapter 2, the characteristics of the target system satisfy both above assumptions.

Several authors have developed theoretical models for the vibrations of fluid-filled pipes. A comprehensive discussion of wave types in thin elastic fluid-filled cylindrical shells was given by Fuller and Fahy [12]. Their model which solves the coupled shell equations using a numerical root searching technique revealed a fairly complex behavior of the shell. A number of simplified models valid in the low frequency region have been proposed in order to simplify the analysis of energy flow in pipes related to practical applications [4,13,14]. All these models become equivalent at very low frequencies. The following discussion is based on Pinnington’s model whose main results are recalled below. The reader is referred to Reference [14] for a complete derivation.

3.1 Propagating waves in fluid-filled pipes

The two main frequency regions of a fluid-filled pipe are separated by the ring frequency, $\omega_0$, defined as $\omega_0 = c_L/a$ where $c_L$ is the compressional wave speed in a plate of same material 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 wave-length in a plate, $\lambda_L = 2\pi c_L/\omega_0$. In order to simplify the shell equations, it is convenient to normalize the excitation frequency $\omega$ by the above ring frequency. The non-dimensional frequency is then defined as $\Omega = \omega/\omega_0 = \omega a/c_L$. For steel pipes of common sizes, the ring frequency is usually much greater than the excitation frequency of potential sources such as pumps (see Tables 2.2). In particular, the low frequency excitation resulting from pressure pulsations will usually occur at very small values of non-dimensional frequency, i.e., $\Omega \ll 1$, and a simplified model is sufficient to estimate the system response.

Assuming a thin cylindrical shell, structural vibrations can be decomposed in a set of circumferential modes of order $n = 0, 1, 2, \ldots$ and associated waves propagating along the shell axis in the positive and
negative directions. The axial distribution of these waves can be expressed as \( \exp[j(\pm k_{s,n}x - \omega t)] \) where \( k_{s,n} \) is the wave-number of the \( s^{th} \) wave associated with circumferential mode \( n \). In the above expression, \( \omega \), \( t \), and \( x \) denote the angular frequency, the time, and the axial location, respectively. Figure 3.1 presents the mode shapes of the first four circumferential modes. The first mode \( n = 0 \) is referred to as the breathing or pulsating mode due to its axisymmetric motion. In the case of the \( n = 1 \) mode (bending mode), the pipe cross section remains undeformed. Higher order modes features lobar type mode shapes with \( 2n \) anti-nodes along the pipe circumference, i.e., the spatial distribution of mode \( n \) takes the form \( \cos(n\theta - \gamma_n) \) where \( \theta \) is the circumferential angle and \( \gamma_n \), a polarization angle. Below the ring frequency, four wave types can propagate energy \[4,14\]. These propagating waves feature a purely real wave-number. 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 above wave types can be further described in terms of their structural and fluid components. The first axisymmetric wave \((s = 1)\) 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. It will be referred to as the longitudinal wave and denoted by the subscript \( l \). The second axisymmetric wave \((s = 2)\) 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 and denoted by the subscript \( a \). The other axisymmetric wave referred to as the torsional wave (subscript \( t \)) 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 \( n = 1 \) flexural or bending wave (subscript \( b \)) 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 \[12\]. Fahy derived approximate formulae for the cut-on frequencies of the first few circumferential modes in the case of an in-vacuo pipe \[15\].

![Figure 3.1: First four circumferential mode shapes](image-url)
the cut-on frequencies of the \( n = 2 \) and \( n = 3 \) modes are approximated as

\[
\begin{align*}
    n = 2, & & \frac{h}{\sqrt{12}a} 2.68f_{\text{ring}} \\
    n = 3, & & \frac{h}{\sqrt{12}a} 7.65f_{\text{ring}}
\end{align*}
\]

Recalling the ring frequencies given in Table 2.2, the \( n = 2 \) and \( n = 3 \) cut-on frequencies of the target system are 1526 and 4355 Hz, respectively, for the outgoing line and above 10 kHz for the incoming line according to equations (3.1) and (3.2). Despite the effect of fluid loading which will slightly reduce the above values by mass loading the pipe, the actual cut-on frequencies are expected to be outside the frequency range of interest. Consequently, the associated modes are not included in this analysis.

All four propagating waves described above do not contribute equally to the sound radiated from the pipe wall. The \( n = 0 \) axisymmetric waves will radiate sound proportionally to the associated wall motion along the radial direction. In the far field, an equivalent source would be a line of monopoles. Due to its purely tangential motion, the torsional wave does not radiate sound. On the other hand both the longitudinal and acoustic waves radiate sound. Note that the longitudinal wave has its main motion along the axial direction. Therefore, for similar amplitudes this wave will radiate less efficiently than the acoustic wave whose main wall motion occurs along the radial direction. The bending wave associated with the \( n = 1 \) mode also radiates sound. In the far field, this wave radiates as a line of dipole with amplitude proportional to the pipe deflection. Dipoles have a smaller radiation efficiency compared to monopoles. However, the pipe deflexion can be several orders of magnitude greater than the radial wall motion associated with the axisymmetric waves.

For the target system considered in this report, the pressure pulsations resulting from the pump and hydraulic engine constitute the main source of vibrations. Therefore, axisymmetric wave types are expected to dominate the system response and the remaining part of the analysis will disregard the torsional and bending waves.

### 3.2 Dispersion relations

The first step towards modeling fluid-filled pipe systems is to obtain the dispersion relations which give for each wave type the associated wave-number in terms of frequency.

For simplicity, an end-excited semi-infinite pipe is considered so that only positive traveling waves are taken into account. The results presented below can be easily extended to an infinite pipe with both negative and positive traveling waves. Figure 3.2 presents the shell coordinate system along with the three orthogonal mid-surface displacements. Using the subscripts mentioned earlier to denote the acoustic and longitudinal waves, the axial and radial mid-plane displacements of the pipe are expressed respectively as

\[
\begin{align*}
    u(x) &= U_a e^{-jka x} + U_l e^{-jk_l x} \\
    w(x) &= W_a e^{-jka x} + W_l e^{-jk_l x}
\end{align*}
\]

where the \( e^{\omega t} \) time dependence has been suppressed for clarity. The axial wave-numbers for the acoustic and longitudinal waves are denoted \( k_a \) and \( k_l \), respectively. The associated axial motion, \( u_f \), and pressure, \( p \), in the fluid are given by

\[
\begin{align*}
    u_f(x) &= U_f a e^{-jka x} + U_f a e^{-jk_l x} \\
    p(x) &= P_f e^{-jka x} + P_l e^{-jk_l x}
\end{align*}
\]
Based on Pinnington’s simplified model, the acoustic and longitudinal wave-numbers are approximated as [14]

\[ k_a = k_f \left( \frac{1 + \beta - \nu^2 - \Omega}{1 - \nu^2 - \Omega^2} \right)^{1/2} \quad (3.7) \]

\[ k_l = k_L \left( \frac{1 - \Omega^2 + \beta}{1 + \beta - \Omega^2 - \nu^2} \right)^{1/2} \quad (3.8) \]

In the above dispersion relations, \( k_f \) denotes the fluid wave-number and \( k_L \), the wave-number of a compressional wave in a flat plate. They are defined respectively as \( k_f = \omega/c_f \) and \( k_L = \omega/c_L \). The sound speed in the fluid, \( c_f \), is expressed in terms of the fluid bulk modulus, \( B \), and density, \( \rho_f \), as

\[ c_f = \left( \frac{B}{\rho_f} \right)^{1/2} \quad (3.9) \]

The plate compressional wave speed is given by

\[ c_L = \left( \frac{E}{\rho_s(1 - \nu^2)} \right)^{1/2} \quad (3.10) \]

where \( E \) is the shell Young’s modulus, \( \rho_s \), its density, and \( \nu \), the Poisson’s ratio. Finally, \( \beta \) represents a fluid loading term given by

\[ \beta = \frac{2Ba}{Eh(1 - \nu^2)} \quad (3.11) \]

where \( h \) denotes the shell thickness.

To illustrate the dispersion relations given in equations (3.7) and (3.8), the wave-number of the acoustic and longitudinal waves are plotted versus frequency in Figures 3.3 and 3.4 for the incoming and outgoing lines (using the properties given in Table 2.1), respectively. The wave-numbers associated with the torsional
Figure 3.3: Incoming line dispersion curves

Figure 3.4: Outgoing line dispersion curves
and bending waves are also shown in the above figures for comparison. They are computed according to the dispersion relations derived by Pavic [4]. Based on the plots shown in Figures 3.3 and 3.4, the wave-length associated with a given propagating wave and frequency can be estimated as $\lambda = 2\pi/k_{l,a,t,b}$.

### 3.3 Relative contribution of the axisymmetric waves

Applying the boundary conditions at the fluid-structure interface, i.e., the fluid particle displacement in the radial direction equals the pipe wall radial displacement, the acoustic pressure amplitudes are related to the radial displacement amplitudes for both wave types as [14]

$$W_a = \frac{a^2}{hE} \left( \frac{1}{1 - \Omega^2/(1 - \nu^2)} \right) P_a$$  \hspace{1cm} (3.12)
$$W_l = -\frac{a}{2B} P_l$$  \hspace{1cm} (3.13)

The relations between the displacements of the acoustic and longitudinal waves can also be found as

$$U_a = -j\frac{\nu}{\omega a} W_a$$  \hspace{1cm} (3.14)
$$U_l = -j\frac{1 + \beta - \Omega^2}{\nu k_l a} W_l$$  \hspace{1cm} (3.15)

Combining equations (3.12), (3.13), (3.14), and (3.15), the ratio of the radial amplitudes for the two waves is obtained in terms of the pressure and axial amplitude ratios as

$$\frac{W_a}{W_l} = \left( \frac{-\beta}{1 - \nu^2 - \Omega^2} \right) \frac{P_a}{P_l}$$  \hspace{1cm} (3.16)
$$\frac{W_a}{W_l} = \left( \frac{1 + \beta - \Omega^2}{\nu^2} \right) \frac{k_a U_a}{k_l U_l}$$  \hspace{1cm} (3.17)

The above ratios are now specialized in the case of two types of excitation applied to the finite end of the pipe. These two cases will give an idea of the contribution of both longitudinal and acoustic waves to the response of straight pipes under various configurations. In the first case, the shell is excited with a pressure release boundary condition applied to the fluid, i.e., $p(x = 0) = 0$ or $P_a = -P_l$. The ratio of radial displacement amplitudes in equation (3.16) then reduces to

$$\frac{W_a}{W_l} = \frac{\beta}{1 - \nu^2 - \Omega^2}$$  \hspace{1cm} (3.18)

In the second case, the shell is free at $x = 0$ while the fluid is excited, i.e., the axial stress in the shell is zero at $x = 0$. The axial stress in the shell is given in terms of axial and radial displacements as [16]

$$\sigma(x) = \frac{E}{1 - \nu^2} \left( \frac{\partial u}{\partial x} + \nu \frac{w}{a} \right)$$  \hspace{1cm} (3.19)

Substituting equations (3.3) and (3.4), the above expression becomes

$$\sigma(x) = -j\rho_s\omega^2 \left( \frac{U_a}{k_a} e^{-jk_ax} + \frac{U_l}{k_l} e^{-jk_lx} \right)$$  \hspace{1cm} (3.20)

Setting $\sigma(x) = 0$ at $x = 0$ yields $U_a/U_l = -k_a/k_l$ which can be substituting into equation (3.17) to give the
ratio of the radial displacement amplitudes as

\[
\frac{W_a}{W_l} = -\left(\frac{k_a}{k_l}\right)^2 \left(\frac{1 + \beta - \Omega^2}{\nu^2}\right)
\] (3.21)

Pinnington and Briscoe also derived estimates of the energies per unit length of the pipe, \(e_a\) and \(e_l\), for the acoustic and longitudinal waves, respectively, as [14]

\[
e_a \approx \frac{\pi a^2 k_a^2}{\rho_f \omega^2} \bar{P}_a^2
\] (3.22)

\[
e_l \approx 2\pi ah\rho_s \omega^2 \bar{U}_l^2
\] (3.23)

where \(\bar{P}_a^2\) and \(\bar{U}_l^2\) are the spatially averages mean-square pressure and axial displacement amplitudes. Assuming \(\Omega \ll 1\) and \(\nu^2 \ll \beta\), the ratio of the energies in the acoustic and longitudinal waves can be estimated by substituting equation (3.12) into (3.22), equation (3.15) into (3.23) and dividing the two resulting expressions to yield

\[
\frac{e_a}{e_l} \approx \frac{\nu^2(1 - \nu^2)^2}{\beta(1 + \beta)^2} \left|\frac{W_a}{W_l}\right|^2
\] (3.24)

The ratios derived above have been calculated using the properties of the incoming and outgoing lines, respectively, as given in Table 2.1. Both fluid and structural excitation cases are presented for two distinct frequencies, 200 and 600 Hz. The results are presented in Table 3.1 for the incoming line and Table 3.2 for the outgoing line. Note that plotting the ratios presented in the two tables versus frequency would reveal that

<table>
<thead>
<tr>
<th>Excitation type</th>
<th>Fluid</th>
<th>Structural</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency (Hz)</td>
<td>200</td>
<td>600</td>
</tr>
<tr>
<td>(</td>
<td>W_a/W_l</td>
<td>)</td>
</tr>
<tr>
<td>(</td>
<td>U_a/U_l</td>
<td>)</td>
</tr>
<tr>
<td>(</td>
<td>P_a/P_l</td>
<td>)</td>
</tr>
<tr>
<td>(e_a/e_l)</td>
<td>79690</td>
<td>79670</td>
</tr>
</tbody>
</table>

Table 3.1: Acoustic and longitudinal wave amplitude ratios for the incoming line

<table>
<thead>
<tr>
<th>Excitation type</th>
<th>Fluid</th>
<th>Structural</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency (Hz)</td>
<td>200</td>
<td>600</td>
</tr>
<tr>
<td>(</td>
<td>W_a/W_l</td>
<td>)</td>
</tr>
<tr>
<td>(</td>
<td>U_a/U_l</td>
<td>)</td>
</tr>
<tr>
<td>(</td>
<td>P_a/P_l</td>
<td>)</td>
</tr>
<tr>
<td>(e_a/e_l)</td>
<td>23940</td>
<td>23930</td>
</tr>
</tbody>
</table>

Table 3.2: Acoustic and longitudinal wave amplitude ratios for the outgoing line

they remain roughly constant across the low frequency region considered in this work.

Examining the amplitude ratios obtained for the incoming line (see Table 3.1), the fluid type excitation yields a dominant acoustic wave. The radial wall motion and fluid pressure exhibit very small contribution from the longitudinal wave. Along the axial direction, the relative displacement associated with the acoustic wave is
not as large compared to that of the longitudinal wave. This result is expected as most of the energy associated with the longitudinal wave is associated with the structural axial motion, while the energy associated with the acoustic wave is mostly contained in the fluid. The energy ratio confirms the above results: only a very small portion of the total energy is contributed by the longitudinal wave. Now considering the structural type excitation, all above ratios are now inverted thereby exhibiting a dominant contribution from the longitudinal wave rather than the acoustic wave. As shown in Table 3.2, similar results are obtained in the case of the outgoing line.

In conclusion, for fluid excitation, almost all energy is contained in the acoustic wave. Thus there is much to gain in controlling the acoustic wave. On the other hand, the structural excitation shifts most of the energy in the longitudinal wave. This wave should then be controlled. As discussed previously, the target system is mainly excited by the pressure pulsations associated with the acoustic wave. Therefore, amplitude ratios close to those obtained for the fluid type excitation are to be expected and the acoustic wave is the main wave to control. Now considering the radial amplitude ratios, it can be seen that exciting the wall will produce larger amplitudes for the acoustic wave than for the longitudinal wave even though the latter will also be excited to some extent. This last result motivates the design of a fluid wave actuator based on radial wall excitation.

It should be noted that additional active or passive devices might also be required in cases where the longitudinal wave amplitude along the axial direction introduces control spillover through wave coupling occurring at discontinuities along the pipe. At such discontinuities, waves are coupled which means that reflected and transmitted waves can be of different types than the incident wave. As an example, a longitudinal wave incident on a pipe bend can couple into a bending wave. This is relevant to sound generation: even though a longitudinal wave does not radiate high levels of sound, the transmitted bending wave resulting from the pipe discontinuity may produce significant sound radiation. Also, the actuator itself constitutes a discontinuity and can therefore excite unwanted waves causing control spillover, i.e., the acoustic wave and associated radiated sound is effectively reduced but a second wave is excited resulting in an overall increase in radiated sound.
Chapter 4

Non-intrusive fluid wave actuator

This chapter discusses the design of the actuator implemented in the experimental testing described in Chapters 5 and 6. After a brief presentation of the actuator concepts, two theoretical models used in predicting the actuator performance are derived. The actuator characteristics and dimensions are then recalled along with results from the numerical simulations. It should be mentioned that the test pipe system used in the experiments including the actuator are based on the outgoing line characteristics of the target system (engine return side) outlined in Section 2.2.

4.1 Concepts

As discussed in the previous chapter, the actuator should excite the axisymmetric acoustic wave such that the resulting pressure pulsations cancel the primary pulsations propagating in the pipe work. This wave is mainly traveling in the fluid. Therefore a possible approach is to use a fluid acoustic source located inside the pipe as the control actuator. Compared to a structural actuator, such an actuator has the advantage of directly driving the fluid, thus increasing its performance as well as reducing the risk of exciting unwanted waves through the structure. The high pressure and small diameter characteristics of the target system, however, make a direct fluid actuator difficult to implement, and a more practical non-intrusive type of actuator was chosen instead. Namely, the acoustic wave is excited by driving the pipe wall along the radial direction in its axisymmetric mode. The induced radial wall motion can in turn drive the fluid through structure/fluid coupling.

The actuator consists of a ring of piezo-electric stacks acting on the pipe outside wall to produce radial motion. Practically, the stack ring is mounted on a short pipe section with reduced wall thickness in order to increase radial wall compliance. This pipe section will be referred to as the active insert in the remaining part of the discussion. In the standard configuration, all piezo-electric stacks are driven in phase in order to apply axisymmetric pressure along the pipe circumference. Piezo-electric materials are materials that change their dimensions when a voltage is applied and produce a charge when pressure is applied. Piezo-electric stacks also commonly referred to as PZT (Plumbum (lead) Zirconate Titanate) actuators can produce very large forces with displacement amplitudes on the order of $10 \mu m$ for stacks approximately 3 cm in height and 2 cm in diameter. Due to the relatively small displacements required to create pressure waves in a fluid medium and the large stiffness of pipes under axisymmetric excitation, this type of actuators was chosen over more conventional types such as electro-magnetic actuators.
4.2 Actuator modeling

Two models are derived in the following section to estimate the generated acoustic fluid pressure for a given stack configuration and applied voltage. The active insert is represented as a finite elastic ring of length $L$ over which the pipe squeeze actuator applies uniform pressure. This flexible ring is connected to two semi-infinite rigid pipe sections. The pipe radial deformation is approximated by the static deformation of the ring generated by the piezo-electric stacks. This assumption is valid as the frequencies of interest are well below the natural frequency of the first flexural modes of the pipe section. The pipe wall motion is assumed axisymmetric due to the assumption of uniform pressure load around the ring circumference. The external pressure, $P_a$, generates a radial wall displacement, $w$, ($w$ oriented inward). The resulting fluid volume displacement yield a fluid particle displacement, $u_f$, and pressure, $P_f$, at the two interfaces between the ring and the connected pipe sections. This simplified system is shown in Figure 4.1.

![Figure 4.1: Elastic ring connected to two semi-infinite rigid pipes](image)

The first model derived below neglects the effect of fluid loading on the dynamic response of the ring. A second model including fluid loading will be presented in the following section.

4.2.1 Stack/ring interaction disregarding fluid loading

Radial wall displacement

According to a simple static model of the ring, the radial deformation is related to the applied pressure as [17]

$$w = \frac{a^2}{Eh} P_a$$  \hspace{1cm} (4.1)

where $E$ denotes the ring Young’s modulus, $a$, its mid-plane radius and $h$, the wall thickness. The external pressure, $P_a$, is expressed in terms of the axial force in each piezo-electric stack, $F_a$, as

$$P_a = \frac{N_a F_a}{2\pi a L}$$ \hspace{1cm} (4.2)

where $N_a$ is the total number of stacks along the ring circumference. Substituting equation (4.2) in (4.1), the ring radial displacement becomes expressed in terms of the total radial force, $N_a F_a$, as

$$w = \frac{N_a F_a}{k_r}$$ \hspace{1cm} (4.3)
where \( k_r \) is the equivalent ring radial stiffness,

\[
k_r = 2\pi L \frac{E h}{a} \quad (4.4)
\]

Let \( k_a \) denote the stack stiffness and \( \Delta l_0 \), the stack unconstrained displacement for a given applied voltage \( (\Delta l_0 > 0 \text{ for positive voltages}) \). The stack stiffness is obtained experimentally from the measured blocked force, \( F_{a,0} \), at a given voltage and the associated unconstrained displacement, \( \Delta l_0 \), as

\[
k_a = \frac{F_{a,0}}{\Delta l_0} \quad (4.5)
\]

Neglecting the influence of the electrode layers in the stack, it can also be estimated from the piezo-electric material properties as [18]

\[
k_a = \frac{A_a}{N h_a S_{33}} \quad (4.6)
\]

where \( A_a \) is the stack cross section area, \( N \), the number of layers, \( h_a \), the thickness of one layer, and \( S_{33} \), the elastic compliance coefficient measured at constant electric field. Note that, in practice, the above theoretical value is reduced by the compliance of the bond joints and electrodes between the ceramic layers. Finally, the unconstrained displacement is obtained in terms of the applied voltage, \( V \), as

\[
\Delta l_0(V) = N d_{33} V \quad (4.7)
\]

where \( d_{33} \) is the piezo-electric constant along the poling direction. When the stack is mounted on an infinitely rigid structure, the force generated in the stack for a voltage, \( V \), is \( F_a = F_{a,0} = k_a \Delta l_0(V) \). If the structure is compliant, its displacement along the stack axis will be non-zero and the generated force will decrease proportionally. Recalling \( w \) denotes the pipe radial displacement oriented inward, the force in each stack becomes

\[
F_a = k_a (\Delta l_0(V) - w) \quad (4.8)
\]

Combining equations (4.3) and (4.8), the pipe radial wall displacement is given as

\[
w = \left( 1 + \left( \frac{k_r}{N a k_a} \right) \right)^{-1} \Delta l_0(V)
= \left( 1 + \frac{2\pi L E h}{a N a k_a} \right)^{-1} \Delta l_0(V) \quad (4.9)
\]

**Fluid volume displacement**

The fluid volume displacement induced by a small radial displacement, \( w \), directed inward is

\[
\Delta V = \pi L \left( (a - w)^2 - a^2 \right) \approx 2\pi a L w, \quad (w \ll a) \quad (4.10)
\]

Assuming plane wave fluid motion, the above volume displacement generates a fluid particle displacement, \( u_f \), across the ring end sections satisfying

\[
\Delta V = 2u_f \pi a^2 \quad (4.11)
\]
Following the same assumption, the generated fluid pressure is expressed as

\[ P_f = j \rho_f c_f \omega u_f \]  

(4.12)

where \( c_f \) is the fluid sound velocity, and \( \rho_f \), the fluid mass density. Therefore, the estimated radial displacement required to produce a pressure perturbation, \( P_f \), is given by

\[ w = \frac{1}{L j \rho_f c_f \omega} P_f \]  

(4.13)

Substituting equation (4.9) in (4.13), the estimated fluid pressure is obtained in terms of applied voltage as

\[ P_f = j \rho_f c_f \omega \left( \frac{L}{a} \left( 1 + \left( \frac{k_r}{N_a k_a} \right) \right)^{-1} \Delta l_0(V) \right) \]

\[ = j \rho_f c_f \omega \left( a/L + \frac{2 \pi E h}{N_a k_a} \right)^{-1} \Delta l_0(V) \]

\[ = j \rho_f c_f \omega \left( \frac{L}{a} \right) \left( 1 + \left( \frac{2 \pi L E h}{a N_a k_a} \right) \right)^{-1} \Delta l_0(V) \]  

(4.14)

### 4.2.2 Stack/ring interaction including fluid loading

The shell radial displacement, \( w \), and fluid particle velocity, \( u_f \), are related to the external applied pressure, \( P_a \), and the fluid pressure, \( P_f \), through the receptance matrix [19] as

\[
\begin{pmatrix}
w \\
u_f
\end{pmatrix} = \begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix} \begin{pmatrix}
P_a \\
P_f
\end{pmatrix}
\]  

(4.15)

Combining equations (4.2) and (4.8), the averaged pressure applied by \( N_a \) piezo-electric stack actuators equally spaced around the ring circumference is expressed in terms of the stack unconstrained displacement and radial wall motion as

\[ P_a = \gamma_c (\Delta l_0(V) - w) \]  

(4.16)

where

\[ \gamma_c = \frac{N_a}{2 \pi a L} k_a \]  

(4.17)

The fluid particle velocity is related to the fluid pressure as

\[ u_f = \alpha_f P_f \]  

(4.18)

where

\[ \alpha_f \approx \frac{1}{j \rho_f c_f \omega} \]  

(4.19)

assuming the ring generates a plane wave inside the pipe.
Equations (4.15), (4.16), and (4.18) can be combined to yield

\[(1 + \alpha_{11}\gamma_c)w - \alpha_{12}P_f = \alpha_{11}\gamma_c\Delta l_0(V)\]  
\[(4.20)\]

\[\alpha_{21}\gamma_c w + (\alpha_f - \alpha_{22})P_f = \alpha_{21}\gamma_c\Delta l_0(V)\]  
\[(4.21)\]

The above linear system is solved for \(w\) and \(P_f\) as

\[w = \left[1 + \frac{1}{\gamma_c} \left(\alpha_{11} + \frac{\alpha_{21}\alpha_{12}}{\alpha_f - \alpha_{22}}\right)\right]^{-1}\Delta l_0(V)\]  
\[(4.22)\]

\[P_f = \frac{\alpha_{21}}{(\alpha_{11} + 1/\gamma_c)(\alpha_f - \alpha_{22}) + \alpha_{12}\alpha_{21}}\Delta l_0(V)\]  
\[(4.23)\]

where the coefficients of the receptance matrix, \(\alpha_{ij}\), are derived below.

The radial displacement is related to the fluid particle displacement as

\[w = \frac{a}{L} u_f\]  
\[(4.24)\]

For an in-vacuo ring under static loads \((P_f = 0)\), the radial displacement is related to the applied pressure by

\[\alpha_{11} = \left.\frac{w}{P_a}\right|_{P_f=0} = \frac{a^2}{Eh}\]  
\[(4.25)\]

From equations (4.24) and (4.25), the \(\alpha_{21}\) receptance term is approximated as

\[\alpha_{21} = \left.\frac{u_f}{P_a}\right|_{P_f=0} = \frac{aL}{Eh}\]  
\[(4.26)\]

Based on the principle of reciprocity, the \(\alpha_{22}\) term is given by

\[\alpha_{22} = \left.\frac{u_f}{P_f}\right|_{P_a=0} = -\alpha_{21}\]  
\[(4.27)\]

Finally, the \(\alpha_{12}\) term is obtained from the above relations as

\[\alpha_{12} = \left.\frac{w}{P_f}\right|_{P_a=0} = \frac{a}{L} \alpha_{22} = -\frac{a^2}{Eh}\]  
\[(4.28)\]

Substituting the above receptance coefficients in equations (4.22) and (4.23), the final expressions for the radial displacement and fluid pressure become respectively

\[w = \left[1 + \left(\frac{2\pi L Eh}{aN_ka}\right)(1 - \left(1 + \frac{1}{\rho_{fc}\omega aL}\right)^{(1)}(1)^{(1)})\right]^{-1}\Delta l_0(V)\]  
\[(4.29)\]

\[P_f = \left[\frac{1}{\rho_{fc}\omega aL}\left(\frac{a}{L}\right) + \frac{2\pi L Eh}{aN_ka}\right]^{-1}\Delta l_0(V)\]  
\[(4.30)\]

Note that equations (4.29) and (4.30) become equivalent to the in-vacuo case presented in equations (4.9) and (4.14) when

\[\rho_{fc}\omega aL \ll 1 \quad \text{and} \quad \rho_{fc}\omega L^2 aN_ka \ll 1\]  
\[(4.31)\]
respectively.

4.3 Actuator specifications

This section gives the specifications of the active insert. The pipe section and the mounting ring of the actuator are first described followed by the characteristics of the PZT stacks.

4.3.1 Pipe section and actuator mounting ring

The active insert consists of a 150 mm long section and a heavy steel ring located in the center of the section to hold the PZT stack actuators. A general drawing of the insert is shown in Figure 4.2.

![Figure 4.2: Actuator general drawing](image)

Six PZT stacks with inner holes are equally spaced around the mounting ring circumference. The stack actuators are held in place with a top and bottom saddle. Both saddles have a shoulder and a dowel to center the stack end plates. A compressing screw applies preload to the stack through the top saddle. The mounting ring and saddles are shown in Figure 4.3 along with their dimensions. Table 4.1 also gives the main dimensions of the actuator mounting ring. Note that the top saddle is made of aluminum while all other actuator parts are in steel.

Figure 4.4 shows a drawing of the pipe section and flanges. The pipe original wall thickness of 2 mm is machined down to approximately 1 mm over a length of about 40 mm centered in the middle of the section. This 1 mm wall thickness remains within the safety limits for the levels of static pressure present on the target system, while increasing the pipe wall compliance.

Both actuator ring and pipe sections were machined at Virginia Tech, Mechanical Engineering Department,
Figure 4.3: Dimensions of the actuator ring
<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>outer diameter (mm)</td>
<td>211.5</td>
</tr>
<tr>
<td>inner diameter (mm)</td>
<td>140.9</td>
</tr>
<tr>
<td>width (mm)</td>
<td>36</td>
</tr>
<tr>
<td>saddle width (mm)</td>
<td>35.1</td>
</tr>
</tbody>
</table>

Table 4.1: Main dimensions of the actuator mounting ring

Figure 4.4: Dimensions of the pipe section
according to the specifications given above. A photograph of the assembled actuator can be seen in Figure 4.5.

![Figure 4.5: Actuator ring, PZT stacks and pipe section](image)

4.3.2 PZT stack actuators

The six PZT stacks implemented in the actuator are manufactured by EDO Corporation, USA (Model E400-P3). They are high-voltage PZT stacks with a maximum driving voltage of 800 Volts peak-to-peak centered at +400 Volts DC. High-voltage stacks were chosen due to their greater stiffness which increases the stack blocked force compared to low-voltage stacks with thinner ceramic layers. Model E400-P3 is rated with an unconstrained axial stroke of 32 \(\mu\)m at 800 Volts. As mentioned earlier, the stack features an inner hole for cooling and positioning purposes. Its length, outer, and inner diameters are 32 mm, 25 mm, and 10 mm, respectively. Figure 4.6 shows a drawing of the stack along with its main properties. The stack is made of EDO EC-98 ceramic (Lead Magnesium Niobate) [18] whose main characteristics are shown in Table 4.2. Finally, Table 4.3 presents the stack performance parameters as provided by the manufacturer [18]. Regarding the operation of the stacks, EDO’s recommendations have been included in Figures 4.7 and 4.8 for reference.

As mentioned earlier, the actuator standard configuration applies axisymmetric pressure along the pipe
Performance Specifications (REV E)

1) Frequency Range: DC to 1000 Hz.
2) Temperature Range: -40°C to 100°C
3) Overall Length: 1.388 +/- 0.005 inches.
4) Overall Diameter: 1.09 inches, maximum.
5) Displacement: +0.0008 inches (nominal) @ 500 VDC.
6) Output Linearity: +/- 5% of Nominal.
7) Peak Actuator Voltage Rating: 800 Volts.
8) Maximum Operating Preload: 1000 pounds.
10) Minimum Preload (Required): 100 pounds.
11) Actuator Weight: Less than 0.50 pounds.
12) Capacitance (1.0 kHz): 0.55 +/- 0.10 µF
13) 24 Awg Lead Length: 17.0 inches +/- 1.0 inches.
14) 300 Series Stainless Steel Endplate, with 100°
    Counter Sink feature as shown, bonded to both ends.
    Thickness of 0.062 +/- 0.010 inches after finishing.
15) Values are referenced to 25°C (room temperature).
16) Built to best commercial practice.
17) Crimp Female Terminals, MOLEX P/N 39-00-0039.

P/N: E400P-3 (REV E)
IMPORTANT: READ THIS BEFORE OPERATING ACTUATOR

August 17, 1993

Dear Actuator Customer,

I have prepared a very brief list of "Do's and Don'ts" type suggestions you may wish to consider as you perform testing and evaluation efforts with your EDO actuator. This list is by no means complete, however, it may serve as a useful reminder in some important areas. Should you have any questions, please feel free to call me directly at (801) 486-7481 x354.

Do's:

1) Always pre-load the actuator(s) in compression to maximize adhesive bond joint life. (The EXO0P-Y actuator configurations use no biasing bolts to support tensile loads.)

2) Always try to bond the actuator in place using a suitable low viscosity epoxy over a very light layer of mold release. Bond line thicknesses of less than 0.001 inches are desirable and achievable using 500 psig preload during the curing process. This bonding process improves coupling by a significant factor. It also serves as an indicator that tensile loads were present if this bond is subsequently broken.

3) Always drive the stack with a +VDC bias applied such that the net displacement of the actuator is always positive. This will prevent unnecessary heating and loss of efficiency.

4) Prior to dynamic testing, apply the maximum anticipated voltage field across the actuator to verify stack integrity (do not exceed +800 volts peak-peak). Audible snaps, pops or arcing may indicate voltage breakdown has taken place. Voltage breakdown occurs due to excessively high voltage fields which are close to the poling voltage or contamination at the exposed surface of the laminated stack assembly. Once an actuator "arcs over", this voltage/field level cannot be surpassed unless the device is repaired. Typically, this involves the removal of a carbon trail which has formed a conductive path between adjacent electrodes of opposite polarity.

Actuator drive voltage should be kept below +800 volts (peak) or +40 volts/mil. (max. total). This might include a DC bias voltage of +420 volts (field = +21 volts/mil) and an AC drive voltage of +/-380 volts (field = +/- 19 volts/mil). The total maximum voltage field across the stack in this case equals +40 volts/mil. or +800 volts.

Figure 4.7: EDO do's and don'ts (1)
Piezoelectric Actuator Do's and Don'ts, Cont.

5) When not in use, always short out the (+) and (-) terminals of the actuator assembly. Piezoelectric materials are capacitors and tend to charge up when compressed, cooled or heated. Touching the (+) and (-) leads will cause a voltage discharge which can be shocking.

Don'ts

1) Don't load the actuator in tension. This is a repeat statement but since this particular actuator uses no biasing bolt, specifically for tensile loads, the bonded joints are moderately weak in tension and this warning bears repeating.

2) Don't apply opposite voltage to the actuator assembly. This reverse voltage can de-pole the actuator stack. Always apply (+) voltage to the red leads (positive) and (-) voltage to the black leads (negative).

3) Never exceed the voltage field rating of the actuator. Exceeding the maximum voltage can damage the actuator and cause partial or total depolarization of the device. A depolarized device loses its piezoelectric properties.

4) Do not touch the (+) and (-) leads of an actuator which has been left idle for any period of time. The piezoelectric device is a capacitor and charges up over time due to changes in temperature and applied stress. Prior to handling, short out the leads to prevent being shocked. If you forget to do this, the actuator will sometimes remind you!

5) Do not drop the actuator onto a hard surface. Ceramic is a tough material in general, however, surface edges are prone to chip when struck bluntly. This chipping action may create open sites where electrode materials are exposed. These are excellent sites for high charge build-up which may cause voltage breakdown at a later time. Typically, the actuators are coated with a suitable encapsulant to prevent surface contamination and prolong operational life.

Good Luck!!

Gordon D. Cook
EDO Corporation

Figure 4.8: EDO do’s and don’ts (2)
density \( (\times 10^3 \text{ kg/m}^3) \) & 7.85 \\
Young’s modulus \( (\times 10^{10} \text{ N/m}^2) \) & 6.1 \\
\( d_{31} \) \( (\times 10^{-12} \text{ m/V}) \) & -312 \\
\( d_{33} \) \( (\times 10^{-12} \text{ m/V}) \) & 730 \\
\( S_3^{E_3} \) \( (\times 10^{-12} \text{ m}^2/\text{N}) \) & 21.1 \\
\hline
\end{tabularx}

Table 4.2: EDO EC-98 ceramic main properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>frequency range</td>
<td>DC to 1000 Hz</td>
</tr>
<tr>
<td>temperature range</td>
<td>(-40^\circ \text{C} ) to (100^\circ \text{C})</td>
</tr>
<tr>
<td>unconstrained displacement</td>
<td>32 (\mu\text{m})</td>
</tr>
<tr>
<td>maximum operating preload</td>
<td>4440 N</td>
</tr>
<tr>
<td>maximum preload</td>
<td>8880 N</td>
</tr>
<tr>
<td>minimum preload</td>
<td>444 N</td>
</tr>
<tr>
<td>weight</td>
<td>225 g</td>
</tr>
<tr>
<td>capacitance ( (1 \text{ kHz}) )</td>
<td>(0.55 \pm 0.10 \mu\text{F})</td>
</tr>
<tr>
<td>layer thickness ( (\text{mm}) )</td>
<td>0.508</td>
</tr>
<tr>
<td>number of layers</td>
<td>(\approx 70)</td>
</tr>
<tr>
<td>measured blocked force at 800 Volts ((\text{kN}))</td>
<td>11</td>
</tr>
</tbody>
</table>

Table 4.3: PZT stack properties and performance specifications

circumference. This axisymmetric excitation is achieved in practice by wiring all six stacks in phase. It is therefore critical to obtain PZT actuators with identical dynamic properties.

4.4 Predicted performances

This section presents the main performance results of the actuator based on the two models derived in Section 4.2 and the properties of the piezo-electric stacks and pipe section given in Section 4.3. For clarity, the corresponding parameters introduced in the model are recalled in Table 4.4. Note that the ring length is based on the width of the active insert saddles (see Figure 4.3). Its material properties are based on standard values

\begin{tabularx}{\textwidth}{|l|X|}
\hline
ring length \( (\text{mm}) \), \( L \) & 35 \\
ring mid-radius \( (\text{mm}) \), \( a \) & 28.65 \\
ring wall thickness \( (\text{mm}) \), \( h \) & 1 \\
ring Young’s modulus \( (\text{GPa}) \), \( E \) & 192 \\
stack stiffness \( (\text{N/}\mu\text{m}) \), \( k_a \) & 344 \\
stack unconstrained displacement \( (\mu\text{m}) \), \( \Delta l_0 \) & 32 \\
number of stacks, \( N_a \) & 6 \\
fluid density \( (\text{kg/m}^3) \), \( \rho_f \) & 871 \\
fluid sound velocity \( (\text{m/s}) \), \( c_f \) & 1193.8 \\
\hline
\end{tabularx}

Table 4.4: Model numerical parameters
for steel. As mentioned previously, the above parameters correspond to the outgoing line of the target system.

The pressure pulsations generated by the active ring are shown versus frequency in Figure 4.9. The solid

![Figure 4.9: Generated fluid pressure at 800 Volts](chart)

line corresponds to the first model neglecting fluid loading (equation (4.14)) and the dashed line to the second model which includes fluid loading (equation (4.30)). In both cases, the stacks are driven to their maximum voltage of 800 Volts. As seen on the plot, the two curves are nearly identical. The frequencies considered here are very small compared to the pipe ring frequency which explains the small influence of fluid loading ($\Omega \ll 1$). As expected, the piezo-electric stacks generate constant radial wall displacement over the frequency range which couples to a fluid pressure amplitude increasing linearly with frequency. The pressure level generated by the actuator is around 0.03 MPa at 200 Hz and 0.09 MPa at 600 Hz. Recalling the pressure pulsations measured on the target system (see Figures 2.3 to 2.7), the theoretical performances computed above show the potential of the proposed fluid wave actuator in controlling unwanted pressure pulsations.

In order to further increase the control authority of the actuator, i.e., the generated fluid pressure, a number of modifications can be made. They include using PZT stacks with increased performance in terms of unconstrained displacement and blocked force, increasing the compliance of the pipe section, or the area of applied mechanical pressure. To illustrate the second approach, Figure 4.10 shows the ratio of the stack actual displacement over its unconstrained stroke versus the ring thickness. As expected, decreasing the pipe thickness increases the pipe compliance which in turn yields an increased radial displacement, i.e., the stack displacement approaches its unconstrained value. Similar results can be obtained using material with a reduced Young’s modulus. However, the above changes are limited by the safety requirements imposed on the pipe section.
Figure 4.10: Stack displacement factor
Chapter 5

Experimental setup

The experimental part of this work is aimed at testing the actuator on a real piping system with characteristics similar to those of the target system outgoing line. The actuator is first characterized in terms of its driving response, and then implemented in the control of fluid pulsations. This chapter presents the experimental setup and procedures used in testing the proposed control system.

5.1 Test system

The pipe system used in the experimental testing was assembled at Karlskronavarvet, AB in the hydraulic laboratory. The test rig is composed of two straight steel pipe sections connected to both ends of the active insert described in Section 4.3. A drawing of the rig is shown in Figure 5.1. The incoming and outgoing sections are 1 m and 1.5 m in length, respectively. They are made of the same material as the outgoing pipe of the target system with same wall thickness and diameter (see Table 2.1). Transmission of vibrations through the support bench is prevented by placing the pipe assembly on two large diameter inner tubes inflated with air as indicated in Figure 5.1. Both sections are connected to the active insert using standard thin joints installed between the flanges. Rubber joints used as passive attenuation device were not tested in the experiments due to lack of time. Figure 5.2 presents the conceptual drawing of such a joint to be installed between the two flanges of connecting pipes in order to attenuate the transmission of vibrations.

The above pipe assembly is connected to the return line of a hydraulic engine using a standard 2 inch reinforced flexible pipe. The hydraulic engine itself is driven by the system main pump. The downstream end of the pipe rig is connected to another section of flexible pipe going back on the return side of the main pump. A simplified schematic of the system is shown in Figure 5.3. The pressure before and after the engine can be adjusted using the main pump speed and two valves on the pressure and return sides of the engine, respectively. The system pressure, engine pressure and return pressure (see figure) are monitored by three gauges, respectively. Additional pressure sensors mounted on the pipe sections upstream and downstream of the active insert are described in the next section. A general view of the pipe rig is shown in Figure 5.4.

5.2 Active insert configuration

As mentioned in Section 4.3.2, preloading screws apply pressure on the PZT stack top saddles (see Figure 4.5). In order to ensure proper operation, EDO recommends a preload within 500 and 4000 N approximately. In practice, the appropriate preload is set by measuring the torque applied to the preloading screws. This torque has been calibrated using a static load cell in place of each one of the six piezo-electric stacks installed in the
both pipes are 60 mm in diameter with a 2 mm wall thickness 
flanges connecting the middle section are DN50 flanges

Figure 5.1: Schematic of the pipe rig and sensor locations
Figure 5.2: Passive isolation joints

Figure 5.3: Schematic of the test system
actuator ring. The applied torque versus measured load is shown in Figure 5.5 for all six preloading screws. All actuator tests use a preloading torque of 8 N.m, i.e., about 3500 N is applied to each stack. It should be
mentioned that preloading requires the stack to be shortened in order to prevent possibly harmful electrical charge buildup (see Section 4.3.2).

Driving all six stacks in phase requires a High-Voltage PZT amplifier capable of driving large capacitive loads over the frequency bandwidth of interest. The capacitance of the six stacks wired in parallel is approximately $6 \times 0.575 = 3.45 \mu F$. Assuming each stack is equivalent to a pure capacitor, the required current is obtained as $i = C \frac{dv}{dt}$, where $i$, $C$, and $v$ denote the current, capacitance, and applied voltage, respectively.

A 1000 Hz pure tone with a 400 Volts peak amplitude yields a 8.6 A peak current amplitude. The above power requirements could not be met by commercially available switching amplifiers and the amplifier used in the experiments was specially designed by BAB Elektronik, AB (Sweden) based on characteristic impedance data of the actuator PZT stacks. The amplifier uses a switching frequency of 35 kHz. An output low-pass filter is implemented to reduce noise in the driving signal above 1 kHz.

One important issue related to the operation of PZT stack actuators at non-zero driving frequencies is heat generation. As outlined above, PZT stacks are reactive loads and therefore require charge and discharge currents that increase with operating frequency. For standard PZT ceramics, the loss factor is on the order of 0.01 to 0.02. This means that up to 2% of the electrical power pumped into the actuator is converted to heat. For large amplitude and high frequency, cooling measures become necessary. For the current actuator, temperatures in excess of 70°C were measured on the outside of the PZT stacks with driving signals above 200 Volts RMS and frequencies above 300 Hz. Cooling was not implemented during the tests and the stacks were only driven to about 75% of their maximum driving voltage. Despite this precaution, overheating lead to the failure of one stack during the control experiments.

5.3 Measurement setup

Both pipe sections connected to the active insert are equipped with ICP pressure sensors from PCB (Model 111A21). This miniature pressure sensor is about 5 mm in diameter and 35 mm in height with a nominal sensitivity of 5.8 mV/kPa (50 mV/psi) and a dynamic range of 689.5 kPa (100 psi). Note that such ICP sensors only measure pressure fluctuations, i.e., the output signal is an AC signal. Each sensor is installed inside a mounting port welded to the pipe. The mounting hole is machined according to the flush installation drawing shown in Figure 5.6. The sensor locations are indicated in Figure 5.1. One sensor is located upstream of the active insert and three additional sensors are located on the downstream section. Two of the pressure sensors can be seen in Figure 5.7 along with the active insert.

Two PCB triaxial accelerometers (Model 356A08) are mounted on each pipe section connected to the active insert in order to estimate the pipe wall acceleration along the radial, axial, and tangential directions. The reader is referred to Figure 5.1 for the transducers location. Each of the three ICP accelerometer channels have a nominal sensitivity of 10 mV/g. Both sensors are glued to the pipe. The pipe outside wall was machined flat over a small area to allow proper mounting of the accelerometer.

The output signal from the pressure and acceleration transducers are conditioned through a multi-channel ICP power unit built at Karlskrona/Ronneby University. Each channel has a nominal gain of one. No calibration could be performed on the pressure sensor and the calibration data provided by PCB was used instead. A hand-held vibration calibrator (PCB, Model 394B06) was used to calibrate each output signal of the triaxial accelerometers.

As explained in the next section, the controller reference signal is based on harmonic tones of the engine rotation speed. To this purpose, an optical sensor is installed next to the engine drive axis as shown in the picture of Figure 5.8. The sensor provides a positive voltage output each time one of the four axis tooth passes
Figure 5.6: Pressure sensor mounting hole specifications
Figure 5.7: Active insert installed on the pipe system

Figure 5.8: Test hydraulic engine and optical sensor
the optical sensor. Thus, the sensor output signal is a square wave of time period one fourth the period of the engine rotation speed. In other words, the engine rotation speed is expressed in revolutions per minute (rpm) as $60 \times f_{opt}/4$ where $f_{opt}$ is the fundamental frequency of the optical sensor output square wave.

All sensor output signals can be analyzed in the frequency domain using a two channel HP analyzer (Model 35665A). Note that all frequency domain spectra estimated on the HP analyzer use 20 averages and a flat top window between 0 and 800 Hz unless otherwise noticed. Compared to other windows, the flat top window is better suited to the type of signals measured on the pipe system as it improves the accuracy of the spectral estimate at frequencies associated with well separated tones.

In addition, the signals can be recorded simultaneously in the time domain using a TEAC 16 channel digital tape recorder. Post-processing of the time domain data is then performed in Matlab. All recorded signals are sampled at 6 kHz.

## 5.4 Control arrangement

A single channel feedforward active control system is used to achieve attenuation of the pressure pulsations. This section describes the controller and its practical implementation.

### 5.4.1 Control algorithm

The control input signal applied to the fluid wave actuator is constructed in real time on a Texas Instrument TMS320C30 digital signal processor (DSP). The code implemented on the DSP is based on a broadband Filtered-$x$ LMS algorithm [20]. Figure 5.9 shows the block diagram of the single channel Filtered-$x$ algorithm. The algorithm implements an adaptive digital filter, $W(z)$, to cancel the plant response due to the primary disturbance, $d(n)$, by feeding forward a reference signal, $x(n)$, correlated with $d(n)$. In the Filtered-$x$ LMS algorithm, the adaptive filter is updated in real time by using an instantaneous estimate of the gradient of a quadratic function of the error signal, $e(n)$. The update equation involves the error signal, $e(n)$, and the reference signal, $x(n)$, filtered through an estimate of the control path transfer function, $\hat{T}_{ce}(z)$. This transfer function represents the filtered-$x$ path. Recalling the pipe system described earlier, the error signal, $e(n)$, is the output of an error sensor related to the pressure pulsations. The plant disturbance path, $T_{de}(z)$ represents the transfer function between the disturbance, i.e., the hydraulic engine, and the error sensor output, and the control path, $T_{ce}(z)$, the transfer function between the control input signal, i.e., the signal driving the

![Figure 5.9: Schematic of the one-channel Filtered-x LMS algorithm](image)
fluid wave actuator, and the error sensor output. Finally, the reference signal should be correlated with the disturbance signal, i.e., related to the engine harmonic tones. The above algorithm has been widely used in active control of infinite and finite systems. It provides fast convergence and increased robustness compared to feedback control techniques in situations where modeling the plant dynamics becomes difficult. Also the algorithm appears to be tolerant of errors made in the estimation of the control path [21].

The controller used in this work implements Finite Impulse Response (FIR) digital filters for both compensator, \( W(z) \), and filtered-x path, \( \hat{T}_{ce}(z) \). The use of FIR filters in the filtered-x path is traditional in active noise control where the control path transfer functions do not present sharp resonance behavior. For the current application, the transfer function between the input signal to the actuator and the generated pressure pulsations in the fluid is expected to be relatively flat which also motivates the use of FIR filters. In this implementation, the filtered-x path FIR filters are modeled by feeding a known signal through the plant control path and measuring the resulting error signal, the plant disturbance being turned off. Off-line computations then allow to approximate the transfer function between the two signals based on the optimal solution of the associated Wiener-Hopf problem. This filtered-x path modeling is performed by the controller DSP prior to turning the controller on. Note that other types of filtered-x path modeling including on-line modeling during control are also available.

5.4.2 Control experimental setup

This section presents the experimental arrangement used in the control tests performed on the pipe system. A general schematic of the control setup is shown in Figure 5.10. The block diagram shows the various components of the system and the equipment used in the experiments. The pipe system is composed of the hydraulic engine, the pipe sections and the active insert. The optical sensor installed on the hydraulic engine provides information to the controller to construct a suitable reference signal. One of the pressure sensors mounted on the pipe sections provides the error signal to be minimized. Finally, the fluid wave actuator installed on the active insert provides the control input to the system. Further details about the reference, control, and error signals, as well as the controller settings are given below.
Reference signal

The output from the optical sensor is fed to a pulse counter. Based on the pulse rate of the signal, the sine wave generator implemented on the DSP then constructs the reference signal, \(x(n)\), containing harmonics of the engine rotation speed. The order and number of the harmonics to include can be set at run time by the user. Note that the generated reference signal is also fed to the output board of the DSP for monitoring purposes.

Control input

As shown in Figure 5.10, the control signal is low-pass filtered through a two channel Kemo digital filter (Model VBF 10M) with cut-off frequency set to 800 Hz in order to eliminate the higher frequency content introduced by the controller D/A converter. This signal is then amplified through the High-Voltage PZT amplifier introduced in Section 5.2.

Error signal

For all control tests, the error signal is taken from pressure sensor 3 installed on the downstream pipe section as indicated in Figure 5.1. The sensor output signal is conditioned through a PCB ICP power unit (Model 442A03) and then band-pass filtered through a second Kemo digital filter (Model VBF 10M). The filter cut-off frequencies are based on the set tones to control for each particular test case. Furthermore, the amplitude of the filter output is adjusted by setting the filter gain in order to use the full dynamic range of the A/D converter of the controller (±2 Volts). It should also be mentioned that the home made ICP power unit used for the remaining ICP transducers (pressure sensors and triaxial accelerometers) could not be used for the error signal in series with the Kemo filter due to impedance mismatch problems. A PCB ICP power unit (Model 443A01) was used instead.

Controller settings

For the purpose of consistency, all control tests use the same controller settings. The sampling frequency is set to 2000 Hz. The compensator and filtered-x path FIR filters use 100 and 350 coefficients, respectively, and the filtered-x path system identification is based on 50 averages.
Chapter 6

Experimental results

This section presents the experimental results obtained on the test system. The response of the system to the fluid wave actuator is first analyzed. Results from the active control experiments are presented in a second part.

6.1 Fluid wave actuator response

6.1.1 In-vacuo response

Prior to installation on the pipe system, the active insert was tested in-vacuo by driving the actuator with pure tones and measuring the resulting radial acceleration of the wall. The tests use a PCB accelerometer (Model 353B68) mounted inside the active insert underneath the stack ring. The acceleration amplitude is estimated with the HP analyzer using a flat-top window and 10 averages. The accelerometer output signal is calibrated using a hand-held calibrator. In the results presented below, the peak-to-peak amplitude of the radial displacement in μm is plotted versus driving voltage amplitude in Volts RMS. The peak-to-peak displacement, $w_{pp}$, is obtained from the measured acceleration RMS amplitude, $\ddot{w}_{RMS}$, as

$$w_{pp} = \frac{2\sqrt{2}}{(2\pi f)^2} \ddot{w}_{RMS}$$

where $f$ represents the driving frequency.

In the first set of measurements, the accelerometer is mounted underneath stack 6 (the six stacks of the actuator are numbered sequentially) and several actuator configurations are driven with a 388 Hz pure tone of increasing amplitude. The measured peak-to-peak displacements shown in Figure 6.1 correspond to four sets of stacks driven in phase. All six preloading screws of the actuator ring are tightened with a 8 N.m torque. In all four configurations, the measured displacement is quasi linear with respect to the driving voltage. As expected, the axisymmetric configuration (all 6 stacks driven in phase) yields the smallest radial displacement. This configuration drives the $n = 0$ mode of the ring which has a relatively small input mobility compared to the higher order modes. The second and third stack configuration (stacks 2, 4, 6, and 3, 6, driven in phase, respectively) correspond to the $n = 3$ and $n = 2$ modes, respectively, which present an increased mobility, thus resulting in larger radial wall displacement amplitudes. For the same reason, the fourth configuration where stack 6 only is driven yields the largest displacement amplitude.

It should be noted that the above displacement amplitudes are not proportional to generated fluid displacements. Despite relatively small radial displacements, the axisymmetric configuration is expected to yield
the largest fluid volume displacement and pressure amplitude. In all other configurations, the total volume displacement of the pipe wall is reduced due to cancellation between adjacent areas along the circumference with out-of-phase motion.

As mentioned previously, the stack maximum driving voltage could not be reached as appropriate cooling of the stacks was not available. The stack maximum driving voltage is 280 Volts RMS. Extrapolating the data shown in Figure 6.1, the radial displacement obtained at 280 Volts RMS is approximately 6 \( \mu \text{m} \) peak-to-peak when driving all six stacks. This value is well below the predicted performance of the actuator estimated in Section 4.4. Recalling the stack displacement factor shown in Figure 4.10, the estimated peak-to-peak amplitude is around 18 \( \mu \text{m} \) assuming a 32 \( \mu \text{m} \) unconstrained displacement and a 1 mm wall thickness. Several factors can explain this discrepancy between predicted and measured displacement amplitudes including actual stack stiffness and unconstrained displacement smaller than the nominal specifications. Also, the actual wall thickness of the active insert is not constant and varies between 1 and 1.5 mm, which decreases the pipe compliance.

The second set of measurements shows the influence of the driving frequency and preloading torque on the actuator response. Stacks 1, 3, and 5 are driven in phase with pure tones of increasing amplitudes at 60 Hz, 200 Hz, 400 Hz, and 600 Hz, respectively. Figures 6.2 and 6.3 present the measured radial displacement underneath stack 5 for the four above tones and torque values of 8 and 4 N.m, respectively. As shown in the previous set of measurements, the radial displacement amplitude is linear with respect to the driving voltage amplitude at all four frequencies and for both preloading torque values. Furthermore, the reduced preloading torque yields a small increase in the measured radial displacement amplitudes of about 10%. Note that the reduced torque value is large enough to ensure proper stack preloading, i.e., above the minimum preload specified by the manufacturer (see Figure 4.6).
Figure 6.2: Measured peak-to-peak displacement - stacks 1, 3, 5 - 8 N.m preload

Figure 6.3: Measured peak-to-peak displacement - stacks 1, 3, 5 - 4 N.m preload
6.1.2 Fluid-filled response

The active insert is now installed in the pipe system as described in Section 5.2. This section presents the response of the system excited by the fluid wave actuator. The fluid pressure levels generated by the actuator under various driving conditions are first discussed. The influence of the fluid static pressure on the actuator performance is then analyzed. The section ends with a discussion of the non-linear characteristics of the actuator response.

Actuator configuration

The following results present the pressure levels generated by the actuator for different stack configurations, driving voltages and frequencies. In all cases, the engine is at rest (closed return valve) and no external pulsations are present in the system. The fluid static pressure level is maintained around 20 bar.

In the first measurement set, all six piezo-electric stacks are driven in phase with a fixed amplitude sine wave at frequencies between 30 and 600 Hz with a 30 Hz increment. The driving voltage is set to 212 Volts RMS (6 Volts peak before the amplifier) which corresponds to 75% of the maximum driving voltage. Figure 6.4 shows the pressure levels measured at the four pressure sensors versus the driving frequency. As expected, the pressure pulsation levels increase as the driving frequency increases. Neglecting the effect of fluid loading, the radial wall displacement of the active insert is quasi constant over the 0–600 Hz frequency range (see previous section). Therefore, the associated fluid displacement is expected to be constant as well which results in increasing acoustic pressure with frequency. In contrast with the case of an ideal one-dimensional infinite medium, however, the pressure amplitude associated with a single driving tone is not constant along the pipe therefore suggesting the presence of additional waves in the fluid. These additional waves can be near-field...
waves as well as reflected waves generated by the discontinuities of the pipe work (pipe bends, valves, changes in wall compliance, etc.). As expected, their influence on the system response increases with frequency as the wave length decreases. In terms of peak-to-peak amplitudes, the measured pressure levels range in average from 0.006 MPa at 200 Hz to 0.015 MPa at 600 Hz. At maximum driving voltage, these levels are expected to be around 0.0075 MPa at 200 Hz and 0.02 MPa at 600 Hz, respectively. For the same reasons mentioned in the previous section, the above pressure levels are significantly smaller than the predicted values shown in Figure 4.9.

The second set of measurements compares the performance of different actuator configurations in terms of generated fluid pressure. Three different sets of stacks are driven successively in phase with a 212 Volts RMS pure tone at frequencies between 60 and 600 Hz with a 60 Hz increment. The three configurations correspond to excitation of all six stacks, stacks 1, 3, and 5 driven in phase, and stack 1, respectively. The measured fluid pressure is shown for sensor 3 in Figure 6.5. The static pressure level measured in each case is indicated on the plot for reference. As expected, the largest fluid pressure amplitude is obtained by driving all six stacks in phase thereby exciting the pipe in its axisymmetric mode. Driving three stacks only yields smaller pressure levels. As mentioned previously, mode \( n = 3 \) presents a greater input mobility compared to mode \( n = 0 \) which results in larger radial displacements. However, the overall volume displacement of the wall is reduced compared to the axisymmetric excitation due to canceling of adjacent areas with out-of-phase motion. With a single stack driving the pipe, despite a larger displacement amplitude (see Figure 6.1), the volume displacement further decreases due to the smaller surface area of the wall moving the fluid. Note that the same trends can be observed on the pressure measured at the remaining three sensors.

The third set of measurements was performed on the five stack actuator used in some of the active control tests after stack 6 had been damaged through overheating (see Section 5.2). In this case, the actuator is driven
at frequencies between 120 and 420 Hz with a 60 Hz increment and the same 212 Volts RMS driving voltage as in the two previous cases. The measured fluid pressure is presented in Figure 6.6 for the four pressure sensors. Interestingly, the pressure levels are slightly greater than in the first case where all 6 stacks are operated in phase (see Figure 6.4). This suggests that the increased input mobility associated with the 5 stack actuator results in a increased overall fluid volume displacement due to relatively small canceling effects. This is in contrast with the previous case based on stacks 1, 3, and 5. In that case, the canceling of adjacent areas results in a smaller volume displacement despite the increased mobility of the wall along the radial direction.

Actuator non-linearities

In this section, the non-linear behavior of the actuator is analyzed. The actuator is driven successively at three frequencies, 180, 300, and 420 Hz, and the fluid pressure and wall acceleration spectra are estimated over the 0–1600 Hz bandwidth. The following figures present bar plots of the system response at the driving frequency and its first harmonics within the above bandwidth. In all cases, the six piezo-electric stacks are driven in phase with a 212 Volts RMS harmonic signal as in the previous sections.

Results for the first driving frequency (180 Hz) are presented in Figure 6.7. The top plot shows the pressure amplitude measured at the four pressure sensors. The radial and axial acceleration at locations 1 and 2 (see Figure 5.1) are shown on the bottom plot. Significant vibration levels can be observed at all harmonics of the driving frequency. The amplitude of some of the acceleration components even increases at some harmonics. However, the non-linear behavior of the actuator does not affect the generated fluid pressure significantly in this case as shown on the top plot. Similar results are obtained at the second driving frequency (300 Hz) as seen in Figure 6.8. Again, the non-linear characteristics of the actuator are mostly visible on the structural response of the pipe, with small influence on the response of the contained fluid. The third driving frequency
Figure 6.7: Pressure and acceleration levels at driving frequency and harmonics - 180 Hz

Figure 6.8: Pressure and acceleration levels at driving frequency and harmonics - 300 Hz
of 420 Hz results in slightly higher non-linear fluid pressure levels compared to the two previous cases (see Figure 6.9).

Note that actuator non-linear behavior can be reduced by imposing smaller tolerances on the different components of the active insert. In particular, the radius of the stack bottom saddle needs to closely match the outside pipe radius, which requires high precision cutting machines. However, non-linearities are likely to remain due to the very high forces developed in each stack. The use of passive attenuation rubber joints installed on each side of the active insert should thus be considered in order to reduce the level of structural vibrations transmitted through the connecting flanges.

**Influence of the static pressure level on the actuator response**

In all measurements presented so far, the fluid static pressure is kept constant around 20 bar. The following results show the influence of the static pressure level on the actuator performance.

In the first set of measurements, the actuator is operated at 240 Hz with a 212 Volts RMS driving voltage. The resulting fluid pressure is measured at the four pressure sensors for various levels of static pressure. The results are shown in Figure 6.10. As seen on the plot, the actuator performance drops as the fluid static pressure decreases. The generated pressure reaches a quasi constant level for static pressure values above 20 bar. Below 20 bar, the pressure fluctuations decrease. At 5 bar, the decrease reaches 50% on average.

To further analyze the influence of the static pressure level on the system response, the transfer function between the input signal to the actuator amplifier and the output of pressure sensor 3 is estimated for various static pressure values. The actuator is driven with a periodic chirp signal between 0 and 800 Hz with a 198 Volts RMS amplitude. The frequency analyzer uses 20 averages and a Hanning window to perform the estimate. Recalling the control setup described in Section 5.4.2, the above transfer function corresponds to the

![Figure 6.9: Pressure and acceleration levels at driving frequency and harmonics - 420 Hz](image)
filtered-\(x\) path of the control system. The magnitude and phase of the estimated transfer function is shown in Figures 6.11 and 6.12 for different static pressure levels. The first set of static pressure levels (0.4, 2.5, 13, and 18.5 bar) yields non negligible variations in the transfer function magnitude and phase, the largest relative discrepancy occurring at 0.4 bar (see Figure 6.11). On the other hand, the second set which includes pressure levels at or above 17.5 bar exhibits very close agreement between each transfer function. Consequently, the filtered-\(x\) path system identification performed prior to control will be accurate as long as the static pressure in the system remains approximately above 15 bar. Below this value, changes in the static pressure will decrease the accuracy of the system identification thereby affecting the controller performance.

### 6.2 Active control results

In this section, results from the active control tests are presented. The active control system introduced in Chapter 5 is applied to the control of pressure pulsations generated by the hydraulic engine downstream of the active insert. As explained in Section 5.4.2, the controller uses a one channel Filtered-\(x\) LMS algorithm. Five separate cases will be discussed successively. In all cases, the error signal is taken from pressure sensor 3 (see Figure 5.1). Each case corresponds to a given engine rotation speed and a set of harmonic tones to control. The active insert implemented as the control actuator uses all six piezo-electric stacks in the first control case presented below. For the remaining four cases, however, the actuator uses stacks 1 to 5 only. As mentioned previously, stack 6 was damaged from overheating and could not be replaced within the time frame of the experiments.
Figure 6.11: System transfer function at pressure sensor 3 (1)

Figure 6.12: System transfer function at pressure sensor 3 (2)
6.2.1 Control case 1

The first control case corresponds to the control of a single tone at 315 Hz. This frequency is the 28th harmonic of the engine rotation speed. The rotation speed is estimated at 675 rpm from the optical sensor output signal as described in Section 5.3. It should be mentioned that all harmonic frequencies associated with the system pressure pulsations are multiple of seven times the engine rotation frequency, seven being the number of pistons in the engine. Therefore, the 28th harmonic corresponds to the 4th harmonic in terms of pressure pulsations. For consistency, the remaining of the discussion will only refer to pressure pulsation rather than engine rotation harmonics. As stated above, this case implements all six piezo-electric stacks of the actuator.

The static pressure levels measured along the system (see Figure 5.3) are given in Table 6.1 along with other important parameters. As indicated in Table 6.1, the error signal is band-pass filtered between 275 and 350 Hz so that harmonic tones dominating the response outside this bandwidth do not deteriorate the estimate of the controller cost function.

The spectra of the measured fluid pressure at the four pressure sensors before and after control are shown in Figures 6.13 to 6.16, respectively. In all four plots, the solid line associated with the uncontrolled system response clearly shows the first five harmonic tones of the engine. The uncontrolled pressure pulsation amplitude at the 4th harmonic frequency is approximately 1.12 kPa. As for the remaining figures, the attenuation of the signals at the controlled harmonic frequencies is indicated on the plots in dB. As expected, the maximum pressure attenuation is achieved at the error sensor, i.e., sensor 3, with 13.3 dB (see Figure 6.15). The control input signal contains a single harmonic frequency in this case and thus no attenuation is achieved at the remaining harmonics present in the error signal frequency bandwidth. Very little pressure attenuation is achieved at sensor 1 (Figure 6.13) while the attenuation measured at sensor 4 is around 5 dB. This suggests that additional waves need to be controlled in order to improve the control of pressure pulsations in this particular case. Considering an infinite system with no fluid loading, a single wave propagates in the fluid at low frequencies, and the attenuation measured downstream of the point of cancellation then remains the same as the attenuation of the error signal. This is in contrast with the current system where the structure and fluid are coupled. As a result, the structural waves can couple to the fluid and decrease the actuator control performances. The active control of additional waves propagating in the system involve the use of additional error sensors capable of observing these waves, as well as appropriate actuators.

The levels of attenuation achieved at the four pressure sensors are summarized in Table 6.2.

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<th>Engine rotation speed (rpm)</th>
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</thead>
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<tr>
<td>Engine static pressure (bar)</td>
<td>38</td>
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<tr>
<td>Return static pressure (bar)</td>
<td>18</td>
</tr>
<tr>
<td>Controlled harmonics</td>
<td>4 (315 Hz)</td>
</tr>
<tr>
<td>Error filter bandwidth (Hz)</td>
<td>275–350</td>
</tr>
</tbody>
</table>

Table 6.1: Control case 1 - Settings

350 Hz so that harmonic tones dominating the response outside this bandwidth do not deteriorate the estimate of the controller cost function.
Figure 6.13: Control case 1 - Pressure sensor 1

Figure 6.14: Control case 1 - Pressure sensor 2
Figure 6.15: Control case 1 - Pressure sensor 3

Figure 6.16: Control case 1 - Pressure sensor 4
<table>
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<td>Harmonic frequency (Hz)</td>
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<tr>
<td>Pressure attenuation - sensor 1 (dB)</td>
<td>1.6</td>
</tr>
<tr>
<td>Pressure attenuation - sensor 2 (dB)</td>
<td>9.1</td>
</tr>
<tr>
<td>Pressure attenuation - sensor 3 (dB)</td>
<td>13.3</td>
</tr>
<tr>
<td>Pressure attenuation - sensor 4 (dB)</td>
<td>4.6</td>
</tr>
</tbody>
</table>

Table 6.2: Control case 1 - Results
6.2.2 Control case 2

The second control case considers the attenuation of the same harmonic tone as in the previous case but with a reduced engine speed in order to lower the frequency and amplitude of the tone. As mentioned earlier, the actuator uses five stacks in this case. The system settings are given in Table 6.3. Note that the results in this case as well as the three following cases are based on time domain data measured with a digital tape recorder and later processed in Matlab. All spectra are estimated based of 50 averages taken over non-overlapping blocks of 4096 samples using a Hanning window. The pressure measured at all four sensors and the axial and radial acceleration measured at location 1 and 2 (see Figure 5.1) are plotted versus frequency for each case.

For comparison with the measurements performed on the target system (see the time histories shown in Figures 2.3 to 2.7), Figure 6.17 shows the pressure fluctuations in the time domain measured at sensor 1. As already observed on the frequency domain spectra presented in case 1, the levels of pressure pulsations are smaller compared to those measured on the target system.

<p>| | |</p>
<table>
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<td>engine rotation speed (rpm)</td>
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<td>return static pressure (bar)</td>
<td>21</td>
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<tr>
<td>controlled harmonics</td>
<td>4 (208 Hz)</td>
</tr>
<tr>
<td>error filter bandwidth (Hz)</td>
<td>190–230</td>
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</tbody>
</table>

Table 6.3: Control case 2 - Settings

![Figure 6.17: Control case 2 - Uncontrolled pressure fluctuations at sensor 1](image-url)
Figures 6.18 to 6.21 show the spectra of the uncontrolled and controlled fluid pressure measured at sensors 1 to 4, respectively. As seen on all four plots, the controlled harmonic at 208 Hz does not dominate the system response making it necessary to band-pass filter the error signal. The main tones are associated with pressure fluctuations of peak-to-peak amplitudes reaching approximately 15.9 kPa in this case. The 4\textsuperscript{th} harmonic presents a 0.9 kPa peak-to-peak amplitude at the error sensor (see Figure 6.20). Note however that despite its lower amplitude, a given pulsation harmonic can yield higher levels of radiated sound depending on its coupling characteristics to the pipe radial wall motion. The controller achieves very little attenuation at the first and second pressure sensors. Much higher attenuation levels can be observed at the error location (sensor 3) and pressure sensor 4 with 13.6 dB and 16.3 dB, respectively. As mentioned earlier, this result is expected from the main direction of wave propagation in the system. The fluid pulses propagate downstream and cancellation can only be achieved past the error sensor assuming an ideal one dimensional infinite medium.

In order to further analyze the properties of the control system, the pipe wall acceleration measured at locations 1 and 2 is presented in the following figures. The axial and radial acceleration measured upstream the active insert before and after control is shown in Figures 6.22 and 6.23, respectively. As seen on the plots, no real attenuation is observed in both axial and radial wall motion at the controlled frequency. Furthermore, the actuator yields increases in the wall motion at frequencies near the controlled tone and its higher harmonics. This behavior is commonly referred to as control spillover. The increase occurring at the controlled harmonics is a direct consequence of the actuator non-linear behavior discussed previously. It should be mentioned that the non-linear characteristics of the actuator are likely to be more important when driving 5 stacks only. In other words, the standard six stack actuator is expected to yield better performance in terms of control spillover. The increase in vibration levels near the controlled tone is induced by the reference signal generator. Examining the auto-spectrum of the actuator input signal, i.e., the reference signal filtered through the control
Figure 6.19: Control case 2 - Pressure sensor 2

Figure 6.20: Control case 2 - Pressure sensor 3
Figure 6.21: Control case 2 - Pressure sensor 4

Figure 6.22: Control case 2 - Axial acceleration 1
compensator, reveals a peak at the controlled frequency which is not as sharp as in the case of a pure tone. The resulting non-zero frequency content near the controlled harmonic causes control spillover in the radial and axial wall motion. The same behavior can be observed in the acceleration components measured at location 2 as shown in Figures 6.24 and 6.25, respectively.

The above results are summarized in Table 6.4. Note that the actuator is only operating at about 50 % of its maximum driving voltage. Therefore, control of pressure pulsations with higher amplitude can potentially be achieved.

```
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<td>harmonic frequency (Hz)</td>
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<td>pressure attenuation - sensor 1 (dB)</td>
<td>0</td>
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<tr>
<td>pressure attenuation - sensor 3 (dB)</td>
<td>13.6</td>
</tr>
<tr>
<td>pressure attenuation - sensor 4 (dB)</td>
<td>16.3</td>
</tr>
<tr>
<td>axial acceleration attenuation - sensor 1 (dB)</td>
<td>0.4</td>
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<td>axial acceleration attenuation - sensor 2 (dB)</td>
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<td>radial acceleration attenuation - sensor 1 (dB)</td>
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<td>radial acceleration attenuation - sensor 2 (dB)</td>
<td>9.2</td>
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<td>control driving voltage (Volts RMS)</td>
<td>138</td>
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Table 6.4: Control case 2 - Results
```
Figure 6.24: Control case 2 - Axial acceleration 2

Figure 6.25: Control case 2 - Radial acceleration 2
6.2.3 Control case 3

The previous two control cases involved the control of a single harmonic tone. Control case 3 now involves three distinct pressure pulsation harmonics, namely, harmonics 4, 6, and 8. The engine rotation speed is slightly lower than in the previous case, while the level of static pressure is maintained around 21 bar as in cases 1 and 2. The system settings are presented in Table 6.5.

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<td>engine rotation speed (rpm)</td>
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<td>system static pressure (bar)</td>
<td>60</td>
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<tr>
<td>engine static pressure (bar)</td>
<td>30</td>
</tr>
<tr>
<td>return static pressure (bar)</td>
<td>21</td>
</tr>
<tr>
<td>controlled harmonics</td>
<td>4 (176 Hz), 6 (264 Hz), 8 (352 Hz)</td>
</tr>
<tr>
<td>error filter bandwidth (Hz)</td>
<td>160–370</td>
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</tbody>
</table>

Table 6.5: Control case 3 - Settings

As in control case 2, the pressure signal measured at sensor 1 before control is shown in Figure 6.26 versus time for comparison with the pressure fluctuations measured on the target system. Maximum peak-to-peak amplitudes of 35 kPa can be observed on the plot.

Figures 6.27 to 6.30 present the fluid pressure before and after control at sensors 1, 2, 3, and 4, respectively. Two tones dominate the system response in terms of pressure pulsations. The first tone around 12 Hz is associated with the second harmonic of the engine rotation speed, i.e., twice the engine rotation frequency. The second tone at 88 Hz is the second pulsation harmonic related to the seven pistons of the engine, i.e.,
Figure 6.27: Control case 3 - Pressure sensor 1

Figure 6.28: Control case 3 - Pressure sensor 2
Figure 6.29: Control case 3 - Pressure sensor 3

Figure 6.30: Control case 3 - Pressure sensor 4
14 times the engine rotation frequency. Both tones have peak-to-peak amplitude around 11 kPa. The three controlled harmonics at 176, 264, and 352 Hz exhibit uncontrolled pressure levels approximately 20 dB down of the two low frequency tones. As observed in the two previous cases, good attenuation is achieved past and at the error sensor except for the second controlled harmonic which is only attenuated by 4 dB at pressure sensor 4. On average, poor attenuation is noticed for pressure sensors 1 and 2 located upstream of the error minimization point.

Now examining the structural acceleration measured at locations 1 and 2 along the radial and axial directions (see Figure 6.31 to 6.34), the actuator yields significant control spillover at all three controlled frequencies. However, the uncontrolled acceleration levels are less than 10 dB above the noise floor and the added vibrations introduced by the actuator remain small. Passive or active attenuation devices must be implemented in order to reduce control spillover. Also note that the first tone at 12 Hz which is dominant on the fluid response is not coupling to pipe wall motion and can therefore be disregarding when controlling pipe radiated noise.

As a summary, Table 6.6 shows the attenuation levels achieved at the three controlled frequencies. It should be noted that the control actuator has relatively small driving voltages in this case as expected from the small amplitudes of the pressure pulsations at the controlled frequencies.

![Figure 6.31: Control case 3 - Axial acceleration 1](image-url)
Figure 6.32: Control case 3 - Radial acceleration 1

Figure 6.33: Control case 3 - Axial acceleration 2
Figure 6.34: Control case 3 - Radial acceleration 2

<table>
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<td>176</td>
<td>3.4</td>
<td>9.1</td>
<td>17.5</td>
<td>12.3</td>
<td>-2.0</td>
<td>-5.8</td>
<td>-14.3</td>
<td>-8.31</td>
<td>60</td>
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<td>264</td>
<td>1.9</td>
<td>-5.1</td>
<td>14.4</td>
<td>4.0</td>
<td>-11.9</td>
<td>-4.0</td>
<td>-18</td>
<td>-12.1</td>
<td>38</td>
</tr>
<tr>
<td>352</td>
<td>4.5</td>
<td>1.7</td>
<td>6.4</td>
<td>6.5</td>
<td>-12.6</td>
<td>-11.2</td>
<td>-17.3</td>
<td>-19.8</td>
<td>10</td>
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</table>

Table 6.6: Control case 3 - Results
6.2.4 Control case 4

The fourth control case considers the attenuation of pressure pulsations at four distinct frequencies associated with harmonics 3, 4, 5, and 6 of the engine pressure pulses (see Table 6.7). The engine rotation speed is increased slightly compared to the previous case by reducing the return static pressure level down to 9 bar. As discussed in Chapter 6.1, the actuator performance decreases at lower static pressure levels and this control case further investigates the effects of low static pressure on the control system.

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<td>system static pressure (bar)</td>
<td>60</td>
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<td>engine static pressure (bar)</td>
<td>19</td>
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<tr>
<td>return static pressure (bar)</td>
<td>9</td>
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<tr>
<td>controlled harmonics</td>
<td>3 (149 Hz), 4 (196 Hz), 5 (246 Hz), 6 (296 Hz)</td>
</tr>
<tr>
<td>error filter bandwidth (Hz)</td>
<td>140–400</td>
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</table>

Table 6.7: Control case 4 - Settings

For consistency with the results from the previous cases, Figure 6.35 presents the uncontrolled pressure fluctuations measured at sensor 1. The pressure fluctuations reach approximately 20 kPa peak-to-peak in this case.

The pressure pulsation spectra measured at sensors 1, 2, 3, and 4 are shown in Figures 6.36 to 6.39. As observed on the above figures, the fluid response now contains all pulsation harmonics associated with the engine seven pistons. This is in contrast with the previous control cases where some of the harmonics...
Figure 6.36: Control case 4 - Pressure sensor 1

Figure 6.37: Control case 4 - Pressure sensor 2
Figure 6.38: Control case 4 - Pressure sensor 3

Figure 6.39: Control case 4 - Pressure sensor 4
were attenuated due to the higher static pressure levels on the engine return side. The controller achieves good attenuation at the error sensor for the second controlled tones (see Figure 6.38). The other three tones are nearly 10 dB down the second tone making it more difficult for the controller to achieve significant attenuation (the cost function is dominated by a single tone). As in the previous cases, poor attenuation is observed at pressure sensors 1 and 2. The fluid pressure at sensor 4 downstream of the minimization point exhibits significant attenuation levels at the dominant second tone while the three remaining tones remain approximatively at their uncontrolled level. Overall, all four controlled tones are kept below 0.9 kPa peak-to-peak. Thus, despite the lower level of static pressure compared to the previous control cases, the actuator still achieves good control performances.

For further analysis, Figures 6.40 to 6.43 present the measured acceleration levels of the uncontrolled and controlled response. Behaviors similar to those observed on the previous control case can be seen on the system structural response. Both radial and axial acceleration levels tend to increase on the controlled response. As observed in Figure 6.40, the ICP channel used for the axial accelerometer behaved unexpectedly by adding high DC voltages to the measurement signal before control.

The attenuation levels achieved by the controller are recalled in Table 6.8 along with the control actuator driving voltages.
Figure 6.41: Control case 4 - Radial acceleration 1

Figure 6.42: Control case 4 - Axial acceleration 2
Figure 6.43: Control case 4 - Radial acceleration 2

Table 6.8: Control case 4 - Results

<table>
<thead>
<tr>
<th>harmonic frequency (Hz)</th>
<th>149</th>
<th>196</th>
<th>246</th>
<th>296</th>
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<td>pressure attenuation - sensor 1 (dB)</td>
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<td>pressure attenuation - sensor 2 (dB)</td>
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<td>4.4</td>
<td>9.7</td>
<td>0.7</td>
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<td>pressure attenuation - sensor 3 (dB)</td>
<td>6.9</td>
<td>14.1</td>
<td>5.3</td>
<td>1.2</td>
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<tr>
<td>pressure attenuation - sensor 4 (dB)</td>
<td>3.8</td>
<td>16.4</td>
<td>1.3</td>
<td>-0.1</td>
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<td>axial acceleration attenuation - sensor 1 (dB)</td>
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<td>3.4</td>
<td>-10.2</td>
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<td>axial acceleration attenuation - sensor 2 (dB)</td>
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<td>-2.6</td>
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<td>radial acceleration attenuation - sensor 1 (dB)</td>
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<td>-18.8</td>
<td>4.1</td>
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<td>radial acceleration attenuation - sensor 2 (dB)</td>
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<td>-4.31</td>
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<td>-4.72</td>
</tr>
<tr>
<td>control driving voltage (Volts RMS)</td>
<td>33</td>
<td>89</td>
<td>41</td>
<td>11</td>
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6.2.5 Control case 5

In the last control case discussed in this report, the controller is set to reduce pressure pulsations associated with harmonics 3, 4, 5, and 6. As shown in Table 6.9, the engine rotation speed reaches 582 rpm thus increasing the controlled frequencies compared to the previous case. The return static pressure is maintained at 10 bar

<table>
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<td>engine static pressure (bar)</td>
<td>25</td>
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<tr>
<td>return static pressure (bar)</td>
<td>10</td>
</tr>
<tr>
<td>controlled harmonics</td>
<td>3 (205 Hz), 4 (272 Hz), 5 (340 Hz), 6 (410 Hz)</td>
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<tr>
<td>error filter bandwidth (Hz)</td>
<td>190–420</td>
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</table>

Table 6.9: Control case 5 - Settings

which is close to the value measured in control case 4.

As for the previous control cases, the measured pressure before control at sensor 1 is shown in the time domain in Figure 6.44. The time domain signal is no longer dominated by two tones as observed previously but exhibits several distinct tones of similar amplitudes instead. Maximum peak-to-peak amplitudes in pressure pulses reach about 30 kPa.

This is confirmed by examining the frequency spectra estimated from the pressure measurements at the four pressure sensors. The spectra are shown for the uncontrolled and controlled response in Figures 6.45 to Figure 6.48. While the second harmonic tone exhibits the largest amplitude, the remaining tones also
Figure 6.45: Control case 5 - Pressure sensor 1

Figure 6.46: Control case 5 - Pressure sensor 2
Figure 6.47: Control case 5 - Pressure sensor 3

Figure 6.48: Control case 5 - Pressure sensor 4
present a significant contribution to the overall response. In terms of pressure attenuation, the controller behaves similarly as in the previous cases. Poor control is observed upstream of the error sensor and relatively good pressure attenuation is achieved at the error sensor and sensor 4 with as much as 15 dB reduction noticed on the second controlled harmonic. Note that the attenuation level achieved at the first controlled harmonic frequency is not as high possibly suggesting poor structure/fluid coupling at this frequency.

The pipe radial and axial wall acceleration is presented in Figures 6.49 to 6.52. Again, large levels of control spillover can be observed in both radial and axial directions. The increase of the controlled response in radial acceleration reaches 24 dB at the third controlled frequency. Interestingly, the control actuator driving voltage at this frequency is only 23 Volts RMS thus suggesting a possible resonance behavior of the system at this frequency. In particular, excitation of bending waves in the pipe is likely due to the non-axisymmetric behavior of the actuator operating 5 out of the 6 stacks installed. As described in Chapter 5, the upstream and downstream pipe sections are connected to flexible pipes and installed on soft mounts to isolate the system from the support bench. Therefore, the bending wave can exhibit a standing wave pattern at certain resonance frequencies due to reflections at the two end connections. The pipe then behaves as a beam in its bending mode and significant radial acceleration can occur. It should also be mentioned that stack 6 is directly opposite from the location of the accelerometers along the circumference. Thus, the non-axisymmetric behavior of the actuator creates a vertical force normal to the pipe axis and located along the same circumferential angle as the accelerometers. The resulting bending motion is then directly related to the measured acceleration along the radial direction.

The results from control case 5 are summarized in Table 6.10 for completeness.
Figure 6.50: Control case 5 - Radial acceleration 1

Figure 6.51: Control case 5 - Axial acceleration 2
Figure 6.52: Control case 5 - Radial acceleration 2

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<td>3.8</td>
<td>-0.8</td>
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<td>pressure attenuation - sensor 2 (dB)</td>
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<td>7.1</td>
<td>7.3</td>
<td>0.6</td>
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<td>pressure attenuation - sensor 3 (dB)</td>
<td>4.9</td>
<td>11.4</td>
<td>9.2</td>
<td>4.2</td>
</tr>
<tr>
<td>pressure attenuation - sensor 4 (dB)</td>
<td>6.3</td>
<td>15.4</td>
<td>8.9</td>
<td>0.7</td>
</tr>
<tr>
<td>axial acceleration attenuation - sensor 1 (dB)</td>
<td>7.1</td>
<td>7.1</td>
<td>-11.0</td>
<td>4.4</td>
</tr>
<tr>
<td>axial acceleration attenuation - sensor 2 (dB)</td>
<td>-3.7</td>
<td>-8.0</td>
<td>-13.7</td>
<td>3.1</td>
</tr>
<tr>
<td>radial acceleration attenuation - sensor 1 (dB)</td>
<td>13.4</td>
<td>-15.5</td>
<td>-24.7</td>
<td>5.28</td>
</tr>
<tr>
<td>radial acceleration attenuation - sensor 2 (dB)</td>
<td>10.1</td>
<td>-20.1</td>
<td>-24.3</td>
<td>-5.6</td>
</tr>
<tr>
<td>control driving voltage (Volts RMS)</td>
<td>98</td>
<td>69</td>
<td>23</td>
<td>8</td>
</tr>
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</table>

Table 6.10: Control case 5 - Results
Chapter 7

Conclusions and recommendations for future work

This report describes a new active control approach for attenuating radiated noise from pipe work systems. For the target system considered in this study, the pressure pulsations generated by the system main pump and hydraulic engine represent the dominant source of noise. As a first step towards achieving sound reduction, the work thus focuses on the control of the pressure pulsations traveling along the pipe line. Traditional methods for controlling pressure ripples in pipe systems involve passive pulsation dampers. The main disadvantages of passive dampers are a relative inefficiency at low frequencies and static pressure drops which reduce system performances. In this work, an active control approach is developed to limit the use of passive dampers and attempt further attenuation of the pressure pulsations. The approach is based on generating secondary pressure pulsations in the fluid. Attenuation of the incoming pulsations is achieved by adapting in real time the amplitude and phase of the secondary pulsations. In practice, the proposed method involves a fluid wave actuator (control actuator), a pressure sensor (error sensor), and a controller driving the control actuator such as to minimize the output of the error sensor. Phase one of the project described in this report focuses on the design, implementation, and testing of the fluid wave actuator.

The proposed fluid wave actuator is implemented as an active insert equipped with a non-intrusive structural actuator. Such an insert can be easily installed on existing pipe work systems. The insert consists of a short pipe section of same diameter as the target pipe line and a pipe squeeze actuator. The pipe squeeze actuator generates radial wall motion over a small section of the pipe insert in order to drive the contained fluid through structure/fluid coupling. The radial wall motion is induced by six piezo-electric stack actuators (PZT stacks) placed along the pipe circumference. All PZT stacks are driven in phase to produce axisymmetric excitation of the pipe wall and in turn generate pressure pulsations in the fluid.

The experimental tests described in the report implement the fluid wave actuator to actively control the pressure pulsations propagating down the discharge line of an oil-driven hydraulic engine. A pressure sensor installed downstream of the active insert provides the error signal to a single channel feedforward controller. The controller uses the output of an optical sensor placed on the engine drive shaft to construct the reference signal containing harmonics of the engine main rotation frequency associated with unwanted pulsation tones. Additional pressure and acceleration sensors are installed upstream and downstream of the active insert to evaluate the performance of the control system.

Prior to the control tests, evaluation of the actuator performance was performed by measuring the generated in-vacuo radial displacement and fluid pressure fluctuations. Results show the proposed configuration yields
radial wall displacement peak-to-peak amplitudes of about 6 μm at maximum driving voltage over the 0–600 Hz frequency range. In terms of fluid pressure fluctuations, average peak-to-peak amplitudes around 6 kPa and 15 kPa were measured at 200 Hz and 600 Hz, respectively, while driving the actuator at 75 % of its maximum voltage rating.

Several control tests were conducted to validate the proposed approach. Cases involved two actuator configurations based on six and five stacks, respectively, and one or several harmonics of the engine main rotation frequency. Results demonstrate the ability of the proposed control system to attenuate incoming pressure pulsations. Pressure reductions between 10 and 15 dB were measured at the error sensor in all cases. Furthermore, reductions ranging from 6 to 15 dB are achieved downstream of the error sensor demonstrating global control of the pressure pulses. Measurements of the structural response, however, exhibit significant increase in pipe wall vibration levels after control. These preliminary results demonstrate the potential of the proposed control strategy for reducing pressure pulsations. Before implementation on the target system, however, a number of important issues remain to be addressed in future work as outlined below.

Further research is needed in order to improve the actuator performances in terms of generated pressure levels. The current levels remain below the ones measured on the target system and thus will not yield complete cancellation of the primary pulsations. Several points can be investigated to increase the actuator efficiency. First, the mechanical design of the actuator can be improved to decrease losses in the transfer of mechanical energy from the stack to the pipe wall. For example, bonding the stacks to the top and bottom saddles can decrease friction losses. Also, smaller tolerances in machining the active section can yield a more uniform pipe wall with reduced thickness which would increase the pipe compliance. The use of alternative materials with increased compliance properties could be investigated as well. The second issue regarding actuator efficiency concerns the cooling of the PZT stacks. At higher frequencies, maximum driving voltage will damage the stacks unless appropriate cooling is implemented. The experiments performed so far did not implement cooling and the actuator could only be operated at 75 % of its maximum driving voltage. Thin tubes wrapped around the stacks and carrying water could be investigated. Finally, the use of different PZT stacks with increased efficiency should be considered. This includes possible custom built stacks that better suit the pipe squeeze actuator concept. Increased efficiency could be achieved by machining the stack end-cap to allow direct bonding to the pipe wall. This approach would suppress the bottom saddle and reduce energy losses through friction.

Another direction of research for future work should address the control of total power flow in the pipe system. As discussed in the previous chapters, fluid-filled pipes are coupled systems where control of the dominant wave may not be sufficient to attenuate the overall radiated sound. In particular, the increase in vibration levels observed on the controlled response (control spillover) needs to be addressed. A first approach would be to implement passive isolation joints to reduce the transmission of the structural vibrations generated by the actuator. On a more general level, further work should investigate the control of additional types of propagating wave. This involves implementing additional sensors and actuators. In particular, PVDF films mounted around the pipe circumference can be used as error sensors to observe different waves. Also, the active control of longitudinal waves (dominant axial motion) could improve total power flow attenuation. An active pipe joint using PZT stacks placed in between flanges to generate longitudinal waves could be investigated. Finally, research can also address the use of alternate fluid wave actuators with increased efficiency compared to the pipe squeeze actuator concept.

A third area of research for future work involves improving the controller used in the active control system. First, the reference signal generator needs to implement multiple sources such that multiple sets of harmonic tones can be controlled. This feature is required for the target system considered in this study as the unwanted
vibrations are correlated to multiple sources (system main pump and engine). Also, control performances can be improved by increasing the accuracy of the filtered-\(x\) path system identification. The system transfer functions are dependent on several factors such as fluid static pressure, velocity, and temperature. Thus the control algorithm should implement on-line update of the system identification during control. Finally, future work should also focus on improving the convergence rate of the controller. The target system is characterized by rapid changes in rotation speed of the main pump and engine. Fast convergence is required in order to follow these changes and ensure continuous control of the unwanted tones. Note that fast convergence also requires good correlation between the reference and error signals. Therefore, the reference signal generator needs to track closely the changes in speed of the system.

Finally, it should be mentioned that the current actuator design is based on the outgoing line characteristics of the target system and a second actuator also needs to be developed and tested for implementation on the incoming line. The smaller diameter and increased thickness of the pipe will have important implications in the actuator design. In particular, the active section will present a reduced radial wall compliance thus requiring larger forces in the pipe squeeze actuator. However, the smaller diameter also reduces the amplitude of radial wall displacement required to produce sufficient levels of fluid volume displacement and associated pressure.
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