Active, Passive and Active/Passive Control
Techniques for Reduction of Vibrational Power
Flow in Fluid Filled Pipes

By
Satish C. Kartha

Thesis submitted to the Faculty of the
Virginia Polytechnic Institute and State University
In partial fulfillment of the requirements for the degree of
Master of Science

In
Mechanical Engineering

APPROVED:

___________________________________________

Dr. Chris R. Fuller, Chairman

___________________________________________

Dr. Daniel G. Inman

___________________________________________

Dr. Martin E. Johnson

February 4, 2000
Blacksburg, Virginia

Copyright 2000, Satish C. Kartha
Active, Passive and Active/Passive Control
Techniques for Reduction of Vibrational Power Flow in Fluid Filled Pipes

By
Satish C. Kartha
Committee Chairman: Chris R. Fuller
Mechanical Engineering

(ABSTRACT)

The coupled nature of vibrational energy flow in fluid filled piping systems makes its control and subsequent reduction a difficult problem. This work experimentally explores the potential of different active, passive and active/passive control methodologies for control of vibrational power flow in fluid filled pipes. Circumferential modal decomposition and measurements of vibrational power carried by individual wave types were carried out experimentally. The importance of dominant structural bending waves and the need to eliminate them in order to obtain meaningful experimental results has been demonstrated. The effectiveness of the rubber isolator in reducing structural waves has been demonstrated. Improved performance of the quarter wavelength tube and Helmholtz resonator was obtained on implementation of the rubber isolator on the experimental rig. Active control experiments using the side-branch actuator and 1/3 piezoelectric composite yielded significant dB reductions revealing their potential for practical applications. A combined active/passive approach was also implemented as part of this work. This approach yielded promising results, which proved that combining advantages of both active and passive approaches was a feasible alternative.
Acknowledgements

First of all, I would like to thank my committee chairman Dr. Chris Fuller for giving me the opportunity to work on this challenging project and also for providing many helpful insights which helped me overcome numerous problems during the course of this project. Special thanks are due to Dr. Marty Johnson for providing me with his able guidance throughout this work. I am also grateful to Dr. Dan Inman for serving on my committee.

I would like to thank all my friends in the Vibrations and Acoustics Laboratories especially Steve Booth for providing me with hands-on knowledge and Dawn Williams for her moral support. I would also like to thank all my friends in Blacksburg for making my stay in the U.S. enjoyable.

Finally I would like to thank my parents and my brother for their deep love and encouragement. Without them I would not have got this opportunity to do my Master’s. I’ll never be able to thank them enough for all they have done,
to Amma, Achan and Binu
# Table of Contents

## 1 Introduction

1.1 Vibrations of fluid-filled cylindrical elastic shells ........................... 3
    1.1.1 Sensing and wave-decomposition in fluid-filled pipes ............... 8
    1.1.2 Passive methods for vibration control in fluid filled pipes ...... 12
    1.1.3 Active control .................................................. 14
    1.1.4 Active control of vibrations in fluid filled pipes ................. 15

1.2 Objective and Approach .................................................. 18

1.3 Organization .............................................................. 20

## 2 Theory of vibrations in cylindrical shell systems ............................ 22

2.1 Wave propagation in fluid filled pipes .................................... 23

2.2 Equations of motion of the coupled system .................................. 25

2.3 Dispersion Curves .......................................................... 32

2.4 Energy flow in fluid filled pipes .......................................... 36
    2.4.1 Structure-borne energy flow ..................................... 37
    2.4.2 Fluid-borne energy flow ......................................... 38
    2.4.3 Relationships between wave amplitudes ............................ 39
    2.4.4 Approximate equations for energy flow per wave type ............ 40

2.5 Experimental method for circumferential mode decomposition .......... 42

## 3 Experimental Setup ......................................................... 46

3.1 Arrangement of the experimental rig ....................................... 47
    3.1.1 Anechoic termination .............................................. 52
    3.1.2 Isolating structural vibrations .................................... 55
3.2 Characterization of experimental rig. ...........................................65

3.2.1 Experiments to determine rig behaviour. .............................. 66

3.2.2 Experimental results and discussion. ................................. 72

4 Experiments using passive control techniques ........................ 81

4.1 Rubber Isolation section. ....................................................... 82

4.1.1 Experiments using rubber isolator section. ......................... 83

4.2 Quarter wavelength tube. ..................................................... 92

4.2.1 Experiments using quarter wavelength tube. ...................... 95

4.3 Helmholtz resonator. ......................................................... 107

4.3.1 Basic Helmholtz resonator experiments. ....................... 115

4.3.2 Tunable Helmholtz resonator tests. .............................. 125

4.4 Summary. ................................................................. 130

5 Active and Active/Passive Control Experiments ..................... 133

5.1 Control arrangement and hardware setup for active control experiments . 138

5.2 Active control experiment using a side-branch actuator. ........ 144

5.3 Active and active-passive experiments using 1-3 piezoelectric composite
actuator. .................................................................................. 177

5.3.1 Active control experiment. ............................................. 182

5.3.2 Active-passive control experiment. ............................... 193

6 Summary and Conclusions ..................................................... 205

References ................................................................. 211
List of Figures

1.1 Schematic diagram of a typical piping system in a ship. ......................... 2
2.1 Circumferential and axial modes for a cylindrical shell. ....................... 26
2.2 Cylindrical co-ordinate system for equations of motion. ..................... 27
2.3 Dispersion curves for an \textit{in vacuo} cylindrical steel shell, h/a=0.143. .... 35
2.4 Dispersion curves for water-filled cylindrical steel shell, h/a=0.143. ........ 35
2.5 Fluid filled piping system with accelerometer mounted at axial locations \(x_1\) and \(x_2\) for measuring energy flow through the pipe. ....................... 41
2.6 Accelerometer configurations for circumferential mode decomposition. ..... 43
2.7 Picture of accelerometers mounted on brass studs at axial locations \(x_1\) and \(x_2\) to measure the pipe wall accelerations in the axial \(u\) direction. ............... 43
3.1 Schematic diagram of experimental test rig. ..................................... 48
3.2 Schematic diagram of the shaker, stinger and diaphragm configuration used for excitation of the test rig. ..................................................... 49
3.3 Photograph of basic experimental rig. .............................................. 50
3.4 Photograph showing the test section mounted on the experimental rig. ..... 50
3.5 White plastic anechoic end referred to as ‘termination 1’. ................. 53
3.6 Transparent plastic end referred to as ‘termination 2’. ....................... 53
3.7 Schematic diagram for anechoic ends comparison test. ...................... 56
3.8 Transfer function and phase between s3 and s4 for both terminations. .... 56
3.9 Asymmetric forcing leading to generation of structural waves. ............. 58
3.10 Structural discontinuity in system leading to generation of bending waves. 58
<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.11</td>
<td>Diaphragm actuator to prevent generation of structural bending waves due to asymmetric forcing.</td>
</tr>
<tr>
<td>3.12</td>
<td>Passive insert in piping system to absorb ( n = 1 ) bending waves.</td>
</tr>
<tr>
<td>3.13</td>
<td>Picture of diaphragm actuator (prevents bending wave generation due to asymmetric forcing).</td>
</tr>
<tr>
<td>3.14</td>
<td>Passive isolating section and rubber pads (inserted between steel flanges).</td>
</tr>
<tr>
<td>3.15</td>
<td>Experimental setup for observing reduction in structural vibrations.</td>
</tr>
<tr>
<td>3.16</td>
<td>Comparison of transfer functions for test rig with and without structural wave dampers installed.</td>
</tr>
<tr>
<td>3.17</td>
<td>Experimental setup for rig characterization tests.</td>
</tr>
<tr>
<td>3.18</td>
<td>Experimental setup of pressure sensors to detect presence of ( n = 1 ) structural bending waves.</td>
</tr>
<tr>
<td>3.19</td>
<td>Plots showing that the pressure sensor readings were not affected by structural waves.</td>
</tr>
<tr>
<td>3.20</td>
<td>Accelerometer configuration for circumferential decomposition.</td>
</tr>
<tr>
<td>3.21</td>
<td>Accelerometers mounted on downstream section at ( X1' ) and ( X2' ) to measure acceleration in the radial (( w )) direction.</td>
</tr>
<tr>
<td>3.22</td>
<td>Radial pipe wall acceleration, ( w ), for the ( n = 0 ) fluid wave.</td>
</tr>
<tr>
<td>3.23</td>
<td>Fluid pressure inside rig measured by pressure sensors and hydrophone.</td>
</tr>
<tr>
<td>3.24</td>
<td>Longitudinal acceleration for the ( n = 0 ) shell extensional wave.</td>
</tr>
<tr>
<td>3.25</td>
<td>Pipe wall acceleration for the ( n = 0 ) torsional shell wave.</td>
</tr>
<tr>
<td>3.26</td>
<td>Pipe wall acceleration for the ( n = 1 ) bending wave.</td>
</tr>
<tr>
<td>3.27</td>
<td>Power carried by the ( n = 0 ) fluid wave through pipe 2.</td>
</tr>
</tbody>
</table>
4.17 Schematic diagram of a simple Helmholtz resonator. .......................... 110
4.18 Schematic diagram of a Helmholtz resonator in a piping system. ............. 110
4.19 Schematic diagram of Basic Helmholtz resonator. ................................. 113
4.20 Picture of the basic Helmholtz resonator mounted on test rig. ................. 113
4.21 Schematic diagram of tunable Helmholtz resonator. .............................. 114
4.22 Tunable Helmholtz resonator mounted on test rig. ............................... 114
4.23 Initial experimental setup for basic Helmholtz resonator tests. ................. 117
4.24 Transfer function plots for experimental rig without rubber isolator. ......... 117
4.25 Improved experimental setup for Helmholtz resonator tests. .................... 119
4.26 Comparision of transfer functions at sensor s4 with and without Helmholtz
   resonator. .......................................................................................... 119
4.27 Transfer function between excitation and s1, Helmholtz resonator tests. .... 121
4.28 Transfer function between excitation and s5, Helmholtz resonator tests. .... 121
4.29 Transfer function between s2 and s3, Helmholtz resonator tests. ............. 122
4.30 Structural wave generation and propagation affecting experimental results for
   Helmholtz resonator tests. ................................................................. 124
4.31 Rubber damper prevents the propagation of structural waves thus reducing their
   dominant presence on the test rig. ..................................................... 124
4.32 Transfer function plots between s2 and s3 for tunable Helmholtz resonator tests .
   ........................................................................................................ 127
4.33 Transfer function between excitation and sensor s1. .............................. 128
4.34 Transfer function between excitation and sensor s4. ............................. 128
4.35 Transfer function between excitation and sensor s4. ............................. 129
5.21 Fluid pressure reduction in dB at sensor s1, 150 Hz (s3 error sensor) ............159
5.22 Fluid pressure reduction in dB at sensor s2, 150 Hz (s3 error sensor) ............159
5.23 Fluid pressure reduction in dB at sensor s3, 150 Hz (s3 error sensor) ............160
5.24 Pressure reduction in dB at sensor s4, 150 Hz (error sensor s3) .................160
5.25 Pressure reduction in dB at sensor s5, 150 Hz (error sensor s3) .................161
5.26 Pressure reduction in dB at hydrophone, 150 Hz (error sensor s3) ..............161
5.27 Pressure reduction at sensor s4 in Pascals, 150 Hz (error s3) ....................163
5.28 Pressure reduction at the hydrophone in Pascal, 150 Hz (error s3) ............163
5.29 Pressure reduction at sensor s1 in dB, 500 Hz (error s3) .......................164
5.30 Pressure reduction at sensor s2 in dB, 500 Hz (error s3) .......................164
5.31 Pressure reduction at sensor s3 in dB, 500 Hz (error s3) .......................165
5.32 Pressure reduction at sensor s4 in dB, 500 Hz (error s3) .......................165
5.33 Pressure reduction at the sensor s5 in dB, 500 Hz (error s3) ....................166
5.34 Pressure reduction at the hydrophone in dB, 500 Hz (error s3) ...............166
5.35 Pressure reduction at sensor s4 in Pascals, 150 Hz (error s3) ...................167
5.36 Pressure reduction at the hydrophone in Pascal, 150 Hz (error s3) ..........167
5.37 Pressure reduction at sensor s1 in dB, 500 Hz (error s3) .......................168
5.38 Pressure reduction at sensor s2 in dB, 500 Hz (error s3) .......................168
5.39 Pressure reduction at sensor s3 in dB, 500 Hz (error s3) .......................169
5.40 Pressure reduction at sensor s4 in dB, 500 Hz (error s3) .......................169
5.41 Pressure reduction at the sensor s5 in dB, 500 Hz (error s3) ....................170
5.42 Pressure reduction at the hydrophone in dB, 500 Hz (error s3) ...............170
5.43 Broadband pressure reduction at sensor s1, 0-500 Hz (error s3) .............173
5.44 Broadband pressure reduction at sensor s2, 0-500 Hz (error s3) ............... 173
5.45 Broadband pressure reduction at sensor s3, 0-500 Hz (error s3) ............... 174
5.46 Broadband pressure reduction in dB at s4, 0-500 Hz (error s3) ............... 174
5.47 Broadband pressure reduction in dB at s5, 0-500 Hz (error s3) ............... 175
5.48 Broadband pressure reduction in dB at hydrophone, 0-500 Hz (error s3) .... 175
5.49 Schematic representation of a 1-3 piezoelectric composite......................... 178
5.50 Picture of actual 1-3 piezoelectric composite used for active control experiment. ................................................................. 178
5.51 Picture of Helmholtz resonator assembly with the 1-3 piezoelectric composite mounted inside it. .............................................. 180
5.52 Schematic diagram showing operation of 1-3 composite as secondary source. ..180
5.53 Rig setup for active control experiment using 1-3 piezoelectric composite. .... 183
5.54 Plot showing that 1-3 composite generates coherent signals only at 1000 Hz and above. ................................................................. 184
5.55 Autospectrum levels at sensor s1, 1500 Hz (error sensor s1) ................... 187
5.56 Autospectrum levels at sensor s2, 1500 Hz (error sensor s1) ................... 187
5.57 Auto spectrum levels at sensor s3, 1500 Hz (error sensor s1) ................... 188
5.58 Auto spectrum levels at sensor s4, 1500 Hz (error sensor s1) ................... 188
5.59 Auto spectrum levels at hydrophone, 1500 Hz (error sensor s1) ............... 189
5.60 Auto spectrum levels at sensor s1, 1500 Hz (error sensors s1 and s2) ....... 189
5.61 Auto spectrum levels at sensor s2, 1500 Hz (error sensors s1 and s2) ....... 190
5.62 Auto spectrum levels at sensor s3, 1500 Hz (error sensors s1 and s2) ....... 190
5.63 Auto spectrum level at sensor s4, 1500 Hz (error sensors s1 and s2) ....... 191
<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.64</td>
<td>Auto spectrum level at hydrophone, 1500 Hz (error sensors s1 and s2)</td>
<td>191</td>
</tr>
<tr>
<td>5.65</td>
<td>Schematic diagram of rig for active-passive control experiment</td>
<td>197</td>
</tr>
<tr>
<td>5.66</td>
<td>Picture of active-passive element on experimental rig.</td>
<td>198</td>
</tr>
<tr>
<td>5.67</td>
<td>Instruments used for active-passive control experiments.</td>
<td>198</td>
</tr>
<tr>
<td>5.68</td>
<td>New Helmholtz resonator and 1-3 actuator positions.</td>
<td>199</td>
</tr>
<tr>
<td>5.69</td>
<td>Helmholtz resonator and 1-3 actuator positions in previous section.</td>
<td>199</td>
</tr>
<tr>
<td>5.70</td>
<td>Autospectrums of pressure sensors and hydrophone.</td>
<td>201</td>
</tr>
</tbody>
</table>
List of Tables

2.1 Shell and fluid properties of test rig used for experiments. ........................ 34

3.1 Distance of the sections for accelerometer locations X1, X2, X1’ and X2’ from
the shaker end of the experimental rig. .............................................. 70

4.1 Table showing location of sensors on the experimental rig and their angular
locations. ....................................................................................... 85

4.2 Table showing the distances of the accelerometer location sections X1, X2, X1’
and X2’ from the shaker end. ......................................................... 88

4.3 Dimensions and resonant frequency of basic Helmholtz resonator. ............. 112

4.4 Resonant frequencies of tunable Helmholtz resonator for different percentages of
open neck area. ............................................................................. 112

4.5 Sensor locations on the experimental rig measured from the shaker end for the
basic Helmholtz resonator tests. .................................................... 116

5.1 Comparison of pressure reduction in dB for active control experiment, 1500Hz. .
......................................................................................................... 186

5.2 Control voltage and pressure reduction for active-passive control experiment. .202
Chapter 1

Introduction

Piping systems play a very important role in various industrial and military applications. They are used in many engineering applications for conveying gases and fluids over a wide range of temperatures and pressures. These applications include hydraulics, fluid transfer, cooling water and fuel supply. Unfortunately the pipe shell and contained fluid are also paths for vibrational energy from the pumping devices to the different receiving structures. These vibrations can cause two kinds of problems, mechanical fatigue and radiated noise. Fatigue failure may result in damage to vital parts of the installation. Hence, engineers have been concerned with this problem in power plants, refineries, oilrigs, etc. The radiated noise from pipes contributes significantly to the noise signature of naval ships and submarines besides causing discomfort. As a result, it is in the best interest of the designers to minimize the radiated noise in order to avoid detection of these military vessels. Vibration in pipes is also a major domestic problem due to noise transmission through the walls by bathroom and kitchen pipes. The pipes used for the experimental rig built as a part of this thesis were industrial 2” steel pipes (internal diameter = 2”, outer diameter = 2.315”, pipe mean radius $a = 1.078”$, pipe wall thickness $h = 0.157”$, thickness-mean radius ratio $h/a = 0.143$).

Piping systems can be excited by a large number of sources associated with the internal fluid path or the structural path. Figure 1.1 shows a typical piping system in a
ship. The internal fluid path can be excited by a source such as a pump. For example, it has been shown that the dominant fluid pulsations measured in a piping system connected to a rotary pump occur at frequencies corresponding to the pump’s blade passing rate and its harmonics. Discontinuities in the flow, e.g. valves and bends, can also create local turbulence, which acts as a noise source in the fluid. Structure borne excitation is generated by direct mechanical excitation of the pipe through vibrating structures such as pumps. Other vibrational sources can reach equipment connected to the piping system, which ultimately transmit vibrational energy to the pipe. Conversely as vibrational energy propagates along the piping system it can excite other components attached to the structure far away from the source.

![Schematic diagram of a typical piping system in a ship.](image)

Figure 1.1: Schematic diagram of a typical piping system in a ship.

The vibrational energy flow in piping systems is through both the fluid and the pipe structure. Due to the coupled nature of the energy transmission path, it is very difficult to reduce the total power flow by solely focussing on either the structural or fluid path. While passive methods are effective at high frequencies, in the low frequency region they are not very practical due to size and weight considerations. Their use is also restricted in critical applications due to the resulting static pressure loss, which decreases
the system performance. Thus, active control is an attractive alternative for vibration control in the low frequency region. An active/passive approach is also employed here to combine the advantages of both these control methodologies.

1.1 Vibrations of fluid filled cylindrical shells

A very good understanding of the basic mechanisms involved in the dynamic behaviour of fluid filled cylindrical shells is required in order, to successfully apply different control techniques to piping systems. More importantly, the different wave types and their relative contribution to the overall system vibrations need to be assessed. In order to understand this, a brief literature review of vibrations in both in vacuo and fluid filled cylindrical shells has been carried out. Literature on various passive devices and active control techniques for vibration control in fluid filled pipes has also been briefly covered.

A number of shell theories have been proposed to describe the vibrational behaviour of circular pipes. A detailed discussion of these theories has been given by Leissa [1]. The characteristic equation of a cylindrical elastic shell in vacuo derived using the three-dimensional equations of linear elasticity is valid for all ranges of wall thickness and frequency. Hermann and Mirsky [2,3,4], Gazis [5], Greenspon [6], Bird, Hart and McClure [7], and Armenakas, Gazis and Hermann [8] have investigated this equation to some extent. Fuller [9] fully described the free vibrations of infinite thin cylindrical shells in vacuo using Flugge shell equations. An attempt was made to characterize the performance of pipe isolators and the effect of attached structural discontinuities by investigating the energy transmission through these discontinuities.
Heckl [10] used a simple shell model for thin walled pipes and obtained the formula for the resonant frequencies and the modal density of the simply supported shell as well as the point impedance of the infinite shell. Kuhn and Morfey [11] showed that for pipes vibrating in vacuo at low frequencies (wavelength greater than the pipe diameter), most of the radiated sound is dominated by pipe bending waves. Furthermore a sharp 90° bend in the pipe causes a significant increase in bending wave excitation when plane waves are incident on the bend. Kumar [12] also conducted a theoretical study on the dispersion of axially symmetric waves in empty and fluid-filled circular cylindrical shells of various wall-thicknesses. Special emphasis was laid on the dispersion spectra. For a specified range of frequency and wavenumber, the number of real, imaginary and complex branches for an empty shell decreased with decreasing wall-thickness. Borgiotti and Rosen [13] analysed the vibration induced by a point excitation on an elastic circular cylindrical shell in vacuum and the power flow associated with it using a state vector technique. This technique provides a systematic approach to the formulation and the solution of shell problems like determination of dispersion curves and forced vibrational response to a point force.

Though the in vacuo solution can be used in cases where the impedance of the internal medium is small compared to that of the pipe (eg. gas in pipes), it cannot be used when the shell is filled with fluid. The behaviour of fluid filled shells has been studied extensively by different authors. Fuller and Fahy [14] have examined the dispersion curves and energy distributions of free waves in fluid filled cylindrical shells. The behaviour of free waves was found to depend strongly on shell wall thickness, and density ratio of shell material to density of contained fluid. An exact equation predicting
energy distribution in fluid-filled shells was developed. The effect of various parametric changes on these equations was studied. It was shown that at low frequencies the energy remained predominantly in the shell or the fluid depending on whether the excitation was structural or acoustic. At higher frequencies the energy could be in the fluid or the shell wall irrespective of the excitation. Fuller [15] studied the vibrational response of an infinite fluid filled cylindrical shell excited by a monopole acoustic source. At low frequencies the shell response is dominated by subsonic modes while at high frequencies the shell radial response is small. It was also found here that the vibrational energy was predominately in the fluid for the monopole acoustic source away from the shell wall. Moving the source closer to the shell wall was found to cause a shift in vibrational energy from the fluid into the shell wall.

Kumar [16] derived frequency equations for the axi-symmetric vibrations of a thin cylindrical elastic shell, filled with nonviscous, compressible fluid by considering the exact three-dimensional equations of motion for the shell. This frequency equation was solved for a higher frequency range and for a greater number of modes. Thomson [17] introduced the effects of Poisson’s ratio and included flexural and axial wave motion and evaluated the phase velocities of the first three axisymmetric “fluid” waves. Lin and Morgan [18] studied the propagation of axisymmetric waves through fluid filled cylindrical elastic shells. The dependence of phase velocity on various physical parameters of the system was analyzed. However their results were restricted to real wave numbers and to circumferential modes of zero order.

Sinha et al. [19,20] investigated the theoretical and experimental aspects of axisymmetric wave propagation in cylindrical shells for different fluid configurations.
Focus was primarily on shell modes, non-cutoff fluid modes and cutoff fluid modes. Pinnington [21] derived the equation of motion for axisymmetric waves in a fluid-filled orthotropic, internally pressurized and axially tensioned tube. The waves transferring energy through the shell and fluid were identified. At any frequency four wave types were found. Under almost all conditions an axial, compressive wave was propagating in the shell. In another related work, Pinnington [22] calculated the input and transfer impedances of flexible fluid-filled pipes using a wave approach. For a plain rubber hose it was found that internal pressure did not greatly affect the stiffness, but for a braided rubber hose there were significant pressure stiffening effects. A computer model was developed to generate the impedance matrix of a pressurized elastic tube.

Kumar [23] derived the frequency equation for vibrations of a fluid-filled cylindrical shell using the exact three-dimensional equations of linear elasticity. These equations were analyzed quantitatively to study the flexural vibrations \( n=1 \) of empty and fluid-filled shells of different thickness. The effect of fluid was negligible for vibrations of thick shells. As the thickness of the shell decreased, the presence of fluid gave extra modes of vibrations. Samsury [24] discussed the phenomenon of liquid-structure coupling in fluid-filled pipes, which results in plane axial waves in the fluid getting converted to flexural beam vibrations of the pipe. A mathematical analysis of liquid-structure coupling in a liquid-filled elbow is presented.

Merkulov, et al. [25] addressed the problem of excitation of a cylindrical fluid-filled shell by an arbitrarily oriented point force applied to its surface. The solution was represented in the form of a normal mode Fourier series. The amplitudes of excited normal modes were expressed as a function of frequency. In another work [26], the same
authors obtained a solution for the problem of excitation of a fluid-filled cylindrical elastic shell by a monopole point source. A numerical calculation of the spectrum and amplitudes of axisymmetric, beam, and other normal-mode configurations was carried out using approximate theory of thin shells described by Kennard’s equations of motion. Fuller [27] derived the force-input mobility of an infinite, elastic, fluid-filled, circular cylindrical shell using the spectral equations of motion. For axisymmetric waves in the shell-fluid system the input mobility is low and it is high when the system behaves in a flexural manner.

White and Sawley [28] focussed on the role of the transmission path connecting the source to the receiver in practical piping systems. An approximate technique for assessing the distribution of acoustical and vibrational energy was presented. Cuschieri [29] investigated the transmission of vibrational power from the piping system to the supporting structure using power flow and structural mobility methods. This approach can be applied to isolated straight pipe sections as well as a number of subsections joined together by components that can be represented by structural mobility terms. Lewis and Roll [30] have addressed the problem of pulsation and vibration in piping systems caused by control valves. The most common cause for this was acoustic resonances. Application guidelines were presented to help the pipe system designer evaluate piping systems for potential acoustic resonances. Several solutions for existing pulsing pipelines were also discussed.
1.1.1 Sensing and wave-decomposition in fluid-filled pipes

In order to control the vibrations in fluid-filled piping systems it is necessary to identify the various propagating waves in the system. To understand the behaviour of piping systems it is normally assumed that the pipe wall can be approximated as a cylindrical shell. In general for frequencies far below the ring frequency \( \omega_o = c_L / a \), where \( c_L \) is the compressional wave speed in a plate of the same material as the pipe and \( a \) is the shell mid-plane radius) of the cylindrical shell i.e. the pipe wall, four different types of waves can propagate energy simultaneously in the system. Three of these are associated with the \( n=0 \) axisymmetric mode of the cylindrical shell and the fourth one corresponds to the \( n=1 \) beam like mode of the shell. Waves of higher circumferential mode orders (\( n=2,3\ldots \)) are evanescent at very low frequencies, i.e. they do not propagate energy much farther from the axial location where they are excited. For higher frequencies however, the propagation of these higher order wave types must be considered. The coupled nature of wave propagation in fluid-filled pipes makes the sensing and identification of these wave types difficult.

Pavic [31] was the first to investigate the wave decomposition problem in fluid filled pipes of finite length with application to acoustic and structural intensity measurement methods. At low frequencies (i.e. non-dimensional), simple rod and beam type laws approximate complicated dispersion laws for waves in pipe. At these low frequencies the fluid filled pipe becomes a homogeneous one-dimensional wave-guide which makes suitable, measurements of energy flow by detecting surface vibrations only. Pavic briefly described an array of accelerometers and strain gauges for extracting wave amplitudes. Recently, de Jong [32] presented other wave decomposition techniques for
fluid-filled pipes, which were based on the use of accelerometers and flush-mounted pressure transducers. The wave decomposition methods described above require very large arrays of sensors with matched sensitivities and their reliability depends on the accuracy of the model providing the wave-number information. Hence defects in the pipe cross section or bubbles at the interface between the shell and the fluid can adversely affect the decomposition. The effect of bubbly mixtures on attenuation of sound velocity has been investigated by Silberman [33]. To avoid this problem, Corrado and Clifton [34] investigated array-processing requirements for experimental determination of wave-numbers. They showed that a large number of sensors are necessary for high resolution of the beam forming process since the subsonic speeds of the n=1 beam bending mode and higher order flexural modes require small sensor spacing. Moreover, the predominance of fluid duct modes at high frequencies will require fine wave-number resolution. Pavic [35] evaluated the expressions for the axial and circumferential component of the energy flow along the shell wall in terms of the axial, radial and tangential components of shell motion. By averaging the energy flow in the circumferential and axial directions, the governing expressions were simplified. In this way the measurements could be accomplished without the need for excessive equipment and processing. Verheij [36] has formulated cross-spectral density methods for measuring the one-dimensional power flow for pipes in the low frequency range carried by bending, longitudinal and torsional waves.

In another work [37], the same author studied the measurement of structure-borne sound energy flow through pipes for the purpose of transmission path ranking. Experiments were performed on two lightly damped pipe sections of a cooling water
circuit to illustrate some practical measuring aspects and procedures for error analysis. Due to the highly reactive fields that are present under practical conditions, the importance of using high quality instrumentation, as well as of careful wave type separation and special phase calibration was stressed. James [38] evaluated the acoustic power radiated, the vibrational response of the shell wall, and the pressure amplitudes of the exterior and interior fluids of an infinite water-filled shell immersed in air for both a mono-pole source and a mechanical point force. de Jong and Verheij [39] developed practical methods to determine acoustic and vibrational energy flow along straight fluid-filled pipes from acceleration measurements on the pipe surface. Approximate expressions were derived for the wave speed and energy flow equations. Appropriate accelerometer configurations were developed to measure the energy flow in the different wave types separately. Preliminary measurements showed that practical energy flow measurements on pipes were feasible, provided reliable wave-type decomposition and accurate phase measurements could be performed. Briscoe and Pinnington [40] presented practical methods for measuring vibrational power transmission in empty or fluid filled pipes using various configurations of axially aligned accelerometers and circumferential piezoelectric sensors. Measurements of energy flow in both the structure and the fluid were successful in mid-frequency ranges. However, finite difference errors limited the high frequency performance. A Statistical Energy Analysis model was developed by Bourget and Fahy [41] to evaluate the transmission of vibrational energy between the main components of the engine, especially along the pipes connecting them. A method for measuring vibrational power flow along a uniform cylindrical thin-walled pipe was described. It was based on the use of both theoretical and experimental results and
required only several simple radial acceleration measurements to be made on the surface of the pipe. Application of this method for evaluating the vibrational power flow along a rocket engine fuel line has been shown.

The use of external sensors for measuring the vibrations in a fluid filled pipe has also received lot of attention. Pinnington and Briscoe [42] investigated an external circumferential piezoelectric transducer, which detects the radial wall motion of the fluid filled pipes. The sensitivity of the transducer to the fluid (s=1) and structural (s=2) axisymmetric n=0 waves is investigated. It was concluded that for many practical cases the transducer is effective for internal fluctuating pressure measurements. For empty pipes however, the transducer measures axial strain in the shell. In another similar work Pinnington and Briscoe [43] designed a circumferential transducer to measure the dynamic pressure of gas or fluid within a pipe. The transducer was primarily a piezoelectric wire wrapped circumferentially around the pipe wall such that it was sensitive only to axisymmetric waves. The sensor was found to measure the magnitude and phase of the fluid borne n=0 wave accurately for fluid based excitation in steel pipes and for any kind of excitation in perspex or rubber pipes. For structural excitation in steel pipes, however, the sensor measurements were contaminated by the n=0 compressional shell wave. Royston [44] experimentally evaluated shaped polyvinylidene fluoride (PVDF) film as a non-intrusive sensor to measure plane-wave sound propagating through water in a steel pipe. The PVDF film was shaped to provide distributed spatial averaging which is not possible with discrete pressure sensors. The film was shaped specifically to respond to the shell breathing, n=0 mode of the pipe and to provide a zero-phase-lag, low-pass-filtered measurement of plane-wave sound pressure. The film response
compared well with hydrophone measurements. Axial strain along the pipe surface can be measured to provide information about the power flow. Lee and O’Sullivan [45] designed a uniaxial strain rate gage, which measured the strain rate only along a specific direction and a pure shear strain rate gage that measured the in-plane shear strain rate. Experimental as well as theoretical results were presented.

1.1.2 Passive methods for vibration control in fluid-filled pipes

The most common cause of noise generation in fluid filled pipes are positive displacement pumps (mostly piston pumps) which produce significant pressure fluctuations (at multiple harmonics of their operating frequency) along the piping system. One of the methods to reduce the fluid-borne noise is to utilize the acoustic waves within the tubing to partially cancel the pressure fluctuations in a manner similar to a reactive automobile muffler. Among these wave-canceling silencers are quarter wavelength resonators. Dodson, et al. [46] experimentally tested three types of quarter wavelength (in-line, flexible side branch, and rigid side branch) in an industrial-scale hydraulic system. It is important to note that the piping system used in these experiments was flexible steel reinforced rubber hose. This setup prevented the generation and transmission of the n=1 bending waves, which was very important as will be discussed later in Chapter 4. All three types were found to possess positive noise reduction characteristics with maximum insertion loss from 10 to 20 dB. The behaviour of rigid side branches could be predicted accurately using one-dimensional acoustic theory. However, the silencers employing flexible hose in their construction did not behave as predicted. The presence of multiple wave types and conversion of wave types at system junctions explained this. Neise and Koopmann [47] experimentally investigated the use
of an acoustic resonator to reduce the blade passing frequency tone produced by turbo-machinery. The mouth of the quarter wavelength resonator was constructed from a series of perforated plates. The resonator was tuned by changing the length via a movable end plug. For appropriate tuning of the resonator, reductions in the blade passing frequency tones of up to 29 dB were observed with corresponding overall sound pressure levels reductions of up to 7 dBA. Parameters that affected the resonator response were the porosity and hole size of the resonator mouth and the flow velocity near the cut-off region. Quincke [48] was the first to propose the use of unequal acoustic path lengths in a branched duct to cause the acoustic waves to destructively interfere with one another thus canceling propagating waves. Selamet, et al. [49] extensively investigated the Hershel Quincke tube analytically, numerically and experimentally. A general expression for the transmission loss characteristics of the Herschel-Quincke tube was developed. A one dimensional finite difference model for attenuation of sound was also studied computationally. Transmission loss predictions from both analytical and computational models were shown to correlate well with experimental data.

Helmholtz resonators are components frequently used in applied acoustics. A simple Helmholtz resonator essentially consists of a cavity of volume V and a neck of area S and length L. The fluid in the neck acts as the mass element while the acoustic pressure inside the cavity provides the stiffness element. At the resonant frequency, the mouth of the Helmholtz resonator radiates sound out of phase with the incident duct field thus canceling it to a zero pressure field. The duct impedance is thus, zero to the propagating wave at that frequency causing most of its energy to be reflected back to the source. Thus the characteristic property of such a resonator is its ability to reactively
reflect sound waves of a particular frequency, the so-called resonant frequency. Rayleigh [50] derived the classical formula for calculation of resonant frequencies of Helmholtz resonators. Alster [51] derived an improved formula for the calculation of Helmholtz resonators. The effect of the inner shape of a resonator on the resonant frequency was demonstrated and a new concept, the form factor of a resonator, was introduced and defined. It provides a means for prediction of resonant frequencies of some open cavities for which methods of calculation were not known. Experimental results and theoretical calculations for a number of resonators show very good agreement.

1.1.3 Active control

The principle behind active control is adding a secondary source(s) in order to cancel the response generated by the primary source or disturbance input. In 1936, Leug [52] first proposed an active control technique to reduce noise using destructive interference of sound waves. With the development of digital signal processing his invention became practically viable. Nelson and Elliot [53] were the first to compile the theory of active control of sound. Initial application of feed-forward control was to one-dimensional acoustic fields as summarized by Warnaka [54]. Feed-forward least-mean-square (LMS) and recursive-least-mean-square (RLMS) adaptive algorithms have been applied to active control of bending motion in infinite or semi-infinite thin beams [55,56,57]. Recently, the simultaneous control of flexural and extensional waves in beams has been demonstrated by a multi-channel LMS approach, in conjunction with specialized piezoceramic transducers [58]. The interested reader is
referred to the textbooks of Nelon, Elliot and Fuller et al for a general introduction and survey of active control of sound and vibration.

1.1.4 Active control of vibrations in fluid-filled pipes

In a piping system connected to a pump, the dominant fluid pulsations are at a frequency corresponding to the pump’s blade passing rate and its harmonics. Flexible bellows section, used to suppress fluid pulsations, tend to stiffen in highly pressurized pipe work systems making them ineffective. Hence 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 feed forward control using a reference signal derived from the speed of the pump [53]. Brennan, et al. [59] designed a non-intrusive fluid-wave actuator and sensor pair for use in an active control system to control fluid-borne vibrations in pipe-work systems. A theoretical framework was developed to integrate the transducers into the pipe and couple them to the motion of the fluid inside the pipe. Experimental results show a reduction in the fluid wave amplitude of around 20 dB though there is an increase in the vibration of the pipe wall in the vicinity of the control actuator.

Berengier and Roure [60] used the results of modal theory, which permits precise expression of the acoustic field generated by a real source in a hard-walled wave-guide, to express the characteristics of different absorbing systems formed by several independent sources. The relations thus expressed lead to the conception of servomechanisms for various systems; mechanisms are indispensable in order to achieve the appropriate absorption over a wide band of frequencies. Micheau, et al. [61] showed
that an oscillating valve positioned in the exhaust duct of an internal combustion engine is capable of reducing the main harmonic of pulsed flow, thus reducing the noise radiated to the outside. Experimentation showed that the adaptive feedback controller defined in the frequency domain was capable of regulating the main harmonic of a periodic disturbance. The adaptability of the control was validated experimentally.

Brevart and Fuller [62] analytically investigated the active control of wave propagation in fluid-filled elastic cylindrical shells. Control was applied as radial line forces on the shell while radial shell vibration was minimized at up to two downstream points. For the \( n=0 \) axisymmetric wave, it was easier to control the total power flow for fluid-type excitation. This was explained as the particular nature of the wave i.e., pressure near field close to the shell wall. However, for the structural-wave type propagation, the control performance severely decreased due to the coupling between the two media. In the \( n=1 \) case, control performance was good below the first acoustic cutoff frequency. Above this frequency, good control of power flow using structural forces is difficult to achieve, as the fluid due to acoustic waves cutting on and propagating carries more energy. Brevart [72] also experimentally demonstrated the potential of active control for control of total energy flow in fluid-filled piping systems. For his experimental setup he used plexiglass pipe as the cylindrical shell. He performed two sets of control experiments, each associated with a particular disturbance type. The first set of experiments was associated with the active control of axisymmetric wave propagation in the cylindrical shell. PVDF (polyvinylidene fluoride) cables wrapped around the shell wall were used as the excitation source, control actuators and sensors. The results indicate that for the axisymmetric case, it was easier to control the power flow
propagating along the coupled system when the disturbance was fluid type incident wave. For structural type wave propagation, the fluid severely decreased the control performance. The second set of experiments was related to the control of shell vibrations excited by a radial point force disturbance. The primary excitation source was an electromagnetic shaker, while the control actuator was a PVDF cable wrapped around the shell and two electromagnetic shakers. A PVDF cable was used to record the axisymmetric shell displacement while a shaped PVDF film monitored the radial displacement. In this case results indicated that the shell wall mainly conveyed the power flow induced by a point force disturbance and hence structural control inputs were very effective at reducing the power flow in the system. Harper and Leug [63] demonstrated that low frequency noise transmission in fluid-filled pipes could be reduced substantially by employing a seven-degree-of-freedom system. The control actuators consisted of six piezoelectric polymer PVDF cables wound helically around a flexible section and a hydro-sounder acting directly into the pipe fluid. Kostek et al., [64] developed a hybrid noise control system by combining an active noise control system with adaptive-passive elements. Specifically, the power requirements of the ANC system was reduced while attenuation occurred over a greater frequency range than the adaptive-passive system alone could achieve. The adaptive-passive element was a tunable Helmholtz resonator. An algorithm, which tunes the resonator in the presence of unmeasurable broadband disturbances, was presented. This enabled the adaptive-passive system to tune in the presence of multiple tones.

Maillard [65] explored the potential of active control for reducing fluid pressure pulses generated by the system pump and hydraulic engines. The target system for his
work was a hydraulic pipe-work on board a marine vessel. Pressure pulses generated in
the pipe-work by a hydraulic pump and engine, led to the generation of high noise levels.
Maillard implemented a non-intrusive structural actuator for the active control of these
pressure pulses. The actuator consisted of a circumferential ring of piezoelectric stacks
acting on the pipe wall to generate an axisymmetric plane wave in the fluid through radial
motion coupling. He demonstrated the effectiveness of the actuator in achieving global
control of the fluid pressure fluctuations. The pressure level generated by the actuator
was however lower than the ones measure on the target system which would not yield
complete cancellation of the primary pulsation. Maillard suggested different steps like
cooling the piezoelectric stacks, smaller machining tolerances, bonding the stacks to the
pipe wall directly to improve the actuator efficiency and thus generate higher-pressure
levels. There was also a significant increase in the pipe wall vibrations. As previously
discussed, this signified the coupled nature of wave propagation in fluid filled pipes.
Thus total control of power flow could not be achieved by solely controlling the fluid
pressure pulsations. Hence another area of research would be to address the total control
of power flow in the pipe system. Implementing passive isolation joints to reduce the
transmission of the structural vibrations generated by the actuator could do this.

1.2 Objective and approach

The goal of this work was to develop passive, active and active/passive devices
and methodology for reduction of vibration and pressure in fluid filled piping systems.
The objective is to simultaneously control the fluid and structural waves, thus controlling
the total vibrational power flow. The work done here is fully experimental and as
discussed above involved the development and integration of components to reduce the structural acoustic power flow in the fluid filled pipes. Ideally, it is desired to keep all the vibrational waves in the system and thus power from propagating beyond a certain position on the pipe. Since the nature of wave propagation in fluid-filled pipes is coupled, this meant that both the structural and fluid waves should be cancelled at the same point.

The final goal is to apply these devices to a practical system, which in this case was a hydraulic pipe-work on board a marine vessel related to Maillard’s work [65] or a system similar to it. To this end an experimental rig approximating a semi-infinite pipe was built. Different vibration reducing devices were then developed and tested on this rig.

In order to actively control the total power flow in fluid-filled pipes, real-time estimation of the amplitude of the propagating waves is needed. This is needed since most practical piping systems are excited by sources whose operating frequency varies with time. Hence to keep track of the change in disturbance frequency, real time estimation of the propagating wave was needed. However, the characteristic equation of a fluid-filled cylindrical shell is significantly more complex. The solutions of the equation may first be decomposed into mutually independent circumferential modes (n=0,1,2…). For each circumferential mode, multiple propagating wave types with dispersion laws of varying complexity exist. The number of propagating wave types increases with frequency. Thus, due to implementation complexity, real time estimation and minimization of the total power flow, as done by Gibbs [66] for thin beams, was not attempted here.
The control approach adopted in this work was much simpler. Given a disturbance and a practical configuration of error sensors and control forces which could sense and actuate the critical modes or waves in the bandwidth of interest with sufficient level, experiments were carried out to determine whether large attenuations of the fluid pressure could be achieved using active control. At low frequencies and for variable excitation frequencies, passive devices are impractical to implement in real piping systems and lose their effectiveness. On the contrary a fully active approach theoretically possesses complete flexibility but in practice is limited by the actuator control authority and power levels. Thus a combined active-passive approach would potentially possess both control flexibility as well as reduced power requirements eliminating disadvantages of both these approaches. Hence a hybrid system combining a passive and an active system was also developed and tested to combine the advantages of both these approaches.

1.3 Organization

In Chapter 2 a comprehensive review of the theory of wave propagation in fluid-filled cylindrical shells is presented. The characteristic equations of the coupled fluid-shell system are derived, dispersion curves analyzed and physical wave behaviour is discussed. Chapter 3 describes the pipe rig used for the experiments. Different methods of isolating the excitation source from the structure are discussed. The different transducers used for sensing the pipe wall vibrations and the fluid pressure are described. Wave decomposition and power flow measurement methods are also discussed. Chapter 4 describes the experimental results for different passive devices. The results of the active
control experiments are then presented in Chapter 5. Finally, based on the results of the experiments presented in the earlier chapters, conclusions are drawn and recommendations made in Chapter 6.
Chapter 2

Theory of Vibrations in Cylindrical Shell Systems

In this chapter the basic mechanisms involved in the vibrations of long fluid-filled cylinder which are representative of piping systems are covered. A thorough analysis of these mechanisms is critical in order to understand the dynamic behavior of piping systems. Several authors have developed theoretical models to describe the vibrations of fluid filled pipes. Wave equations for pipes or cylinders can be found in Leissa [1]. Fuller and Fahy [14] comprehensively discussed the propagation of different wave types in thin elastic fluid-filled cylindrical shells. In the same work, the authors also evaluated the expressions for energy flow along the shell’s wall, in terms of axial, tangential and radial components of shell wall motion. Esparcieux confirmed some of these wave-types in wave-number spectra obtained from pipe surface acceleration measurements [67]. Pavic showed that at low frequencies the complex dispersion laws for waves in pipes simplify and the energy flow in both the wall and the fluid can be evaluated from surface vibrations only [31].

The chapter begins with a description of the propagation of waves in fluid filled pipes. Then the characteristic equation of a cylindrical elastic shell filled with fluid is derived. The dispersion curves for a steel-shell, *in vacuo* and filled with water, are then presented. Based on de Jong and Verheij’s [39] work, practical methods to determine
acoustic and vibrational energy flow are also discussed. Approximate expressions for power flow through the pipe are given.

### 2.1 Wave propagation in fluid-filled pipes

There are two main frequency regions in a fluid-filled pipe, a low and a high frequency region, which are separated by the ring frequency, $\omega_r$. The ring frequency is given by $\omega_r = \omega_c L / a$ where $c_L$ is the compressional wave speed in a plate of the same material as the pipe and $a$ is the shell mid-plane radius. At the ring frequency, the pipe vibrates in a breathing mode and the pipe circumference equals the associated compressional wavelength in a plate, $\lambda_L = 2\pi c_L / \omega_r$. In order to simplify the coupled fluid-shell equations, the excitation frequency $\omega$ is normalized by the ring frequency. The resulting non-dimensional frequency is then defined as $\Omega = \omega / \omega_r = \omega c L / c_c$. It has been found that at very low frequencies, i.e. $\Omega << 1$, the characteristic equation of the fluid filled pipe is simplified [31]. For steel pipes of common sizes the ring frequency is very high (e.g. 40Khz for a 2” industrial steel pipe, mean radius $a = 1.078”$) compared to the excitation caused by low frequency pressure pulsations from noise sources such as pumps (e.g. 70-150 Hz for a typical pump). Hence the simplified equations of motion for such cylindrical shells can be utilized in the present studies.

For frequencies well below the ring frequency of the pipe ($\Omega << 1$), the energy is carried by four types of propagating waves with real wavenumbers [31, 42]. The first three are axisymmetric wave types with circumferential mode number $n=0$. The fourth wave type is the $n=1$ flexural or bending wave. The first $n=0$ axisymmetric wave, termed $s=1$, is a predominantly acoustic wave in the fluid with some radial wall motion associated with the shell compliance. The second axisymmetric wave ($s=2$) is
predominantly a compressional wave in the shell with some associated radial wall motion influenced by the Poisson ratio and the fluid bulk modulus of elasticity. The third axisymmetric wave, \( s=0 \), is a torsional wave primarily in the shell and uncoupled from the fluid with insignificant pipe wall radial motion. The fourth type of wave, is the \( n=1 \) flexural or bending wave characterized by the lateral motion of the pipe in such a way that its cross-section remains unchanged i.e. vibrates like a rod. Real wave number solutions of the equations of motion for \( n>1 \) show cut-on frequencies \( f_{cn} \), below which no wave of circumferential mode \( n \) can propagate. These cut-on frequencies for fluid-filled pipes are given approximately by [39],

\[
f_{cn} = f_{ring} \sqrt{\frac{\beta^2 n^2 (n^2 - 1)^2}{1 + n^2 + \eta n}}
\]

where \( f_{ring} \) is the shell ring frequency, \( \beta^2 = \frac{h^2}{12a^2} \) the shell thickness parameter and \( \eta = \frac{a \rho_f h \rho_s}{a} \) the fluid-shell mass ratio. For our case i.e. industrial 2” steel pipe (pipe mean radius \( a = 1.078" \), pipe wall thickness \( h = 0.157" \), \( \beta = 0.002 \) and \( \eta = 0.91 \)) the above equation gives the cut-on frequencies for the \( n=2 \) and the \( n=3 \) waves as 3735 Hz and 14942 Hz, respectively. However, it can be recalled that the frequency of pressure pulsations caused by the hydraulic pump is much lower compared to the cut-on frequencies of the higher modes. Hence these modes are not considered in the analysis. Only the \( n=0 \) axisymmetric and \( n=1 \) bending modes will be considered for the propagation of energy.

All four propagating waves described above do not contribute equally to the sound radiated from the pipe wall. The \( n=0 \) axisymmetric wave radiates sound proportional to the radial wall motion. In the far field it would be equivalent to a line of
monopoles. The torsional wave is tangential to shell cross-sectional and hence does not radiate sound. The \( n=1 \) bending wave also radiates sound. In the far field it is equivalent to a line of dipoles with amplitude proportional to the pipe deflection.

Figure 2.1 represents the mode shapes of the first six circumferential and corresponding axial modes [1]. The first circumferential mode \( n=0 \) is referred to as the breathing or pulsating mode. The \( n=1 \) mode or the flexural mode doesn’t feature a change in the cross-sectional shape of the shell. Higher order modes feature lobar type mode shapes with \( 2n \) anti-nodes along the pipe circumference, i.e., the spatial distribution of mode \( n \) takes the form \( \cos(n\theta - \gamma_b) \) where \( \theta \) is the circumferential angle and \( \gamma_b \), a polarization angle [64]. The circumferential modes \( n \geq 2 \) are not considered in this work for the previously stated reason.

2.2 Equations of motion of the coupled system

The Donnell-Mushtari equations are very convenient to represent the harmonic vibrations of thin walled cylindrical shells containing an acoustic field. Several authors in a number of books and technical papers used these equations. The Donnell-Mushtari equations which are a simplified form of Kennard’s shell equations [69] do not give reliable results at very low frequencies (\( \Omega \to 0 \)). Kennard’s shell equations give good results at very low frequencies. Kennard’s equations are valid for thin-walled shells and exclude the effects of rotary inertia and transverse shear stresses. However at low frequencies the shell thickness is not an important factor. Hence Kennard’s shell equations will be used in this analysis.
Figure 2.1: Circumferential and axial modes for a cylindrical shell
Figure 2.2: Cylindrical co-ordinate system for equations of motion
The cylindrical coordinate system used in this analysis is shown in Figure 2.2. \( u, v \) and \( w \) are the shell axial, tangential and radial displacements respectively while \( r, \theta \) and \( x \) represent the cylindrical coordinates. \( a \) is the shell mid-plane radius while \( h \) the thickness of the shell. The inhomogenous form of Kennard’s shell equations, including a pressure fluid loading, \( p_a(\theta,x) \), and an external pressure load \( p_0 \) with axial, tangential and radial components \( p_0^x, p_0^\theta \) and \( p_0^r \) respectively, is written as:

\[
\frac{\partial^2 u}{\partial x^2} + \frac{1-v}{2a^2} \frac{\partial^2 u}{\partial \theta^2} + \frac{1+v}{2a} \frac{\partial^2 v}{\partial x \partial \theta} + \frac{v}{a} \frac{\partial w}{\partial x} - \frac{\ddot{u}}{c_L^2} = \frac{p_0^x(1-v^2)}{Eh},
\]

\[
\frac{1+v}{2a} \frac{\partial^2 u}{\partial x \partial \theta} + \frac{1-v}{2a^2} \frac{\partial^2 v}{\partial x^2} + \frac{1}{a^2} \frac{\partial^2 v}{\partial \theta^2} + \frac{1}{a^2} \frac{\partial w}{\partial \theta} + \frac{h^2}{8a} \frac{1-v}{\beta^2} \frac{\partial^2 w}{\partial \theta^2} \frac{\partial w}{\partial \theta} - \frac{\ddot{v}}{c_L^2} = \frac{p_0^\theta(1-v^2)}{Eh},
\]

\[
\frac{\nu}{a} \frac{\partial u}{\partial x} + \frac{1}{a^2} \frac{\partial v}{\partial \theta} + \frac{w}{a^2} + \beta^2 \left( a^2 \frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial \theta^2} + \frac{1}{a^2} \frac{\partial^4 w}{\partial \theta^4} + \frac{4}{2} \frac{1}{a^2} \frac{\partial^2 w}{\partial \theta^2} + \frac{2}{2(1-v)} \frac{\partial w}{\partial \theta} \right) + \frac{\ddot{w}}{c_L^2} - \frac{p_a(1-v^2)}{Eh} = \frac{p_0^r(1-v^2)}{Eh}.
\]

where \( \beta^2 = h^2 / 12a^2 \), is the shell thickness parameter and \( u, v \) and \( w \) are the axial, tangential and radial components of the shell wall displacement. The extensional phase speed in the shell material is given by \( c_L = \sqrt{E/\rho_s(1-v)^2} \), where \( E, \nu \) and \( \rho_s \) are the Young’s modulus, the Poisson’s ratio and the specific mass of the shell material, respectively. Note that the pressure fluid loading, \( p_a(\theta,x) \), acts normally to the shell surface and hence appears only in the third equation.
The assumed form of solution for the displacement of the shell wall, associated with an axial wave-number $k_{ns}$, which satisfies the equations of motion for the cylindrical shell are

$$u = \sum_{n=0}^{\infty} \sum_{s=1}^{\infty} U_{ns} \cos(n\theta) \exp[i k_{ns} x - i \omega t - i \pi/2], \quad (2.4a)$$

$$v = \sum_{n=0}^{\infty} \sum_{s=1}^{\infty} V_{ns} \sin(n\theta) \exp[i k_{ns} x - i \omega t], \quad (2.4b)$$

$$w = \sum_{n=0}^{\infty} \sum_{s=1}^{\infty} W_{ns} \cos(n\theta) \exp[i k_{ns} x - i \omega t]. \quad (2.4c)$$

where $U_{ns}$, $V_{ns}$ and $W_{ns}$ are the axial, tangential and radial displacement amplitudes. The subscript ‘ns’ refers to a particular wave type s of circumferential order n.

The assumed form of the pressure field in the contained fluid, which satisfies the acoustic wave equation in cylindrical coordinates is,

$$p = \sum_{n=0}^{\infty} \sum_{s=1}^{\infty} P_{ns} \cos(n\theta) J_n(k^r_{s} r) \exp[i k_{ns} x - i \omega t], \quad (2.5)$$

where $P_{ns}$ is the pressure amplitude and $k^r_{s}$ is the radial wave-number given by the vector relation

$$(k^r_{s})^2 = k_f^2 - k_{ns}^2, \quad (2.6)$$

where $k_f = \omega/c_f$, is the fluid wave-number.

Substitution of these forms in the shell equations results in the equations of motion of the coupled system in terms of the amplitudes of the three displacements and the acoustic pressure. In equations (2.4) and (2.5), $k_{ns}$ and $k^r_{s}$ are the axial and radial
wavenumbers respectively, \( n \) is the circumferential mode number and subscript \( s \) denotes a particular branch of the dispersion curves.

The motion of the system can be represented in matrix form. However before this is done, it is convenient to apply the boundary condition at the shell wall. For continuity of contact between shell and fluid surface, the shell radial velocity and radial fluid velocity must be equal. Thus for a particular mode \((n, s)\) the radial velocity of the fluid at the shell wall, given by the momentum equation, is

\[
\nu_{e}(r = a) = -\left(\frac{1}{i\rho_f \omega}\right) \frac{\partial p}{\partial r} = -\frac{k_s^r J_n(k_s^r a)}{i\rho_f \omega} P_{ns} \cos(n\theta) \exp[i\omega t - ik_{ns} x], \quad (2.7)
\]

Equating \( \nu_{e}(r = a) \) to the shell radial velocity derived from equation (2.4c) enables the fluid pressure amplitude to be written in terms of the shell radial displacement amplitude as

\[
P_{ns} = [\omega^2 \rho_f / k_s^r J_n(k_s^r a)] W_{ns}. \quad (2.8)
\]

The pressure fluid loading at the shell wall can thus be expressed in terms of the shell radial vibration as

\[
p_{ns}(\theta, x) = \sum_{n=0}^{\infty} \sum_{x=0}^{\infty} W_{ns} \frac{\rho_f \omega^2 J_n(k_s^r a)}{k_s^r J_n(k_s^r a)} \cos(n\theta) \exp[ik_{ns} x - i\omega t]. \quad (2.9)
\]

The free vibrations of the coupled system can be represented in matrix form as

\[
\begin{bmatrix}
L_{11} & L_{12} & L_{13} \\
L_{21} & L_{22} & L_{23} \\
L_{31} & L_{32} & L_{33}
\end{bmatrix}
\begin{bmatrix}
U_{ns} \\
V_{ns} \\
W_{ns}
\end{bmatrix}
= \begin{bmatrix}
0 \\
0 \\
0
\end{bmatrix} \quad (2.10)
\]

where

\[
L_{11} = -\Omega^2 + (k_{ns} a)^2 + \frac{1}{2}(1 - \nu)n^2, \quad L_{12} = \frac{1}{2}(1 + \nu)n(k_{ns} a), \quad L_{13} = \nu(k_{ns} a), \quad (2.11a-c)
\]
\[ L_{21} = L_{12}, \quad L_{22} = -\Omega^2 + \frac{1}{2}(1-v)(k_{ns}a)^2 + n^2, \quad L_{23} = n - \frac{h^2}{8a^2} \frac{v}{1-v} n(n^2-1), \quad (2.11d-f) \]

\[ L_{31} = L_{31}, \quad L_{32} = n, \quad (2.11g,h) \]

\[ L_{33} = -\Omega^2 + 1 + \beta^2 \left\{ \left[(k_{ns}a)^2 + n^2 \right] - \frac{4-v}{2(1-v)} n^2 + \frac{2+v}{2(1-v)} \right\} - FL \quad (2.11i) \]

where \( \Omega \) is the non-dimensional frequency, defined as \( \Omega = \alpha a / c_L \). The extensional phase speed of the shell material is \( c_L \) and \( v \) is the Poisson’s ratio of the shell material.

\( FL \) is the fluid loading term due to the presence of the fluid acoustic field given by,

\[ FL = \Omega^2 \frac{\rho_f}{\rho_s} \left( \frac{h}{a} \right)^{-1} \frac{J_n'(k_s'a)}{(k_s'a) J_n(k_s'a)}. \quad (2.12) \]

The radial fluid wave-number \( k_s'a \) is related to the axial wave-number \( k_{ns}a \) by the usual vector relation, written in terms of the shell non-dimensional frequency \( \Omega \) as

\[ k_s'a = \pm \left[ \Omega^2 (c_L/c_f)^2 - (k_{ns}a)^2 \right]^{1/2}, \quad (2.13) \]

where, \( c_f \) is the free wave speed in the fluid.

Equation (2.12) provides immediate insight into the effect of the contained fluid on the shell response [14]. Varying the fluid loading term can be seen to directly influence the behaviour of the system. For very low frequencies (\( \Omega \to 0 \)) the fluid loading term is small for shell waves and thus one would expect the fluid-filled shell response to be close to that of an in vacuo shell. Similarly when \( J_n'(k_s'a) = 0 \), the boundary condition for a rigid walled duct mode, the fluid loading term is large and the system behaviour will approach that of an acoustic wave in a rigid walled tube. It can also be seen from equation (2.12) that increasing the shell thickness or decreasing the
density ratio, $\rho_f / \rho_s$, decreases the effect of the contained fluid on the shell response and leads to a decrease in the coupling of the shell and fluid.

For the homogenous system of equations (2.10) to have a solution the determinant of the matrix $L$ should be zero,

$$|L| = 0. \quad (2.14)$$

Equation (2.14) provides the characteristic equation of the coupled system. Solving this equation for $k_{ns}a$, for various non-dimensional frequencies $\Omega$, yields the dispersion curves of the system.

### 2.3 Dispersion Curves

To obtain a better understanding of fluid-filled pipe behaviour it is rewarding to plot the dispersion curves for each wave type. A dispersion curve represents the associated wave number of each wave type versus frequency. In this analysis, dispersion curves are plotted only for real wave numbers i.e. propagating waves, since it is these waves, which carry energy through the piping system. The following approximate expressions can be derived for non-dimensional wave-numbers $k_{L,F,T,B}$ where subscripts $L, F, T$ and $B$ denote the $n=0$ longitudinal shell wave, acoustic fluid wave and torsional shell wave and the $n=1$ bending wave respectively [39],

$$k_L^2 = (1 + \Delta)\Omega^2 = \zeta_L^2 \Omega^2, \quad (2.15a)$$

$$k_F^2 = \left(\Psi + \frac{2\eta + v^2}{1-v^2}\right)\Omega^2 = \zeta_F^2 \Omega^2, \quad (2.15b)$$

$$k_T^2 = \frac{2}{1-v}\Omega^2 = \zeta_T^2 \Omega^2, \quad (2.15c)$$
\[ k_B^2 = (2 + \eta)^{1/2} \Omega = \zeta^2 \Omega, \tag{2.15d} \]

where \( \zeta \) is a wave-number factor and \( \eta = a \rho_f / \rho_s \) the fluid-shell mass ratio. \( \Omega \) and \( \Delta \) are given by, \( \Psi = (c_L / c_F)^2 \) and \( \Delta = v^2 (\Psi - 1) / [(\Psi - 1)(1 - v^2) + 2\eta + v^2] \). The first three wave numbers were derived by Pavic [70]. The bending wave number was derived by Pavic in a related publication [31]. Table 2.1 gives the pipe dimensions, material properties and other derived characteristics of the test rig built for carrying out experiments, which will be discussed in the Chapters 3, 4 and 5. The values given in this table are used to calculate the longitudinal, fluid, torsional and bending wave numbers from equations (2.15 a-d). These non-dimensional wave numbers are then plotted against frequency to obtain the dispersion curves of the system. It is to be noted that dispersion curves are plotted here for only real wave number \( k \) i.e. those waves, which propagate.

Figures 2.3 and 2.4 show the dispersion curves of the piping system in vacuo and filled with water. It can be seen from the dispersion curves that at low frequencies only four types of waves have real wave numbers and propagate through the test rig for both the in vacuo and the fluid-filled cases. Considering the in vacuo shell, Figure 2.3, there are three \( n=0 \) axisymmetric waves and one \( n=1 \) bending wave propagating through the pipe. The first \( n=0 \) axisymmetric wave propagating through the system is a torsional wave denoted by \( s=0 \). It is uncoupled from radial shell wall motion and hence is unaffected by fluid loading [14]. The second \( n=0 \) axisymmetric mode (\( s=1 \)) is a fluid wave which is similar to the acoustic plane wave in a rigid-walled tube at low frequencies. This wave is affected by fluid loading. The final \( n=0 \) axisymmetric mode, denoted as \( s=2 \), is extensional in nature and is generally referred to as the longitudinal
<table>
<thead>
<tr>
<th>Material</th>
<th>Steel</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe mean radius a (m) for 2” industrial pipe</td>
<td>0.028</td>
<td>-</td>
</tr>
<tr>
<td>Pipe wall thickness h (m) for 2” industrial pipe</td>
<td>0.004</td>
<td>-</td>
</tr>
<tr>
<td>Thickness-radius ratio (h/a)</td>
<td>0.143</td>
<td>-</td>
</tr>
<tr>
<td>Ring frequency (Hz)</td>
<td>39442</td>
<td>-</td>
</tr>
<tr>
<td>Young’s modulus (N/m$^8$)</td>
<td>$19.2 \times 10^{10}$</td>
<td>-</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.3</td>
<td>-</td>
</tr>
<tr>
<td>Density (kg/m$^3$)</td>
<td>7700</td>
<td>1000</td>
</tr>
<tr>
<td>Free wave speed (m/s)</td>
<td>5200</td>
<td>1500</td>
</tr>
</tbody>
</table>

Table 2.1: Shell and fluid properties of test rig used for experiments
Figure 2.3: Dispersion curves for an *in vacuo* cylindrical steel shell, h/a=0.143

Dispersion curves for the coupled shell-fluid system

Figure 2.4: Dispersion curves for water-filled cylindrical steel shell, h/a=0.143
wave. The longitudinal $s=2$ wave propagates mainly in the pipe wall. For the $n=1$ beam mode, only one wave propagates undamped at low frequencies and the shell motion for this wave type approaches that of a simple rod [9]. For the water filled shell, Figure 2.4, the dispersion curves are very similar to the \textit{in vacuo} shell. It can be observed that the dispersion curves remain almost the same for the axisymmetric $n=0$ torsional and extensional wave ($s=0$ and $s=2$) indicating that these waves remain unaffected by the fluid loading at low frequencies. This can be explained since these waves exist in the pipe wall i.e. they are predominantly structural. For the axisymmetric $n=0$ fluid wave ($s=1$) the increase in wave number with frequency is less compared to the \textit{in vacuo} case which is expected since the speed of sound in water is more than that in air. The fluid loading for this wave appears as a mass loading added on the inner side of the structure [14]. In the water filled pipe too there is only one propagating wave type for the $n=1$ bending mode and is very similar to the bending wave propagating through the \textit{in vacuo} shell.

2.4 Energy flow in fluid-filled pipes

The total axial energy flow $P_x$ can be obtained from the sum of the axial structure-borne energy flow $P_{xs}$ and the axial fluid-borne energy flow $P_{sf}$ [39],

$$P_x = P_{xs} + P_{sf} \quad (2.16)$$

Due to the symmetry of the cylindrical shell, wave propagation occurs in circumferential modes of various orders $n$. The shell mid-plane displacements $u$, $v$, $w$ and acoustic pressure $p$ can be expanded for each mode [1, 14, 35]. Thus each mode $n$ has its own modal amplitudes $u_n(x,t)$, $v_n(x,t)$, $w_n(x,t)$ and $p_n(x,r,t)$ and polarization angles $\theta_n(x,t)$ where $r$, $x$ and $\theta$ are cylindrical co-ordinates [39]. Hence the total energy flow, obtained by the summation of the individual modes can be written as,
2.4.1 Structure-borne energy flow

The axial vibrational energy flow in the shell structure can be described using the axial component of the structural intensity vector [14]. The instantaneous axial structural intensity \( I_{xs} \) may be expressed in terms of the classical resultant forces \( N_x, N_{x\theta} \) and shell mid-plane displacements \( u, v, w \) (Figure 2.2), where the dot denotes a time derivative [39],

\[
I_{xs} = -N_x \dot{u} - N_{x\theta} \dot{u}
\]  

(2.18)

Using the Donnell-Mushtary shell equations [1], the intensity can be written in terms of shell mid-plane displacements:

\[
N_x = D \left( \frac{\partial \nu}{\partial x} + \frac{v}{a} \frac{\partial \nu}{\partial \theta} + \frac{w}{a} \right), \quad N_{x\theta} = \frac{1 - \nu}{2} D \left( \frac{\partial \nu}{\partial x} + \frac{1}{a} \frac{\partial u}{\partial \theta} \right)
\]  

(2.19a,b)

where \( D = \frac{E \: h}{(1 - \nu^2)} \) is the membrane stiffness of the shell. The total axial structure-borne energy flow in a time-averaged sense is obtained from integration of the instantaneous shell intensity over the shell cross-section [39],

\[
P_{xs} = \frac{2\pi}{\langle I_{xs} \rangle, a} d\theta.
\]  

(2.20)

From equations (2.18), (2.19) and (2.20) the axial structural energy flow can be expressed as
\[ P_{ss} = \frac{2\pi}{a} \left\{ -D \left( \frac{\partial u}{\partial x} + \frac{v}{a} \frac{\partial u}{\partial \theta} + \frac{w}{a} \right) \hat{u} - \frac{1 - \nu}{2} D \left( \frac{\partial v}{\partial x} + \frac{1}{a} \frac{\partial v}{\partial \theta} \right) \right\} a d\theta \]  

(2.21)

where \( \langle \_\_ \rangle_t \) indicates a temporal averaging. Temporal averaging in the time domain has equivalent operations in the frequency domain [36]. Hence the Fourier transform of equation (2.21) gives the modal structure-borne energy flow in the frequency domain as [39],

\[
(P_{ss})_n = \frac{\varepsilon_n}{2\pi a} D \omega \left\{ \text{Im} \left( \frac{\partial u_n}{\partial x} \right)^* \cdot \vec{u}_n \right\} + \frac{1 - \nu}{2} \text{Im} \left[ \frac{\partial \sigma_n}{\partial x} \right] \cdot \vec{v}_n \right\} + \nu \text{Im}[\vec{w}_n \cdot \vec{u}_n] + \frac{1 + \nu n}{2} \text{Im}[\vec{\sigma}_n \cdot \vec{u}_n]
\]

(2.22)

where the factor \( \varepsilon_n \) equals 2 if \( n=0 \) and equals 1 if \( n>0 \), ‘\(-\)’ indicates a Fourier transform, * indicates a complex conjugate and \( \text{Im} \) denotes the imaginary part of a complex quantity. \( u_n \), \( v_n \), and \( w_n \) are the shell mid-plane displacements for the respective mode of order \( n \), \( D \) is shell membrane stiffness and \( \nu \) is the Poisson’s ratio of the shell material.

### 2.4.2 Fluid-borne energy flow

The axial energy flow in the fluid can be determined using the definition of acoustic intensity [14],

\[
P_{xf} = \frac{2\pi a}{4} \rho f D \iint_0^0 \langle p v_x \rangle r dr d\theta.
\]

(2.23)

where \( p \) and \( v_x \) are the acoustic fluid pressure and the acoustic axial fluid velocity. The Fourier transform of the fluid-borne energy flow can be written as,

\[
(P_{xf})_n = \frac{\varepsilon_n \pi a^2}{4 \rho f \omega} \left\{ J'_n(k_n a) \right\}^2 + \left\{ 1 - \left( \frac{n}{k_n a} \right)^2 \right\} J_n(k_n a)^2 \right\} \text{Im} \left[ \rho_n^* \frac{\partial \rho_n}{\partial x} \right]
\]

(2.24)
where the factor \( \varepsilon_n \) equals 2 if \( n=0 \) and equals 1 if \( n>0 \). \( k_n^r \) is the radial wave-number and \( J_n' \) denotes the derivative of the Bessel function \( J_n \) with respect to its argument. \( p_n \) is the acoustic pressure associated with the circumferential mode of order \( n \). The acoustic pressure \( p \) in the fluid can be expressed in terms of the shell wall radial displacement \( w \) as [39],

\[
\overline{p}_n = \frac{\rho \omega^2}{k_n^r J_n'(k_n^r a)}
\]  

(2.25)

Equation 2.25 indicates that the fluid pressure and hence the fluid-borne energy flow can be determined from measurements of the radial shell wall displacements [39].

### 2.4.3 Relationships between wave amplitudes

The displacement amplitudes \( U, V, W \) and the pressure amplitudes \( P \) stand in a firm relationship for any pair, which results from the coupled equations of pipe motion. For low values of frequency \( \Omega \), the displacement relationships simplify to [31]

\[
\begin{align*}
U_{l\pm} &= \Gamma_1 \Omega W_{l\pm}, & U_{a\pm} &= \Gamma_a \Omega W_{a\pm}, & U_{b\pm} &= \pm \xi_b \sqrt{\Omega} W_{b\pm}, & V_{b\pm} &= -W_{b\pm}, \\
\end{align*}
\]  

(2.26a-d)

where \( \Gamma = \frac{\nu \xi}{(\xi^2 - 1)} \).

From equations (2.26) it is shown that the amplitudes of axial wave components of the longitudinal mode, \( U_{l\pm} \), have to be much larger than the amplitudes of the corresponding radial components \( W_{l\pm} \), since \( \Omega << 1 \). The opposite applies to the bending mode where radial (and thus tangential) motions dominate. This implies that measurements of longitudinal and bending modes can be achieved by detecting the axial and radial motions respectively [31].
2.4.4 Approximate equations for energy flow per wave type

The equations (2.22), (2.24) and (2.25) provide the theoretical basis for the energy flow measurements [39]. If the circumferential modes $n$ can be separated experimentally, then the structure-borne energy flow per mode $n$ can be determined from cross-spectral measurements of the shell displacements [39]. The spatial derivatives that occur in equation (2.22) can be determined by a finite difference method as was used by Verheij [36]. For example,

$$\omega \text{Im} \left[ \left( \frac{\partial \tilde{u}_n}{\partial x} \right)^* \cdot \tilde{u}_n \right] = \frac{1}{\omega^2 \Delta x} \text{Im} \left[ S(\ddot{u}_n(x_1), \ddot{u}_n(x_2)) \right]$$

(2.27)

where $S$ denotes the cross-spectrum between the modal axial accelerations at the axial positions along the pipe surface, $x_I$ and $x_2$ with $\Delta x = x_2 - x_I$ [39].

The amplitude relationships in equation (2.26) and the finite difference approximation shown in equation (2.27) can be applied to the energy flow equations (2.22), (2.24) and (2.25) to determine simplified equations for the structure-borne and fluid-borne energy flow per wave type. These energy flow equations for the longitudinal, acoustic, torsional and bending waves can be expressed in terms of single cross-spectral measurements as [39],

$$ (\overline{P}_{sx})_L = \frac{1}{\zeta^2} \frac{\pi a D}{\omega^2 \Delta x} \text{Im} \left[ S(\ddot{u}_0(x_1), \ddot{u}_0(x_2)) \right], \quad (2.28a) $$

$$ (\overline{P}_{sf})_L = \frac{2}{(\xi_F^2 - \Psi)^2} \frac{\pi a^2 \rho_F c_T^2}{\omega^2 \Delta x} \text{Im} \left[ S(\ddot{w}_0(x_1), \ddot{w}_0(x_2)) \right], \quad (2.28b) $$

$$ (\overline{P}_{sv})_F = \frac{1-v}{2} \frac{\pi a D}{\omega^2 \Delta x} \text{Im} \left[ S(\ddot{v}_0(x_1), \ddot{v}_0(x_2)) \right], \quad (2.28c) $$
\[
(P_{ss})_B = \pi \alpha h E_y (1 - k_B^2) \frac{k_B^2}{\omega^2} \Im[S(\ddot{w}_i(x_1), \ddot{w}_i(x_2))].
\]  

(2.28d)

where $\zeta_s$ and $\zeta_F$ are wave number factors used in equation (2.15), $\Psi = \left(\frac{c_L}{c_F}\right)^2$ and $k_B$ is the bending wave number given by equation (2.15). $S$ denotes the cross spectral density between the accelerations at axial locations $x_1$ and $x_2$ as shown in Figure 2.5 and $\Im$ is the imaginary part of the cross spectral density. Figure 2.5 shows an arbitrary fluid-filled piping system excited by a positive displacement pump. $x_1$ and $x_2$ are axial locations along a straight section of the pipe where the accelerometers are mounted to record the cross spectral density for circumferential modes $n=0$ and $n=1$. The distance between $x_1$ and $x_2$ (i.e. $\Delta x$), should be no larger than one quarter of a wavelength of the smallest axial wavelength considered [39].

![Fluid filled piping system](image)

Figure 2.5: Fluid filled piping system with accelerometer mounted at axial locations $x_1$ and $x_2$ for measuring energy flow through the pipe

The techniques for decomposing the circumferential modes and obtaining the modal accelerations for each wave type will be discussed in detail in the next section. Once the modal accelerations are obtained the cross-spectral densities between the appropriate modal accelerations can be evaluated and substituted in the corresponding energy flow equations (2.28). For example, the cross-spectral density between the modal
accelerations at \( x_1 \) and \( x_2 \) due to the \( n=0 \) torsional mode can be substituted in equation (2.28c) to obtain the energy carried by the \( n=0 \) torsional wave. Thus the energy carried by each wave type can be calculated [39].

2.5 Experimental method for circumferential mode decomposition

For frequencies below the cut-on frequency of the \( n=2 \) waves the circumferential modes can be easily separated due to the simple \( \theta \) dependence of the shell displacements [39]. For example,

\[
\ddot{u}(x,t) = \ddot{u}_0(x,t) + \ddot{u}_1(x,t)\cos[\theta + \theta_1(x,t)]
\]

(2.29)

where \( \ddot{u} \) denotes the total axial acceleration along the pipe surface while \( \ddot{u}_0 \) and \( \ddot{u}_1 \) are the axial accelerations on the pipe surface due to the \( n=0 \) axisymmetric and \( n=1 \) bending mode respectively and \( \theta_1 \) is the orientation with respect to \( \theta \). Equation 2.29 shows that the total axial acceleration on the pipe wall surface can be determined from the axial accelerations due to the \( n=0 \) axisymmetric and \( n=1 \) bending modes and the circumferential angle \( \theta \). The total tangential and radial accelerations on the pipe wall surface can be determined similarly. Thus two amplitudes in equation (2.29) i.e. \( \ddot{u}_0 \) and \( \ddot{u}_1 \) and one phase angle \( \theta_1 \) have to be determined from measurements. In practice the modal accelerations and polarization angles are determined according to Figure 2.5. Figure 2.6 shows the actual experimental setup for circumferential mode decomposition. As shown in Figure 2.6 four accelerometers at \( 90^0 \) angles are used for each \( n=0 \) axisymmetric wave type i.e. the torsional, extensional and fluid waves and the \( n=1 \)
Figure 2.6: Accelerometer configurations for circumferential mode decomposition

Figure 2.7: Accelerometers mounted on brass studs at axial locations $x_1$ and $x_2$ to measure the pipe wall accelerations in the axial $u$ direction
bending mode. The accelerometers are mounted on the brass studs as shown in Figure 2.7 such that they can record the pipe wall displacements in the axial u, tangential v and radial w directions. Four accelerometers are mounted at each cross section \( x_1 \) and \( x_2 \) to record the accelerations in either the axial u, tangential v or radial w directions. The four accelerometer locations at the cross section \( x_1 \) denoted by points A, B, C, and D are 90\(^\circ\) apart as shown in Figure 2.6. The following equations are then used to determine the shell modal accelerations in the axial u, tangential v and radial w directions for the \( n=0 \) axisymmetric and the \( n=1 \) bending mode [39],

\[
\ddot{u}_0 = \frac{1}{4}(\ddot{u}_A + \ddot{u}_B + \ddot{u}_C + \ddot{u}_D), \quad (2.30a)
\]

\[
\ddot{v}_0 = \frac{1}{4}(\ddot{v}_A + \ddot{v}_B + \ddot{v}_C + \ddot{v}_D), \quad (2.30b)
\]

\[
\ddot{w}_0 = \frac{1}{4}(\ddot{w}_A + \ddot{w}_B + \ddot{w}_C + \ddot{w}_D), \quad (2.30c)
\]

\[
\ddot{w}_1 \cos(\theta_1) = \frac{1}{2}(\ddot{w}_A - \ddot{w}_C), \quad (2.30d)
\]

\[
\ddot{w}_1 \sin(\theta_1) = \frac{1}{2}(\ddot{w}_B - \ddot{w}_D). \quad (2.30e)
\]

where subscripts 0 and 1 indicate the respective circumferential mode while subscripts A, B, C and D denote the accelerations at locations A, B, C and D on the pipe circumference as shown in Figure 2.6. Thus the accelerations at axial locations \( x_1 \) and \( x_2 \) due to each wave type can be determined from equations (2.30a-e). The cross-spectral density between the accelerations at locations, \( x_1 \) and \( x_2 \) are evaluated for each wave type using Matlab. These cross spectral densities are then substituted in the appropriate energy flow.
equations (2.28a-d) to determine the energy carried by individual wave types through the fluid-filled pipe.
Chapter 3

Experimental Setup

As stated in Chapter 1, the main goal of this work was to develop a device for reducing fluid-borne noise, which could be practically implemented on an actual piping system onboard a Swedish marine vessel. Basically the piping system on board the ship is excited due to the pressure fluctuations generated by a reciprocating piston pump. Thus the excitation source is fluid based. Hence a purely experimental study was carried out in order to test various methodologies for fluid pressure pulsation and vibration reduction in fluid filled piping systems excited by an acoustic source. Three approaches were tried for achieving the goal of piping system noise control. These three techniques were passive control, active control and active/passive control. These approaches and their results are described in detail in Chapters 4 and 5. In order to test these approaches it was necessary to build an experimental rig. In this chapter the experimental rig used for testing the different approaches for reduction of fluid borne noise and vibrations in fluid filled piping systems is described in detail. The chapter begins with a description of the basic experimental setup. The different anechoic terminations built and tested are described and their performance compared. One of the biggest problems in terms of achieving desired results was the presence of structural waves whose effects will be discussed in Chapter 4. Hence it was very important to reduce if not eliminate the dominant presence of structural
waves in the experimental rig. The different reasons for the generation of structural waves and the methods adopted to tackle them are discussed in this chapter. The chapter finally deals with the characterization of the experimental rig by determining its response for broadband excitation (0-1000 Hz). The rig characterization is very important because it determines how the rig behaves, what kind of wave types are propagating through it and how much power is carried by each of them. Characterizing the rig behaviour is crucial to quantifying the effects of various noise and vibration control devices.

3.1 Arrangement of the experimental rig

This section describes the experimental rig used for testing the different approaches for reducing fluid-borne noise and vibrations in fluid filled piping system. Figure 3.1 shows a schematic diagram of the experimental setup, which will be referred to as the test rig from now on. Figures 3.3 and 3.4 are pictures of the actual experimental rig and the test section whose function is explained in the next paragraph.

The cylindrical pipe used for building the experimental rig was 2" inner diameter, standard steel pipe. For more information on the pipe material and properties the reader is referred to Table 2.1 in Chapter 2. As can be seen in the schematic diagram Figure 3.1 the entire pipe rig could be divided into four sections. The first is the upstream section, which was a 2" standard steel pipe, 1.67 m long. The upstream section was connected to the excitation source, which was a 50 lb. Ling Dynamics shaker. The upstream section had a circular brass plate, 0.05" thick, attached to its excitation end by steel flanges as shown in Figure 3.2. This brass plate which served as a speaker diaphragm was connected to the excitation shaker by a stinger. The second section, which followed the upstream section,
Figure 3.1: Schematic diagram of experimental test rig

- **Shaker Test Section (steel)**
- **Hydrophone (mobile)**
- **Downstream section (steel)**
- **Upstream section (steel)**
- **Pressure Sensors**
- **Removable steel plug**
- **Steel flange**
- **Anechoic termination (plastic)**
- **Inflatable rubber tube**
- **Water**
Figure 3.2: Schematic diagram of the shaker, stinger and diaphragm configuration used for excitation of the test rig.

- Circular brass plate used as diaphragm (2.15” diameter and 0.05” thick)
- Fluid-filled pipe
- Shaker
- Stinger
Figure 3.3: Photograph of basic experimental rig

Figure 3.4: Photograph showing the test section mounted on the experimental rig
was the test section which was a 2” steel ‘T’ approximately 0.45 m long and fitted with a removable steel plug as shown in Figure 3.4. This removable steel plug, could be replaced by the fluid borne noise and vibration-reducing device which was to be tested. The main purpose behind such a re-configurable test section is to compare the response of the test rig with the test device in the test section and with the steel plug in the test section. This would enable one to estimate the effect of the test device on the rig response. The next section is the downstream section, which incorporated a standard 2” inner diameter steel pipe, 2.13 m long. It was desired to minimize the reflection of fluid waves at the downstream end of the test rig by building an anechoic termination. This was necessary to prevent the reflection of fluid waves leading to the formation of standing waves inside the steel pipe. The presence of standing waves inside the steel pipe could dominate the fluid response thus contaminating the experimental results. Hence the fourth and final section is an anechoic termination partly shown in Figure 3.1 (not shown in Figure 3.3 and Figure 3.4). Two separate anechoic terminations were built and tested on the experimental test rig. These tests are described in detail in the next section. Both terminations were standard 2” inner diameter pipe.

Ball valves were connected to the shaker end of the upstream section and the anechoic end of the downstream section to drain water from the rig. Water was fed to the test rig from the anechoic end and drained out from both the ball valves on the upstream and downstream sections. The junction at the downstream section and the anechoic termination was kept at a higher elevation compared to the shaker end of the upstream section. This was done to remove air bubbles trapped inside the test rig while filling it with water, through the ball valve on the downstream section as it was at a higher level.
compared to the ball valve on the upstream section. The acoustic pressure inside the fluid was monitored using a B&K hydrophone (type 8103) mounted inside the fluid-filled pipe through a hollow standard 3/8” steel rod, 2.38m long. The rod was supported at the hydrophone end by a small brass bracket and guided out of the rig by a hollow brass plug on the anechoic termination. The hydrophone could be traversed inside the pipe by manually moving the portion of the steel rod jutting outside the anechoic end. The steel rod supporting the hydrophone was marked in inches to determine the exact location of the hydrophone inside the steel pipe. PCB pressure sensors (type S112A22) were mounted on the upstream and downstream sections at the locations shown in Figure 3.1 to measure the fluid pressure inside the test rig. The PCB sensors measured the fluid pressure at the pipe wall and fluid interface while the hydrophone measured fluid pressure at the center of the pipe. Thus the hydrophone and PCB sensor reading could be compared to ensure if the fluid pressure was uniform throughout the pipe cross section. It should be noted that the PCB pressure sensors could be located at other positions too. The surface vibrations along the pipe wall were also measured using PCB accelerometers.

The upstream and downstream portions of the rig were supported by inflatable rubber tubes on wooden stands, to isolate the system from possible vibrations transmitted along the floor. The rubber tube on the downstream end of the test rig was inflated to a bigger size compared to the rubber tube on the upstream end to elevate the downstream end of the rig to a higher level and facilitate the removal of air bubbles from the test rig.

3.1.1 Anechoic Termination

It was necessary to minimize if not eliminate the fluid wave reflections at the downstream section of the rig for reasons mentioned earlier. This could be achieved by
Figure 3.5: White plastic anechoic end referred to as 'termination 1'.

Figure 3.6: Transparent plastic end referred to as 'termination 2'.
building an anechoic termination and connecting it to the downstream section of the test rig. Two anechoic terminations were built and tested in the course of this project. Both were standard 2" inner diameter pipes. One of them was made of hard white plastic and will be referred to as termination 1 from now on. The second anechoic end was made from compliant, transparent plastic and will be referred to as termination 2. Figures 3.5 and 3.6 show pictures of both terminations. Termination 1 was 50 feet long while termination 2 was approximately 35 feet long. Water was filled in the experimental rig through the water supply hose connected to the end of terminations 1 and 2. As shown in Figures 3.5 and 3.6 the ends of both, termination 1 and 2, were elevated to a much higher level compared to the experimental rig. Water level inside terminations 1 and 2 were maintained at a higher elevation compared to the experimental rig. This ensured that the experimental rig was always filled with water. The concept behind the working of the anechoic terminations 1 and 2 was that a travelling fluid wave would lose most of its energy by the time it had reached the end of the termination (either 1 or 2) thus causing a very small part of it to be reflected.

It was necessary to determine which of the two terminations was working better i.e. had lower reflectivity. The two-microphone technique developed by Chung and Blaser [70] was used to measure the reflectivity for both terminations 1 and 2. According to this technique for an ideal anechoic end i.e. with no reflections, the transfer function between two PCB pressure sensors mounted on the pipe walls of the downstream section at a known distance should be unity, or the logarithm of the magnitude of the transfer function should be zero. Furthermore the phase of the transfer function between the two microphones gives the fluid wave propagation speed inside the pipe. Using this
technique, experiments were carried out to determine which anechoic termination worked more efficiently so that it could be used for the rest of the experiments. Figure 3.7 shows the schematic diagram of the experimental setup for this experiment. PCB pressure sensors s1 and s2 were mounted on the upstream section while s3 and s4 were mounted on the downstream section. The sensors were mounted at a known distance as shown in Figure 3.7. It is to be noted that the test section wasn’t used in this experiment. The test rig was filled with water, care being taken to remove air bubbles. A 0-1000 Hz white noise signal was supplied to the electromagnetic shaker, which drove the brass plate connected to the upstream rig section. Thus the brass diaphragm acted as the acoustic source. Transfer functions were recorded between pressure sensors s1 and s2 for the upstream section, and s3 and s4 for the downstream section. Figure 3.8 shows results of these experiments. The logarithms of the transfer functions between pressure sensors s3 and s4 mounted on the downstream pipe section are plotted against frequency. The phase of the transfer function is also plotted against frequency. From the plots it can be seen that for termination 2, the logarithm of transfer function is closer to 0 for the frequency range 0-1000 Hz compared to termination 1. Thus termination 2 was working better as an anechoic end as it had lower reflectivity than termination 1. Hence termination 2 (made from clear, compliant plastic 2” standard pipe diameter) was used as the anechoic termination for all the experiments.

3.1.2 Isolating structural vibrations

In most fluid filled piping systems the fluid is transferred from one point to another by centrifugal or reciprocating pump. As already discussed in Chapter 1 the pump acts like an acoustic source generating pressure pulsations in the fluid at harmonics
Figure 3.7: Schematic diagram for anechoic ends comparison test

Figure 3.8: Transfer function and phase between $S_3$ and $S_4$ for both terminations
corresponding to its operating frequency. Thus the excitation source is fluid based. It was desired to create an experimental rig similar to such a piping system excited by an acoustic source with only fluid waves propagating through the rig. But this was difficult to achieve in practice due to the reasons shown in Figure 3.9 and Figure 3.10. From Figure 3.9 it can be seen that the line of action for the excitation source is not exactly aligned with the pipe central axis. This is very similar to the shaker-stinger system connected to the rig. It is very difficult if not impossible to align the shaker and stinger exactly with the central axis of the upstream pipe section. This leads to excitation of the structural bending waves along with the n=0 fluid wave. Figure 3.9 illustrates how a 90° bend in a piping system, results in increased coupling between an incoming fluid wave and the pipe wall resulting in an outgoing fluid and structural bending wave. There was a similar 90° elbow at the junction of the downstream section and the anechoic end. This led to coupling between the fluid and pipe wall. Thus a purely fluid wave (n=0) coupled with the pipe-wall, causing energy exchange between the fluid and the pipe wall. This lead to generation of the n=1 structural waves which propagated downstream along with the n=0 fluid wave. Thus these n=1 structural waves traveled from the upstream section of the test rig to the downstream section. For broad band excitation it was observed that structural waves dominated the response of the test rig. As a result, the output of the sensors, mounted on the rig to monitor the fluid response, were contaminated by the structural waves i.e. the sensors were recording the rig response due to structural waves rather than the fluid wave. Thus the desired performance of a fluid wave controlling device (e.g. a quarter wavelength tube), could not be obtained since the sensors
Figure 3.9: Asymmetric forcing leading to generation of structural waves.

Figure 3.10: Structural discontinuity in system leading to generation of bending waves.
monitoring fluid behaviour were affected by the structural waves. Experiments showing how the dominance of structural waves affects the experimental results for fluid wave canceling devices are described in Chapter 4.

Hence it was important to reduce if not eliminate the structural vibrations. Figures 3.11 and 3.12 show the schematic diagram of the components, a diaphragm actuator and a passive insert respectively, developed to eliminate structural bending waves due to asymmetric forcing and discontinuities in the piping system. Figures 3.13 and 3.14 show the actual pictures of these devices. Figure 3.11 shows a diaphragm actuator, which reduces generation of structural bending waves due to asymmetric forcing by isolating the excitation source from the pipe walls. It consists of a Class 3C pre-convoluted rolling diaphragm (courtesy BELLOFRAM). The BRD (Bellofram Rolling Diaphragm) diaphragm material is essentially a layer of specially woven fabric, impregnated with a thin layer of elastomer, which makes it flexible and help isolate the excitation from the pipe wall. The diaphragm is 0.03” thick. The fabric lends high tensile strength to the diaphragm. For more information on the diaphragm properties the reader is referred to the Bellofram Diaphragm Design Manual [70]. Two cylindrical aluminium discs were mounted on either side of the diaphragm and screwed together. The diaphragm actuator was then mounted between the two flanges on the excitation end. The stinger was screwed on to the center of one of the aluminium discs. The flexible diaphragm thus, isolates the excitation source i.e. the shaker, from the structure and prevents the generation of structural bending waves in spite of asymmetric forcing.

The other component developed was the passive insert used to reduce transmission of structural vibrations. It consists of a fabric reinforced rubber hose
Figure 3.11: Diaphragm actuator to prevent generation of structural bending waves due to asymmetric forcing.

Figure 3.12: Passive insert in piping system to absorb $n=1$ bending waves.
Figure 3.13: Picture of diaphragm actuator (prevents bending wave generation due to asymmetric forcing)

Figure 3.14: Passive isolating section and rubber pads (inserted between steel flanges)
clamped on two 2” standard steel pipes, 6” long, fitted with flanges at both ends. The rubber section inserted between the steel pipes represents a change in the structural impedance. This change in material leads to poor wave coupling induced by a change in extensional phase velocities of both materials [9]. This leads to reflection of the incident structural wave at the steel-rubber interface. Fuller [9] showed that the maximum reflection of the incident wave occurred between the ring frequencies of the materials used. The passive insert is bolted on to the piping system with rubber pads inserted between the steel flanges. The rubber pads were inserted to prevent transmission of structural waves between the steel flanges. Figure 3.15 shows the setup for the experiments carried out to compare the response of the rig after inserting the structural dampers discussed above. The shaker was excited by white noise (0-1000 Hz). Transfer functions between the input signal to the shaker and the hydrophone mounted inside the water filled pipe were recorded for the setup with and without the structural wave dampers. Figure 3.16 is a plot showing transfer functions between the excitation signal and the hydrophone for the test rig with and without the structural wave dampers i.e. the diaphragm actuator, the passive insert and the rubber pads. It can be clearly seen that the transfer function plot for the rig with structural dampers is less resonant compared to the rig without dampers. Thus the use of structural dampers resulted in less resonant structural behaviour thus implying damping or reflection of structural waves towards the source. Reduction of the structural wave presence was necessary to obtain desired experimental results as will be shown in Chapter 4.
Figure 3.15: Experimental setup for observing reduction in structural vibrations

Figure 3.16: Comparision of transfer functions for test rig with and without structural wave dampers installed
3.2 Characterization of experimental rig

It was necessary to characterize the rig behaviour in order to understand what kinds of waves were propagating in the system. It should be noted that the characterization of the rig was very difficult because of many factors. One of the biggest problems in understanding the rig behaviour was the presence of air bubbles inside the pipe. The effect of bubbly mixtures on wave propagation has been discussed by Silberman [33]. He found that presence of air bubbles in a standing wave tube filled with water attenuated the propagation of sound waves inside the tube. Velocity measurements could not be obtained near the natural frequency of the air bubbles because of high attenuation of standing waves inside the water filled tube. Hence sufficient care was taken to bleed the air out of the system by elevating one end of the rig to a higher level compared to the other end. However air bubbles were still present. Ordinary water has soluble oxygen in it, which separates itself from water after a short period of time. Thus despite efforts to make the rig free of air bubbles, air could always be found trapped in the system. The problem with air bubbles inside a fluid is that it changes the fluid impedance. Furthermore the behaviour of bubbles cannot be predicted because they are of different shapes and sizes. Hence the rig response to the same excitation forces was found to differ in experiments performed with an interval of time between them. The effect of structural waves on the rig characterization experiments could not be ruled out. There was also the problem of rust forming inside the steel pipes, which could affect experimental results. Thus characterization of the rig was a very tough problem. These problems notwithstanding, a series of experiments were carried out in an attempt to get a rough idea of the rig behaviour.
3.2.1 Experiments to determine rig behaviour

Figure 3.17 shows the experimental setup used in the experiments for characterization of the rig. The electromagnetic shaker was used as the excitation source and a 0-1000 Hz white noise signal was applied to it. As shown in the Figure, the experimental rig consisted of three sections. The first, the upstream section, was the passive insert used for reducing the transmission of structural waves through the experimental rig. The second section, the downstream section, was the standard 2” steel pipe used for the anechoic termination tests. The rubber diaphragm actuator was fixed to the shaker end of the upstream section. The other end of the upstream section was bolted to the downstream section with rubber pads inserted between the connecting flanges (to reduce structural wave transmission through the steel flanges). The anechoic termination described in the earlier section was the third section and was connected to the end of the downstream section. Two PCB pressure sensors, S1 and S2 were mounted on the upstream section while four more, S3 to S6, were mounted on the downstream section. The pressure sensors on pipe 2 i.e. S3, S4 and S5, S6 were mounted on opposite sides as shown in Figure 3.18 to determine if the pressure sensors were affected by the n=1 structural bending waves. This could be verified by averaging the signals of the sensors mounted opposite each other (e.g. S3 and S4) and comparing it to either one of them (i.e. S3 (or S4). If there is no bending wave propagating through the rig then the signals will be similar. However, if n=1 bending waves do propagate through the system then the signals will not be the same. Figure 3.18 shows a schematic diagram of the pressure sensor setup to measure the effect of the n=1 structural bending wave. S3 and S4 are the
Figure 3.17: Experimental setup for rig characterization tests

- Shaker
- Rubber pads
- Anechoic termination
- Inflatable rubber tube

Accelerometer locations:
- S1
- S2
- S3
- S4
- S5
- S6
- X1
- X2
- X1'
- X2'
pressure sensors mounted opposite each other. According to the method already
discussed the average of the sensor signals i.e \((s3+s4)/2\) is compared to either s3 or s4.
Figure 3.19 shows the results of the experiments to determine if the pressure sensor
readings were affected by the \(n=1\) structural waves. As can be seen from the plots the
average of the sum of the pressure sensor readings of s3 and s4 were very similar to that
of s3. From these plots it could be concluded that the pressure sensor readings were
unaffected by the \(n=1\) structural bending waves.

A B&K hydrophone, type 8103, was supported inside the downstream section by
a steel rod and a brass bracket. The hydrophone was mounted at exactly the same cross-
section as sensors s3 and s4 in order to compare their outputs. Cross-sections X1, X2 and
X1’, X2’ are the accelerometer locations for the circumferential mode decomposition and
power flow measurement method discussed in Chapter 2. Figure 3.20 and Figure 3.21
show four brass studs screwed on the circumference of the steel pipes at each of these
cross sections at an interval of 90°. As shown in the pictures in Figure 3.20 and Figure
3.21 the studs were cuboid shaped with faces perpendicular to the axial (u), tangential (v)
and radial (w) directions. Accelerometers were mounted on the faces of the studs
perpendicular to the axial (u), tangential (v), and radial (w) directions, to record the
accelerations in these directions as shown in Figure 3.20. The accelerometer signals in
the axial (u), tangential (v) and radial (w) directions were recorded at sections X1, X2 on
the upstream section and at X1’, X2’ on the downstream section. Table 3.1 gives the
exact location of the accelerometers on the experimental rig. From the table it can be seen
that the distance between X1 and X2 as well as X1’ and X2’ was 0.15m. The PCB
pressure sensor and hydrophone signals were also recorded. A PCB signal conditioner
Figure 3.18: Experimental setup of pressure sensors to detect presence of \( n=1 \) structural bending waves

Figure 3.19: Plots showing that the pressure sensor readings were not affected by structural waves
Figure 3.20: Accelerometer configuration for circumferential decomposition

Figure 3.21: Accelerometers mounted on downstream section at X1’ and X2’ to measure acceleration in the radial (w) direction
Table 3.1: Distance of the sections for accelerometer locations $X_1$, $X_2$, $X_1'$ and $X_2'$ from the shaker end of the experimental rig

<table>
<thead>
<tr>
<th>Distance of accelerometer locations from shaker end</th>
<th>$X_1$ (m)</th>
<th>$X_2$ (m)</th>
<th>$X_1'$ (m)</th>
<th>$X_2'$ (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.96</td>
<td>1.11</td>
<td>2.2</td>
<td>2.35</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: Distance of the sections for accelerometer locations $X_1$, $X_2$, $X_1'$ and $X_2'$ from the shaker end of the experimental rig
amplified the accelerometer and pressure sensor signals while a B&K signal conditioner amplified the hydrophone signal. The accelerometer, pressure sensor and hydrophone signals were recorded as frequency spectrum and time signals, on a 16-channel Data Acquisition System implemented using Labview. The accelerometer signals were summed together according to the equations 2.30a - 2.30c in Chapter 2 to give the accelerations for the corresponding circumferential modes. Cross-spectrums of the accelerometer signals between sections X1 and X2 and between sections X1’ and X2’ for each individual wave types (i.e. n=0 fluid, extensional and torsional and n=1 bending) was evaluated. These cross spectrums were substituted in the equations 2.28a – 2.28d for power flow in Chapter 2 to determine the power carried by the individual wave types in both, the upstream and downstream sections. The next section discusses the experimental results of wave decomposition and power flow measurements.

3.2.2 Experimental results and discussion

A 0-1000 Hz random noise signal supplied to the electromagnetic shaker excited the experimental rig. The accelerometer, pressure sensor and hydrophone signals were recorded using the data acquisition system. The sampling frequency was 4000 Hz (four times the maximum frequency of interest of 1000 Hz) to prevent anti-aliasing. For frequencies much below the ring frequency of the pipe only four types of waves exist and propagate energy [31,42]. The first three types are axi-symmetric n=0 fluid, longitudinal and torsional waves. The fourth wave type is the n=1 bending wave. The ring frequency for the 2” standard steel pipe that was used in the experiment is approximately 40 Khz as derived in Chapter 2. The experiments were carried out in a range of 0-1000 Hz, which is very low compared to the ring frequency. Hence only the n=0 axi-symmetric and n=1
bending waves were considered in this analysis. The circumferential decomposition and power flow measurement method described by de Jong and Verheij [39] has been used in this analysis. This method has been briefly reviewed in Chapter 2. The accelerometer signals in the axial (u), tangential (v) and radial (w) directions at all the four studs were summed for the respective n=0 axisymmetric modes (since the pipe wall undergoes a breathing mode and all accelerations have the same phase in one circumferential plane). The difference between the accelerometer signals was considered for the n=1 bending wave (since the phase of the accelerations are in the opposite direction for diagonally opposed location).

The results of the circumferential mode decomposition are shown in Figures 3.22 to 3.26. All the figures were plotted for the accelerometers on the downstream section. Figure 3.22 shows the average absolute acceleration of the pipe wall for the n=0 fluid wave. As seen from Figure 3.22 the accelerometer response revealed a resonance at 135 Hz. This meant that at this frequency there was an axial acoustic resonance inside the test rig. The axial length of the water column in the test rig was approximately 12m. The speed of sound in water is 1500 m/s i.e. 1500/12 = 125Hz, which was very close to the resonant frequency at 135 Hz. It could be concluded that there was an acoustic axial standing wave inside the test rig at 135 Hz. Hence even though the experimental rig was excited using a broad band disturbance there was a dominant fluid response at 135 Hz (i.e. the resonance at 135 Hz was an acoustic mode). Figure 3.23 shows the pressure at sensors s1, s2, s4 and the hydrophone. From figure 3.23 it could be observed that the sensor S4 which was mounted at the same cross section as the hydrophone, recorded the same signal as the hydrophone. Thus the PCB pressure sensors were sensing the fluid
wave and were not affected by the structural waves. It was very important to ascertain this because the PCB sensors were used as error sensors in active control experiments for fluid pressure reduction discussed later in Chapter 5.

Another important observation was that despite low coupling between the fluid and pipe wall at low frequencies, the fluid wave at 135 Hz was sensed by the accelerometers. Thus from the results of the n=0 fluid wave decomposition, it could be concluded that a dominant n=0 acoustic mode existed inside the test rig at 135 Hz. Sharp spikes can be observed in the accelerometer signals in Figure 3.22. This was due to the electrical noise at 60 Hz and its harmonics. Significant efforts were made to filter out these signals using band pass filters with limited success. Possible reasons could have been unshielded cables, electrical noise generated by acquisition channels, etc. Results at these frequencies must be considered suspect. Next it was desired to decompose the structural n=0 axisymmetric waves i.e. the extensional and the torsional waves. Figure 3.24 shows the results of the extensional mode decomposition. From the figure, an axial resonance at approximately 853 Hz can be observed. This meant that an extensional shell wave was dominant in the pipe wall at 853 Hz i.e. at this frequency the response of the shell wall in the longitudinal direction was dominant compared to other frequencies. As a result, even though the test rig was excited by a broad band (0-1000 Hz) signal, a dominant response was recorded only at the extensional resonant frequency of 853 Hz. Thus it could be concluded that at 853 Hz there was an n=0 extensional mode in the pipe wall. It has been shown that shell modes are purely a function of shell geometry [72].

Figure 3.25 shows the results of the torsional mode decomposition. In this case the accelerometer signals revealed two dominant resonances at 174 Hz and 465 Hz. This
meant that the torsional shell wave was dominant in the pipe wall at 174 Hz and 465 Hz i.e. the response of the shell wall in the torsional direction at these frequencies was greater compared to other frequencies. Thus it could be concluded that at 174 Hz and 465 Hz n=0 torsional modes existed in the pipe wall which caused the resonance peaks in the accelerometer signals. As was stated before the shell modes are purely a function of shell geometry [72]. Figure 3.26 shows the modal decomposition results for the n=1 bending wave. The accelerometer signals show two distinct resonances at 181 Hz and its harmonic at 362 Hz. This meant that the n=1 bending wave was dominant on the test rig at 181 Hz and 362 Hz i.e. the out of plane displacement of the pipe wall at these frequencies was greater compared to other frequencies. From these observations it could be concluded that n=1 bending modes were present on the test rig at 181 Hz and 362 Hz which caused resonances in the accelerometer signals.

To summarize the modal decomposition results it can be said that the n=0 acoustic wave had an axial mode inside the test rig at 135 Hz. The structural n=0 axisymmetric waves i.e. the extensional and torsional shell waves had axial modes at 853 Hz and 174 Hz, 465 Hz respectively. The n=1 bending wave had axial modes that existed at 181 Hz and 362 Hz on the test rig. Thus it could be concluded that the modal response of the test rig has been decomposed into the dominant individual wave types that exist at low frequencies, i.e. the dynamic response of the rig was comprehensively characterized. Figures 3.27 to 3.30 show the contribution of the different wave types to the power flow (calculated using equations 2.28a – 2.28d in Chapter 2) through the rig. The frequencies have been marked for clarity to show the correspondence between the acceleration signals and the power carried at the same frequency. The sharp spikes in the power plot
Figure 3.22: Radial pipe wall acceleration, \( w \), for the \( n=0 \) fluid wave

Figure 3.23: Fluid pressure inside rig measured by pressure sensors and hydrophone
Figure 3.24: Longitudinal acceleration for the n=0 shell extensional wave

Pipe 2 wall acceleration for the extensional n=0 wave, basic rig

853 Hz

Extensional acceleration in m/s²

Pipe 2 wall acceleration for the n=0 torsional wave, basic rig

174 Hz

465 Hz

Torsional Acceleration in m/s²

Figure 3.25: Pipe wall acceleration for the n=0 torsional shell wave
Figure 3.26: Pipe wall acceleration for the n=1 bending wave, basic rig

Pipe 2 wall acceleration for the n=1 bending wave, basic rig

Bending Acceleration in m/s²

Frequency (Hz)

0 200 400 600 800 1000

0 0.5 1 1.5 2 2.5 3 3.5 4

181 Hz

362 Hz

Figure 3.26: Pipe wall acceleration for the n=1 bending wave
Figure 3.27: Power carried by the n=0 fluid wave through pipe 2

Figure 3.28: Power carried by the n=0 extensional shell wave through pipe 2
Figure 3.29: Power carried by the n=0 torsional shell wave through pipe 2

Figure 3.30: Power carried by the n=1 bending wave through pipe 2
are due to harmonics of the 60 Hz electrical signal discussed previously. As explained earlier efforts were made to remove the electrical noise by band passing the signals. But the electrical noise was still present which could be attributed to unshielded cables or electrical noise generated at acquisition channels. It can be seen from the figures that the maximum power is carried by the n=0 fluid. This result is expected because the excitation source in this experiment is fluid based. Thus for low frequency excitation major portion of the energy remains in the fluid which is similar to the results obtained by Fuller [14]. Thus the energy carried by the n=0 extensional and torsional waves, which are purely shell waves, is small compared to the energy carried by the n=0 fluid wave. The energy carried by the n=1 bending wave, which is not a purely fluid wave, can be explained by the phenomenon of fluid-structure coupling. This phenomenon causes energy exchange between the fluid and the shell as discussed in Chapter 2. The n=0 extensional and torsional shell waves are uncoupled from the fluid wave and hence they carry less energy compared to the n=1 bending wave which is coupled to the n=0 fluid wave.
Chapter 4

Experiments using passive control techniques

A primary source of environmental noise in industries is the fluid supply system. A majority of fluid supply systems employ positive displacement pumps (mostly piston pumps) which produce significant pressure fluctuation transmitted along the fluid in the pipe. Due to coupling between the fluid and the pipe wall, piping systems transmit unwanted energy in the form of structural and acoustical vibration. Discontinuities in the piping system reflect or absorb some of the incident wave thus reducing the levels of the transmitted wave. Thus discontinuities can be built into the system to reduce transmission of fluid and structure borne noise and vibration. These discontinuities can be in the form of an isolator section or a fluid pulsation canceling device (e.g. quarter wavelength tube or Helmholtz resonator) each of which will be described in detail in the next sections. All these discontinuities can be generally classified as passive devices. It can be recalled that the piping system onboard the Swedish marine vessel had a reciprocating piston pump as the prime mover [65]. The pump generated pressure pulsations inside the fluid at fundamental and harmonics of the operating frequency of the pump. The fundamental frequency and harmonics can be calculated by the relation, $f = \text{rpm} \times N \times n / 60$ where rpm is the speed of the pump, $N$ is the number of pump blades ($N = 1$ for a piston pump) and $n$ is harmonic order ($n = 1, 2, 3, \ldots$). For the pump described in Maillard’s report [65]
the fundamental frequency was 64 Hz. The fundamental frequency will be referred to as the pump operating frequency from now on. Due to fluid-structure coupling the fluid borne noise is transmitted to the pipe structure and vice versa causing noise problems in the passenger cabin of the ship [65]. Thus the passive devices discussed above could be incorporated into the piping system to reduce fluid and structure borne noise. It was necessary to test these devices on a laboratory rig before they could be implemented on the actual piping system on the ship described in reference [65]. Different passive devices were built and tested on the experimental rig described in Chapter 3. The aim of these experiments was to determine if the passive devices at their resonant frequency were effective at canceling the fluid pulsations inside the experimental rig. This chapter discusses the results of the experiments using passive devices and their significance.

4.1 Rubber isolating section

One of the most common methods to reduce vibration transmission in fluid filled piping systems is by introducing an impedance change in the piping system. The most common method of doing this is by introducing a rubber section into the piping system. Passive rubber sections have been used effectively to reduce the fluid pressure pulsations propagating through piping systems [65]. The effect of such rubber isolation sections on wave propagation has been studied in detail by Fuller [9]. He found that the transmission loss due to a change in material depends profoundly on the poor wave coupling induced by a change in extensional phase velocities of both the materials. It was found that the maximum transmission loss occurred between the ring frequencies of the different material piping systems (of the same diameter). Thus for transmission loss over a broad frequency band it is advisable to use materials for pipes such that the difference in ring
frequencies of the pipe (different material pipe of same diameter) is large [9]. Such a
detailed analysis is not attempted here. Experiments were carried out to determine the rig
response with and without the rubber isolator section described in Chapter 3. The rubber
isolator consists of a fabric reinforced rubber hose clamped on two 2” standard steel
pipes, 6” long, fitted with flanges at both ends. The rubber section inserted between the
steel pipes represents a change in the structural impedance. This change in material leads
to poor wave coupling induced by a change in extensional phase velocities of both
materials [9]. This leads to reflection of the incident structural wave at the steel-rubber
interface. The results of the rubber isolator experiments and their implications are
discussed in the next section.

4.1.1 Experiments using rubber isolator section

The rubber isolator section was built primarily to reduce the transmission of
structural waves through the experimental rig as was explained in Chapter 3. A picture of
the rubber isolator section has been shown in Chapter 3 in Figure 3.14 and hence has not
been repeated here. It was seen in Chapter 3 that the use of the rubber isolator on the
experimental rig made the rig less structurally resonant by creating a change in
impedance and thus reflecting the incident structural waves. This reduced the dominance
of the structural n=1 bending waves which were affecting the experimental results as will
be shown in the section on quarter wavelength tubes. It was necessary to ascertain more
accurately the effect of the rubber isolator on wave propagation in the rig (i.e. what wave
types were most affected by the rubber isolator). Experiments were carried out to
determine this.
Figure 4.1 shows a schematic diagram of the experimental setup used to test the rubber isolator. The experimental rig consisted of four main sections as can be seen in Figure 4.1. The first section, the upstream section, was a 2” standard steel pipe, 1.22m long. As shown in Figure 4.1 two PCB pressure sensors s1 and s2 were mounted on this section to monitor the fluid pressure. The second section was the rubber isolator section. The rubber isolator section consisted basically of a wire reinforced rubber tube with two 2” standard steel pipes, 6” long, clamped on both its ends. Flanges were screwed on both the steel pipes to bolt the rubber isolator between the upstream and downstream section (third section) of the experimental rig. The third section i.e. the downstream section, was a 2” standard steel pipe, 2.17m long. PCB pressure sensors s3, s4, s5 and s6 were mounted on the downstream section. Table 4.1 shows the exact location and angular positions of the pressure sensors (sensor s1 has been considered as the reference). Sensors s3, s4 and s5, s6 were mounted on opposite sides of each other to ascertain whether bending waves were affecting the pressure sensor readings. The clear plastic anechoic termination described in Chapter 3 was the fourth and final section. The rig was filled with water with the usual care being taken to bleed air out of the system. A 0-1000 Hz white noise signal supplied to the electro-magnetic shaker was used to excite the experimental rig. X1, X2 and X1’, X2’ were the accelerometer locations on the upstream and downstream sections respectively. Table 4.2 gives the exact locations of the accelerometers on the upstream and downstream sections. The accelerometers were mounted in the appropriate configurations as described in Chapter 2 and their time signals recorded using a TI digital data acquisition system. Cross spectra between the
<table>
<thead>
<tr>
<th>Distance from shaker end (m)</th>
<th>Pressure sensor, s1</th>
<th>Pressure sensor, s2</th>
<th>Pressure sensor, s3</th>
<th>Pressure sensor, s4</th>
<th>Pressure sensor, s5</th>
<th>Pressure sensor, s6</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.98</td>
<td>3.32</td>
<td>3.32</td>
<td>3.92</td>
<td>3.92</td>
<td></td>
</tr>
<tr>
<td>Angular position on the pipe (degrees)</td>
<td>0°</td>
<td>0°</td>
<td>0°</td>
<td>180°</td>
<td>0°</td>
<td>180°</td>
</tr>
</tbody>
</table>

Table 4.1: Table showing location of sensors on the experimental rig and their angular locations
Figure 4.1: Experimental setup for rubber isolator tests

- Shaker
- Steel flange
- Inflatable rubber tube
- Rubber isolation
- Accelerometer locations
- Anechoic termination
- Rubber insert
- S1, S2, S3, S4, S5, S6
- X1, X2, X1', X2'
accelerometers was evaluated and substituted in the equations 2.28a – 2.28d for power flow in Chapter 2 to determine the power carried by individual wave types in the upstream and downstream sections. The transmission loss for a particular incident wave (a) on the discontinuity, in this case the rubber isolator, is then evaluated according to the following formula given by Fuller[9],

\[
\text{Transmission loss (dB), } TL = 10\log_{10} \frac{\text{Power transmitted, wave (a)}}{\text{Incident power, wave (a)}}
\]  

(4.1)

Using equation (4.1) above the transmission losses were calculated for the experimental rig with and without the rubber isolator. Note that a negative TL implies a power reduction. Figures 4.2 to 4.5 show plots comparing transmission losses for the rig with and without the rubber isolator for the n=0 fluid, extensional, torsional and the n=1 bending wave respectively. From the figures it is clear that the effect of the rubber isolator is much more pronounced for the n=1 bending waves than the n=0 axisymmetric waves. This result shows that rubber isolators are effective in isolating the out of plane motion of the pipe rather than the in plane motion. Even among the n=0 axisymmetric waves the rubber isolator is seen to cause more transmission loss for the torsional and extensional waves compared to the purely fluid wave.

This can be explained by the fact that the torsional and extensional waves are predominantly shell waves. As a result the change in shell material from steel to rubber causes more reflection and absorption for these waves compared to the fluid wave which propagates only through the fluid. Rubber isolation will be good at controlling the n=1 structural waves but will be probably poor for the n=0 fluid wave, since this carried via the fluid path. The only attenuation of this wave will be due to absorption at the walls of the rubber section, which is likely to be low due to the long acoustic wavelength. The n=0
Table 4.2: Table showing the distances of the accelerometer location sections X1, X2, X1’, and X2’ from the shaker end

<table>
<thead>
<tr>
<th></th>
<th>Upstream section</th>
<th>Downstream section</th>
</tr>
</thead>
<tbody>
<tr>
<td>X1</td>
<td>0.5</td>
<td>3.42</td>
</tr>
<tr>
<td>X2</td>
<td>0.65</td>
<td>3.57</td>
</tr>
<tr>
<td>Distance of accelerometer</td>
<td></td>
<td></td>
</tr>
<tr>
<td>locations from shaker end (m)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
structural wave is attenuated less than the n=1 structural wave as its wavelength is much longer at the same frequency.

Thus rubber isolators can be successfully implemented for damping the waves propagating through the pipe wall or reducing structural vibrations. However they are ineffective for damping fluid pulsations. Rubber dampers also cause static pressure loss [65], which is a disadvantage in piping systems where static pressure has to be maintained at a constant value. Moreover in high pressure piping systems, rubber damper sections tend to stiffen losing their flexibility and their ability to damp transmission of structural vibration through the pipe wall [9].
Figure 4.2: Transmission loss for rig with and without damper, n=0 fluid wave

Figure 4.3: Transmission loss comparison, n=0 extensional wave
Figure 4.4: Transmission loss comparison, \( n=0 \) torsional wave

Figure 4.5: Transmission loss for rig with and without damper, \( n=1 \) bending wave
4.2 Quarter wavelength tube

A primary reason for environmental noise in industries is the fluid supply system. A major contributor to the fluid supply system noise is the hydraulic pumping system. A majority of these systems employ positive displacement pumps (e.g. piston pumps) for providing pressure for transporting the fluid from one point to another. These pumps also act as acoustic sources by generating pressure fluctuations, which are transmitted along the fluid path. Conventional rubber dampers are ineffective in reducing the fluid pressure pulsations as was shown in the last section. One method of reducing the fluid borne noise is by using multiple paths of unequal length for the acoustic waves within the piping system to partially cancel the pressure fluctuations. Quincke [48] first introduced the idea of unequal acoustic path lengths in a branched duct to cancel the propagating waves by destructive interference with the incoming acoustic wave. This approach is essentially a “reactive” effect as the pressure zero impedance change at the branched duct outlet causes total reflection of sound towards the source. The other passive devices discussed below (quarter wavelength tube and Helmholtz resonator) are also reactive passive devices.

A quarter wavelength tube is a device, which can be introduced into a piping system to cancel fluid pressure pulsations occurring at the resonant frequency of the device. Figure 4.6 shows a quarter wavelength tube in a piping system. The length of a quarter wavelength tube is chosen to be an odd integer multiple of one quarter wavelength of the acoustic disturbance to be controlled (e.g. if the acoustic disturbance has a wavelength of 10 meters then the quarter wavelength tube would be 2.5 meters long.
Figure 4.6: Schematic diagram showing principle of operation of a quarter wavelength tube

L = odd integer multiple of one quarter wavelength of incident acoustic wave
or odd integer multiples of 2.5 meters long). As shown in Figure 4.6 the incident acoustic
wave in the quarter wave tube is reflected off the end of the side branch tube and travels
back to the junction point, out of phase with the incident wave (since total path length in
the side branch is half a wavelength), thus canceling the incoming wave at the side
branch location. The zero impedance at the side branch location thus causes total
reflection of the incident sound towards the source and zero transmitted power. Note this
implies that a quarter wavelength tube will work well at a single design frequency. Thus
quarter wavelength tubes can be used to achieve high noise reduction in hydraulic
systems where the pressure pulsations are of a tonal nature and occur at the fundamental
and harmonics of the pump operating frequency. However a quarter wavelength tubes
effectiveness in producing a high attenuation of the acoustic disturbance is limited to
narrow frequency bands as shown in Figure 4.8. Thus if the excitation frequency is no
longer within the operating bandwidth of the quarter wavelength then the quarter tube
becomes ineffective. Another limitation of quarter wavelength tubes is that for
attenuation of low frequency acoustic waves their dimensions (i.e. length) become very
large and they are not very practical to implement.

Experimental investigation of quarter wavelength tubes in hydraulic systems has
been carried out by Dodson, et. al [42]. Three different designs of quarter wavelength
tubes were tested. It was found that quarter tubes could effectively reduce fluid-borne
noise. A very important feature of these experiments was that flexible rubber hose was
employed for the piping system used for these experiments. This almost eliminated the
dominance of structural waves in the piping system, which can adversely affect the
experimental results, as will be shown later in this section.
As was discussed earlier in Chapter 4, the piping system onboard the Swedish marine vessel had a reciprocating piston pump generating pressure pulsations at the fundamental and harmonics of the pump operating frequency [65]. Theoretically a quarter wavelength tube whose length was a quarter wavelength of the acoustic wave generated by the pump on the Swedish marine vessel could give large attenuation of pressure at one of these tonal frequencies. Multiple length quarter wave tubes could be used to simultaneously attenuate multiple pump tones. Hence it was necessary to determine if a quarter wavelength tube could be implemented successfully on such a system to attenuate pump vibrations. Tests were carried out using a quarter wavelength tube on the experimental rig discussed previously to determine the potential of such devices for controlling the pump induced piping vibrations. The tests and their results are described in detail in the following section.

### 4.2.1 Experiments using quarter wavelength tube

Figure 4.7 shows a picture of the quarter wavelength tube used in the experiments. It was built from a 2” standard, industrial steel pipe (see Chapter 2 for specifications), 0.835 m long, connected to a 2” standard steel tee. A steel end cap was fitted at the end of the steel tube and flanges were screwed onto the steel tee to enable the quarter tube to be bolted into the experimental rig. A valve was fitted at the end of the steel tube to remove air bubbles as shown in the picture. The fundamental resonant frequency of the quarter wavelength tube, estimated from the length of the resonator tube \( f = c/4 \times l \), where \( c \) is the speed of sound in water and \( l \) the length of the quarter wave tube, was 434 Hz. Figure 4.7 shows the expected behaviour of the quarter tube at the
resonant frequency [46]. If the quarter wavelength tube with a resonant frequency of 434 Hz is introduced inside a duct system, then the acoustic wave travelling inside the duct at 434 Hz is reflected off the end of the quarter wave tube and travels back to the junction of resonator as was shown previously in Figure 4.6. Since the quarter wave tube is a quarter wavelength long the reflected wave is half a wavelength long or 180° out of phase with the incident wave and thus cancels or attenuates it. Figure 4.8 shows the expected pressure drop at 434 Hz inside the duct system with a quarter wavelength tube (resonant frequency, 434 Hz) incorporated in it. Theoretically, at this frequency the incident wave is reflected from the end cap and returns to the junction 180° out of phase with the incoming wave, leading to its cancellation at the junction and downstream of it.

Figure 4.9 shows the schematic diagram of the experimental setup for the quarter wavelength tube test. The upstream and downstream steel pipes were both 2” standard steel pipes 1.53m and 2.17m long respectively. The quarter wavelength tube section was connected between the upstream and downstream steel pipes. The anechoic termination described in Chapter 3 was connected to the downstream section (not shown in Figure 4.9). The hydrophone was positioned at the junction of the ‘T’ to monitor the pressure at the mouth of the quarter wave tube. It is very important to note here that the rubber damper sections built for reducing the dominance of structural waves on the experimental rig as discussed in section 3.1.2 of Chapter 3 were not incorporated in the experimental setup for these first tests.

The rig was then filled with water. A 0-1000 Hz white noise signal was used to excite the electromagnetic shaker connected to the upstream steel pipe. The transfer function between the excitation and the hydrophone signal was recorded. The rig was
Figure 4.7: Picture of the quarter wavelength tube used in the experiments

Figure 4.8: Theoretical behaviour of the quarter wavelength tube at its resonant frequency
then emptied and the resonator tube in the tee section was replaced with a steel plug similar to the one shown in Figure 3.3 in Chapter 3. The rig was again filled with water and the experiments were repeated exactly as above. Figure 4.10 shows the transfer function plots between the excitation and the hydrophone for the rig with and without the quarter wavelength tube. It can be observed that the transfer functions for both cases contains many resonant peaks and also the expected pressure drop at 434 Hz cannot be seen.

Ideally it is expected that for low frequency excitation of fluid inside a pipe, a plane, propagating wave be generated inside the experimental rig for frequencies below the cut off frequency of the pipe (1,0) mode i.e. 15440 Hz for 2” industrial steel pipe (approximately given by $f_c = \frac{1.81 \times c}{2\pi \times a}$ for hard walls). Hence the transfer function should ideally show resonant peaks only at the propagating frequency of the fluid wave (i.e. large response at axial resonances). It was suspected that n=1 structural bending waves were dominating the response of the experimental rig. The n=1 structural waves could be generated by the asymmetric excitation of the experimental rig as was discussed in section 3.1.2 of Chapter 3.

In fact the levels near the resonant frequency (434 Hz) appear to be higher for the rig with the quarter wavelength tube in place. Since the structural system is asymmetric around the pipe axis it could be well excited by n=1 incident waves generated by the excitation shaker misalignment problem described in section 3.1.2. Once excited, the structural quarter wave tube and contained water system vibrate and also re-excite the piping system at various resonance frequencies. The net result is large levels of structural vibration at various frequencies even though the source is dominantly symmetric around
the pipe axis. Figure 4.11 shows the quarter wavelength tube vibrating in a cantilever mode. Thus the n=1 structural waves were generated inside the experimental rig and were probably dominating the rig response. The desired effect of the quarter wavelength tube on the fluid wave at 434 Hz could not be observed since the fluid response was dominated by the n=1 structural waves. It was necessary to prevent the generation and thus reduce the domination of the structural n=1 waves on the rig response. Hence another set of experiments was carried out using the rubber isolating section before the quarter wavelength tube section to prevent the generation of structural waves as was done in section 3.1.2 of Chapter 3.

Figure 4.12 shows a schematic diagram of the new experimental setup. The rubber damper section discussed in the first section of this chapter was inserted before the quarter wavelength tube. The hydrophone was mounted at the mouth of the resonator. The rig was then filled with water. The experimental rig was excited by a 0-1000 Hz white noise signal, supplied to the electromagnetic shaker. Transfer functions between the excitation and the hydrophone were recorded. The quarter wave tube was then replaced with a steel plug. The experiments were repeated again with the hydrophone mounted inside the pipe at the steel plug. Transfer functions between the excitation signal and the hydrophone were recorded. Figure 4.13 shows the transfer function plots for the rig with and without the quarter wavelength tube. From the transfer function plot one can clearly observe the anti-resonance at approximately 454 Hz, which is close to the theoretically predicted frequency of 434 Hz. Thus the quarter wavelength tube was showing the expected behaviour by causing a pressure drop at its resonant frequency. In this case the hydrophone was mounted at the mouth of the quarter wave tube. Another
Figure 4.9: Initial experimental setup for quarter wavelength tube experiments

Figure 4.10: Comparison of transfer functions for initial experimental rig without rubber isolator section
Figure 4.11: Quarter wavelength tube vibrating as a cantilever beam leading to generation of dominant structural waves

Structural $n=1$ bending waves generated by asymmetric shaker excitation propagating along the experimental rig

Side branch tube vibrating as a cantilever beam
Figure 4.12: Improved experimental setup with rubber isolator

Figure 4.13: Comparision of transfer functions for improved experimental setup
measurement was taken with the hydrophone 32” downstream of the mouth of the quarter wavelength tube. The results for this measurement are shown in Figure 4.14. In this case also comparison of the rig response with and without the quarter wavelength tube clearly shows the drop in pressure at approximately 454 Hz. Thus it could be deduced from this result that the quarter wavelength tube was reducing pressure not only at the mouth of the tube but also further downstream of its location i.e the quarter wavelength tube was causing global reduction of pressure levels at its resonant frequency. Thus the quarter wavelength tube was implemented on the test rig successfully.

One important observation from these experiments was the demonstration of the fact that the dominating presence of structural waves on the original experimental rig adversely affected the experimental results. As shown in Figure 4.14, for the quarter wavelength tube experiments without the rubber damper section incorporated on the experimental rig, the shaker was exciting the n=1 structural bending waves along with the fluid waves. It is also possible that the quarter wavelength tube was behaving as a cantilever beam and was excited by the structural waves generated at the shaker end further increasing the presence of structural waves on the experimental rig. The problem with the presence of structural waves on the experimental rig was that it was dominating the rig response i.e. it was not possible to observe the fluid response due to the presence of structural waves on the rig. As a result the expected behaviour of a fluid wave-canceling device like the quarter wavelength tube could not be observed. Thus it was necessary to reduce the dominance of the n=1 structural bending waves if not eliminate them completely to obtain good experimental results. This was done by incorporating a
Figure 4.14: Plot showing transfer function between excitation and hydrophone mounted 32” downstream of quarter wave tube mouth
rubber damper section (described in section 3.2.1 of Chapter 3) as the upstream section on the experimental rig.

As shown in Figure 4.15 the rubber isolator section served as an impedance change on the experimental rig causing the structural waves to be reflected and thus preventing their propagation on the experimental rig [9]. The fluid waves would not be affected by the introduction of the rubber isolator section because at low frequencies there is minimal coupling between the fluid and the pipe walls and hence the change in structural impedance would produce no effect on the fluid wave propagation [14]. As was shown earlier in this section, after implementation of the rubber isolator section on the experimental rig, the quarter wavelength tube tests showed the desired results. Thus these experiment showed that the presence of $n=1$ structural bending waves on the experimental rig was a major obstacle in achieving good experimental results. Interestingly Dodson et. al. [46] also used flexible rubber hose for their investigation on quarter wavelength tubes though they have not mentioned the reason for doing so. Their work showed good attenuation of fluid waves at the quarter wavelength tube, design frequency.

Thus it could be concluded from the present experiments that it was necessary to reduce the presence of structural waves on the experimental rig to obtain desired performance of fluid pressure reducing devices. Hence the test rig with the rubber isolator as the upstream section was used for the rest of the experiments.
Figure 4.15: Structural waves generated by asymmetric excitation at the shaker end

- Asymmetric excitation due to shaker misalignment
- Fluid wave propagating inside experimental rig
- Quarter wavelength tube vibrating as a cantilever beam due to structural waves propagating along the experimental rig

Rubber isolator section, which prevents n=1 structural bending waves from propagating on the experimental rig

Figure 4.16: Rubber isolator section working as a n=1 structural bending wave absorber

- Structural n=1 bending waves
- Propagating n=0 fluid waves which are not affected by the presence of the rubber isolator section
4.3 Helmholtz resonator

A Helmholtz resonator is a simple element frequently used in acoustic applications. As shown in Figure 4.12, the Helmholtz resonator basically consists of a cavity of large volume connected to the outside space through a relatively narrow neck or opening. For wavelengths much larger than the dimensions of the resonator, the resonator volume acts as a stiffness element while the fluid in the neck acts as a mass element [70]. The resonator can thus be modeled as a mass-spring system. At the resonant frequency of the Helmholtz resonator the fluid mass inside the neck of the resonator moves with high velocity amplitudes. At the resonant frequency of the Helmholtz resonator the mouth of the resonator radiates sound out of phase with the incident duct field thus canceling it to a zero pressure field. The duct impedance is thus zero to the propagating wave at that frequency causing most of its energy to be reflected back to the source. Thus the characteristic property of such a resonator is its ability to reactively reflect sound waves of a particular frequency known as the resonant frequency of the device. This frequency is given by the following equation derived by Helmholtz [51],

\[
f = \frac{c}{2\pi} \sqrt{\frac{F_N}{V(l_N + \alpha)}}
\]  

(4.2)

where \( F_N \) is the area of the neck, \( V \) volume of the resonator, \( l_N \) length of resonator neck and \( \alpha \) the end correction factor. The fluid length in the neck element of the resonator is longer than the physical length of the neck i.e. the effective length of the neck is longer than the physical length because of its radiation-mass loading [70]. The factor \( \alpha \) is an end correction used to include the effective length of the neck in equation...
Following are the formulas for the end correction factor assuming that the mass loading at inner end of the neck is equivalent to a flanged termination [70],

\[ \alpha = 1.7a \text{ (outer end flanged)} \]  
\[ \alpha = 1.5a \text{ (outer end unflanged)} \]

where \( a \) is the radius of the neck. One of the advantages that a Helmholtz resonator holds over conventional passive rubber isolators is that it can be used to reactively control low frequency fluid wave propagation. Also, compared to quarter wavelength tubes, its size is much more compact at low frequencies. However the working and design of a Helmholtz resonator is much more complicated than that of a passive rubber damper or a quarter wavelength tube. For example if the area of the resonator neck is small, then predicted values of resonant frequencies and measured values are not in agreement due to dissipative forces resulting from viscosity [70]. It must be noted that the equation (4.2) does not consider the shape of the resonator cavity when deriving the resonant frequency.

Alster [51] showed that the shape of the resonator cavity is an important factor for deriving the resonant frequency and gave an improved formula for the calculation of resonant frequencies. The improved formula for calculating the resonant frequency of a Helmholtz resonator is as follows [51],

\[
f = \frac{c}{2\pi} \times \left[ 1.21(V + V_N) \frac{V}{V + V_N + V_{01}} \frac{h}{h + l_N + l_{01}} \left[ l_N + (l_{01} + l_{01}) \left( 1 + \frac{1}{2} \left( V_N + V_{01} + l_N + l_{01} \right) \left( \frac{1}{h} + \frac{1}{3} \left( \frac{V_N + V_{01} + l_N + l_{01}}{h} \right) \right) \right) + l_{02} \right] \right]
\]

where,

\( f \) = resonant frequency,

\( c \) = speed of sound in fluid,
\( F_N \) = area of the neck,
\( l_N \) = length of the neck,
\( V_N \) = volume of the neck (=\( F_N l_N \))
\( V \) = volume of the resonator without the neck
\( l_F \) = form factor (for more details refer to [51])
\( l_{01}, l_{02} \) = two parts of end correction lengths due to motion of gas particles outside the resonator (generally \( l_{01}=l_{02}=0.24r \) where \( r \) is the radius of the neck)

Figure 4.13 shows a schematic Helmholtz resonator in a fluid supply system. In order to control the pipe acoustic waves, the resonator is designed such that its resonant frequency matches the fundamental frequency or harmonics of the pump operating speed e.g. if the pump operating speed is 550 rpm then the corresponding frequency is given by \( \omega = 2\pi(550)/60 = 58 \) rad/sec. Thus, theoretically a Helmholtz resonator with a resonant frequency of 58 rad/sec, would reflect the pressure pulsations propagating through the piping system at its resonant frequency thus reducing the pressure pulsations inside the piping system. The target piping system onboard the Swedish marine vessel employed a positive displacement pump for circulating the engine oil [65]. Pressure fluctuations generated by the pump were transmitted by the fluid through the piping system at the operating frequency of the pump. Thus a Helmholtz resonator with its resonant frequency same as the fundamental frequency of the pump operating speed could be installed in the piping system, to reduce pressure fluctuations.

Experiments were carried out using Helmholtz resonators installed on the test rig. Two Helmholtz resonator configurations were tested. Figures 4.14 and 4.15 show a
Figure 4.17: Schematic diagram of a simple Helmholtz resonator

Figure 4.18: Schematic diagram of a Helmholtz resonator in a piping system
schematic diagram and picture of the first resonator configuration, which will be referred to as the basic Helmholtz resonator. The resonator volume was a 3” standard diameter cast iron pipe, 5.5” long, with a cast iron end cap at one end. The other end was connected to a cast iron reducer (3” to 2” diameter). This subassembly formed the volume of the resonator. A 2” standard diameter steel pipe, 1” long, was connected to the 2” end of the cast iron reducer section. This formed the neck part of the resonator. The neck and volume section together will be referred to as the resonator. The neck section of the resonator was connected to a 2” standard diameter steel tee similar to the one used for the quarter wavelength tube section. Table 4.3 gives the values of resonator dimensions and the equation used for calculating the resonant frequency of the device.

PCB pressure sensors, s2 and s3 were mounted at the junction of the ‘tee’ and volume element of the resonator respectively as shown in Figure 4.14. At the resonant frequency of the Helmholtz resonator the pressure at the mouth of the resonator should be theoretically very low while pressure inside the resonator cavity would be high. Hence a transfer function between pressure sensors s2 and s3 should show a resonance at the resonant frequency of the Helmholtz resonator, since it is a ratio of the output at s3 to the input at s2. This was the reasoning behind mounting pressure sensors s2 and s3 at their respective locations.

Figures 4.16 and 4.17 show a schematic diagram and a picture of the second Helmholtz configuration, which will be referred to as a tunable Helmholtz resonator. For this configuration the cast iron reducer section was connected to the steel tee through a standard 2” brass gate valve. It was assumed that the valve cross section would act as the neck in this case. Rotating the valve would enable one to control the mass of fluid in the
### Table 4.3: Dimensions and resonant frequency of basic Helmholtz resonator

<table>
<thead>
<tr>
<th>Volume of the resonator, $V$ (m$^3$)</th>
<th>Effective length of resonator neck, $L_e$ (m)</th>
<th>Area of resonator neck, $S$ (m$^2$)</th>
<th>Resonant frequency in Hz, ( f_n = \frac{c}{2\pi} \sqrt{\frac{S}{L_e \cdot V}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.036</td>
<td>0.015</td>
<td>0.002</td>
<td>460</td>
</tr>
</tbody>
</table>

### Table 4.4: Resonant frequencies of tunable Helmholtz resonator for different percentages of open neck area

<table>
<thead>
<tr>
<th>Effective length of resonator neck, $L_e$ (m)</th>
<th>Percentage of neck area open, $\alpha$ (%)</th>
<th>Resonant frequency in Hz, ( f_n = \frac{c}{2\pi} \sqrt{\frac{\alpha \cdot S}{L_e \cdot V}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.053</td>
<td>100</td>
<td>244</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>173</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>77</td>
</tr>
</tbody>
</table>
Figure 4.19: Schematic diagram of Basic Helmholtz resonator

Figure 4.20: Picture of the basic Helmholtz resonator mounted on test rig
Figure 4.21: Schematic diagram of tunable Helmholtz resonator

Figure 4.22: Tunable Helmholtz resonator mounted on test rig
neck and thus the resonant frequency. Hence this configuration is referred to as a tunable Helmholtz resonator. Table 4.4 gives the resonant frequency of the tunable Helmholtz resonator for different neck areas. For this configuration also pressure sensors s2 and s3 were mounted at the same locations as for the basic Helmholtz resonator configuration. Experiments were carried out using both these configurations and will be described in the following two sections.

4.3.1 Basic Helmholtz resonator experiments

Figure 4.18 shows a schematic diagram of the experimental setup for the basic Helmholtz resonator tests. The basic Helmholtz resonator section (theoretical resonant frequency 460 Hz) described earlier was bolted in between the upstream and downstream experimental rig test sections. Similar to the quarter wavelength tube experiments, the initial experimental rig for the basic Helmholtz resonator tests did not have the rubber damper section incorporated on the test rig, i.e. the upstream section was a steel pipe, 2” inner diameter, and 1.22 m long, as shown in Figure 4.18. The downstream section was a 2” standard diameter steel pipe, 2.17m long. The clear plastic anechoic termination described in Chapter 3 was connected to the downstream pipe. PCB pressure sensors, s1 to s5 were mounted at the positions shown in Figure 4.18. The sensors s2 and s3 were mounted at the mouth and on the body of the resonator respectively. Sensors s4 and s5 were mounted downstream of the resonator to monitor the effect of the resonator on the propagating fluid waves. Table 4.5 gives the exact location of the pressure sensors on the experimental rig. The rig was then filled with water. A 0-1000 Hz random noise signal supplied to the electromagnetic shaker was the excitation source for the experimental rig.
Table 4.5: Sensor locations on the experimental rig measured from the shaker end for the basic Helmholtz resonator tests

<table>
<thead>
<tr>
<th>Distance of sensors from shaker end (m)</th>
<th>S1</th>
<th>S2</th>
<th>S3</th>
<th>S4</th>
<th>S5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.1</td>
<td>1.4</td>
<td>1.4</td>
<td>2.26</td>
<td>2.72</td>
</tr>
</tbody>
</table>
Figure 4.23: Initial experimental setup for basic Helmholtz resonator tests

Figure 4.24: Transfer function plots for experimental rig without rubber isolator
The transfer functions between the excitation signal and all the pressure sensor outputs were recorded. In addition transfer functions between s2 and s3 were recorded.

The water was then drained out from the rig. The resonator was removed from the steel tee and replaced with a steel plug. The same experiment was carried out for the new setup and transfer functions between excitation and pressure sensor signals were recorded. Figure 4.19 is a plot showing the transfer function between excitation and sensor s4. From the plot it can be observed that the desired drop in pressure at the resonant frequency of the Helmholtz resonator is not visible. As has been already discussed in section 4.2 of this chapter it was suspected that structural waves were again dominant on the experimental rig and contaminating the fluid response to excitation. This prevented the acquisition of desired results for the basic Helmholtz resonator test, which in this case was a pressure drop at the resonant frequency of the resonator. Hence as was done for the quarter wavelength tube tests it was decided to use the rubber damper as the upstream section to reduce transmission of structural vibrations generated at the excitation end.

Figure 4.20 shows the improved experimental setup with the rubber damper used for the quarter wavelength tube experiments as the upstream section. The experimental procedure was similar to the experiments with the steel pipe as the upstream section and hence has not been repeated. Transfer functions were recorded again between the excitation and all the pressure sensors. Figures 4.21 shows the plots of transfer function between excitation and pressure sensor s4 for the experimental rig with and without the Helmholtz resonator. It can now be seen that the resonance at 490 Hz for the test rig without the Helmholtz resonator in place is reduced significantly for the setup with the
Figure 4.25: Improved experimental setup for Helmholtz resonator tests

Figure 4.26: Comparison of transfer functions at sensor s4 with and without Helmholtz resonator

Transfer function between excitation and sensor s4

Transfer function, dB

Autospectrum level, dB

Pressure at downstream sensor s4

With resonator

Without resonator

Frequency, Hz

0 100 200 300 400 500 600 700 800

-90 -80 -70 -60 -50 -40 -30

Transfer function, dB

Figure 4.26: Comparison of transfer functions at sensor s4 with and without Helmholtz resonator
It could be concluded that addition of the Helmholtz resonator caused the propagating fluid wave at 490 Hz to be reflected leading to its attenuation downstream of the resonator. In addition a resonant peak at approximately 100 Hz is observed in Figure 4.21. This was the n=0 fluid acoustic resonance inside the experimental rig as was discussed earlier in Chapter 3. To further determine the effect of adding the Helmholtz resonator, transfer function plots for sensors s1 and s5 are also shown in Figures 4.22 and 4.23. Figure 4.22, which is a transfer function plot for sensor s1 shows no difference in the sensor output for both rig setups i.e. without and with the Helmholtz resonator. This was expected since sensor s1 was mounted upstream of the Helmholtz resonator and hence there should be no pressure drop at s1. However Figure 4.23, which is a transfer function plot of sensor s5 shows a drop in pressure at the resonant frequency of the Helmholtz resonator, i.e. the propagating fluid wave at 490 Hz was being reflected at the Helmholtz resonator and hence there was a pressure drop at sensors s5. Thus transfer functions at sensors s4 and s5 showed reductions in pressure at 490 Hz while there was no effect of the Helmholtz resonator at sensor s1. Hence it could be ascertained that the Helmholtz resonator was causing global pressure reduction downstream of the test rig at its resonant frequency.

Figure 4.24 shows transfer function between s2 and s3. At the resonant frequency of the Helmholtz resonator the pressure inside the resonator cavity is high while the pressure at the mouth of the resonator is low. Hence the transfer function between s2 and s3 which is essentially a ratio of output of sensor s3 to that of sensor s2 should show a resonance at 490 Hz. This resonance can be seen in Figure 4.24. This meant that at approximately 490 Hz, the pressure inside the resonator cavity was high while the
Figure 4.27: Transfer function between excitation and s1, Helmholtz resonator tests

Figure 4.28: Transfer function between excitation and s5, Helmholtz resonator tests
Figure 4.29: Transfer function between s2 and s3, Helmholtz resonator tests
pressure at the mouth of the resonator was low i.e. the resonator was reflecting the incident acoustic wave at 490 Hz back to its source and reducing the pressure downstream of the resonator.

Thus similar to the quarter wavelength tests, incorporating the rubber isolator on the test rig led to better results for the Helmholtz resonator experiments. The experiments on quarter wavelength tube and the Helmholtz resonator comprehensively proved the importance of reducing if not eliminating the presence of structural waves on the experimental rig for demonstrating the performance of acoustic passive control devices. Figure 4.25 shows how structural waves could have been generated on the experimental rig with a steel pipe as the upstream section for the Helmholtz resonator tests. As shown in the Figure 4.25 the asymmetric excitation due to improper alignment of the shaker with the center of the pipe causes structural $n=1$ bending waves to be generated along with the $n=0$ fluid waves. The structural waves dominated the rig response i.e. they subdued the response of the fluid inside the rig to excitation. As a result the desired performance of a fluid wave reflecting device like the Helmholtz resonator could not be obtained. Thus to demonstrate the desired response of the Helmholtz resonator it was necessary to eliminate the presence of structural waves on the experimental rig. This was done as shown in Figure 4.26 by replacing the upstream steel pipe with a rubber damper section described in section 4.1 of this chapter.

As shown in Figure 4.26 the rubber isolator presented a change in impedance to the propagating structural waves causing them to be reflected and thus reducing their dominant presence on the test rig. The propagating $n=0$ fluid waves would not be affected by the rubber damper since the coupling between the fluid and structural waves is
Figure 4.30: Structural wave generation and propagation affecting experimental results for Helmholtz resonator tests

Asymmetric excitation due to shaker misalignment

Dominant propagating $n=1$ structural bending waves generated due to asymmetric excitation

Subdued fluid wave propagating inside experimental rig

Rubber damper section, which prevents $n=1$ structural bending waves from propagating on the experimental rig

Structural $n=1$ bending waves

Propagating $n=0$ fluid waves which are not affected by the presence of the rubber damper section

Figure 4.31: Rubber damper prevents the propagation of structural waves thus reducing their dominant presence on the test rig
insignificant at low frequencies [14, 35]. Thus addition of the rubber damper reduced the dominance of structural waves which in turn caused the fluid response to be more pronounced. This enabled the acquisition of desired results for experiments on the fluid wave reflecting device i.e. the Helmholtz resonator.

Thus it can be concluded that in order to demonstrate the behaviour of fluid wave controlling devices in experiments it is very important to eliminate the presence of structural waves on the test rig. Generation of structural waves and their propagation on the test rig was a very important factor affecting the acquisition of desired results. One can also conclude that for total control of power in the piping system, a system would have to simultaneously act on both the fluid waves and structural waves.

### 4.3.2 Tunable Helmholtz resonator tests

In practical piping systems the exciting pressure pulsations do not always occur at constant frequencies i.e. the frequency of the excitation is variable. Most piping systems employ positive displacement pumps, which generate pressure pulsations at fundamental and harmonics of their operating speed [65]. Often the operating speed of the pump and hence the excitation frequency varies depending on the load on the pump and other external conditions. Hence a conventional Helmholtz resonator which is designed for a single resonant frequency would become ineffective if the excitation frequency were not the same as the designed resonant frequency of the Helmholtz resonator. If the excitation frequency were not the same as the resonant frequency then the propagating fluid wave at the excitation frequency, would not be reflected by the Helmholtz resonator. Thus the propagating wave would not be attenuated downstream of the Helmholtz resonator rendering the resonator ineffective.
A Helmholtz resonator with a tunable resonant frequency could adapt to the changing excitation frequencies. The resonant frequency for a Helmholtz resonator is given by,

\[
f_n = \frac{c}{2\pi} \sqrt{\frac{S}{L_e \cdot V}}
\]  

(4.5)

where \( S \) is the area of the resonator neck, \( L_e \) the effective length of the neck and \( V \) the volume of the resonator body. Changing the area of the neck or volume of the resonator cavity in equation 4.5 can change the resonant frequency of the Helmholtz resonator. A metallic gate valve, to change the neck area of the resonator, was used for the tunable Helmholtz resonator shown in Figure 4.15. By rotation of the valve, it was possible to reduce or increase the neck area leading to a corresponding decrease or increase in the resonant frequency of the Helmholtz resonator. The theoretical range of frequencies for the tunable resonator was from 77 Hz to 278 Hz (neck 10% to 100% open) as was shown in Table 4.4. The experimental setup was the same as that for the basic Helmholtz resonator tests discussed in section 4.3.1 and shown in Figure 4.20 and hence has not been repeated. Excitation was a 0-1000 Hz random noise signal supplied to the electromagnetic shaker. Transfer functions between s2 and s3 were recorded for different valve positions. Transfer functions were also recorded between excitation and sensors s1, s2 and s5. Figure 4.27 shows the transfer function plots between s2 and s3 for three different cases of open neck area. From the figure it can be seen that there is not any significant change in the transfer function plots for different neck openings. Figures 4.28, 4.29 and 4.30 show transfer function plots for sensors s1, s4 and s5 respectively. These plots also do not any change in the rig response by varying the cross-sectional area of the Helmholtz resonator neck.
Figure 4.32: Transfer function plots between s2 and s3 for tunable Helmholtz resonator tests
Figure 4.33: Transfer function between excitation and sensor s1 for different valve positions

Figure 4.34: Transfer function between excitation and sensor s4 for different valve positions

100% open
50% open
10% open
Figure 4.35: Transfer function between excitation and sensor s4
Two possible explanations for this are possible. One of them could be the fact that air bubbles were trapped inside the groove of the metallic gate valve, which prevented the acquisition of desired experimental results. Air bubbles present an impedance change to the propagating fluid wave. This could lead to reflection of the propagating waves from the neck area of the resonator itself, i.e. the propagating waves would not be able to attain high pressures inside the volume of the resonator which is a must for the working of the Helmholtz resonator. The presence of air bubbles also changes the speed of sound in the fluid and thus changes the resonant frequency of the Helmholtz resonator given by equation 4.5. Another possible reason could be that the neck cross section was only 0.015m wide. Hence the mass of fluid in the neck was very small. For large neck areas and thus large fluid mass, it has been found that theoretically predicted values of resonant frequency match very well with measured values [70]. For small neck areas and thus small fluid mass, the theoretical and measured values of resonant frequencies do not match. This is due to increased dissipative forces arising out of viscosity for small narrow necks [70]. Similarly for these experiments it could be argued that changing such a small mass did not provide the desired results due to increased dissipative forces resulting from viscosity. As a result changing the neck cross sectional area did not provide the expected change in the resonant frequency of the tunable Helmholtz resonator.

4.4 Summary

Experiments were carried out to test the performance of different passive devices in this chapter. Experiments were performed using a rubber isolator section, quarter
wavelength tube and Helmholtz resonator (basic and tunable). The following is a summary of the main results of these experiments:

- The rubber isolator tests showed that the isolator section is very effective in controlling the \( n=1 \) structural waves and to some degree maybe the \( n=0 \) structural waves i.e. the torsional and extensional waves. The rubber isolator section represented a change in structural impedance causing the above mentioned structural waves to be reflected back towards the source. The rubber isolator would be probably poor for the \( n=0 \) fluid wave, since this wave travels mainly through the fluid path and also the coupling between the fluid and pipe walls is small at low frequencies [35].

- The quarter wavelength tube experiments did not give the desired results for the initial experimental rig without the rubber isolator section incorporated upstream of the device in the test rig. However once the rubber damper section was installed on the experimental rig global reduction in pressure was observed downstream of the quarter wavelength tube at its resonant frequency. These experiments showed how important it was to remove the \( n=1 \) structural bending waves on the test rig in order to demonstrate the performance of the device.

- Similar to the quarter wavelength tube tests, the basic Helmholtz resonator tests also did not give the desired pressure drop at the resonant frequency of the resonator. However once the rubber isolator section was incorporated on the test rig, global reduction in pressure at the resonant frequency of the Helmholtz resonator was obtained. The tunable Helmholtz resonator tests however did not give the desired results. Presence of air bubbles inside the valve and the small neck size of the resonator could be the possible reasons for failure to obtain desired results.
A general but very important result from all the above experiments was that structural waves on the test rig dominate the response of the rig to excitation, subduing the fluid response. Hence to get meaningful results for experiments on fluid wave controlling devices it is of the utmost importance, to reduce the presence of structural waves if not eliminate them completely. This also implies that reduction of total power flow will require a device that acts on both structural and fluid waves simultaneously.
Chapter 5

Active and Active/passive Control Experiments

Passive devices such as rubber isolator sections for vibration damping as investigated in Chapter 4 are effective only at high frequencies. For low frequencies they tend to be oversized and impractical to implement in actual piping systems. Rubber dampers also can cause loss of static pressure in highly pressurized fluid piping systems making them unfit for use in such systems. These dampers also tend to stiffen at high pressures reducing their efficiency and have little effect on fluid borne waves and are useful for controlling structural waves [65]. Other reactive passive devices such as quarter wavelength tubes and Helmholtz resonators discussed in the previous chapter are designed to operate in narrow frequency bands and control of fluid waves. However any variation in the excitation frequency can render these devices ineffective. Thus if a system has a tonal source whose frequency varies with operation then these devices will be ineffective. In addition these reactive devices are only effective at controlling fluid borne waves and do not give any useful reduction of structure borne waves.

The above-mentioned disadvantages of passive control devices, makes active control of fluid-borne noise an attractive alternative. Leug [71] was the first to implement an active control system for noise control in a duct. In feed forward active control of sound, the primary noise source is detected using transducers. This information is then
provided to the controller in the form of a reference signal. The controller then generates a secondary noise field of the same amplitude but exactly out of phase with the primary noise source, which is used to drive the active or secondary source. The superposition of the primary and secondary noise fields causes attenuation of the primary source field [71].

The actual physical components of a single-channel feedforward controller are shown in Figure 5.1 [71]. The excitation signal $x$, is the primary input to the mechanical system. The electrical controller $H$, is driven by an estimate of the excitation signal $x$. The original excitation is assumed to influence the mechanical system by the force, $f_p$, transmitted through the primary path $P$. The electrical controller signal influences the mechanical system by the secondary force $f_s$. The net excitation of the mechanical system is assumed to be proportional to the difference between the primary and secondary force ($f_p - f_s$) [71]. Figure 5.2 shows the equivalent block diagram of the feedforward control system where all the signals are represented by their Laplace transforms and the responses of the various components by their transfer functions [71]. From Figure 5.2 the Laplace transform of the response of the mechanical system can be expressed as

$$E(s) = G(s) [P(s) - H(s)] X(s) \quad (5.1)$$

Assuming that the response of the mechanical system is only due to the primary and secondary force it is possible, in principle, to drive the response of the system to zero by exactly matching the primary force $F_p$ and the secondary force $F_s$. If the controller response exactly matched that of the primary path then the Laplace transform of the response of the mechanical system would be zero, i.e.,

$$\text{if } H(s) = P(s) \text{ then } E(s) = 0. \quad (5.2)$$
Figure 5.1: Components of a feedforward control system [71]

Figure 5.2: Equivalent block diagram of a feedforward control system [71]
The fluid borne noise in fluid filled piping systems is mostly caused by positive
displacement pumps at the fundamental frequency or harmonics of the pump operating
speed i.e. the noise source is deterministic. The operating frequency of the pumping
device could be used as a reference signal to the controller, which in turn would generate
secondary signals out of phase with the pump signals to effect cancellation of fluid borne
noise. Hence application of active control for reducing the fluid-borne noise in the piping
system aboard the marine vessel appeared to be feasible.

Figure 5.3 shows a schematic diagram of a single channel feed forward active
control system applied to a fluid filled piping system. It can be recalled from Chapter 1
that the goal of this project was to develop vibration control systems that could be
incorporated on a practical piping system on board a Swedish marine vessel. The piping
system onboard the marine vessel employed a positive displacement pump for
transporting the fluid [65]. The pressure pulsations generated inside the piping system
were generated at the fundamental frequency and harmonics of the pump operating speed
[65] (i.e. the excitation source was deterministic). Thus an active control system could be
developed which utilized the operating speed of the pump as a reference signal to the
controller which would then drive the control actuator to generate signals of same
amplitude but 180° out of phase with the primary disturbance. Critical aspects are
actuators, which can provide the necessary control signals in terms of amplitude, and
frequency content, and error sensors, which provide coherent information (with the
reference signal) and levels, related to the variable desired to be controlled. This chapter
will investigate the possibilities of active noise control for reducing the total power flow
in piping systems filled with fluid.
Figure 5.3: Schematic diagram of an active control applied to a fluid filled piping system
5.1 Control arrangement and hardware setup for active control experiments

The experimental setup used for active control experiments in this thesis was very similar to the one used by Maillard [65] except that the excitation source was an electromagnetic shaker and the reference signal was obtained from an accelerometer mounted on the shaker diaphragm. Figure 5.4 shows a general schematic diagram of an active control setup applied to the experimental rig in this project. As shown in the Figure 5.4 the primary disturbance was an electromagnetic shaker connected to a piston diaphragm positioned the upstream section as discussed in section 3.1.2. The reference signal was obtained from an accelerometer mounted on the excitation diaphragm shown in Figure 3.2 of Chapter 3. Sensor s3 was used as the error sensor. The reference and the error signal were inputs to the controller, which drove the secondary (control) actuator. The control signal applied to the secondary actuator was constructed in real time on a TMS320C40 digital signal processor board. The code implemented on the C40 board is based on a broadband Filtered-x LMS algorithm [71]. The concept behind application of active control to the test rig was that the controller would drive the secondary source such that it generated fluid waves of the same amplitude but 180° out of phase with the fluid waves generated by the primary source. Hence superposition of the primary and secondary wave fields would theoretically lead to cancellation of the noise field downstream of the actuator. Two very important assumptions are made for an active control system i.e. the response of the mechanical system (in this case the test rig) is linear and the reference signal is well correlated to the primary disturbance.
Figure 5.4: Schematic diagram of an active control system applied to the experimental rig
Figure 5.5 shows a schematic diagram of the digital control system used for the single channel feedforward active control experiments [65]. As shown in Figure 5.5 the algorithm feeds forward a reference signal \( x(n) \) (correlated to the disturbance \( d(n) \)), which in this case was the output of the accelerometer mounted on the excitation diaphragm to a digitally implemented adaptive filter, \( W(z) \). The filter generates a signal \( u(n) \) to cancel the plant response due to the primary disturbance, \( d(n) \) in this case the signal to the electromagnetic shaker. In the filtered-x LMS algorithm described in equation 5.3 [71], the adaptive filter is updated in real time by using an instantaneous estimate of the gradient of a quadratic function of the error signal, \( e(n) \), which was the output from sensor s3 on the test rig.

\[
h(n+1) = \sum_{i=0}^{I-1} h_i(n) - \alpha e(n) r(n-i)
\]  

(5.3)

where variable \( h_i \) denotes the filter coefficients which weight the current and previous \( I-1 \) input samples of the reference signal \( x(n) \). \( r(n) \) is called the filtered reference signal and is the sequence generated by passing the excitation signal \( x(n) \) through the digital filter \( T'_{ce}(z) \). \( \alpha = 2\mu \) is a convergence coefficient. This equation is known as the filtered-x LMS algorithm, since \( r(n) \) is obtained by filtering the reference signal \( x(n) \).

The update equation 5.3, thus includes the error signal \( e(n) \), and the reference signal \( x(n) \) filtered through an estimate of the control path transfer function, \( T'_{ce}(z) \). This transfer function is called the filtered-x path. \( T_{de}(z) \), the plant disturbance path, is the transfer function between the disturbance i.e. the signal to the electromagnetic shaker and the error signal \( e(n) \). \( T_{ce}(z) \) represents the transfer function between the control signal i.e. the signal driving the secondary active source and the output of the error sensor s3. The
Figure 5.5: Schematic diagram of a single channel filtered-x LMS algorithm [65]
controller used for the active control experiments implements Finite Impulse Response (FIR) digital filters for both, compensator $W(z)$ and filtered-$x$ path $T'_{ce}(z)$. In practice, the filtered reference signal is obtained using a digital filter, $T'_{ce}(z)$, whose response is an approximation to the true secondary path, $T_{ce}(z)$ [71]. $H(z)$ is a dummy adaptive filter, driven by $r(n)$ and adapted to minimize $e^2(n)$, whose coefficients are copied in the controller $W(z)$. The plant is identified when the error signal is driven to zero. This is known as *system identification*.

Figure 5.6 shows a schematic diagram of the controller hardware and experimental arrangement used for the active control experiments on the test rig. It shows the various components of the system and the equipment used for the experiments. The output of the accelerometer mounted on the excitation diaphragm, conditioned by an ICP power unit and band pass filtered through a 2-channel Ithaco filter, provided the reference signal $x(n)$ to the controller. The output of the pressure sensor s3, conditioned by an ICP power unit and band pass filtered through a 2-channel Ithaco filter, was the error signal $e(n)$ to the controller. Both, the reference and error signal were converted to digital signals using an A/D converter. The sampling frequency for the controller was set to four times the maximum frequency of interest for all the experiments. 70 filter coefficients were used for both the compensator and filtered-$x$ path FIR filters. The control signal was led through a D/A converter and band pass filtered by a 2-channel Ithaco filter. The band pass filtered control signal was amplified using a Raine power amplifier and then fed to the control actuator.

Various devices were used as control actuator. The side-branch actuator, discussed in the next section was driven using the output of the Raine power amplifier.
Figure 5.6: Block diagram of control setup [65]
However, for driving the piezoelectric actuator which will be discussed in section 5.3, the output of the Raine amplifier was fed to a step up transformer, step-up ratio 1:27, to drive the piezoelectric actuator. This was done because the piezoelectric actuator needed much higher drive voltages than what the Raine power amplifiers could supply. In the next section we discuss active control with these devices.

5.2 Active control experiment using a side-branch actuator

Figures 5.7 and 5.8 show a schematic diagram and picture of the side-branch actuator built for the active control experiments. As shown in Figure 5.7 the side-branch resonator consisted of a 2” standard steel pipe, approximately 6” in length connected to a standard 2” steel tee junction. A flexible brass diaphragm was fitted to the end of the side-branch and connected to the secondary shaker (active source) through a stinger. Henceforth the entire assembly consisting of the secondary shaker, the side-branch and the steel tee will be referred to as the side-branch actuator.

Figure 5.9 shows the principle behind the working of the side-branch actuator. The side-branch actuator described above was bolted in between the upstream and downstream sections of the test rig. The disturbance source was a large electromagnetic shaker and the control source was a smaller shaker as shown in Figure 5.9. The reference signal \( x(n) \) was the output of an accelerometer mounted on the excitation diaphragm at the primary shaker end. The error signal \( e(n) \) was obtained from a pressure sensor mounted downstream of the side-branch actuator. As shown in Figure 5.9 the reference signal was input to the controller discussed in section 5.1 of this chapter. The controller then generated the control signal \( u(n) \) which was input to the control shaker. The control shaker was connected by a stinger to the diaphragm mounted on the end of the side-
branch actuator. The diaphragm was mounted at the end of the side branch so that the diaphragm would behave as a secondary noise source similar to the primary disturbance source inside the piping system. Since the side branch is a resonant element, driving it with an active source near the resonant frequency at its end will lead to generation of large sound levels with a very low control voltage (power) level. In addition, one could also consider the end of the quarter wave tube as having variable impedance as dictated by the control signal to the brass diaphragm. The resonant frequency can thus be “tuned” by varying the control signal and thus end impedance. The control shaker would drive the diaphragm such that it generated fluid waves of the same amplitude as the fluid waves generated by the disturbance shaker but $180^\circ$ out of phase.

Assuming that the test rig was a linear system, superposition of the disturbance and control source generated wave fields would lead to the cancellation of the wave field generated by the disturbance. Thus the propagating waves generated by the disturbance source would be attenuated downstream of the side branch actuator as long as the error sensor was positioned in this region. The side branch built as part of these experiments was very compact in design and could be implemented very easily on a practical piping system. However, a drawback of the side branch actuator was that the diaphragm end of the side-branch had to be compliant enough to drive the fluid inside the side-branch tube of the actuator. Designing such a compliant end could be a problem in highly pressurized piping systems.

Figure 5.10 shows the experimental setup for the active control experiments using the side-branch actuator. The entire side-branch actuator assembly, comprising the tee junction and the side-branch, was bolted in between the upstream and downstream
Figure 5.7: Schematic diagram of side-branch actuator built for active control experiments
Figure 5.8: Picture of the side-branch actuator

Figure 5.9: Principle of working of side-branch actuator
Figure 5.10: Experimental setup for active control experiments using side-branch actuator
sections as shown in Figure 5.10. As was already discussed in Chapter 3 and 4 presence of structural n=1 bending waves on the test rig was a very important issue. The structural waves had a dominant presence on the test rig and they dominated the fluid response. Hence it was important to eliminate the presence of the n=1 structural waves on the test rig and the rubber damper section described in section 4.1 of Chapter 4 was used as the upstream section for reducing the presence of structural waves on the test rig. The downstream section was a 2” standard industrial steel pipe, 2.17m long. The clear plastic anechoic termination described in Chapter 3 was connected to the downstream section. A small 5 lb. electromagnetic shaker was connected to the diaphragm on the side branch actuator, comprising the secondary/active source for active control. Pressure sensors s1 to s5 were mounted on the experimental rig at the locations shown in Figure 5.2. A hydrophone was mounted midway between sensors s4 and s5. The excitation disturbance source was the electromagnetic shaker (50lb) connected to the upstream section. This shaker will be referred to as the primary shaker. The reference signal was obtained from an accelerometer mounted on the source diaphragm at the primary shaker end. The sensor s3 in the downstream section was used as the error sensor. Active control experiments were carried out at three discrete frequencies of 70 Hz, 150 Hz and 500 Hz.

As was discussed earlier in this chapter the goal of this work was to develop and finally implement an active control system on a piping system onboard a Swedish marine vessel. The fundamental frequency of operating speed for the pump on the marine vessel was 64 Hz. The pressure pulsations inside the piping system onboard the marine vessel also propagated at the fundamental frequency and harmonics of the pump operating speed. It was desired to see if active control using the side-branch actuator could be
successfully implemented at frequencies close to the fundamental excitation frequency inside the piping system on the Swedish marine vessel. Hence it was decided to perform active control experiments at a frequency of 70 Hz. In the test rig characterization experiments carried out in Chapter 3, it was observed there was an $n=0$ acoustic resonance in the test rig close to 150 Hz. Thus there was a need to determine the effectiveness of the side-branch actuator in controlling this acoustic resonance inside the test rig. Hence active control experiments using the side-branch actuator were carried out at 150 Hz.

Pressure pulsations in practical piping system occur at very low frequencies compared to the ring frequency of the pipe and they normally occur below a frequency of 500 Hz [65]. Hence to determine if the side-branch actuator was effective within these frequencies, active control experiments were also carried out at a discrete frequency of 500 Hz. In many practical systems the disturbance frequency is not necessarily discrete and can be spread over a broad band of frequencies. Hence active control experiments using the side-branch actuator were carried out using broad band excitation of 0-500 Hz since the pressure pulsations in most piping systems are generated at very low frequencies [65].

Figures 5.11 to 5.16 show the autospectra levels (dB scale, relative to 1 V) of the pressure sensor $s_1$, $s_2$, $s_3$ (error sensor) and the downstream sensors $s_4$, $s_5$ and the hydrophone before and after control for active control experiments at 70 Hz. It is to be noted that the excitation voltage level to the primary source in all the test cases was kept constant. There is very little change in the response of the sensors $s_1$ and $s_2$ before and after control which is expected since both the sensors are mounted upstream of the side-
Figure 5.11: Control at sensor s1, 70 Hz (s3 error sensor)

Figure 5.12: Control at sensor s2, 70 Hz (s3 error sensor)
Figure 5.13: Pressure reduction in dB for sensor s3 at 70 Hz (s3 error sensor)

Figure 5.14: Pressure reduction in dB for sensor s4 at 70 Hz (s3 error sensor)
Figure 5.15: Pressure reduction in dB, for sensor s5 at 70 Hz (s3 error sensor)

Figure 5.16: Pressure reduction in dB, for hydrophone at 70 Hz (s3 error sensor)
Figure 5.17: Fluid pressure at sensor s3 before and after control, 70 Hz.

Figure 5.18: Fluid pressure at sensor s4 before and after control, 70 Hz.
branch actuator and the control source generates waves that go upstream resulting in an upstream standing wave. In fact this was a good result because this meant that there was no control spillover which occurs when there is feedback from the control actuator to the detection sensors i.e. mechanical disturbances caused by the secondary source find their way to the detection sensor through the primary path. This is a common problem normally faced in feed forward control [71]. Figures 5.13 to Figure 5.16 show a reduction of approximately 5 dB at the error sensor s3, s4, s5 and the hydrophone. This meant that reduction was achieved not only at the error sensor but also at the other downstream sensors i.e. the control was achieved throughout the downstream section of the test rig. Thus though low dB reduction was achieved at the error sensor s3, global control was achieved at the other pressure sensors downstream of the actuators, which can be considered a positive result. The low dB reduction at the error sensor could be attributed to the fact that the reference signal obtained from the accelerometer was not well correlated with the disturbance [71]. One of the assumptions in feed forward control is that the physical system i.e. in this case the test rig, responds linearly. Thus, non-linear behaviour of the rig cannot be ruled out as one of the reasons for low dB reduction or also poor signal-noise ratio.

Figures 5.17 and 5.18 show the actual fluid pressure reduction at the sensors s3 and s4 due to active control at 70 Hz. The pressure diagrams were plotted here to get an idea of the kind of fluid pressures that could be actually controlled using the side-branch actuator. As seen it the plots the pressure at sensor s3 before control is 45KPa and after control is about 25KPa. At sensor s4 the pressures before and after control are roughly 38KPa and 22KPa. The piping system onboard the marine vessel that Maillard [65]
Figure 5.19: Control at sensor s3, 70 Hz (error sensors s3 and s4)

Figure 5.20: Control at sensor s4, 70 Hz (error sensors s3 and s4)
described in his research report had static pressure levels of 2-20MPa and the pressure pulsations had a peak to peak amplitude of 0.1MPa. This meant that if the side-branch actuator was suitably redesigned to withstand the high pressures of a practical piping system it could be used effectively to reduce fluid pressure pulsations inside a practical fluid filled piping system.

Active control experiments were also carried out at 70 Hz using two error sensors, s3 and s4 to see if better dB reductions could be obtained instead of using just one error sensor. Figures 5.19 and 5.20 show the auto spectrum levels at both the error sensors s3 and s4 before and after control. Both, sensors s3 and s4, recorded roughly a 5 dB reduction in pressure after control. Thus there was no improvement over the case with one error sensor. One of the possible reasons might be that the actuator did not possess enough control authority at 70 Hz i.e. it did not have enough control power to reduce the pressures at sensors s3 and s4 at 70 Hz.

As was discussed earlier in the characterization experiments of Chapter 3, an n=0 axial acoustic resonance was measured in the experimental rig, at roughly 150 Hz. Hence active control experiments were carried out at 150 Hz, to determine if the side-branch actuator could be used to control the n=0 acoustic resonance. The experimental setup was very similar to the one used for the active control experiments at 70 Hz. Sensor s3, was the error sensor and the reference signal was obtained from an accelerometer, mounted on the excitation diaphragm. Figures 5.21 to 5.26 show dB reductions in the auto spectrum levels at sensors s1, s2, s3, s4, s5 and the hydrophone. The auto spectrum plots for s1 and s2 show that there is no difference in their spectrums at 150 Hz with and without control. This is a good result as was mentioned earlier because it meant that the secondary
actuator i.e. the active source was not causing control spillover which is an important issue in feed forward control. Control spillover can result in a rise in the spectrum levels of the monitoring sensors, after control due to disturbances generated by the secondary source [71].

The auto spectrum plot for the error sensor s3 shows roughly a 17dB reduction in pressure levels after control. Thus better reduction was achieved at the error sensor in this case compared to 70 Hz. This meant that the side-branch actuator possibly had better control authority at 150 Hz, than at 70 Hz. In other words, its capacity to drive the fluid inside the rig and thus control pressure at the error sensor was better at 150 Hz than at 70 Hz. This might be due to the fact that it was more difficult for the secondary shaker to drive the diaphragm (mounted on the end of the side-branch) efficiently at 70 Hz than at 150 Hz due to the fluid resistance. The fluid resistance would be more at 70 Hz because the diaphragm would have to displace more fluid at & 70 Hz than at 150 Hz. 150 Hz is closer to the natural frequency of the ¼ wave tube, hence requires less control voltage. Auto spectrums of the pressure sensors s4, s5 and the hydrophone show reductions of 15dB, 8dB and 8dB respectively. Thus global control was achieved throughout the downstream section of the test rig. The dB reductions at the downstream pressure sensors s4, s5 and hydrophone for the 150 Hz case were also more than those for the active control experiments at 70 Hz. Thus better global control was achieved at 150 Hz than at 70 Hz. This re-affirmed the explanation that the actuator had better control authority at 150 Hz than at 70 Hz.

One important feature of all the plots was the uncontrolled spectra at 60 Hz and multiples of 60 Hz. This was the electrical 60 Hz noise and its harmonics. All the signals
Figure 5.21: Fluid pressure reduction in dB at sensor s1, 150 Hz (s3 error sensor)

Figure 5.22: Fluid pressure reduction in dB at sensor s2, 150 Hz (s3 error sensor)
Figure 5.23: Fluid pressure reduction in dB at sensor s3, 150 Hz (s3 error sensor)

Figure 5.24: Pressure reduction in dB at sensor s4, 150 Hz (error sensor s3)
Figure 5.25: Pressure reduction in dB at sensor s5, 150 Hz (error sensor s3)

Figure 5.26: Pressure reduction in dB at hydrophone, 150 Hz (error sensor s3)
were band-pass filtered to eliminate the 60 Hz signal. But it was still present probably because some of the cables were not shielded properly, or due to electrical noise generated by the equipment itself, etc. Since there was no reference signal for electrical noise these spectra remained uncontrolled in the active control experiments. Figure 5.27 and 5.28 show the actual pressure reduction in Pascal at sensor s4 and the hydrophone. A reduction of roughly 20KPa and 22KPa were recorded at s4 and the hydrophone respectively. Thus from the pressure plots at 70 Hz and 150 Hz it could be said that the side-branch actuator could be used to control pressure pulsations in the range 20-45KPa for the test rig.

To ascertain whether better control authority could be gained by increasing the control frequency, active control experiments were also carried out at 500 Hz. These experiments were similar to the ones performed at 70 Hz and 150 Hz and hence description of the setup has not been repeated. Figures 5.29 to 5.34 show pressure reduction in dB at pressure sensors s1, s2, error sensor s3, s4, s5 and the hydrophone for active control at 500 Hz respectively. Auto spectrums of upstream sensors s1 and s2 do not show any change before and after control. This was as expected and it also meant that there was no control spillover from the secondary actuator. Auto spectrum levels of the error sensor show a 25 dB drop in fluid pressure levels thus indicating that the secondary actuator had better control authority at 500 Hz compared to both the 70 Hz and 150 Hz cases. The auto spectrums of the sensors s4, s5 and the hydrophone show an average reduction of 15 dB each. Thus much better global control was achieved at this frequency compared to both the 70Hz and 150 Hz experiments. Thus it could be concluded that the control authority of the side-branch actuator increased with the increase in drive
Figure 5.27: Pressure reduction at sensor s4 in Pascals, 150 Hz (error s3)

Figure 5.28: Pressure reduction at the hydrophone in Pascal, 150 Hz (error s3)
Figure 5.29: Pressure reduction at sensor s1 in dB, 500 Hz (error s3)

Figure 5.30: Pressure reduction at sensor s2 in dB, 500 Hz (error s3)
Figure 5.31: Pressure reduction at sensor s3 in dB, 500 Hz (error s3)

Figure 5.32: Pressure reduction at sensor s4 in dB, 500 Hz (error s3)
Figure 5.33: Pressure reduction at the sensor s5 in dB, 500 Hz (error s3)

Control at s5, 500 Hz

Figure 5.34: Pressure reduction at the hydrophone in dB, 500 Hz (error s3)

Control at hydrophone, 500 Hz
Figure 5.35: Pressure reduction at sensor s4 in Pascals, 150 Hz (error s3)

Figure 5.36: Pressure reduction at the hydrophone in Pascal, 150 Hz (error s3)
Figure 5.37: Pressure reduction at sensor s1 in dB, 500 Hz (error s3)

Figure 5.38: Pressure reduction at sensor s2 in dB, 500 Hz (error s3)
Figure 5.39: Pressure reduction at sensor s3 in dB, 500 Hz (error s3)

Figure 5.40: Pressure reduction at sensor s4 in dB, 500 Hz (error s3)
Figure 5.41: Pressure reduction at the sensor s5 in dB, 500 Hz (error s3)

Control at hydrophone, 500 Hz

Figure 5.42: Pressure reduction at the hydrophone in dB, 500 Hz (error s3)
frequency as it was closer to the ¼ wave tube resonant frequency. This can also be explained by the fact that for higher drive frequencies the stroke of the diaphragm connected to the end of the side-branch (refer Figure 5.1) would be less i.e. it would have to displace less fluid and it would thus face lower resistance, improving the efficiency of the actuator.

Thus the side-branch actuator tests yielded global control for each discrete frequency case. As the frequency of excitation was increased it was observed that better control was achieved i.e. the control authority of the active source increased. For each of the discrete frequencies there was no control spillover. Hence it can be concluded that the active control experiments were successful at single frequencies. This was an important result because pressure pulsations in fluid filled piping systems occur at harmonics of the pump operating frequency. The lowest pulsation frequency, present in the piping system onboard the Swedish marine vessel described by Maillard [65], was at 64 Hz. This is very close to the lowest frequency, i.e. 70 Hz, at which active control experiments were conducted using the side-branch actuator. This meant that the active control experiments performed as part of this thesis could be implemented on a practical piping system with positive displacement pumps provided that the side-branch actuator was redesigned to withstand high pressures and the active source could generate high enough pressure levels.

In many industrial systems, the piping is often subjected to broad band frequency disturbances. These disturbances can occur for a variety of reasons. For example bends in piping system appear as discontinuities to the fluid flow causing vortices in flow. These flow vortices can act as disturbance sources, which excite the fluid inside the piping
system over a broad band of frequencies. Hence it was necessary to test the effectiveness of the side-branch actuator for broad band frequency control. In most practical piping systems the frequency of excitation is very low compared to the ring frequency of the pipe [35]. The highest frequency tone generated the pump in the piping system described by Maillard [65] in his research report was at 248 Hz. Hence the it was decided to perform the active control experiments on the side-branch actuator in the range 0-500 Hz. Moreover at frequencies higher than 500 Hz conventional rubber dampers are efficient in controlling sound radiation from fluid-filled pipes [65]. The physical setup of the test rig was the same as that used for the tests at discrete frequencies and has not been repeated. However the control bandwidth on the controller and the error sensor filter was set from 0-500 Hz and the sampling frequency was \( f_s = 4 \times f_{\max} = 2000 \) Hz, where \( f_{\max} \) was the maximum frequency of interest. Sensor s3 was the error sensor again and the reference signal was the output of the accelerometer mounted on the excitation diaphragm at the shaker end.

Figures 5.35 to 5.40 show the auto spectrum levels at the sensors s1, s2, s3, s4, s5 and the hydrophone. The auto spectrum plot of the sensor s1 reveals that for the frequency range from 200-350 Hz there is an increase in the pressure levels after the control actuator was turned on. This could be due to the phenomenon of control spillover, which is the feedback of the secondary source to the primary disturbance path or the reflection of sound towards the source[71]. Similarly sensor s2 also shows an increase in pressure levels for the frequency range 90-300 after control is applied. This also might have occurred due to the feedback from the side-branch actuator to the sensor s2. The auto spectrum plot of the error sensor reveals an average reduction in pressure levels.
Figure 5.43: Broadband pressure reduction at sensor s1, 0-500 Hz (error s3)

Figure 5.44: Broadband pressure reduction at sensor s2, 0-500 Hz (error s3)
Figure 5.45: Broadband pressure reduction at sensor s3, 0-500 Hz (error s3)

Figure 5.46: Broadband pressure reduction in dB at s4, 0-500 Hz (error s3)
Figure 5.47: Broadband pressure reduction in dB at s5, 0-500 Hz (error s3)

Figure 5.48: Broadband pressure reduction in dB at hydrophone, 0-500 Hz (error s3)
between 100 to 300 Hz by 20dB. It can be recalled from Chapter 3 that an acoustic resonance was measured inside the test rig close to 150 Hz. The auto spectrum plot for sensor s3 shows a 30dB reduction in the resonant peak at this frequency. The auto spectrum plot for sensor s4 shows an average reduction in pressure levels between 120 and 220 Hz by 20 dB. In this case the acoustic resonance at 150 Hz was reduced by roughly 25 dB. The auto spectrum levels for sensor s5 show an average reduction of 10dB over the frequency range 120 to 250 Hz. In this case the acoustic resonance at 150 Hz was reduced by 10 dB after active control was applied. For the hydrophone an average 15dB reduction was recorded between 150 to 230 Hz. The acoustic resonance at 150 Hz was reduced by roughly 30 dB at the hydrophone. Thus the active control experiments reveal that side-branch actuator caused global reductions downstream of its location. Also the side-branch actuator was able to reduce the pressure levels at the acoustic resonance at 150 Hz. The actuator however created some unwanted disturbances, which lead to an increase in the pressure levels at sensors s1 and s2 i.e. the actuator caused control spillover [71]. However considering the fact that global reductions from 25-15dB could be achieved at all sensors and the capability of the actuator in controlling the n=0 acoustic resonance inside the test rig it can be concluded that the side-branch actuator tests were moderately successful for broadband frequency excitation. A similar side-branch actuator, which could withstand higher pressures and generate higher control sound levels could be built and tested on practical fluid-filled piping systems.
5.3 Active and active-passive control experiments using 1-3 piezoelectric composite actuator

In the previous section active control experiments were carried out using a side-branch actuator. It is difficult to install such an actuator on a practical piping system because the diaphragm part of the side-branch actuator might not be able to withstand the high fluid pressures prevalent inside the system. Thus it was decided to investigate active control experiments with a purely fluid-based actuator [14,35]. This was done using a 1-3 piezoelectric composite shown in Figures 5.18 and 5.19. The characteristic property of piezoelectric materials is that when subjected to stresses they produce electrical charge. Conversely when an electrical field is applied to such materials they change in shape and size. These properties of piezoelectric materials have led to their increased use as actuators and sensors in many applications [71]. Most of the current applications of piezoelectricity use polycrystalline ceramics instead of natural piezoelectric crystals since they are harder, denser, and can be manufactured to required shapes and produce higher strain rates. Hence these crystals are also referred to as composites since they are manufactured from a mixture of different crystals.

Figure 5.18 shows a schematic diagram of the 1-3 piezoelectric composite used for the current experiments. Three axes 1, 2 and 3 corresponding to $x$, $y$ and $z$ of the rectangular co-ordinate system are used to identify directions in the piezoelectric element. These axes are set by applying large d.c. voltages to the piezoelectric element for extended periods. Piezoelectric coefficients, which relate applied voltage and resultant strain are described by double subscripts e.g. $d_{13}$. The first subscript gives the direction of the electrical field associated with applied voltage and the second subscript gives the
Figure 5.49: Schematic representation of a 1-3 piezoelectric composite [71]

Figure 5.50: Picture of actual 1-3 piezoelectric composite used for active control experiment
direction of the mechanical strain. The 1-3 composite was constructed from piezoelectric rods embedded in a transparent urethane block, which had the same acoustic impedance as water, the fluid used in the experiments. A normal piezoelectric rod will shrink in cross section (due to Poisson’s coupling) as it expands and thus its net volume displacement is low. However if the rods are embedded in a urethane matrix then the composite Poisson’s ratio is small so the sides hardly shrink when the surface expands. As a result the net volume displacement is much higher than the rods themselves. The resulting device is a 1-3 composite. A 1-3 composite implies that the voltage is applied in the 1 direction and the resultant force or displacement is predominantly in the 3 direction. These piezoelectric actuators generate relatively large forces when their natural expansion is constrained. Thus such an actuator could be used as an active source to generate secondary pulses for an active control system installed on a fluid-filled piping system. Hence it was decided to use such an actuator for active control experiments in this thesis. The maximum voltage that could be applied to the piezoelectric composite was 530V_{rms}. It had a stable response up to 250Khz and a capacitance of 5.9nF.

Figure 5.20 shows a picture of the basic Helmholtz resonator with the 1-3 piezoelectric composite mounted inside (not visible). Figure 5.21 shows a schematic diagram of the piezoelectric actuator mounted inside the Helmholtz resonator and its principle of operation. It must be noted here that the quarter wavelength tube was not used for this experiment because the physical dimensions of the Helmholtz resonator made it easier to incorporate the 1-3 composite in it as compared to the quarter wave tube. As was discussed earlier applying an electrical voltage to a piezoelectric composite causes it to undergo physical strain. Now variation in the physical dimensions of the
Figure 5.51: Picture of Helmholtz resonator assembly with the 1-3 piezoelectric composite mounted inside it

Figure 5.52: Schematic diagram showing operation of 1-3 composite as secondary source
composite causes the fluid around it to be displaced generating pressure pulses inside the fluid. The principles of active control, which were discussed in section 5.1 of this chapter, could be applied to implement the piezoelectric composite as a secondary actuator. Thus if an electrical voltage were applied to the composite such that the pressure pulses generated by it were of the same amplitude but $180^0$ out of phase with the disturbance pressure pulsations at the error sensors then there would be cancellation of the primary pressure field. This was the physical principle behind implementing the piezoelectric composite for active control experiments.

The advantage of using the piezoelectric composite was that it was purely fluid based i.e. it could be mounted inside the piping system for generating pressure pulses. It did not need a diaphragm for exciting the fluid like the side-branch actuator, which could be difficult to design for a highly pressurized system. Hence the 1-3 composite could be incorporated in highly pressurized practical piping systems without involving complex piping design. Another very important advantage of the piezoelectric composite was that it generated fluid pressure pulses without exciting the pipe wall in any way. This was important exciting the fluid through the pipe wall can cause unwanted disturbances to propagate along the pipe wall and couple with the fluid leading to an increase in the pipe vibration [65]. A drawback of using the piezoelectric composite, as a secondary actuator was the high voltage required to drive the actuator (maximum drive voltage, $530 V_{\text{rms}}$). However compared to the advantages the actuator had, high voltage requirement seemed a small price to pay.
5.3.1 Active control experiment

Figure 5.22 shows a schematic diagram of the experimental setup for the active control experiments using the 1-3 piezoelectric composite actuator. The basic arrangement consisted of an upstream rubber damper section (to prevent transmission of structural vibrations), a test section, the downstream steel section and the anechoic termination (not shown). The upstream section was the rubber damper section described in section 3.1 of Chapter 3. The rubber damper reduced the presence of the n=1 structural waves on the rig. This was very crucial because as seen in Chapter 4 the structural waves dominated the rig response thus subduing the response of the fluid to excitation. In Chapter 4 it was seen that to get desired experimental results for fluid wave canceling devices like the quarter wavelength tube it was necessary to reduce the presence of the n=1 structural waves by incorporating the rubber damper on the test rig. Hence to observe the effectiveness of the 1-3 piezoelectric composite in reducing the fluid pressure levels it was important to install the rubber damper on the test rig to reduce the dominance of the n=1 structural waves. The test section was the basic Helmholtz resonator used for experiments in section 4.3.1 of Chapter 4. The 1-3 composite active actuator was mounted inside the resonator in the position shown in Figure 5.22 due to size constraints. The dimensions of the actuator were 4”×2” while the inner diameter of the Helmholtz resonator was 3”. Hence it was possible to accommodate the actuator only across the length of the resonator.

A PCB pressure sensor s1 was mounted at the junction of the steel tee to monitor the pressure at the mouth of the resonator. Sensors s2, s3 and s4 were mounted on the downstream steel pipe 1.84m, 2.1m and 2.45m from the shaker end. The hydrophone was
Figure 5.53: Rig setup for active control experiment using 1-3 piezoelectric composite.

- **Shaker**
- **Test Section**
- **Pressure Sensors**
  - S1
  - S2
  - S3
  - S4
- **Upstream section**
- **Piezoelectric actuator**
- **Downstream section**
- **Hydrophone (mobile)**
Figure 5.54: Plot showing that 1-3 composite generates coherent signals only at 1000 Hz and above
mounted between sensor s2 and the test section. The idea behind mounting the 1-3 piezoelectric composite inside the Helmholtz resonator was to test an active-passive system, which will be explained in detail in the next section. However initially the actuator could not be driven at frequencies below 1500 Hz i.e. the pressure sensors recorded a coherent signal only at 1500 Hz and beyond as shown in Figure 5.23. At lower frequencies the sensor signals were measuring electrical noise. The presence of bubbles inside the resonator was suspected. It should be noted that the resonant frequency of the Helmholtz resonator was 460 Hz, which was much lower than the frequency at which the 1-3 piezoelectric composite could be driven. Despite this drawback, it was necessary to find if active control experiments could be carried out successfully using the 1-3 piezoelectric composite. Hence a preliminary active control experiment were carried out at 1500 Hz. To this end the shaker was excited at a single frequency, 1500 Hz. An accelerometer was mounted on the diaphragm at the primary shaker end. This accelerometer output was used as the reference signal to the controller, which drove the piezoelectric actuator through a step up transformer (1:27 step up ratio). Care was taken not to exceed the voltage limit of the piezoelectric actuator ($530V_{\text{rms}}$) by constantly monitoring the voltage through a digital multi-meter connected to the power supply.

Two control experiments were conducted. In the first experiment only the sensor s1 was used as an error sensor while in the second experiment, s1 and s2 were used as error sensors. Figures 5.24 to 5.33 show the auto spectrum plots (dB scale relative to 1 V) at the sensors s1, s2, s3, s4 and the hydrophone before and after control for both the cases i.e. s1 as error sensor and s1, s2 as error sensors. Table 5.1 shows the results of both these experiments. The results for sensor s3 are not available since it appeared to be recording
<table>
<thead>
<tr>
<th>Sensors</th>
<th>dB reduction (s1 error sensor)</th>
<th>dB reduction (s1, s2 error sensor)</th>
</tr>
</thead>
<tbody>
<tr>
<td>s1</td>
<td>30</td>
<td>35</td>
</tr>
<tr>
<td>s2</td>
<td>4.5</td>
<td>35</td>
</tr>
<tr>
<td>Hydrophone</td>
<td>0.33</td>
<td>12</td>
</tr>
<tr>
<td>s4</td>
<td>0.05</td>
<td>12</td>
</tr>
</tbody>
</table>

Table 5.1: Comparison of pressure reduction in dB for active control experiment, 1500Hz
Figure 5.55: Autospectrum levels at sensor s1, 1500 Hz (error sensor s1)

Figure 5.56: Autospectrum levels at sensor s2, 1500 Hz (error sensor s1)
Figure 5.57: Auto spectrum levels at sensor s3, 1500 Hz (error sensor s1)

Control at sensor s3, 1500 Hz

Figure 5.58: Auto spectrum levels at sensor s4, 1500 Hz (error sensor s1)

Control at sensor s4, 1500 Hz
Figure 5.59: Auto spectrum levels at hydrophone, 1500 Hz (error sensor s1)

Figure 5.60: Auto spectrum levels at sensor s1, 1500 Hz (error sensors s1 and s2)
Figure 5.61: Auto spectrum levels at sensor s2, 1500 Hz (error sensors s1 and s2)

Figure 5.62: Auto spectrum levels at sensor s3, 1500 Hz (error sensors s1 and s2)
Figure 5.63: Auto spectrum level at sensor s4, 1500 Hz (error sensors s1 and s2)

Figure 5.64: Auto spectrum level at hydrophone, 1500 Hz (error sensors s1 and s2)
random noise as shown in Figure 5.26 and Figure 5.31. Possible reasons could have been a bad cable connection or electrical noise affecting the measurement channel.

For the case using a single error sensor, a reduction of 30 dB is recorded at the error sensor s1. The other sensors s2, s4 and the hydrophone record reductions of 4.5, 0.05 and 0.33 dB respectively. This showed that very good control was achieved at one sensor i.e. the error sensor s1. However very low reductions were recorded at the other downstream sensors. This could be due to the fact that the system identification that the controller performed using a single error sensor s1 was not a good representation of the physical system between the control actuator and the other monitoring sensors. The controller however modeled the path between the control actuator and the error sensor s1 very well. As a result though a 30dB reduction was recorded at s1 global reductions in pressure levels could not be achieved at the other sensor.

To get a better representation of the physical system between the control actuator and the downstream sensors, sensors s1 and s2 were used as error sensors. For the case with s1 and s2 as error sensors there was a reduction of 35 dB each at sensors s1 and s2. In this case the sensor s4 and the hydrophone recorded reductions of 12 dB each. Thus using two error sensors led to better reduction at the downstream sensors s2, s4 and the hydrophone (35 dB at sensor s2 and 12 dB at both sensor s4 and hydrophone) compared to the case using a single error sensor (0.05 dB at s4 and 0.33 at the hydrophone). This could be due to the fact that increasing the number of error sensors gave a better representation of the physical system i.e. the downstream section of the test rig, to the controller. This meant that the controller could model the path between the control actuator and the sensors better. As a result the controller could drive the secondary
actuator i.e. the 1-3 composite, much more effectively to cancel the disturbance pressure pulses which resulted in better overall dB reductions at all the sensors. Thus it could be concluded from these experiments that increasing the number of error sensors led to better global reduction of fluid pressure throughout the downstream section of the experimental rig.

5.2.2 Active-passive control experiment

Passive devices such as quarter wavelength tubes and Helmholtz resonators are effective only in a narrow frequency bandwidth. They lose their effectiveness if the disturbance frequency changes and is no longer within the resonant frequency range of the passive device [63]. Moreover other passive devices such as rubber isolators tend to stiffen under high fluid pressure and lose their effectiveness [65]. At very low frequencies (e.g. below 200 Hz) the wavelength of propagating fluid waves is very long. Passive devices such as quarter wavelength tubes, which act by reflecting fluid waves to effect wave cancellation at the mouth of the tube, become large at these frequencies. For example to cancel a fluid wave at 100, Hz a quarter wavelength tube 4m long would be needed. Hence such a device would be difficult to implement in practical piping systems. On the contrary a fully active approach theoretically possesses complete flexibility but in practice is limited by the actuator control authority and power levels. The actuator control authority can be described as the ability of the secondary actuator to drive the physical system effectively to similar levels to that of the disturbance field (i.e. in this case the ability of the 1-3 piezoelectric composite to drive the fluid inside the pipe). A control actuator with good control authority would be able to generate a secondary noise field
strong enough to cancel the primary disturbance noise field. Typically high voltages and thus high power levels are required to drive piezoelectric devices used as secondary actuators in active control experiments. For example the piezoelectric stacks Maillard [65] used for active control experiments needed voltages of 800 V$_{p-p}$ to drive them effectively.

This situation tends to suggest that a combined active-passive approach might have possibilities in overcoming the drawbacks of these individual approaches (i.e. active and passive control) and harness their advantages. A passive device could be built with the secondary actuator incorporated in it and driven at the resonant frequency of the passive device. Theoretically at the resonant frequency of the passive device there should be a reduction of pressure in the system. The control actuator would have to attenuate reduced system pressures now and hence the voltage/power required to drive the control actuator would be lower. Another reasoning is that it is easier to excite a system close to its resonant frequency. Much larger vibration levels could thus be obtained with much lower control voltages than in a non-dynamic system. Thus a combined active-passive approach would possess advantages of both active and passive methodologies i.e. control flexibility as well as reduced power requirements, eliminating disadvantages of both these approaches.

A similar approach was adopted here by using a combination of a Helmholtz resonator and an active 1-3 piezoelectric composite actuator. In the previous section the experiments were carried out using a piezoelectric composite mounted inside a Helmholtz resonator. However the active control experiments were carried out at 1500 Hz whereas the resonant frequency of the Helmholtz resonator was approximately 460
Hz. Thus the control frequency was not in the range of the resonant frequency of the Helmholtz resonator. Hence the reduction of pressure levels at the pressure sensors was purely due to the action of the 1-3 piezoelectric composite and not due to a combined effect of the Helmholtz resonator and the 1-3 piezoelectric composite (i.e. the experiments in the previous section were purely active control experiments). In this section however, the 1-3 piezoelectric composite mounted inside the Helmholtz resonator was driven at or near the resonant frequency of the Helmholtz resonator. As explained earlier in this section, it was desired to be seen if there was a reduction in the electrical power required to drive the 1-3 composite actuator by exciting it at the resonant frequency of the Helmholtz resonator. Hence the experiments performed in this section are referred to as active-passive control experiments.

Figure 5.34 shows a schematic diagram of the experimental setup used for the active-passive control experiments. Figures 5.35 and 5.36 show the pictures of the experimental rig and the equipment used for the tests. From Figure 5.34 it can be seen that the basic arrangement of the rig was the same as the previous section. However there was a change in the orientation of the Helmholtz resonator and the position of the piezoelectric actuator inside. Figure 5.37 shows a diagram of the actuator orientation inside the Helmholtz resonator. As shown in Figure 5.37 the Helmholtz resonator was positioned upside down to remove air bubbles through a brass fitting on top of the resonator. As was discussed in section 5.2.1 the dimensions of the 1-3 composite actuator prevented it from being mounted in the most efficient position. The strain direction for the actuator was in the 3-direction (z-axis) as discussed in section 5.2. Hence the best position was to mount the actuator such that its x y-plane was parallel to plane of the
resonator mouth as shown in Figure 5.36. In this position the pressure pulses generated by the 1-3 composite actuator would directly travel to the mouth of the resonator instead of being reflected from the resonator walls which was the case for section 5.2.1 as shown in Figure 5.38. Thus the actuator would be able to control the pressure pulses generated by the primary disturbance much effectively in this new position as it was able to generate stronger pressure pulses at the mouth of the resonator compared to the earlier configuration. The dimensions of the 1-3 actuator was reduced to fit the Helmholtz resonator in the new position by removing some of the urethane (refer Figure 5.19) at the edges of the 1-3 actuator. The rubber isolator (see section 4.1) was used as the upstream section to prevent transmission of structural vibrations as previously. The pressure sensor s1 was mounted on the test section while sensors s2, s3, s4 and s5 were mounted on the downstream steel pipe 1.84m, 2.1m, 2.45m and 2.75m from the shaker end respectively as shown in Figure 5.34. The internal hydrophone was mounted at approximately the same location as pressure sensor s2.

Before the active-passive control experiments could be done, it was necessary to determine at what frequencies the Helmholtz resonator was acoustically resonant. Once these frequencies were determined, active control could then be carried out at or near these frequencies to reduce the electrical power (i.e. control voltage) required for driving the control actuators. To determine these frequencies, the 1-3 piezoelectric composite positioned inside the Helmholtz resonator was driven with a broadband random signal from 0-1000 Hz and the pressure sensor and hydrophone signals recorded. Figure 5.38 shows the auto spectrum levels (relative to 1V) of the pressure sensors and hydrophone in dB. Two resonances at 135 Hz and 480 Hz can be seen clearly in these plots. The
Figure 5.65: Schematic diagram of rig for active-passive control experiment.

- **Shaker**
- **Test Section**
- **Piezoelectric actuator**
- **Pressure Sensors**: S2, S3, S4, S5
- **Upstream section**
- **Downstream section**
- **Hydrophone (mobile)**
Figure 5.66: Picture of active-passive element on experimental rig

Figure 5.67: Instruments used for active-passive control experiments
Figure 5.68: New Helmholtz resonator and 1-3 actuator positions

Figure 5.69: Helmholtz resonator and 1-3 actuator positions in previous section
theoretical resonance of the Helmholtz resonator was 460 Hz as calculated in Chapter 4. The other resonance at 135 Hz is much more dominant and it was an axial acoustic resonance inside the test rig as discussed in Chapter 3. Active control experiments were carried out at these resonances, i.e. 135 Hz and 480 Hz, as well as at a non-resonant frequency, 400 Hz.

As was discussed earlier it was desired to see if there was a reduction in the control voltage required to drive the 1-3 actuator at the resonant frequency of the Helmholtz resonator. Hence a control experiment was carried out at 480 Hz. Experiments also had to be carried out at a non-resonant frequency to compare the control voltage with the case where it was driven at the resonant frequency of the Helmholtz resonator. An arbitrary non-resonant frequency of 400 Hz was chosen for these experiments. The active control experiments at 135 Hz were carried out solely to observe what would be the control achieved at the acoustic resonance of the test rig. In these tests the excitation signal supplied to the disturbance shaker was used as the reference signal to the controller and of transducers s2 and s3 were used as error sensors. The control voltage supplied to the piezoelectric actuator was monitored constantly for each of these experiments while the excitation voltage was kept constant.

Table 5.2 shows the reduction in dB and the control voltage required in rms volts for the experiments at each frequency. It can be clearly seen that for the most dominant acoustic resonance at 135 Hz the control voltage required (approximately 100 V) is the least and the overall pressure reduction at all pressure sensors is the greatest. This could be due the fact that at 135 Hz there was an acoustic resonance inside the test rig and as
Figure 5.70: Autospectrums of pressure sensors and hydrophone

Broadband excitation of control actuator mounted inside resonator

Autospectrum levels of the sensors in dB

Frequency (Hz)

Autospectrum levels of the sensors in dB

Hydrophone
S2
S3
S4
S5
<table>
<thead>
<tr>
<th>Reduction at 480 Hz, dB Error Sensors, S2 &amp; S3</th>
<th>Control Voltage V rms</th>
<th>Hydrophone</th>
<th>S1</th>
<th>S2</th>
<th>S3</th>
<th>S4</th>
<th>S5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>240</td>
<td>-15.4</td>
<td>37.7</td>
<td>-37</td>
<td>-26.2</td>
<td>-26</td>
<td>-20</td>
</tr>
<tr>
<td>Reduction at 135 Hz, dB Error Sensors, S2 &amp; S3</td>
<td>100</td>
<td>-34.2</td>
<td>-14.6</td>
<td>-43.3</td>
<td>-37.5</td>
<td>-31.1</td>
<td>-16.5</td>
</tr>
<tr>
<td>Reduction at 400 Hz, dB Error Sensors, S2 &amp; S3</td>
<td>610</td>
<td>-3.8</td>
<td>34</td>
<td>-4.88</td>
<td>-5.7</td>
<td>-5.9</td>
<td>-5.33</td>
</tr>
</tbody>
</table>

Table 5.2: Control voltage and pressure reduction for active-passive control experiment
such it would be easier for the 1-3 actuator to generate pressure pulses at this frequency (i.e. at 135 Hz the 1-3 actuator needed less voltage to drive the system). Thus the 1-3 actuator had better control authority and was able to effectively reduce the pressure levels at all the sensors.

At 480 Hz the control voltage at 240 V is higher and the overall reduction is less compared to the experiment at 135 Hz. This could be due to the fact that at 480 Hz, there was no acoustic resonance inside the test rig. Hence the 1-3 actuator needed higher drive voltages to generate pressures inside the test rig at this frequency. However the drive voltage required at 480 Hz was much less than that required at the non-resonant frequency 400 Hz. This could be attributed to the fact that at 480 Hz (i.e. the resonant frequency of the Helmholtz resonator) the pressures inside the test rig was low since most of the incident wave at that frequency is reflected (refer section 4.3). Also as discussed earlier in this section it is easier to excite a system at a frequency close to resonance. Thus much higher vibration levels are obtained with much lower control voltages than in a non-dynamic system. At this frequency the voltage required to drive the 1-3 actuator was less compared to that at the non-resonant frequency 400 Hz i.e. the 1-3 actuator had better control authority at 480 Hz compared to 400 Hz and hence better reductions in pressure level could be obtained at the pressure sensors. The maximum control voltage at 610 V was recorded for the 400 Hz case with not much reduction in pressure at the sensors. This was due to the fact that 400 Hz was a non-resonant frequency. Hence at this frequency the 1-3 actuator had to be driven harder to generate pressure pulses inside the test rig (i.e. the 1-3 actuator had poor control authority).
Thus it could be concluded that at resonant frequencies of the passive devices the power requirement for the control actuator is much less than at non-resonant frequencies. A combined active-passive approach for reduction of fluid pulsations in piping systems appeared to be a viable option as long as the passive dynamics has resonances near the frequencies to be controlled.
Chapter 6

Summary and conclusions

The primary objective of this work was the investigation, evaluation and development of an effective system to for reducing pressure pulsations due to pumps in fluid filled piping systems. These fluid pulsations couple with the pipe wall and excite the pipe and structures connected to it leading to noise radiation from the piping system. Eventually this system is to be implemented on the target system, which was piping system onboard a Swedish marine vessel [65]. Different passive, active, and active-passive systems were built and tested to achieve this goal. The following is a summary of the work performed and main conclusion found towards achieving this goal.

An experimental rig was built from 2” industrial steel pipe for carrying out tests on the vibration and pressure reducing devices. The rig was then filled with water, which was the working fluid in this case. An electromagnetic shaker was used to generate acoustic disturbances in the fluid similar to a positive displacement pump in an actual piping system. It should be however be noted that the water inside the experimental rig was under atmospheric pressure unlike actual piping systems. Two anechoic terminations were built and tested for their effectiveness in preventing reflections on the experimental rig. An experimental method for circumferential wave decomposition and power flow measurement developed by de Jong and Verheij [39] was implemented for the characterization of the experimental rig.
The presence and importance of the n=1 structural waves on the experimental rig despite a fluid based excitation was demonstrated. The possible reasons for the generation of structural waves could be due to asymmetric excitation by the shaker or structural discontinuities on the experimental rig as described in section 3.1.2 of Chapter 3. Different methods of reducing the structural waves were developed and tested successfully. One of them is a rubber isolator whose performance was studied using the methods developed by de Jong and Verheij [39]. The analysis of the performance revealed that the rubber isolator caused the maximum attenuation for the n=1 structural waves. Even among the n=0 axisymmetric modes, the damper caused more attenuation for the torsional and extensional waves which were structurally transmitted compared to the n=0 fluid wave. This showed that the rubber isolator caused impedance reflection of the structural waves back to the source but did not have any effect on the n=0 fluid wave. Thus rubber isolators could be used to prevent structural wave transmission in fluid-filled pipes.

The degradation in performance of quarter wavelength tube due to the structural waves has been demonstrated. The improved results for the quarter wavelength tube tests after implementing the rubber isolator has been shown. This proved the importance of structural waves in fluid filled pipe experiments. Experiments using the Helmholtz resonator showed that at the resonant frequency of the Helmholtz resonator the incident fluid wave was reflected at the mouth of the resonator causing reduction in pressure levels inside the test rig downstream of the Helmholtz resonator. The experiments using the tunable Helmholtz resonator did not give the expected results however. This could have been either because of air bubbles or the omnipresent n=1 structural waves. Air
bubbles trapped inside the rig and structural waves were the biggest obstacles in characterizing the performance of passive devices, which made their good performance, still seem doubtful.

Active control experiments using the side branch actuator carried out at three different frequencies 70, 150 and 500 Hz showed dB reductions from 15-30 dB. Broadband experiments from 0-500 Hz showed dB reductions from 10-20 dB at the pressure sensors. This is especially important because many practical systems are subjected to broadband frequency disturbances. Active control experiments carried out at 1500 Hz using the fluid based 1-3 piezoelectric composite also demonstrated better control authority by increasing the number of error sensors.

Active-passive control experiments were conducted using a combination of the Helmholtz resonator and the 1-3 piezoelectric composite. These experiments showed that at the resonant frequency of the passive device the control voltage and hence the power required to drive the piezoelectric actuator decreased and there was increased pressure reduction at the pressure sensors. Thus a combined active-passive approach seemed very feasible as it had the advantages of both active and passive methods and at the same time eliminated their disadvantages.

The important observations and main conclusions that could be drawn from the experiments performed as a part of this thesis have been presented as follows:

• One of the most important observation and conclusion that could be drawn from the experiments performed was that the presence of \( n=1 \) structural bending waves dominated the rig response. It was necessary to eliminate the structural waves in order
to obtain good experimental results to demonstrate the effectiveness of the fluid pressure reduction devices (quarter wavelength tube and Helmholtz resonator). The n=1 structural waves could be generated either by asymmetric excitation at the shaker end or due to a discontinuity on the test rig. Employing a diaphragm actuator described in Chapter 3 eliminated the structural wave generation due to asymmetric excitation. Employing a rubber isolator thus prevented the transmission of structural waves on the test rig.

- The tests on the rubber isolator showed that it was more effective in attenuating the structural n=1 bending and n=0 torsional and extensional waves as compared to the n=0 fluid wave. Thus rubber isolators could be employed to reduce vibration levels in structurally excited piping systems.

- The quarter wavelength tube tests produced pressure reduction at the mouth of the resonator and downstream of it at the resonant frequency. It is important to note here that these results were obtained only after the rubber isolator was incorporated on the test rig highlighting the importance of reducing structural wave dominance to obtain good experimental results. Thus a quarter wavelength tube could be implemented on a piping system which was excited by a source at a constant frequency.

- Helmholtz resonator experiments showed reduction in pressure levels at the resonant frequency of the resonator. In this case similar to the quarter wavelength tube tests expected results were obtained only when the rubber isolator was incorporated on the test rig once again emphasizing the importance of reducing structural dominance on the test rig. Hence a Helmholtz resonator could be incorporated on a piping system
excited by a source whose frequency was same as the resonant frequency of the Helmholtz resonator.

- The adaptive Helmholtz resonator tests did not give the desired change in resonant frequencies with a change in valve positions. The small neck area or bubbles inside the resonator could be a possible reason. The volume of the resonator should be varied instead of the neck area since this would have a greater effect on the resonant frequency than varying the small neck area.

- Active control experiments using the side-branch actuator showed dB reduction from 5 dB at 70 Hz to 30 dB at 500 Hz. Reductions of 10-20 dB were recorded for broadband excitation. Thus the side-branch actuator showed promise in controlling low frequency as well as broad band excitation. With improved design such that the diaphragm end of the actuator could withstand high pressures, the side-branch actuator could be installed in a practical piping system.

- Active control experiments at 1500 Hz using the 1-3 composite actuator showed that using more error sensors provided a better estimation of the physical system of the controller which enabled it to achieve better pressure reductions. Thus increasing the number of error sensors led to better global reduction of fluid pressure inside the test rig.

- Active-passive control experiments using a combination of the Helmholtz resonator and 1-3 piezoelectric composite actuator showed that, at the resonant frequency of the Helmholtz resonator, the actuator required less drive voltage compared to that required when it was driven at a non-resonant frequency. The dB reduction in pressure was also more when the actuator was driven at the resonant frequency. It can
be concluded that a combined active-passive approach led to better control authority and reduced power levels for the control actuator. Thus a combined active-passive approach appeared to be feasible for application on a practical fluid-filled piping system.

In all these experiments the working fluid (water in this case) was at atmospheric pressure and stationary. In practical piping systems the fluid is under high pressure and flowing, which can degrade the performance of all the elements tested in the course of this work. Flow in practical piping system occurs at very low Mach numbers and typically would not affect the characteristics of a piping system [17]. However at high pressures conventional vibration absorbers like the rubber isolator used in this thesis, tend to stiffen making them ineffective as structural vibration absorbers [65]. Hence as future research work these devices should be implemented on a real piping system with a positive displacement pump as the prime mover and the fluid under high pressure. An adaptive Helmholtz resonator with a tunable resonant frequency, which can be achieved by varying the volume of the resonator, should be built and tested. A more detailed analysis of the active-passive device (incorporating the piezoelectric actuator inside the Helmholtz resonator) should be carried out for the optimal design of these elements.
References


Vita

Satish Kartha was born on September 30th, 1974 near the beautiful port of Cochin, India. He was brought up in the historical city of Baroda, India. He finished his undergraduate studies in Mechanical Engineering from the Maharaja SayajiRao University of Baroda in June 1996. He worked with a multinational company Chicago Pneumatic Tools Ltd. from July 1996 to June 1997 where he developed an interest in the field of vibrations and acoustics. In August 1997 he enrolled as a graduate student at Virginia Tech. He worked on his Masters degree in the Vibration and Acoustics Laboratories under the guidance of Dr. Chris Fuller. He has been working with Macrosonix Corp. in Richmond, Virginia since August 1999.