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In-Situ Modal Response Characterization of Pipe-Structures Through Reynolds Number Variation

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IN-SITU MODAL RESPONSE CHARACTERIZATION OF PIPE-STRUCTURES THROUGH
REYNOLDS NUMBER VARIATION

by

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A THESIS

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Abstract

In this investigation, an in-situ method of system excitation is explored experimentally. The modal characteristics of externally-supported pipe structures are investigated by varying the flow Reynolds number (Re_d) with hydrodynamic pressure fluctuations due to fully developed turbulent pipe-flow providing a varying excitation source on the internal pipe wall.

During experiments, time series records of single-point fluctuating wall pressure and multi-point wall vibrations are collected. Power spectral density functions of both wall pressure fluctuation and wall vibration are computed at each discrete Reynolds number. Visualization of the computed power spectral density functions with flow Reynolds number are then used for system characterization.

A comparative analysis of the data sets collected for both acrylic and ABS pipe show that the pressure spectra are similar, while the vibration spectra change significantly. Pressure spectra reveal a character whereby the magnitude of the spectra increase with increasing Reynolds number. A comparison of in-situ results to those obtained using traditional impact response tests show that the vibration spectra collected through Reynolds number variation successfully capture the modal characteristics of the pipe-structure. Both acoustic analysis to determine the vibration source and preliminary health diagnostics investigations for both loss-of-fluid and loss-of-material events are performed.

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Chapter One: INTRODUCTION

1.1 Background Introduction

Oil and gas are important energy sources for Canada, and in 2016 these sources represented over 50% of total energy usage [1]. In 2016, Canada produced 218.2 million tonnes of oil of which 100.9 million tonnes were consumed internally, and it produced 152 billion cubic metres of natural gas of which 99.9 billion cubic meters were consumed internally [1]. Energy pipelines currently are the safest method of transportation of oil and gas products from production fields to refineries and petrochemical plants. According to the Canadian Energy Pipeline Association (CEPA), 99.99% of oil and gas products were transported safely through their member pipelines in 2017. Therefore, it is clear that pipelines represent one of the most important industrial systems influencing both the national economy and people's livelihood.

Modal analysis represents an efficient method by which the state of a structure, including pipe structures, can be monitored so as to further ensure a long and safe service life. Modal analysis is significant given that it can be extended and employed in many engineering applications including the verification of analytical models, the identification of vibration and acoustic source problems, structural modification and sensitivity analysis and vibration-based inspection techniques. Given that mechanical damage to a structure, such as through cracking, corrosion, erosion, or denting can cause changes to mechanical properties that include mass, damping and stiffness, the vibrational modes (modal characteristics) of the structure will change. This forms the basis for this thesis work.

Modal analysis can be divided into two categories, model-based and experiment-based. Given that model-based analysis requires significant time to both develop and run models, with the advent of digital Fast Fourier Transform (FFT) spectrum analyzers in the 1970's,

experimental modal analysis attracted more widespread adoption [2]. Most often, impulse response tests (bump testing) or shaker tests represent the methods by which the modal characteristics of a structure are determined. However, these tests are more applicable for the laboratory environment given they both require a known input where the structure needs to be temporarily separated and excited by only either a hammer or a shaker. As a result, experimental modal analysis has not seen widespread adoption for large installed structures including bridges, offshore platforms or pipelines located either in the ocean or buried underground.

In the 1970's [3], modal characteristic extraction from a structure in an ambient environment under operating conditions started to be investigated, and these methods are called Ambient Modal Identification (AMI) or Operational Modal Analysis (OMA). In this test method, a structure can be subjected to a variety of excitation sources that are not measured but rather are assumed to be "broadband random." Until now, this technique has been applied to large civil structures such as bridges [4] [5] and offshore structures [6].

As for pipeline applications, over the last two decades some vibration-based investigations (mainly inspection techniques) have been performed involving pipes without fluid [7] [8] [9] or with internal pressurized fluid but the fluid not moving [10] [11] [12]. Therefore, in order to better understand the actual modal characteristics of a pipe under operational conditions, the research reported in this thesis investigates pipe structure modal response with internal flow at varying Reynolds numbers.

1.2 Research Objectives

The main objective of this research is to investigate the modal characteristics (dynamic response) of a pipe under its operational conditions (Operational Modal Analysis). The objectives can be summarized as follows:

1. Investigate the pipe wall pressure fluctuations at varying Reynolds numbers, which represents the unmeasured excitation source.
2. Collect pipe vibrations at different spatial points and by varying Reynolds number. This is referred to flow rate sweep tests in the in-situ modal characterization method investigated. Both acrylic and ABS pipe material are used to quantify their modal characteristics. Suitable analysis methods are used to interpret the resulting data.
3. Conduct impulse response tests at the same spatial points and compare with the flow rate sweep test results.
4. Apply the method to pipeline health monitoring for loss of fluid and loss of material events.

1.3 Thesis Outline

This thesis is organized in the following manner. Chapter 2 presents a brief literature review on pressure fluctuations for wall-bounded shear flow and turbulent boundary layer (TBL) induced vibration of pipe structures. Chapter 3 describes the experimental apparatus and details of the experiments performed including flow rate sweep tests, impulse response tests, phase information investigations and applications involving loss-of-fluid and loss-of-material detection. Chapter 4 presents the experimental results and discussion. Chapter 5 presents the conclusions and future work for this thesis work. Appendix A provides some relevant descriptions about the experimental equipment.

Chapter Two: LITERATURE REVIEW

2.1 Chapter Overview

In this chapter, a brief literature review is given about the pressure fluctuations beneath a fully-developed turbulent boundary layer (pipe wall) and its induced vibration on a pipe structure. Previous investigations into turbulent boundary layer pressure fluctuations will be reviewed including discussion about experimental methods and data processing techniques. As for pipe vibrations induced by turbulent pipe flow, the classification of pipe vibration and mechanism studies including investigation development will be reviewed.

2.2 Turbulent Pressure Fluctuations for Wall-Bounded Shear Flows

2.2.1 The investigated history

The character of pressure fluctuations along the wall of a turbulent boundary layer (TBL) is a research topic that has been studied for over fifty years and has been driven by a number of important engineering applications. The study of turbulent wall pressure fluctuations began in the 1950s, with early investigations being somewhat restricted by the experimental measurement systems available at that time for both sensing wall pressure fluctuations and examining the signals. In 1951 Batchelor [13] was one of the first to investigate turbulent pressure fluctuations, but for the restrictive case of isotropic turbulence. A few years later the first investigations of pressure fluctuations in turbulent boundary layers were performed, with Kraichnan [14] in 1956 investigating the wall pressure fluctuations on a flat plate, and Willmarth [15] and Bakewell et al. [16] investigating pressure fluctuations on the wall of a wind tunnel in 1956 and 1962, respectively. With the increasing growth of commercial aviation during this period, an increasing number of researchers started to study specific aspects of wall pressure fluctuations due to turbulent boundary layers. In 1962 Willmarth et al. [17] investigated wall pressure fluctuations

beneath thick turbulent boundary layers in a wind tunnel where boundary layer thickness was controlled by altering wall roughness. In 1964 Corcos [18] considered an array of points along the wall and studied the turbulent pressure field in attached turbulent shear flows. Interest was also growing for applications involving semi-rigid walls, with Dinkelacker [19] conducting a preliminary investigation of the influence of flexible walls on the flow-induced pressure field, and Robert et al. [20] investigated wall pressure spectra in tube flow with stenosis for biomedical applications.

In 1975 Willmarth [21] wrote the first review paper on pressure fluctuations on a wall due to a turbulent boundary layer, and two decades later the body of knowledge was updated in a review article by M. K. Bull [22]. Given the wide body of knowledge that currently exists in this field and the experimental nature of the work performed in this thesis, the literature review in this thesis work will mainly focus on experimental investigations of wall pressure fluctuations in both wall-bounded shear flow and turbulent pipe flow.

2.2.2 The mechanism investigation

Understanding how fully-developed internal pipe flow, specifically the structure of turbulent pipe flow, induce pipe-wall pressure fluctuations and in turn how these pressure fluctuations induce pipe vibrations is central to the current investigation. In reality, this is a very complex topic involving fluid mechanics, structural mechanics and their fluid-solid interactions [23], as well as the acoustic radiation produced by both the turbulent flow and the vibrating pipe, in addition to background facility noise [19] [23] [24].

Pressure fluctuations in turbulent pipe flow have been investigated both theoretically and experimentally, but the latter goes deeper than the former. Essentially, the inertia force in a fluid is proportional to the fluid density and the square of the fluid velocity. When the inertia forces in

the fluid are relatively large in comparison to the viscous forces, the viscous forces are not able to suppress the small velocity disturbances that exist along streamlines in a flow. As a result, these small velocity disturbances grow, and through a non-linear growth process the fluid becomes chaotic and eventually turbulent eddies are generated, at which point the velocity contains fluctuations that are no longer small (usually less than 2% the mean velocity) but can be significant (from a few percent up to around 10% the mean velocity).

According to the continuity equation in viscous incompressible flow (divergence-free), we have

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_x)}{\partial x} + \frac{\partial(\rho v_y)}{\partial y} + \frac{\partial(\rho v_z)}{\partial z} = \frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} + \frac{\partial v_z}{\partial z} = \nabla \cdot \vec{u} = 0 \quad 2.1$$

where ∇ is the nabla operator (divergence operator). The Navier-Stokes Equations (NSE) can also be expressed in the form of both nabla (∇) and Laplacian operators ($\Delta = (\frac{\partial^2}{\partial^2 x} + \frac{\partial^2}{\partial^2 y} + \frac{\partial^2}{\partial^2 z})$).

$$\nabla \cdot \left(\frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} \right) = \nabla \cdot \left[-\nabla \left(\frac{p}{\rho} \right) + \nu \Delta \vec{u} + f \right] \quad 2.2$$

The LHS of equation 2.2 can be simplified to:

$$\nabla \cdot \left(\frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} \right) = \frac{\partial}{\partial t} (\nabla \cdot \vec{u}) + \nabla \cdot [(\vec{u} \cdot \nabla) \vec{u}] = \nabla \cdot (\vec{u} \cdot \nabla \vec{u}) \quad 2.3$$

and the RHS of equation 2.2 can be simplified to:

$$\nabla \cdot \left[-\nabla \left(\frac{p}{\rho} \right) + \nu \Delta \vec{u} + f \right] = \nu \Delta (\nabla \cdot \vec{u}) - \Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f = -\Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f \quad 2.4$$

By rearranging these equations (2.2, 2.3 and 2.4), the pressure field can be expressed as:

$$\Delta \left(\frac{p}{\rho} \right) = \nabla \cdot f - \nabla \cdot (\vec{u} \cdot \nabla \vec{u}) \quad 2.5$$

Given the definition for the vorticity vector, $\vec{\omega} \equiv \nabla \times \vec{u}$, equation 2.5 can be further rewritten as:

$$\Delta\left(\frac{p}{\rho}\right) = \nabla \cdot f - \frac{1}{2}\Delta(\vec{u} \cdot \vec{u}) + \vec{u} \cdot \Delta\vec{u} + \vec{\omega} \cdot \vec{\omega} \quad 2.6$$

The detailed derivation of Equation 2.6 can be found in Appendix B. Equation 2.6 is known as the Pressure Poisson's Equation (PPE), showing that mean pressure is related to both the body force and the mean velocity field, and given that body forces typically do not change with time, the pressure fluctuations are related to turbulent velocity fluctuations. Because the turbulent velocity field cannot be determined theoretically, and pressure fluctuations at one point on the wall will be affected by velocity fluctuations in the neighbouring region above the wall [25], the wall pressure field varies irregularly at any instant. Therefore, direct pressure fluctuation measurements on the pipe wall beneath a boundary layer are often analyzed using statistical analysis techniques. These statistical analysis techniques include overall RMS pressure [15] [16] [18] [20] [26], frequency spectra [19] [27] [28], wavenumber spectra [29], normalized spectra [16] [26] [30], space-time correlations parallel to the streamline and transverse to the streamline [18] [26] [27], and space-frequency correlations [30].

Over the last few decades, many researchers have made significant contributions towards developing an understanding for the inner mechanism of the wall pressure fluctuation. Willmarth [15] reported that the ratio of the RMS of the wall pressure (fluctuations) to the free stream dynamic pressure is a constant (0.0035) over the Reynolds number range 1.5×10^6 to 20×10^6 and Mach number range 0.2 to 0.8. Other investigators reported a slightly higher constant of 0.01 [16] [26]. In reality, the measurement results will be affected by both Reynolds number and surface roughness [16]. This does, however, illustrate that the intensity of the pressure fluctuation is relatively small in comparison to the free stream dynamic pressure. The presence of the wall tap leading to the pressure transducer and other facility-related noise will affect the

characteristics of the pressure fluctuation signal. Assessing the magnitude of the pressure coefficient can assist in examining if these other factors are negligible or not. Dinkelacker [19] performed a series of experiments in pipe flow facilities and found that the wall pressure fluctuation spectra in turbulent pipe flow will be affected by the pipe material, the number of experiments performed (repeatability), flutter of the pipe, air bubbles in the flow and the existence of junctions (referred to as wall steps) between pipe sections. For a better comparison, the non-dimensional power spectral density function can be used to calculate the mean square pressure fluctuation in each frequency band and this can be plotted against Strouhal number [16] [26]. A particular region of the power spectral density function, known as the linear overlap region, can be fitted with a $-7/3$ slope [30]. Willmarth [17] found that the pressure fluctuations on a rough surface increased at all frequencies by the ratio of the mean-square pressure measured on rough surfaces to that on smooth surfaces. Jonathan [27] investigated the wall pressure spectrum beneath rough-wall turbulent boundary layers and found that increasing wall roughness will cause the spectra to rise in the low frequency region, reducing the range of the linear overlap region. Comparing to experiments performed by J.M.Clinch [26] investigating the wall pressure field of a smooth-walled pipe (tolerance of 0.002 in) with fully-developed turbulent flow at various Reynolds numbers, the maximum point in the spectra is found at the lowest frequencies with spectral magnitude then decreasing with increasing frequency.

2.2.3 Recent development of the pressure fluctuation investigation

2.2.3.1 Experimental methods

Some articles can be found reporting on investigations of methods to improve the experimental measurements, i.e. reducing noise, improving the experimental design and investigating the effect of transducer size on measurements. J.M.Clinch [26] used a low-pass

acoustic filter in experiments involving a 2000 gallon pressure vessel with a rubber hose connected to each side of a pipe flow facility in order to suppress pump-induced acoustical noise from propagating both upstream and downstream. The low-pass acoustic filter had an upper cut-off frequency of 10 Hz and was able to reduce overall pump noise by around 30dB. William et al. [29] used a large reservoir at positive elevation to provide sufficient hydrostatic pressure to drive water through a test section, thereby removing pump noise. Durant [23] employed acoustic mufflers on either side of a test section and accelerated the flow gradually using several contraction sections in an effort to minimize acoustic contamination and ambient noise. Willmarth [15] [21] found that the wall pressure fluctuation will attenuate due to the finite size of a pressure transducer mounted on the pipe wall, and he pointed out that the transducer should be sealed to produce a smooth surface without gaps in order to prevent resonance peaks in the high frequency region. McKeon et al. [31] reported that a measurement error will be incurred due to the size of a pressure tap (an open hole in the wall used to measure pressure) and they developed a corresponding correction term that is a fraction of the pipe wall stress. Based on their experiments, the pressure tap should have a length-to-diameter ratio larger than two and the tap-diameter to pipe-diameter ratio should be small. In addition, sensor calibration techniques can be found in the literature, such as a sensor switching technique [32] involving switching locations of a sensor pair to determine calibration coefficients.

2.2.3.2 Data processing methods

Some techniques have been developed to post-process pressure fluctuation data. J.Y.Chung [33] used a coherence function method which would also be applicable to pipe systems involving processing of the output from three transducers. The method was able to reject interference noise caused by flow over a transducer under the assumption that the flow noise

local to each of the three transducers was mutually uncorrelated. This method rejects the local turbulence signal and captures the facility noise that is in common to all three transducers. Once isolated, facility noise can then be removed from the power spectra at each transducer resulting in a turbulence spectra that is due to the flow itself.

The problem of low-frequency signal contamination is due to either an acoustic mode or a structural vibration [24], or often times both. Durant et al. [23] and Horne et al. [24] utilized a Wiener filter to remove peaks due to an acoustic mode that occurred in the low-frequency range by processing signals from two wall-mounted pressure transducers separated in the span-wise direction where the acoustic noise field is more coherent than the turbulence [29]. R.Camussi et al. [30] used the spectral conditioning technique proposed by M.Carley et al. [34] to reduce background noise in the acquired pressure signals. The technique was found to increase the signal-to-noise ratio by removing the coherent background noise from the pressure fluctuation frequency spectra. William et al. [29] used several groups of sensor arrays to measure pressure fluctuation signals and remove unwanted noise using a conditional spectral densities processing technique [35]. These above mentioned techniques use multiple sensors to acquire data from different spatial locations, and distinguish them based on their coherence. The desired and uncontaminated pressure fluctuation data is only correlated over a very limited spatial region [29] [35] due to the relatively small and chaotic eddies present in the turbulent boundary layer, while the noise sources that propagate from the pump can have coherence lengths that last for a few meters, depending on the bandwidth of the noise source and the type of fluid.

2.3 Response of Pipe Structure to Turbulent Pipe Flow

2.3.1 Classification of pipe structure vibration

Vibration processes in pipe flow can include both external and internal flow-induced vibration. Configurations involving external flow induced vibration often consist of vortex-shedding-induced vibration and fluid-elastic instability. Long, flexible piping systems exposed to ocean currents [36], such as in marine risers [37], will vibrate due to vortex shedding. Tube bundles in nuclear reactor heat exchangers and steam generators [38] will vibrate with excessive amplitudes after the external cross flow reaches/exceeds a critical velocity, the so called fluid-elastic instability. However, there is no equivalent fluid-elastic instability for cases such as oil transportation through long distance pipelines at which a critical internal fluid velocity will lead to excessive lateral vibration amplitude [39]. As for internal flow induced vibration, researchers use this expression to loosely describe two kinds of flow scenarios. The first is the dynamic response of a pipe structure to excitation under a variety of conditions, generally called transient pipe flows [40] [41], including water hammer [42] [43], cavitation [44] [45], multiphase flow [46], pipe whip [47], flow separation in elbows [44] [48] and mechanical components that include valves and supports [49] [50]. The large-scale transient processes in each of these flows have the potential to result in strong pipe vibrations. The second is pipe vibration induced by a turbulent boundary layer. Wall pressure fluctuations due to the turbulent boundary layer, although weaker in magnitude than the first scenario, can cause pipe-wall vibrations.

2.3.2 The mechanism investigation

As discussed earlier in this chapter, pipe flow can become turbulent when the inertial force becomes dominant over the viscous force. In other words, the momentum force becomes dominant over the frictional shearing stress. Within the turbulent boundary layer eddies of

varying scales will constantly be generated by the shear flow. This process results in random velocity fluctuations that result in the fluctuating pressures that act on the structure surface. Once excited, the structure can vibrate, and produce affects back on the flow. These pipe-liquid interaction mechanisms can be distinguished into three types including friction coupling, Poisson coupling and junction coupling ([41] [43] [51] [52]). As to the pressure field on the pipe wall, some researchers have ignored this influence as the RMS value of the wall displacement was found to be much lower than the laminar sublayer thickness and the turbulent pressure sources were actually located outside the sublayer [23]. Also, both the turbulent motion of the flow and the vibration of the structure will produce acoustic radiation propagating in both the internal [53] and external [23] directions. A more specific description of the contribution of pressure fluctuations to pipe vibration is provided in the following.

The measured wall pressure wavenumber spectrum can be divided into an acoustic domain, a low-wavenumber domain and a convective domain. Both structural vibrations and acoustic emissions are induced by pressure fluctuations located in the low-wavenumber domain [54] [29]. Vibrations themselves will be unique due to the specific structural characteristics of the pipe system as the structure acts as a mechanical filter and is excited only by pressure fluctuations over the frequency range that are close to the structural modes [26]. Important parameters for pipe-wall vibration include the elastic modulus of the structural material, the damping, the mass/density and pipe boundary conditions [40].

2.3.3 The development of TBL induced vibration

In the present work, the vibration of a pipe induced by internal turbulent flow will be studied experimentally. Predictive models such as the transfer matrix method (TMM) for three-dimensional pipes [55] with bends and elastic constraints [56], the eliminated-element Galerkin

method for natural frequency analysis [57], the vibration evaluation method for large-diameter piping with an elbow [58], the structural impedance method [59] and Glimm's method for transient analysis of piping systems containing fluid flows [60] are outside the scope of the current work and are only mentioned here for reference purposes.

Flow induced vibration in pipes has been researched since the 1950's [39]. Clinch [26], in 1970, was one of the first to predict the magnitude of pipe vibrations induced by internal turbulent flow. In the beginning many investigations considered pipe vibration due to transient flow processes such as vortex shedding [36] [37] given that vortex shedding was both a source of noise and structural failure [23]. Later, the characteristics of fluid induced vibration was better understood and started to be employed by many researches to solve engineering problems. Some investigations have considered the use of vibration sensors as non-intrusive mass flow meters for mass flow rate measurement. Kim [61] used three accelerometers to measure vibration signals in steel pipes conveying water excited by an external mechanical shaker and showed that the estimated flow rates were within 12% of the actual flow rates. Matthew [40] [62] used one accelerometer to measure vibration signals in pipes involving different materials and sizes and showed that there was a definite relationship between pipe vibration and flow rate. Amjad [63] proposed an ad hoc wireless network coupled with accelerometers to monitor flow rate. Yanfeng et al. [64] performed multiphase flow experiments and found that the standard deviation of the vibration signal and the gas-liquid flow rate can be approximated as a function of gas mass fraction. Pettigrew [38] noted that in nuclear reactors the control rod vibration amplitude varied in proportion to the magnitude of wall pressure fluctuations. William [29] addressed the low wavenumber turbulent boundary layer wall-pressure measurements from the vibration data by utilising a modal analysis technique. Thompson [65] utilized vibration signals to detect for

internal leakage of gases through valves. Starting around 13 years ago, a vibration-based technique was used to detect pipeline integrity, mainly based on the natural frequency [10] [11] [66] [67]. More recent investigations that use a similar approach have also been reported in the literature [68] [69] [70] [71].

Until now, many researchers have investigated specific cases. Faal et al. [72] considered flow-induced vibration of a pipe with elastic support. Dae et al. [73] investigated flow-induced vibration in two phase flows. Copeland [74] studied the chaotic flow-induced vibration of a flexible tube with end mass. Within these investigations, most often accelerometers have been used to measure vibration signals [61] [40] [63] [29], while strain gauges are used to characterize the nature of the vibration, such as by showing natural frequencies [37]. When using an array of sensors, Kim [61] found that for certain sensor spacing, sensor position will lead to a magnitude and phase mismatch between the accelerometers used to measure the vibration signal of the pipe and he compensated for these mismatches by referring measurements to one reference measurement. For data processing, power spectral density functions are often used to analyse vibration signals [37]. There are other methods, such as one developed by Kim [61] involving wave decomposition theory that is used to measure the change of wave number in a pipe carrying water using vibrational signals and decomposing the signals into propagating waves. Martin [75] used the Pulse system made by the Brüel & Kjaer company to extract both natural frequency and modal shapes from vibration signals that were obtained experimentally.

In this thesis, vibration measurements of a pipe with varying internal flow rates were performed assuming that the variable flow rate was the excitation source. Guo et al. [37] performed related experiments by performing impact response tests of a vertical riser with varying internal fluid velocities. They found that as the internal flow velocity increases the

natural frequency decreases and the amplitude of each mode increases. A similar result was found by Dai [55], who investigated the natural frequency response of a horizontal pipe with varying internal flow velocities and both ends supported in the air. The difference between these investigations and the current investigations is that Guo et al. [37] and Dai [55] considered the pipe and its internal flowing fluid as one single system, while the current investigation treats the internal flowing fluid as the excitation source for the pipe facility (pipe and supports).

Chapter Three: EXPERIMENTAL APPARATUS

3.1 Chapter Overview

This chapter provides an overview of the experimental facility used and details of the experiments conducted during this study. The first subsection describes the pipe-flow facility used during the experiments. This is followed by a description of the different experiments performed including *flow rate sweep tests*, *impulse response tests* and *phase information investigations*. A final subsection introduces the experimental configurations for two applications, one a loss-of-fluid experiment and the other a loss-of-material experiment, both using the pipe structural health diagnostics method developed in this thesis. For each experiment a description of the experimental facility configuration, the data acquisition system used, and the experimental procedure is provided.

3.2 Pipe-Flow Facility

3.2.1 Pump and flow meter

Experiments introduced in this thesis work were conducted in a pipe-flow facility at the University of Calgary shown in Figure 3.2.1. The pump used to supply the flow facility is a centrifugal pump connected to a 15-horsepower three-phase AC motor (Model No. 215TTFW7022 as shown in Table A.1 of the Appendix). The pump motor was controlled by a variable frequency drive (Lenze AC Tech M32150B) with two analog inputs (0-10 VDC and 4-20 mA) that can be used for speed reference, a PID set point reference, or PID feedback. A speed potentiometer (10,000 ohm) can be used with the 0-10 VDC input. There are also two analog outputs; one is proportional to speed (frequency) and the other to load. The drive has three programmable outputs for status indication, one form C relay and two open-collector outputs.

The connections of the drive will be shown in more detail later in the “pump drive system” subsection.

The flowmeter used was a KROHNE IFC020D, which is an electromagnetic flowmeter that can be set to show flowrate in L/s, gallon/min or m³/hr. In this experiment, a 250 Ohm resistor was used to convert the 4 - 20 mA output current signal into a 1 – 5 V voltage signal.

Figure 3.2.1: Pipe-Flow Facility

Before performing an experiment, a calibration test was conducted as described in the following:

1. Operate the pump at the maximum input voltage (5 V) for several minutes to fill the flow facility piping with water and in the process purge air from the system as much as possible, then reduce the input voltage back to 0 V.

2. Operate the pump at a set input voltage and acquire the flowmeter output voltage at a sample rate of 100 Hz. Calculate the average flowmeter output voltage over a 1-minute period of operation and record the flowrate value displayed in L/s.
3. Increase the pump input voltage in increments of 0.2 V and wait for the system to attain a steady operating state. Record the pump input voltage and the corresponding pump frequency.
4. Repeat steps 2 and 3 until the measured flowrate does not increase with an increase of pump input voltage.

Calibration curves for both the pump and the flowmeter can be obtained for each specific experimental facility configuration (pressure drop), as shown in Figure 3.2.2. The flowmeter calibration curve is always linear, whereas the pump calibration curve displays some non-linearity. For the lower flowrate settings, even though the pump is on and operating, the flowrate remains at zero until the pump input voltage exceeds 1.8V. This result is due to the hydrostatic head in the flow loop that the pump needs to overcome. Above this voltage the pump generates sufficient pressure to overcome the hydrostatic head and flow begins at 1.37 L/s. The pump calibration curve shows that the maximum flowrate for the facility configuration is 4.77 L/s. Through numerous calibrations, it was found that the flowmeter calibration curve does not change with time while the pump calibration curve can change slightly (by up to 0.03 L/s) which might be affected by temperature or other disturbances. Although it was possible to use PID control to set the pump flowrate, this feature was not used during experiments as it was desired to maintain the pump rotational frequency fixed during each test. During the flow rate sweep tests, the flow rate varied from 1.5 L/s to 4.5 L/s with an increment of 0.1 L/s.

Figure 3.2.2: Calibration Curve for Flowmeter (left) and Pump (right)

3.2.2 HP 3562A dynamic signal analyser

The HP 3562A two channel dynamic signal analyser used for some of the experiments can be operated through both the front panel and remotely programmed via GPIB link. The analyser performs low-pass filtering of signals, auto/cross power spectral and correlation analysis, and other math operations of the sampled signals. The HP 3562A settings used during the experiments performed in this research are shown in Table A.2 of the Appendix.

The HP 3562A was used to perform both spectral and coherence analysis, as outlined in the following equations. The power spectrum is computed using:

$$G_{xx}(f) = X(f)X^*(f) \quad 3.1$$

where: $X(f)$ is the linear spectrum (FFT or Fast Fourier Transform) of channel 1

$X^*(f)$ is its complex conjugate

The cross spectrum is computed using:

$$G_{yx}(f) = Y(f)X^*(f) \quad 3.2$$

where: $Y(f)$ is the linear spectrum of channel 2 (FFT of the signal)

$X^*(f)$ is the complex conjugate of the linear spectrum of channel 1

The coherence function is computed using:

$$\gamma^2 = \frac{G_{xy}(f)G_{xy}^*(f)}{G_{xx}(f)G_{yy}(f)} \quad 3.3$$

where: $G_{xy}(f)$ is the cross spectrum

$G_{xy}^*(f)$ is the complex conjugate of the cross spectrum

$G_{xx}(f)$ is the power spectrum (PSD) of channel 1

$G_{yy}(f)$ is the power spectrum (PSD) of channel 2

3.2.3 NI USB-6009 data acquisition card

The National Instruments (NI) USB-6009 is a multifunction data acquisition (DAQ) USB device that is powered over the USB interface. It provides either eight single-ended 13-bit analog input (AI) channels with a $\pm 10V$ measurement range or four differential-ended 14-bit analog input channels with user selectable measurement ranges ($\pm 10V$, $\pm 5V$, $\pm 4V$, $\pm 2.5V$, $\pm 2V$, $\pm 1.25V$, $\pm 1V$). It also has two analog output (AO) channels, 12 digital input/output (DIO) channels, and a 32-bit counter. Table A.3 in the Appendix provides a brief description of the NI USB-6009.

3.2.4 Pipe vibration and pressure fluctuation measurement

Pipe vibration was measured using an accelerometer (PCB Piezotronics Model 333B50) and the output signal from the accelerometer was amplified by a 4-channel signal conditioner (PCB Model 442B104). The accelerometer was secured to the pipe surface by direct adhesive mounting and the accelerometer was used to measure mechanical vibrations in the pipe structure from 0.5 to 3000 Hz with a sensitivity of 1071 (mV/g). A wet-wet differential pressure transducer, Validyne DP103, was used to measure the pressure fluctuation on the pipe wall. This sensor has interchangeable diaphragms (shown in Table A.4 of the Appendix) that can measure

over standard ranges from as low as ± 55 Pa to as high as $\pm 86,200$ Pa. The diaphragm pressure range corresponds to a -10 to +10 Vdc output from the Validyne CD280 multi-channel carrier demodulator used to power the DP103. The Validyne DP103 is capable of measuring high frequency fluctuations with a frequency response of up to 1000 Hz and an accuracy of $\pm 0.25\%$ at full scale output. In the current investigation, diaphragm No. 8-32 was used, which measures pressure over the range of ± 13790 Pa (± 2 psi) with a tolerance of 138 Pa. In certain experiments more than two signals were acquired, and for these experiments a system other than the two-channel HP 3562A dynamic signal analyzer had to be used. For these experiments the National Instruments USB-6009 was used and low-pass (anti-aliasing) filtering was performed using a TSI IFA 100 with four Model 157 signal conditioner modules set at a low-pass frequency of 400 Hz. The signal conditioner in the IFA 100 uses a third-order (-18 dB/octave) Sallen-Key type filter.

3.2.5 Other sensor systems

A National Instruments (NI) PCI-6024E with SCXI-1302 terminal block was used for both controlling the pump and acquiring data from the flowmeter. The NI PCI-6024E is a multifunction I/O device that is equipped with 16 channels of analog input, two channels of analog output, and eight lines of digital and timing I/O through a 68-pin connector.

3.2.6 Pump drive system

This subsection describes the pump drive system including the hardware, connections and the software GUI developed using LabVIEW. Figure 3.2.3 shows the configuration of the pump drive system. A computer with a National Instruments PCI-6024E data acquisition card and LabVIEW software connected to a NI signal terminal block (NI SCXI 1302) mounted in a NI SCXI chassis (NI SCXI 1000) was used to send an analog output signal to the variable frequency

drive system used to control the pump. The actual flow rate from the pump was monitored using the output from the Krohne flowmeter and sampled using the PCI-6024E.

Figure 3.2.3: Configuration of the Pump Driving System

A LabVIEW program was developed, and the GUI is shown in Figure 3.2.4. In the software, the user can enter the desired flow rate in terms of L/s, Reynolds number or m/s. A strip-chart waveform shows the actual real-time flow rate over a period of time.

Figure 3.2.4: Pump Flow Rate Setting GUI

3.3 Flow Rate Sweep Tests

The purpose of the investigation was to characterize the dynamic response of the pipe structure to pipe flow turbulence. Given that the strength of turbulence increases with Reynolds number, it is possible to vary the strength of the excitation source by varying the flowrate. Additionally, given that turbulence exists over a broad range of frequencies, the excitation source also covers a broad range of frequencies. Flow rate sweep tests are used to perform investigations by varying the flow rate in small increments from one test to the next. In the experiments reported here, tests were conducted using both acrylic and ABS pipe with flow rate increasing from 1.5 L/s to 4.5 L/s in increments of 0.1 L/s. This corresponded to a Reynolds number ranging from 52,000 to 155,000, assuming a water temperature of 20°C. Accelerometers mounted at three different locations (referred to as Locations #1, #2 and #3) were used to measure pipe vibration (the dynamic response) and a pressure transducer was used to measure pipe-wall pressure fluctuations.

3.3.1 Acrylic pipe

Two sets of flow rate sweep experiments were performed, one using an acrylic pipe and the second using an ABS pipe. Pictures of the flow loop facility with the acrylic pipe are shown in Figure 3.3.1, with the flow entering from the left as indicated by the red arrow. Measurements were performed on the section of pipe near the centre of the picture directly behind the red tool cabinet. The pipe test section was mechanically isolated from the pump supply line using flexible hose and elastomer couplings; however, it was challenging to completely isolate the pipe test section from all pump vibrations.

A more detailed schematic of this section of pipe is shown in Figure 3.3.2. Mechanical vibration measurements were made using three PCB Piezotronics 333B50 one-axis accelerometers mounted to the top of the pipe at three different spatial locations:

Location #1: Upstream location near a pipe mount;

Location #2: Midway between locations #1 and #3;

Location #3: Downstream location near the pressure transducer, positioned between two flexible couplings.

The pipe-wall pressure fluctuation measurement was made using the Validyne DP103 wet-wet pressure transducer and physically connected to the pipe using a 210-mm long 6.5-mm ($\frac{1}{4}$ ") inner diameter semi-rigid plastic tubing (Nylaflow). Only one spatial location (Location #3) was investigated using the pressure transducer.

Figure 3.3.1: Experimental Facility, Overview (top) and Zoom-in (bottom) – Acrylic Pipe

Figure 3.3.2: Acrylic Pipe Experiments

The position of the accelerometer at Location #1 was sufficiently far enough downstream of the pipe entrance to ensure that the measurement was made in the fully developed flow region. This was determined using an equation for entrance length as a function of Reynolds number in turbulent pipe flow [76]:

$$\frac{l_e}{D} = 4.4Re^{1/6} \quad 3.4$$

where l_e is the length of the entrance region and D is the inner diameter of the pipe. Based on the pipe diameter (approximately 38 mm) and Reynolds number range investigated (up to 155,000), Location #1 should at least 1.25m from the pipe entrance.

3.3.2 ABS pipe

The flow rate sweep tests were performed using ABS pipe shown in Figure 3.3.3 and its schematic in Figure 3.3.4, with the flow entering from the right as indicated by the red arrow. The ABS pipe was slightly longer than the acrylic pipe, and thus the flexible coupling located directly upstream of Location #3 was removed when testing with ABS pipe.

Figure 3.3.3: Experimental Facility – ABS Pipe

Figure 3.3.4: ABS Pipe Experiments

3.3.3 Data acquisition system

The data acquisition system shown in Figure 3.3.5 was used during the flow rate sweep tests. Data was collected from three accelerometers and a pressure transducer, with the accelerometer signal conditioned by an amplifier and the pressure transducer signal conditioned using the CD 280 demodulator. For these experiments, all signals were first low-pass filtered (for anti-aliasing) using a TSI IFA-100 and then time histories were acquired using the USB-6009

data acquisition system. The collected time signals were then post processed using both LabVIEW and MATLAB.

A LabVIEW program was developed for the flow rate sweep tests and its GUI is shown in Figure 3.3.6. This program uses the NI USB 6009 to record and display time history data and the HP 3562A to record and display power spectra of the time history data, and then saves the time history data into files that can be imported into MATLAB later for further analysis. A sampling rate of 1000 Hz was used when acquiring with the NI USB 6009 and the power spectra computed using 100 ensemble averages for spectral smoothing.

Figure 3.3.5: Configuration of the Data Acquisition System

Figure 3.3.6: Frequency Measurements GUI

3.3.4 Experimental procedure

The flow rate sweep tests were performed using the following steps:

1. Check the integrity of the hardware connection, turn on all measurement systems and set the suitable parameters in the LabVIEW GUI.
2. Operate the pump at a relatively high flow rate for around 10 minutes to purge the water in the system of air bubbles.
3. Operate the pump to provide a flow rate of 4.5 L/s (upper limit), adjust the back pressure on the pressure transducer, and set the gain factor (X1, X10, X100) of the accelerometer signal conditioner (PCB Model 442B104). Be careful not to exceed the range of the A/D converter, otherwise clipping (i.e. railing) of the signals may occur.

4. After the system has been operating for several minutes and the flow becomes steady, start to capture data. After collecting the data, save the data into Excel files (tab-delimited ASCII).
5. Decrease the flow rate in 0.1 L/s decrements and repeat step 4 until the flow rate is decreased to 1.5 L/s (the lower limit). Adjust the back-pressure on the pressure transducer to guarantee that the magnitude of the signal does not exceed the output range of the pressure transducer.

3.4 Impulse Response Test

The impulse response test is an experiment that acquires the dynamic response (frequency response function) of the system corresponding to a transient input signal, called an impulse. The impulse is mathematically described as a Dirac delta function, a function that is equal to zero everywhere with the exception of time zero, and it contains amplitude at all frequencies and hence is spectrally broad. When the system is assumed to be linear and time invariant, its frequency response function defines the response to all frequencies, the so called system descriptor. A typical impulse response test uses an impact hammer to generate an impact (normally 10^{-3} s in duration) and the response is measured using an accelerometer. Based on the characteristics of the investigated system, a suitable sensor with a relatively small mass should be chosen to maintain the original properties of the system, and a suitable impact hammer should be selected to generate an excitation that covers a suitable range of frequencies. The impulse response test was performed at different locations (Location #1, #2 and #3) and with the flow facility filled with water.

3.4.1 Data acquisition system

The data acquisition configuration used for the impulse response test is as shown in Figure 3.3.5. The impact hammer (PCB Model 086C03) shown to the upper right in Figure 3.4.1

was used to generate the impact that excited the structure. The frequency distribution and energy level of an impact are interrelated, and both will be affected by the configuration of the hammer. The impact tip affects the hammer impulse frequency distribution and the extender, a mass on the hammer head, affects the energy level. The hammer impact velocity will also have an impact on both frequency distribution and energy level. For a higher frequency response, a stiffer tip without an extender mass should be used. For a lower frequency response, a softer tip with an extender mass should be selected. To increase signal energy, either the impact velocity and/or the hammer mass is increased. According to the performance curves for the impact hammer, a medium tip with vinyl cover and extender mass were selected to generate an impact covering a frequency range from 0 to 400 Hz.

Figure 3.4.1: Performance Curve of Impact Hammer Model 086C03 [77]

For the impulse response test, a LabVIEW program was developed for both data capture and analysis. The GUI is shown in Figure 3.4.2 and it consists of a data entry section for specifying data processing functions and a data visualization section. The data acquisition program was designed to be triggered by the amplitude of the impact enabling the test to be performed by a single person for high efficiency. The time duration of the impact can be tailored depending on the specific situation in order to avoid unexpected noise and extraneous input. Finally, a peak detection function was added to highlight the frequency response properties.

Figure 3.4.2: Impulse Response Measurements GUI

3.4.2 Experimental procedure

The impulse response test was performed through the following steps:

1. Verify the integrity of the hardware connections, power on all systems with suitable settings and enter these settings into the LabVIEW GUI.

2. Click the Start DAQ button and strike the pipe with the impact hammer near Locations #1, #2 or #3. If the data acquisition procedure was not triggered by the impact, strike the pipe with the hammer again. If a “double impact” occurs, clear all of the graphs and strike the pipe with the impact hammer again.
3. After successfully acquiring data, adjust the time window of both the impact hammer and vibration response signals, and then adjust the detection threshold to a suitable level for peak identification in the frequency response function.

3.5 Phase Information Investigation

The phase information investigation involves analysis of the propagation of acoustic signals between two locations in the media under investigation (either the pipe or the water inside of the pipe). It involves examining both the direction and propagation speed of an acoustic wave in either media. Phase information investigations are normally performed using microphones in air, hydrophones in water or accelerometers in structures. It is worth noting that the signal collected during phase information investigations is easily contaminated by external noise sources, and as a result some form of calibration technique is often required. In this thesis work, the investigation of the phase information was performed by analysing signals from two accelerometers mounted on the pipe at two locations, A and B. The purpose of the investigation was to determine direction in which the signal propagated and the media through which the signal propagated (propagation path).

3.5.1 Experimental facility

The phase information investigation was conducted in the experimental facility presented earlier in Figure 3.3.3 using ABS piping. A more detailed schematic is shown in Figure 3.5.1. For acoustic propagation in a media, the speed, frequency, and wavelength are related by:

$$v = f \cdot \lambda \quad 3.5$$

where v is the speed of sound in the media in m/s, f is the frequency in cycles/s (Hz), and λ is the wavelength in m/cycle. Due to the fact that the speed of sound in both ABS and water is greater than 1000 m/s, the frequency range analysed is from 0 to 400 Hz, and assuming nondispersive acoustic propagation, the wavelengths of interest for this investigation are 2.5 m or larger. Therefore, to obtain a pronounced phase difference, the distance between the two accelerometers was set to the maximum possible for the facility, 1.5 m.

Figure 3.5.1: Schematic of the Experimental Facility for Phase Information Investigation

3.5.2 Data acquisition system

The GUI of the LabVIEW program developed for this experiment is shown in Figure 3.5.2. The program computes auto spectra, cross spectra, cross-correlation function, coherence function and phase relationship for two accelerometer signals. A second LabVIEW program, shown in Figure 3.5.3, was developed to extract the slope of the phase relationship over a specified range of frequencies. The program was designed to be interactive, providing a zoom-in function to improve functionality and an auto-slope calculation function with adjustable

frequency range. The experimental procedure is as described earlier for the flow rate sweep tests.

Figure 3.5.2: Phase Information GUI

Figure 3.5.3: Phase Slope Calculation GUI

3.6 Applications to Pipe Structure Health Diagnostics

3.6.1 Loss-of-fluid experiment

The loss-of-fluid experiment was performed in the experimental facility described in Figure 3.3.3, but modified using the pipe section shown in Figure 3.6.1. In this experiment, water flowed from left to right and four accelerometers (labeled as #0, #1, #2, and #3) were spaced 170 mm apart and centered about the location where the fluid was withdrawn from the pipe. A flip valve, visible at the bottom of the pipe in Figure 3.6.1, was connected to the gate valve shown inset in Figure 3.6.1 by 60 cm of Tygon tubing. The flip valve provided binary flow control, while the gate valve provided finer control over the fluid removal rate. By rotating the gate valve in half-turn increments, it was possible to exert fine control over the amount of fluid withdrawn from the pipe. The data acquisition system was the same as used in the flow rate sweep tests reported earlier with the exception of the pressure transducer being replaced by the fourth accelerometer.

Figure 3.6.1: Pipe Section for Loss-of-Fluid Experiment – ABS pipe

The purpose of this experiment was to examine the influence of a controlled loss-of-fluid event (a pipeline leak) on pipe vibration. During the experiment the pipe Reynolds number was maintained constant (at 95,000 for the flow rate of 3 L/s) while the leaked fluid flow rate was increased incrementally by rotating the gate valve in half-turn increments. Figure 3.6.2 shows the relationship between the flow rate of the leaked fluid and valve opening. Each whole number value denotes a 360-degree rotation of the gate valve.

Figure 3.6.2: Relationship of the Valve Opening and Leaked Fluid Flow Rate (turns)

The loss-of-fluid experiments were performed using the following steps:

1. The integrity of the hardware connections was checked before turning on all measurement systems. Set values were entered into the LabVIEW GUI.
2. The pump was operated at a relatively high flow rate for around 10 minutes to purge the system of air bubbles.
3. The pump was then operated at a flow rate of 3 L/s and the accelerometer amplifier was adjusted to a suitable gain factor (X1, X10, or X100). Care was taken not to exceed the range of the A/D converter, otherwise signal clipping may occur.

4. After the system was operated for about one minute the flow becomes steady, and the process of data capture begins. After collecting the data, data files are saved in Excel format.
5. The valve opening is increased a half rotation and step 4 is repeated until the valve is fully opened.

3.6.2 Loss-of-material experiment

The loss-of-material experiment investigated how mass removed from the pipe wall influenced pipe wall vibration. The pipe layout, shown in Figure 3.6.3, has water flowing from left to right with accelerometers numbered #0, #1, #2, and #3. Detailed dimensions for the experiment are shown in Figure 3.6.4. The data acquisition system was the same as used in the loss-of-fluid experiment.

Figure 3.6.3: Pipe Section for Loss-of-Material Experiment – ABS pipe

Figure 3.6.4: Schematic of the Pipe Section for Loss-of-Material Experiment – ABS pipe

As shown in Figure 3.6.5, the material was removed layer by layer using a Dremel high-speed rotary tool. Including the reference case of no mass removed, a total of five cases were investigated. Referring to the cross-section shown to the right in Figure 3.6.5, the percentage of mass removed was computed by dividing the cross-sectional area of material removed by the cross-sectional area of the original pipe wall. The depth of material removed plotted as a function of percentage of material removed is shown in Figure 3.6.6.

Figure 3.6.5: Pipe Material Removed (top) and Schematic (bottom)

Figure 3.6.6: The Relationship between Mass Removed and Removal Depth

The procedure for the loss-of-material experiments was performed using the following steps:

1. Check the integrity of the hardware connection, turn on all measurement systems and set the suitable parameters in the LabVIEW GUI.
2. Operate the pump at a relatively high flow rate for around 10 minutes to purge the system of air bubbles from the flow.
3. Operate the pump to provide a flow rate of 4 L/s (the upper limit was decreased to 4 L/s for these experiments because of increased system back pressure) and adjust the amplifier with a suitable gain factor (X1, X10, or X100). Be careful not to exceed the range of the A/D converter, otherwise clipping of the signal may occur.
4. After the system has operated for several minutes and the flow becomes steady, capture data. After collecting the data, save the data into Excel files.
5. Decrease the flow rate in 0.1 L/s decrements and repeat step 4 until the flow rate is decreased to 1.5 L/s (the lower limit).
6. Stop the pump and use the Dremel tool to remove layers of material to reach the expected mass losses percentage (increments of 1.4%, 4.4%, 7.4%, and 10.4%). Repeat steps 2 through 6 until all mass removal percentages have been tested.

Chapter Four: EXPERIMENTAL RESULTS AND DISCUSSION

4.1 Chapter Overview

This chapter includes the experimental results and discussion. The chapter starts with the background noise measurements. The second subsection examines spectral analysis results for both the fluctuating pressure and vibration signals, considering the data both quantitatively and qualitatively (visually). The third subsection compares the results from the flow rate sweep tests with results from the impact response test. The fourth subsection investigates the relationship between pipe vibrational energy and Reynolds number. This is followed by a subsection that examines the phase relationship of the vibration signal. The final subsection examines application of the developed methodologies to pipe structure health diagnostics including both loss-of-fluid and loss-of-material events.

4.2 Background Noise Measurements

Before each experiment was performed, the background noise of the laboratory environment was quantified by collecting spectral data with the pump off. The results can be seen in Figure 4.2.1 where the no-flow fluctuating pressure data is plotted on the left and the no-flow vibration data is plotted on the right.

Figure 4.2.1: Background Noise Spectra – Pressure (left) and Vibration (right) - Acrylic Pipe

The plots show that the power in both the pressure fluctuation and vibration signals are mainly concentrated in a frequency range that is less than 40 Hz. The source of energy in these signals is due to background environmental noise. The similarity of the two signals demonstrates the sensitivity of the pressure transducer to mechanical vibration. As will be shown later for experiments with the flow on, the magnitudes of the no-flow signals are low, and the results indicate that the influence of environmental noise on the experiment is almost negligible. These results are for acrylic pipe, and similar results can be obtained for ABS pipe.

4.3 Fluctuating Pressure and Vibration Spectral Analysis

4.3.1 Qualitative analysis

In order to develop a preliminary understanding of the nature of fluctuating pressure and vibration signals in the acrylic pipe, spectral data was collected at seven flow rates, starting at 1.5 L/s ($Re = 51,000$) and increasing up to 4.5 L/s ($Re = 154,000$) in increments of 0.5 L/s.

Figure 4.3.1: Pressure Fluctuation (log scale) – Acrylic Pipe

Given that the Validye DP103 is a differential pressure transducer, the transducer reference pressure was adjusted at each flow rate so as to remove any DC offset from the pressure fluctuation signal. This is performed by connecting the reference pressure tap on one side of the differential pressure transducer to an acrylic cylinder that can be pressurized using a syringe. The pressure fluctuation signal is analysed by calculating its power spectral density function. Pressure fluctuation spectra collected from the acrylic pipe with flow on are shown in Figure 4.3.1. The amplitude of the power spectra is seen to increase with increasing Reynolds number.

Each pressure spectra follows a characteristic pattern, increasing in magnitude in the low frequency range (0.5-2 Hz) and then starting to decay between 2-10 Hz where several broader peaks are observed. The spectral plot reveals an almost linear region with a slope of $-7/3$ from 2-20 Hz. Thereafter, the power spectral amplitude decreases to a frequency of about 60 Hz where the presence of a “hump” starts to appear. The energy in this “hump” is seen to increase with Reynolds number. At frequencies above where the “hump” reaches a maximum value (at

approximately 100 Hz), the power spectral amplitude decreases until the transducer noise floor is reached.

It is observed that the spectra have the greatest energy at the lowest frequencies, and this corresponds to the largest eddies in the flow. In contrast, at higher frequencies the energy in the flow decreases and this corresponds to the smallest eddies in the flow. These smallest eddies are in the process of decay where kinetic energy is being converted into thermal energy through the action of viscous shear [78].

Internal pipe wall roughness is known to have an influence on the power spectra. Given that the inner pipe wall of the plastic pipe used in this experiment has not been smoothed, it has a certain amount of roughness. A rough surface will increase the wall shear and the magnitude of turbulence which in turn increases the magnitude of pressure fluctuations [17]. This is observed in the spectra in Figure 4.3.1 where energy in the low frequency region (0.5-2 Hz) increases before reaching a maximum at around 2 Hz. It has been observed [27] that inner pipe wall roughness contaminates and narrows the linear ($-7/3$ slope) region of the spectra, and it has also been found that the linear slope region is more prominent for smooth wall flows. The presence of other energy sources (broad peaks and humps) in this region make it even more difficult to observe the linear slope region.

Power spectral density curves for pressure fluctuations in turbulent flow tend to collapse as one approaches the higher frequencies. In Figure 4.3.1, however, this is not the case due to the presence of the “hump” that begins to appear at around 100 Hz and grows in magnitude with increasing Reynolds number. As shown in Figure 4.3.2, there are two possible explanations for the presence of this hump. The first is due to the pipe-section transition from a longer pipe section to the shorter pipe section used to house the pressure tap. The disruption from one pipe

section to the next has the potential to generate flow separation and in extreme misalignment cases, vortex shedding. This induces pressure fluctuations into the flow and mechanical vibrations in the pipe. As shown in Figure 4.3.2 (top), the short pipe section that holds the pressure transducer tap in place is connected to the other pipe sections using two elastomer flexible couplings on either side of the pressure tap section. Although the use of elastomer couplings helps to insulate the pressure tap section from mechanical vibration, it results in the small wall step from one pipe section to the next as shown in Figure 4.3.2 (bottom). Wall steps due to minor misalignment and on the order of 1 mm or smaller can impact pressure fluctuation measurements, resulting in the atypical pressure spectra. According to Dinkelacker [19] and as shown in Figure 4.3.3, both symmetric and asymmetric wall steps can contribute to the “hump” observed in the high frequency range.

Figure 4.3.2 (bottom) shows a second possible cause for the spectral hump, and this is due to the existence of the pressure tap itself. During experiments it was observed that small air bubbles in the flow would become trapped in the cavity at the pressure tap. The low pressure region associated with the recirculation zone that is created next to the pressure tap is what causes these small air bubbles to become trapped, gradually forming an air cavity.

Figure 4.3.2: Pressure Transducer Tap (top) and Pipe Misalignment (bottom)

Figure 4.3.3: Wall Pressure Fluctuation with Small Symmetric Step of 0.05 mm Height (left), with Asymmetric Steps of 0.5 mm Height (right). $\Delta - \Delta$: with, $\circ - \circ$: without. [19]

The broad peaks in the low frequency range (from 2-10Hz) are believed to be due to acoustic modes excited by both pipe flow turbulence and facility background noise. These peaks can be partially removed by post-processing signals obtained from two pressure transducers mounted in the span-wise direction of the pipe [23] [24]. The work of filtering out these peaks is not included here. There are also additional peaks observed in the frequency range from 20 Hz to 40 Hz. These peaks appear to shift in both frequency and magnitude with increasing Reynolds

number. Through a number of tests, these peaks were found to be due to the pump blade passage frequency and this will be discussed in more detail later in this chapter. The spikes in the high frequency region are harmonics of 60 Hz and thus are attributed to electronic noise.

In order to examine how the pressure fluctuation scales with Reynolds number, a fluctuating pressure coefficient is calculated as shown in Figure 4.3.4 (left). The pressure coefficient is normally calculated from a time series, computing the RMS of the pressure fluctuation, and dividing by the dynamic pressure. Given that the signals are directly processed by the HP3562A spectrum analyzer, time series information is not saved but rather only the final power spectral density (PSD) data is stored. Consequently, the fluctuating pressure coefficient is calculated by integrating the PSD data across all frequency bins and then dividing by the dynamic pressure. In performing this calculation, the centreline velocity for turbulent pipe flow is used to compute dynamic pressure which is assumed to be 1.2 times the mean velocity. This is an approximation as the factor of 1.2 will change as the Reynolds number changes.

Figure 4.3.4: Pressure Coefficient - Acrylic Pipe (left), Reference from [26] (right)

Figure 4.3.4 (right) shows the results of Clinch [26], indicating that the pressure coefficient should be constant with Reynolds number and with a value less than 0.01. Clinch [26]

obtained these results using a steel pipe and employing two kinds of filters, one a mechanical low-pass acoustic filter used to suppress extraneous pump noise and the other an electrical filter. As shown in Figure 4.3.4 (left), the fluctuating pressure coefficient is approximately constant with Reynolds number, varying from 0.06 to 0.09 over the entire Reynolds number range investigated. The pressure fluctuation coefficient measured in the current experiments is, however, an order-of-magnitude higher than the results reported by Clinch, most likely due to the fact that Clinch performed significant filtering in order to remove unwanted pump noise, while in the current investigation only anti-aliasing filtering was used.

Figure 4.3.5 shows vibrational power spectra with a linear frequency scale over seven flow rates, increasing from 1.5 L/s to 4.5 L/s in increments of 0.5 L/s. The most obvious conclusion that can be drawn is that the vibrational energy increases with increasing Reynolds number. This is understandable given that the magnitude of pressure spectra also increase with Reynolds number and increased pressure fluctuation excitation is expected to result in larger vibration. The vibrational spectra also reveal that some spectral spikes remain fixed for all Reynolds numbers, while other spectral spikes change with Reynolds number. An example of a fixed peak is the peak at 10.5 Hz which occurs for all three locations and at all flow rates. Peaks that change with Reynolds number are also evident, and these are indicated by red ellipses in the figures.

Figure 4.3.5: Vibration for Location #1(upper left), #2(upper right), #3(lower middle) – Acrylic Pipe

4.3.2 Data visualization

In order to better understand the dynamics of the system as the Reynolds number increases from 51,000 to 154,000, it was decided to repeat the flow sweep experiments, only now with the Reynolds number increasing in smaller increments (flow rate increments of 0.1 L/s). This results in a larger number of spectra (31 in total) for both the fluctuating pressure and vibration signals. In order to visualize these larger data sets, both 2-D color contour and 3-D waterfall plots are used with frequency plotted on the abscissa, Reynolds number on the ordinate, and power spectral amplitude corresponding to either the colour of the contour or vertical height.

Figure 4.3.6 shows one of the 3-D waterfall plots. Several features are evident in both the pressure spectra on the left and the vibration spectra on the right. These include spectral spikes that do not change with Reynolds number, values that increase in magnitude with Reynolds number but remain at the same frequency, and values that change in both magnitude and frequency with Reynolds number.

Figure 4.3.6: Pressure Fluctuation (left) and Vibration (right) for Location #1 - Acrylic Pipe

Figure 4.3.7 shows 2-D contour maps for pressure spectra collected from the acrylic pipe using a log scale (left) and a linear scale (right). With the exception of the “hump” located at around 100 Hz, all of the spectra show the spectral amplitude decreasing with increasing frequency. The spectra also reveal a trend whereby the spectral amplitude increases with Reynolds number, and the largest increase occurs for frequencies below 150 Hz. Vertical lines that remain constant for all collected spectra are also noted in Figure 4.3.7 (right), and these are attributed to background noise. The fact that these spikes appear at 120 Hz, 180 Hz, 240 Hz, 300 Hz, and 360 Hz (all multiples of 60 Hz) are indicative of electronic noise. These same spikes are also evident in the background noise spectra in Figure 4.2.1. At a Reynolds number of 96,000 (flow rate of 2.8 L/s), the broad spectral “hump” at around 100 Hz that was discussed when

Figure 4.3.1 was presented starts to appear. This hump becomes broader as the Reynolds number increases, with the centre frequency shifting from 100 Hz to 120 Hz by a Reynolds number of 154,000 (flow rate of 4.5 L/s).

Figure 4.3.7: Pressure Fluctuation Contour Map – x-axis log (left) and x-axis linear (right) - Acrylic Pipe

The pressure fluctuation contour plot (Figure 4.3.7 - left) reveals a spectral spike that is seen to change with Reynolds number. This spike can be seen starting at a Reynolds number of 51,000 (flow rate of 1.5 L/s) and a frequency of 20 Hz, and it then increases as the Reynolds number increases. It is difficult to see this spectral spike above a Reynolds number of 82,000 (flow rate of 2.4 L/s) as the pressure fluctuation signal due to flow turbulence starts to become stronger, masking the spike by a Reynolds number of about 85,000.

Figure 4.3.8: Vibration Contour Maps for Location #1 (upper left), #2 (upper right), #3 (lower middle) - Acrylic Pipe

Figure 4.3.8 shows the vibrational spectral response in the form of a contour map collected with accelerometers mounted at Locations #1, #2, and #3 on the acrylic pipe. Several fixed spectral spikes (at approximately 15 Hz and 30 Hz) are noted at all three accelerometer locations. These same spectral spikes were noted in the background noise spectra shown in Figure 4.2.1 (right). All three accelerometer locations reveal somewhat similar spectral responses; however, the broad spectral distribution evident at Location #3 becomes narrower as the accelerometer mounting location moves closer towards a structural mount (at Location #1). The structural response is expected to become stiffer as the mounting location moves toward a

structural mount. At Location #3, on the other hand, the elastomer couplings have a tendency to dampen out mechanical excitations. By comparing the three contour plots in Figure 4.3.8, spectral spikes that change with Reynolds number are observed at all three locations. In the next subsection it will be shown that these spikes are attributed to the pump blade passage frequency. The contour plots demonstrate that the spectral spikes become stronger (brighter red in colour) the closer that an accelerometer is placed to a structural mount (moving from Location #3 towards Location #1). It is also observed that the highest spectral levels occur across frequency bands where the structure responds most favourably to external excitation (modal locations), which is to be expected. Finally, the magnitude of the spectra is seen to increase as the Reynolds number increases.

4.3.3 Evidence of pump blade passage frequency

By tracking the spectral spikes, it was found that there was a linear relationship between the frequency of a spectral spike and Reynolds number. This led to speculation that these spikes were attributed to the pump, and the relationship between the spectral spikes and pump rotational speed is examined in this subsection. The pump harmonic frequencies were superimposed onto the log-scale pressure fluctuation contour map from Figure 4.3.7, as shown in Figure 4.3.9 (right). In doing this, it is found that the spectral spikes fit the second harmonic curve. It was also found that the amplitude of the spike increased as the Reynolds number increased.

Figure 4.3.9: Pressure Fluctuation Contour Map without (left) and with (right) Pump Harmonic Frequencies Shown as Dashed Lines - Acrylic Pipe

Figure 4.3.10 shows a magnified contour plot (from 0 to 100 Hz) of the vibration signal collected at Locations #1, #2 and #3. Also shown on these plots are lines corresponding to the first seven harmonics of the pump rotational speed. When looking at these plots it becomes evident that some of the spectral spikes correspond directly to harmonics of the pump rotational frequency.

Figure 4.3.10: Vibration Spectra 0-100 Hz – First Seven Harmonics of Pump Frequency Location #1 (upper left), #2 (upper right) and #3 (lower middle) – Acrylic Pipe

Figure 4.3.11 plots the entire spectra up to 400 Hz for the vibration signal collected at Location #1, with the first seven harmonics of the pump rotational frequency superimposed. It is noted that the most distinct band of spectral spikes correspond to the seventh harmonic of the pump rotational frequency. This is most likely the blade passage frequency of the pump, assuming that the centrifugal pump used in the facility has seven impeller blades.

Figure 4.3.11: Vibration Spectra 0-400 Hz - Location #1 With First Seven Pump Harmonics – Acrylic Pipe

4.3.4 Acrylic versus ABS pipe

The experiments performed with the acrylic pipe are repeated in this subsection using ABS pipe. Given that the inner diameter of the ABS pipe is 39.9 mm and for the acrylic pipe it is 37.1 mm, the Reynolds number computed for a given volumetric flow rate varies slightly between the two types of pipe. Figure 4.3.12 compares the fluctuating pressure response collected by flow sweep for Acrylic pipe (left) and ABS pipe (right). The pressure contour plots between acrylic and ABS pipe are somewhat similar, with the “hump” in the pressure spectra being narrower and at a slightly higher frequency for the ABS pipe.

Figure 4.3.12: Pressure Fluctuation Contour Plots (top) and Pressure Fluctuation Power Spectra (bottom), Acrylic Pipe (left) and ABS Pipe (right)

Figure 4.3.13 compares the vibration response for Acrylic pipe (left) and ABS pipe (right) as collected through a flow sweep. For both pipe materials, the broad humps that are very apparent at Location #3 become narrower as the accelerometer mounting point moves closer to the structural support near Location #1. In a general sense, the contour plots collected at each location are qualitatively similar between the two pipe materials. It is noted that experiments performed on ABS pipe demonstrate energy at higher frequencies than for the experiments performed on Acrylic pipe. The Location #3 contour plots show the greatest difference, most likely due to the fact that for Acrylic pipe this location had two elastomer flexible couplings mounted on either side of the short pressure tap pipe section, whereas for ABS pipe there was only one elastomer coupling. The two-coupling system results in a broader distribution of energy, reflecting that the system is less stiff with two couplings than it is with only one coupling.

Figure 4.3.13: Vibration Contour Plots, Location #1 (upper left), #2 (middle left), #3 (lower left) - Acrylic Pipe, Location #1 (upper right), #2 (middle right), #3 (lower right) - ABS Pipe

4.4 Impact Response Test versus Flow Rate Sweep Tests

The system dynamic response was investigated using an impact hammer to provide excitation and measuring the time history of both the impact and the structural response. These tests were performed on ABS pipe with the pump off. In this configuration the water had not been drained from the system and consequently the ABS pipe was filled with water during the tests.

Figure 4.4.1: Impact Response Test Plots, Impact Force (upper left), Impact Response (upper right), FFT of Impact Force (lower left), Frequency Response Function (lower right) - ABS Pipe – Location #1

In doing this, it is assumed that the impact from the hammer will impose a Dirac delta forcing function onto the structure, and in turn all frequencies are excited given that the Fourier

transform of a delta function has energy at all frequencies. In practice, a short impact duration is required (on the order of 10^{-3} s) in order to provide an energy distribution that is evenly distributed over a wide range of frequencies. As shown in Figure 4.4.1, the impact time is around 0.003 s and provides excitation up to 500 Hz. To ensure that the system is sufficiently excited, the frequency response is analysed only from 0 to 400 Hz.

Figure 4.4.2: Impact Frequency Response Function (top) Compare with Flow Rate Sweeps Contour Plot (bottom) - ABS Pipe – Location #1

Figure 4.4.3: Impact Frequency Response Function (top) Compare with Flow Rate Sweeps Contour plot (bottom) - ABS Pipe – Location#2

Figure 4.4.4: Impact Frequency Response Function (top) Compare with Flow Rate Sweeps Contour plot (bottom) - ABS Pipe – Location #3

The Location #1 frequency response function is shown in Figure 4.4.2 (upper), from which it can be seen that the natural frequencies are mainly around 190 Hz, 270 Hz and 370 Hz. In the corresponding vibration contour map collected from the flow sweep test and shown in Figure 4.4.2 (lower), the energy is seen to exist at these same frequencies. In addition, this spectral energy does not change as the Reynolds number increases other than growing in magnitude. These results as flow turbulence provides broad excitation to the structure through pressure fluctuations, and the vibration contour map is able to capture the modal response of the structure. Similar results are observed for Location #2 in Figure 4.4.3 and Location #3 in Figure 4.4.4. The results at all three locations indicate that the shape of the energy distribution in each contour plot due to a flow sweep and the shape of the frequency response function match quite well. This provides indication that a flow sweep can reveal the modal characteristics of a pipe structure under operational conditions without the need to strike the pipe with an impact hammer, something that would be impossible if the pipe were to be buried. At some frequencies, such as from 300 to 350 Hz at Location #3 in Figure 4.4.4, it is noted that energy exists in the contour map that does not exist in the frequency response function curve. This energy is

probably induced by coupling effects and suggests that the flow sweep contour map could be used to determine other dynamic characteristics, such as damping. Although not performed in this thesis, if additional dynamic properties are to be extracted from the contour map, Natural Excitation Techniques (NExT) and Ambient Modal Analysis Techniques [79] [80] [81] [82] could be employed.

4.5 Power Variance Analysis

In this thesis work, energy from the vibration power spectra at each flow rate was integrated from 0 to 400 Hz for all cases investigated. This enables the signal variance (the vibration level) to be quantified, as shown in Figure 4.5.1. A power fit to the data is performed resulting in the following equation:

$$A'(g) = 4.326 \times 10^{-4} + 2.296 \times 10^{-16} Re^{2.607} \quad 4.1$$

and as shown by the dashed line in Figure 4.5.1.

Figure 4.5.1: Scaling of Pipe Vibration

The data plotted in Figure 4.5.1 shows that the vibration level in the pipe increases with increasing Reynolds number. In order to further investigate this relationship, the standard deviation of the pipe vibration is calculated. Figure 4.5.2 shows a comparison between the data

collected in the current investigation and that collected by Pittard et al. (2004) [62] from which it can be seen that pipes made from Acrylic, ABS, Steel, or Aluminum all behave in a similar manner. The absolute magnitude of the two data sets shown in Figure 4.5.2 left and right is different due to the fact that the pipe support structure is different between the two investigations. A power fit to the data was performed:

$$A' \left(\frac{\rho_m}{\rho_w} \right)^{\frac{1}{2}} \left(\frac{t}{D} \right) = 3.982 \times 10^{-5} + 5.126 \times 10^{-16} Re^{2.34} \quad 4.2$$

where ρ_m is the density of the pipe material, ρ_w is the density of the fluid, t is the thickness of the pipe, and D is the inner diameter of the pipe. For ABS pipe the density is 1047.19 kg/m³ and for acrylic pipe it is 1180 kg/m³. Assuming that the water is at 20°C, the density is 998.23 kg/m³. The thickness of the ABS pipe is 4.3 mm and 3.2 mm for the acrylic pipe, and the diameter of the ABS pipe is 39.9 mm and 37.1 mm for the acrylic pipe. This power fit is shown as the dashed line in Figure 4.5.2 (left).

Figure 4.5.2: Scaling of Standard Deviation of Pipe Vibration for Current Investigation (left) and Pittard et al. 2004 [62] (right)

4.6 Acoustic Analysis

The results of an acoustic analysis are discussed in this subsection. The vibration contour maps shown in Figure 4.6.1 (left) and (right) respectively represent the modal characteristics collected during a flow sweep with accelerometers mounted at locations A and B, as described in the discussion pertaining to Figure 3.5.1.

Figure 4.6.1: Vibration Contour Plots for Location A (left), B (right) - ABS Pipe

In order to investigate acoustic propagation, cross spectra and coherence functions between locations A and B are computed and contour plots generated, as shown in Figure 4.6.2 top (cross spectra) and middle (coherence). Areas with higher coherence indicate that locations A and B are correlated at a specific Reynolds number and frequency. High correlation, however, does not specify the direction of acoustic signal propagation (from A to B or B to A). To further determine propagation direction, relative phase information is calculated. Writing the one-sided cross spectra of two signals in complex form:

$$G_{xy}(f) = C_{xy}(f) - jQ_{xy}(f) \quad 4.3$$

The phase information can be computed using:

$$\phi_{xy}(f) = \tan^{-1}[Q_{xy}(f)/C_{xy}(f)] \quad 4.4$$

Figure 4.6.2: Cross Spectrum Plot (top), Coherent Contour Plot (middle) and Phase Information (bottom) for Location A and B - ABS Pipe

The contour plot of the phase information is shown in Figure 4.6.2 (bottom). Normally, the phase information for a propagating wave should have high coherence and a linear phase

relationship that either increases or decreases, depending on propagation direction. Given that the slope in this linear region can be calculated knowing the propagation speed, propagation direction can be determined. According to [35], the slope of this linear region has the following relationship when expressed in radians:

$$\text{slope} = 2 * \pi * \Delta t \quad 4.5$$

where $\Delta t = d/v$, d is the distance between location A and B (1.5 m for the current experiment), and v is the speed of sound in the propagating path. For water at 20 °C this is 1480 m/s and for ABS pipe material this is computed to be 1170 m/s using:

$$v = \sqrt{\frac{E}{\rho}} \quad 4.6$$

where E is the Young's modulus and ρ the density of the ABS material. For ABS material $E = 1.433 \text{ GPa}$.

The results from the experiment, however, do not seem to verify this theorem, i.e., frequencies from 40 to 80 Hz are highly coherent yet phase information over this frequency range is heavily contaminated. Only over a narrow frequency band from around 160 to 170 Hz does the coherence plot and phase information match each other. To illustrate this, the phase relationship at a Reynolds number of 95,000 (flow rate of 3 L/s) is shown in Figure 4.6.3 (assuming that the sound wave is propagating downstream).

Figure 4.6.3: Phase Relationship between Location A and B – ABS Pipe at $Re=95,000$

It is found that the phase information is difficult to interpret. Combined with their coherence plot, this suggests that the portion of sound from 160 to 170 Hz may be propagating downstream, but it is still hard to determine transmission path. Ignoring other environmental sources, when the wall is absolutely rigid, turbulent flow and associated hydrodynamic pressure fluctuations would be the only other source of acoustical energy [15]. The elastomer couplings and elastic supports in the experiment and the low stiffness of the plastic pipe would filter out most acoustic signals. Other investigators have attempted similar experiments and found that it is difficult to perform acoustic analysis in plastic pipes given that the acoustic signals in these pipes are heavily attenuated, generally narrow band, and at relatively low frequency [71].

To perform acoustic analysis successfully, the pipe should be assumed to be both straight and infinite without the discontinuities that can reflect the wave in the fluid [71]. Given that the flow loop used in the current experiment has a pipe riser segment and several 90-degree elbows downstream of the investigation region, it is possible for acoustic signals to propagate both forward and backward through the facility piping and thereby contaminate the phase information.

4.7 Applications in Pipe Structure Health Diagnostics

This thesis has introduced a method for investigating the modal characteristics of a pipe structure under operational conditions by using pressure fluctuations due to turbulent pipe flow as the excitation source and mechanical vibrations as the diagnostic signal. In this subsection the method is applied to pipe structural health diagnostics including loss of fluid and loss of material.

4.7.1 Loss-of-fluid experiment

The experiments were conducted at a bulk flow rate of 3 L/s ($Re = 95,000$), and data was collected as the fluid-removal flow rate was increased in discrete steps by turning the gate valve used to control flow rate in discrete half-turn increments. From the valve closed position (0 turns) to the fully-opened position (9.5 turns), a total of 20 data sets were collected and plotted as color contour plots. Contour plots generated by the original data set in either linear or log scale at different fluid-removal flow rates always showed the same result. In order to highlight the change due to fluid loss, all data was first transformed to a log scale and then subtracted from the no leak case to make the changes more pronounced, as shown in Figure 4.7.1.

Figure 4.7.1: Vibration Contour Plot with Leakage at Location #0 (upper-left), #1 (upper-right), #2 (lower-left), #3 (lower-right)

In the contour plot, the zero value on the legend color-bar scale (the yellow-green color) indicates that the amplitude of the pipe vibration under leak conditions was the same as the no-leak condition (the reference case). Colors that are redder represent a greater vibration amplitude with respect to the no-leak reference, and colors that are bluer indicate a weaker vibration amplitude. By processing the data in this manner, it was found that it was possible to detect a fluid loss starting at 0.5% the bulk flow rate. It is noted that data collected from all four accelerometer locations was similar in character, with some responses being stronger in the 150 – 200 Hz and 270 – 330 Hz ranges, while other responses were weaker at frequencies below 100 Hz and between 150 – 200 Hz. This suggests that leak indicators may be independent of location. An alternative explanation is that the measurement locations may not have been separated enough from each other. In order to understand this better, further experimentation needs to be performed. For Location #3, the contour plot is somewhat different from the other locations, and may be due to the data being influenced by the elastomer coupling upstream of the accelerometer location.

4.7.2 Loss-of-material experiment

A second application of the measurement technique involved loss of material from a pipe structure, simulating material loss due to corrosion. The experiments were conducted by performing flow rate sweep tests for each mass removal amount, and vibration signals were collected from four evenly distributed accelerometer locations (Location #0, #1, #2, and #3 as mentioned in Chapter 3). It was found that different locations have their own vibration contour maps due to unique modal characteristics at each location, while increasing removal amounts from a single location can result in similar contour maps. The latter result may be due to the fact that the changes with increased material loss may be too small to directly visualize using the current data processing technique. Figure 4.7.2 and Figure 4.7.3 respectively show the vibration contour plots for the reference case (0% material loss) and the largest material loss case (10.4% removal). The 4.4% and 7.4% material loss scenarios are not shown given their similarity.

Figure 4.7.2: Vibration Contour Plot with 0% Material Loss at Location #0 (upper-left), #1 (upper-right), #2 (lower-left), #3 (lower-right)

Figure 4.7.3: Vibration Contour Plot with 10.4% Material Loss at Location #0 (upper-left), #1 (upper-right), #2 (lower-left), #3 (lower-right)

To qualitatively analyse the changes caused by a loss of material, a suitable method for highlighting the differences needed to be identified. The following was found to be a method to amplify the difference between the material loss case (C_x) and the no-loss reference case (C_0). The data was further normalized to the range [0,1] by dividing by the difference between the maximum and minimum values measured.

$$\frac{[\ln(C_x) - \ln(C_0)] - C_{min}}{C_{max} - C_{min}}, C_{min} \leq \ln(C_x) - \ln(C_0) \leq C_{max} \quad 4.7$$

The results of all cases are shown in Figures 4.6.4 through 4.6.7. In each plot, the figure on the left side is generated using the above formula, with the color scale ranging from 0 (blue) to 1 (red). The figures on the right side correspond to the plots from the left side, only now having had a threshold of 0.8 imposed. When comparing different material loss scenarios collected from the same location, it is found that the area of the red spots increases as the amount of material removed increases. When comparing the same amount of material removed but collected from different locations, it is noted that the plots look similar to one another. Based on these two results, two possibilities arise. The first is that the change in vibration power spectral density (PSD) function caused by loss of material is independent of location. The second is that the accelerometers were not distributed far enough apart in order to result in a discernable difference. Further experiments need to be performed in order to further explore application of this method to loss of material. From the preliminary results performed to date, it is believed that material loss can be detected using the method developed, and that further refinements to the technique are possible.

Figure 4.7.4: Vibration Contour Plot of Material Losses at Location #0, from Top to Bottom Are 1.4%, 4.4%, 7.4%, 10.4%, Left Are the Normalization Contour Plot and the Right Are the Plot above Threshold of 0.8.

Figure 4.7.5: Vibration Contour Plot of Material Losses at Location #1, from Top to Bottom Are 1.4%, 4.4%, 7.4%, 10.4%, Left Are the Normalization Contour Plot and the Right Are the Plot above Threshold of 0.8.

Figure 4.7.6: Vibration Contour Plot of Material Losses at Location #2, from Top to Bottom Are 1.4%, 4.4%, 7.4%, 10.4%, Left Are the Normalization Contour Plot and the Right Are the Plot above Threshold of 0.8.

Figure 4.7.7: Vibration Contour Plot of Material Losses at Location #3, from Top to Bottom Are 1.4%, 4.4%, 7.4%, 10.4%, Left Are the Normalization Contour Plot and the Right Are the Plot above Threshold of 0.8.

Chapter Five: CONCLUSIONS

This thesis has experimentally investigated modal characteristics of pipe structures made of both acrylic and ABS material. Modal characteristics of the pipe structures were quantified using flow rate sweep tests and the results were compared to those obtained using traditional impact response tests. The flow rate sweep tests were conducted by increasing Reynolds number (flow rate) in small increments from 51,000 to 154,000. At each Reynolds number, measurements were made using both a pressure transducer and accelerometers. Data acquisition and analysis for both flow rate sweep tests and impact response tests were performed using LabVIEW.

Frequency domain analysis was performed on the time series collected from both the pressure transducer and accelerometers, mainly by computing power spectra for single signals and cross-spectra for multiple signals. Power spectra of the pressure fluctuation and vibration signals were visualized using contour plots, yielding both qualitative and quantitative information about how modal characteristics changed with increasing Reynolds number (flow rate).

A comparison of results from flow rate sweep tests was made with results from impulse response tests, and the two tests were shown to yield consistent information. This indicates that it is possible to measure pipe structure modal characteristics by varying flow Reynolds number, a condition that exists as part of normal pipeline operations (fluid density variation during fluid batching, for example). This removes the need to conduct a pipe excavation, as would be required in a Direct Assessment procedure, for instance.

An acoustic propagation analysis was performed on the vibration signals in an attempt to determine both propagation speed and path. Propagation speed through the pipe wall material

would be different from propagation through the flowing water in the pipe. Results of the acoustic propagation analysis were inconclusive, and it is presumed that this was due to multiple internal reflections within the relatively short pipe sections used in the experimental apparatus.

Two applications of flow sweep tests were performed to diagnose the pipe structure health in loss-of-fluid and loss-of-material situations. From the preliminary results performed to date, it is believed that both cases can be detected using the method developed, and that further refinements to the technique are possible.

5.1 Summarized Main Findings

The main findings of this thesis work consist of the following:

1. Wall pressure fluctuations in fully-developed turbulent pipe flow were investigated experimentally. Understanding how the pressure spectra vary with Reynolds number represents an important aspect to this investigation given that pressure fluctuations due to pipe flow turbulence are the main excitation source for pipe structural vibration. Energy distribution in the pressure fluctuation spectra was concentrated in the lower frequency region. Certain features observed in the pressure spectra were attributed to pump blade passage frequency and the methods used to join pipe sections (elastomer couplings and wall steps).
2. Flow rate sweep tests were performed with accelerometers mounted at different spatial locations on the pipe structure for both acrylic and ABS pipe. Contour plots generated from the flow rate sweep tests revealed that vibration response varies with Reynolds number and accelerometer mounting location. Strong agreement was found when modal frequencies determined using both flow rate sweep tests and impact response tests were compared. The modal shapes collected with the two tests were also in qualitative agreement. The flow rate

sweep tests yielded energy at certain frequencies that were not observed in the impact response tests, and it was hypothesized that these may have been due to mode coupling effects. Energy attributed to the pump rotational rate was clearly evident in the contour plots generated during flow rate sweep tests. All of these results indicate that flow rate sweep tests can provide dynamic characteristics of pipe structures under operational conditions in a passive manner. This has the advantage that pipe structural health can be monitored continuously and in a non-intrusive manner. Preliminary results obtained from loss of fluid and loss of material investigations did prove that this was technically feasible.

3. The acoustic analysis demonstrated inconclusive results as frequency ranges with high coherency did not exhibit a linear phase relationship. It is speculated that this resulted due to the low stiffness of the plastic pipe used in the facility, and other factors such as the elastomer couplings used to join pipe sections, the elastic supports used to support pipe sections, and the existence of elbows and risers at either ends of the pipe section. Some of these factors reduced wave propagation along the pipe, while others caused waves to reflect internally, complicating data interpretation. In order to perform acoustic analysis using plastic pipe, the pipe facility should be designed with continuous, straight and long pipe sections that ensure acoustic propagation without transitions or barriers.

5.2 Future Work

This thesis work experimentally investigated pipe wall pressure fluctuations and the pipe vibrations that result due to these pressure fluctuations for turbulent pipe flow. It was demonstrated that pipe structure modal characteristics can be quantified through flow rate sweep tests. Further investigations that could be performed include:

1. Theoretically investigate the mechanism by which turbulent pressure fluctuations induce pipe vibration. This will help to better understand the theoretical coupling between pressure fluctuations at the pipe wall (the input) and the resulting pipe vibration (the output).
2. Considering the shortcomings of the current flow facility, a new experimental pipe flow facility is suggested with longer single section pipe runs. Pipe materials other than plastic, including carbon steel and copper could be explored. A hydraulic pressure gauge should also be installed near the measurement location in order to quantify gage pressure, provided that it does not significantly interfere with the measurement process.
3. Further develop this thesis work into a more robust in-situ vibration response technique for pipeline health monitoring. Modal characteristics can be treated as a unique feature of a mechanical system. Changes to the structural properties (mass, damping and stiffness) will result in modal changes (natural frequencies, mode shapes and damping). Therefore, the condition of the pipe in response to damage including dents, internal and external corrosion, erosion and leakage, can be diagnosed through its dynamics response. Developing signal analysis techniques that are able to extract and identify these unique features (such as contour plots combined with artificial intelligence) is a promising area of research that will be performed in future investigations.

5.3 Open Questions

As with all investigations, a number of open questions remain. These include:

1. Is the method investigated sensitive to damage such as dents, cracks or internal corrosion?
2. How will soil both above and around the pipe impact the results? Will the vibration signal be strong enough to detect using commercially available accelerometers? Surrounding soil

will dampen mechanical vibrations, and at present it is unclear how this dampening effect will impact the measurement results.

3. How many pipe diameters away from a source of damage is it possible for an accelerometer to assess the health of a pipe?
4. Is it necessary for the accelerometer to be physically mounted to the pipe or is an accelerometer located in the adjacent soil sufficient enough to assess the health of a buried pipe?
5. Experiments performed to date have involved Acrylic and ABS material. It remains to be determined how the method will perform when using more common pipes made of steel with more significant internal roughness characteristics.

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APPENDIX

Appendix A: Information of Apparatus

Table A.1: M417 Information

General information:			
Brand	Marathon electric motor	Catalog No:	M417
Model:	215TTFW7022	Category:	Other Purpose
Part Description:	15, 3600, TEFC, 215JM, 3/60/208-230/460	Stock:	No
Nameplate Specifications:			
Output HP:	15 Hp	Output KW:	11.2 KW
Frequency:	60 Hz	Voltage:	208-230/460 V
Current:	40-37/18.5 A	Speed:	3520 rpm
Service Factor:	1.15	Phases:	3
Efficiency:	88.5%	Duty:	CONTINUOUS
Insulation Class:	B	Design Code:	B
KVA Code:	G	Frame:	215JM
Enclosure:	TEFC	Overload Protector:	NOT
Ambient Temperature:	40 °C	Drive End Bearing Size:	6309
Opp Drive End Bearing Size	6206	UL:	Recognized
CSA:	Y	CE:	Y
IP Code:	43		

Technical Specification:			
Electrical Type:	SQ CAGE IND RUN	Starting Method:	ACROSS THE LINE
Poles:	2	Rotation:	REV
Mounting:	RIGID	Motor Orientation:	HORIZONTAL
Drive End Bearing:	BALL	Opp Drive End Bearing:	BALL
Frame Material:	ROLLED STEEL	Shaft Type:	JM
Overall Length:	22.09 in	Frame Length:	11.15 in
Shaft Diameter:	.88 in	Shaft Extension:	4.25 in
Assembly/Box Mounting:	F1/F2 CAPABLE		

Table A.2: HP 3562A Settings Used During the Experiments

Setting	Explanation	Setting	Explanation
REMOTE	Operation through programming	PSP1	Display power spectrum channel 1
MSMD	Measure mode	PSP2	Display power spectrum channel 2
LNRS	Linear resolutions	ICPL	Input couple
SMES	Select measurement	C2AC1	AC coupling in channel 2

FRSP	Frequency response measurement	UNIT	Units
CH1	Active channel 1	PSUN	Power spectrum units
CH12	Active channel 1 and 2	V2HZ	V ² /Hz
WNDO	Window	CORD	Coordinate
HANN	Hanning window	MGLG	Log in magnitude
AVG100	Average 100	LOGX	Log in x axis
STBL	Stable(mean)	FRS 0 ,400HZ	Frequency from 0-400Hz
MDSP	Measure display		

Table A.3: Description of NI USB-6009

Analog input (AI 0-7)	Floating signal sources	Other than ± 10 V
	Referenced single-ended (RSE) signal sources (Ground)	± 10 V
Analog output (AO 0-1)	0-5 V, 5 mA	
Digital pinouts (P0.0-1.3) (Can be chosen as input and output mode)	Input	-0.3 – 0.8V (low)
		2.0 – 5.8 V (high)
	Output	Up to 0.8 V (low)
		2.0 – 3.5 V (high)
External Voltage	+5 V (max 200 mA)	
	+2.5 V (max 1 mA)	

Event counter (PFI 0)	Max input frequency 5 MHz
	Input voltage: 0.8 – 2.0 V
	Resolution: 32 bits

Table A.4: Diaphragm Ranges for DP 103

Diaphragm No. for DP 103	Pressure Range(psi)
8-08	± 0.008
8-10	± 0.0125
8-12	± 0.02
8-14	± 0.032
8-16	± 0.05
8-18	± 0.08
8-20	± 0.125
8-22	± 0.2
8-24	± 0.32
8-26	± 0.5
8-28	± 0.8
8-30	± 1.25
8-32	± 2.0
8-34	± 3.2
8-36	± 5.0
8-38	± 8.0
8-40	± 12.5

Appendix B: Derivation of PPE (pressure Poisson's Equation)

According to the continuity equation (conservation of mass) we have:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_x)}{\partial x} + \frac{\partial(\rho v_y)}{\partial y} + \frac{\partial(\rho v_z)}{\partial z} = 0 \quad \text{B.1}$$

where ρ is the density, $\frac{\partial \rho}{\partial t}$ is the accumulation (or loss) of the system, and $\frac{\partial(\rho v_x)}{\partial x} + \frac{\partial(\rho v_y)}{\partial y} + \frac{\partial(\rho v_z)}{\partial z}$ is the difference between flow in and flow out (net “flux” of the quantity through the surface of the control volume). In incompressible flow, $\rho = \text{constant}$ and therefore $\frac{\partial \rho}{\partial t} = 0$. As a result,

Equation B.1 can be re-written to:

$$\frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} + \frac{\partial v_z}{\partial z} = \nabla \cdot \vec{u} = 0 \quad \text{B.2}$$

where $\vec{u} = v_x \vec{e}_x + v_y \vec{e}_y + v_z \vec{e}_z$ and ∇ is the nabla operator (the divergence operator). $\nabla \cdot \vec{u} = \text{div } \vec{u}$, and the divergence operator acting on a vector field produces a scalar value, which if positive means expansion and negative means compression. Equation B.2 means that there is no compression or expansion of the control volume, and thus is called divergence-free.

In viscous incompressible flow, the Navier-Stokes Equations (NSE) can also be expressed in the form of both the nabla operator ($\nabla = \frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z}$) and the Laplacian operator

$$(\Delta = \frac{\partial^2}{\partial^2 x} + \frac{\partial^2}{\partial^2 y} + \frac{\partial^2}{\partial^2 z}).$$

$$\frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} = -\nabla \left(\frac{p}{\rho} \right) + \nu \Delta \vec{u} + f \quad \text{B.3}$$

where ν is the kinematic viscosity and f is the external source (body forces). Equation B.3 can be re-formatted to:

$$\nabla \cdot \left(\frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} \right) = \nabla \cdot \left[-\nabla \left(\frac{p}{\rho} \right) + \nu \Delta \vec{u} + f \right] \quad \text{B.4}$$

The LHS of equation B.4 can be simplified to:

$$\nabla \cdot \left(\frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} \right) = \frac{\partial}{\partial t} (\nabla \cdot \vec{u}) + \nabla \cdot [(\vec{u} \cdot \nabla) \vec{u}] = \nabla \cdot [(\vec{u} \cdot \nabla) \vec{u}] \quad \text{B.5}$$

and the RHS of equation B.4 can be simplified to:

$$\begin{aligned} \nabla \cdot \left[-\nabla \left(\frac{p}{\rho} \right) + v \Delta \vec{u} + f \right] &= \nabla \cdot v \Delta \vec{u} - \nabla \cdot \nabla \left(\frac{p}{\rho} \right) + \nabla \cdot f \\ &= v \nabla \cdot [\nabla \cdot (\nabla \vec{u})] - \Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f \quad \text{Rule 1} \\ &= v \nabla \cdot [\nabla (\nabla \cdot \vec{u})] - \Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f \\ &= v \Delta (\nabla \cdot \vec{u}) - \Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f \\ &= -\Delta \left(\frac{p}{\rho} \right) + \nabla \cdot f \quad \text{B.6} \end{aligned}$$

By rearranging these equations (B.4, B.5 and B.6), the pressure field can be expressed as:

$$\Delta \left(\frac{p}{\rho} \right) = \nabla \cdot f - \nabla \cdot [(\vec{u} \cdot \nabla) \vec{u}] = \nabla \cdot f - \nabla \cdot (\vec{u} \cdot \nabla \vec{u}) \quad \text{Rule 2} \quad \text{B.7}$$

Given the definition for the vorticity vector, $\vec{\omega} \equiv \nabla \times \vec{u}$, equation B.7 can be further rewritten as:

$$\begin{aligned} \Delta \left(\frac{p}{\rho} \right) &= \nabla \cdot f - \nabla \cdot (\vec{u} \cdot \nabla \vec{u}) \quad \text{Rule 3} \\ &= \nabla \cdot f - \nabla \cdot \left(\frac{1}{2} \nabla (\vec{u} \cdot \vec{u}) - \vec{u} \times (\nabla \times \vec{u}) \right) \\ &= \nabla \cdot f - \frac{1}{2} \Delta (\vec{u} \cdot \vec{u}) + \nabla \cdot (\vec{u} \times \vec{\omega}) \quad \text{Rule 4} \\ &= \nabla \cdot f - \frac{1}{2} \Delta (\vec{u} \cdot \vec{u}) + \vec{\omega} \cdot (\nabla \times \vec{u}) - \vec{u} \cdot (\nabla \times \vec{\omega}) \\ &= \nabla \cdot f - \frac{1}{2} \Delta (\vec{u} \cdot \vec{u}) + \vec{\omega} \cdot \vec{\omega} - \vec{u} \cdot (\nabla \times (\nabla \times \vec{u})) \quad \text{Rule 5} \\ &= \nabla \cdot f - \frac{1}{2} \Delta (\vec{u} \cdot \vec{u}) + \vec{\omega} \cdot \vec{\omega} - \vec{u} \cdot (\nabla (\nabla \cdot \vec{u}) - \Delta \vec{u}) \end{aligned}$$

Because $\nabla \cdot \vec{u} = 0$:

$$\Delta\left(\frac{p}{\rho}\right) = \nabla \cdot f - \frac{1}{2}\Delta(\vec{u} \cdot \vec{u}) + \vec{u} \cdot \Delta\vec{u} + \vec{\omega} \cdot \vec{\omega} \quad \text{B.8}$$

Equation B.8 is known as Pressure Poisson's Equation (PPE).

Notes:

Rule 1: $\nabla \cdot (\nabla \vec{u}) = \nabla(\nabla \cdot \vec{u})$

Rule 2: $(\vec{u} \cdot \nabla)\vec{u} = \vec{u} \cdot \nabla \vec{u}$

Rule 3: $\nabla(\vec{u} \cdot \vec{v}) = (\vec{u} \cdot \nabla)\vec{v} + (\vec{v} \cdot \nabla)\vec{u} + \vec{u} \times (\nabla \times \vec{v}) + \vec{v} \times (\nabla \times \vec{u})$

Rule 4: $\nabla \cdot (\vec{u} \times \vec{v}) = \vec{v} \cdot (\nabla \times \vec{u}) - \vec{u} \cdot (\nabla \times \vec{v})$

Rule 5: $\nabla \times (\nabla \times \vec{u}) = \nabla(\nabla \cdot \vec{u}) - \nabla^2 \vec{u} = \nabla(\nabla \cdot \vec{u}) - \Delta \vec{u}$