Vibration Control of a Pipe Submerged in Water

A Thesis

Presented to

the Faculty of the Department of Mechanical Engineering

University of Houston

In Partial Fulfillment

of the Requirements for the Degree

Master of Science

in

Mechanical Engineering

by

Jinwei Jiang

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VIBRATION CONTROL OF A PIPE SUBMERGED IN WATER

Jinwei Jiang

Approved:

Chair of the Committee
Gangbing Song, Professor
Mechanical Engineering

Committee Members:

Li Sun, Associate Professor
Mechanical Engineering

Yi-Lung Mo, Professor
Civil and Environmental Engineering

Suresh K. Khator, Associate Dean
Cullen College of Engineering

Pradeep Sharma, Professor and Chair
Mechanical Engineering

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Abstract

This thesis, for the first time, studies experimental vibration control of pipe like structures submerged in water by using a Pounding Tuned Mass Damper (PTMD). The PTMD is a novel damper that was recently invented in the Smart Materials and Structures Laboratory at the University of Houston. This new damper innovatively uses collision/impact with viscoelastic materials to dissipate vibration energy. To facilitate the experimental study, a particular submerged vibrating system consisting of four springs and a short-pipe to mimic a jumper was designed. A method to estimate the natural frequencies of a submerged structure was proposed and experimentally verified. Therefore, a PTMD was designed and fabricated to control the vibration of the submerged system. Extensive experimental results clearly demonstrated the effectiveness of the PTMD in vibration mitigation of a submerged structure. Moreover, the submerged PTMD is robust with respect to the natural frequency variation of the structure to be controlled.

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1. Introduction

1.1 Motivation and Objectives

A subsea jumper is typically subject to vortex induced vibration (VIV), which may reduce its fatigue life and may eventually lead to catastrophic failure if the vibration is not mitigated. The remoteness of the jumper makes inspection and servicing difficult, which means that a passive vibration control device will be more feasible in realizing vibration control of the subsea jumpers without requiring any external power sources. This thesis continues the research of a novel passive vibration control device, Pounding Tuned Mass Damper (PTMD), which was recently invented in the Smart Materials and Structure Laboratory (SMSL) at the University of Houston. This new damper innovatively uses collision/impact with viscoelastic materials to dissipate vibration energy. Extensive studies of the PTMD and its applications to vibration control were carried out at the SMSL in the past, however, none of the studies experimentally involved a submerged structure. This motivates this thesis research that experimentally studies vibration control of a submerged structure using the PTMD technology.

The general objective of this thesis is to experimentally verify the vibration control of a submerged structure using the PTMD. Considering the cost of the experiment and time limitation of the thesis study, the specific objective of this research is the demonstration of the effectiveness of a PTMD to mitigate the induced vibration of a submerged short pipe that mimics a jumper. The short-pipe simulates the vibration motion of subsea jumpers in cross-line direction. In general, the excitation in the cross-flow direction is potentially more dangerous than that in in-line since amplitudes of the response are much greater than those associated with in-line motion. Moreover, to ensure the proper design of a PTMD, a direct method to estimate the natural frequencies of a PTMD and a pipe submerged in water will be proposed and experimentally studied. In addition, the robustness of the PTMD in water will be experimentally investigated. This thesis is expected to contribute to a better understanding of the PTMD characteristics and its application for vibration control of subsea structures.

1.2 Contribution

The contribution of this thesis is multifold:

This thesis offers the first experimental study of PTMD vibration control of a submerged structure. This study verified the PTMD's effectiveness in the vibration control of an underwater pipe.

Secondly, a method is derived in this thesis to estimate the natural frequencies of the pipe and PTMD submerged in water. This method helps to ensure that the natural frequency of the PTMD is tuned to that of the structure to be controlled in water, and thus realizing the goal of the effective vibration control.

Moreover, this research conducts the first robustness study of a PTMD in controlling vibration of a submerged structure. We investigated the effect of mistuned on the resonant frequency on the performance of the vibration control of PTMD for the pipe in water. From our observations, we have discovered the proper frequency range to verify the relative robustness of the PTMD for the vibration control of the submerged pipe, which will be of benefit to better understand the PTMD behavior in water.

1.3 Thesis Scope and Organization

The thesis is organized in eight chapters:

Chapter one introduces the motivation and objective of this thesis. The background of the subsea field, the concepts of the subsea jumper and vertex-induced-vibration (VIV) are presented in Chapter two.

Chapter three briefly introduces the concept of vibration and the main methods to reduce the vibration used in engineering. A vibration control system mainly includes active control, passive control, or semi-active control. Furthermore, the mechanisms of a tuned mass damper (TMD) are discussed.

Chapter four presents an experiment for testing the natural frequency of the real-scale jumper in field. The experimental results were verified by the simulation results by finite element modeling (FEM) using ABAQUS software.

Chapter five, a direct method for determining natural frequencies of vibration for the pipe and PTMD submerged in water is presented. A comparison of the experimental results and the estimated results showed that the estimated natural frequencies are close to the natural frequencies measured from the experiments.

Chapter six describes the realization of vibration suppression through a PTMD mounted on the pipe submerged in water. A new vibrating system with a short pipe and a new specific PTMD device were designed and built. As observed in the experimental results, the effectiveness of the PTMD for vibration control was successfully realized on a short-pipe submerged in water.

Chapter seven summaries the effect of mistuning on the resonant frequency on the performance of the vibration control of PTMD to verify its robustness in water. We summarized the results of the performance of vibration control of the PTMD in water during a particular frequency range. Thus, the conclusion and future work related to this thesis are summarized in Chapter eight.

2 Subsea Jumpers and Vortex Induced Vibrations

2.1 Offshore Oil and Gas Exploration

The offshore oil and gas industry started in 1947 when Kerr-McGee completed the first successful offshore well in the Gulf of Mexico (GoM) [1]. The concept of subsea field development was first suggested in the early 1970s through installing subsea wellhead and production equipment on the sea floor with some other components encapsulated in a sealed chamber [2]. The hydrocarbon are extracted at the seabed, and then can be tied-back to a nearby processing facility, either on land or on an existing offshore platform, limited by the distance or offset [3]. Generally, subsea completions in less than 1,000 ft (305 m) water depths are considered to be shallow-water completions, whereas those at depths greater than 1,000 ft (305 m) are considered to be deep-water completions. In the past 40 years, with the depletion of onshore and offshore shallow-water reserves, the exploration and production of oil in deep water has become a challenge to the offshore industry [4]. Subsea systems have advanced from shallow-water, manually operated systems into systems capable of operating via remote control at water depths of up to 3,000 meters (10,000 ft), which involves many challenges including pressure containment, deep water depths, remote operation, maintenance, flow assurance, and oil recovery infrastructure [5] [6].

The subsea technology used for offshore oil and gas production is a highly specialized field of application that places high demands on engineering ingenuity [7]. The subsea production system carries some unique aspects related to the inaccessibility of the installation and its operation and servicing. These special aspects make subsea production a highly specialized engineering discipline.

2.2 The Subsea Jumpers

A subsea production system consists of a subsea completed well, seabed wellhead, subsea production tree, subsea tie-in to flowline system, and subsea equipment and control facilities to operate the well. It can range in complexity from a single satellite well with a flowline linked to a fixed platform, FPSO (Floating Production, Storage and Offloading), or onshore facilities, to several wells on a template or clustered around a manifold that transfer to a fixed or floating facility or directly to onshore facilities [8].

A subsea jumper is an M or U shaped pipe connector used to transport production fluids between two subsea components, for example, one tree and one manifold, one manifold and one manifold, and one manifold to one sled, as shown in the Figure 2-1 [1] [9]. It may also connect other subsea components such as PLEM/PLETs and riser bases.

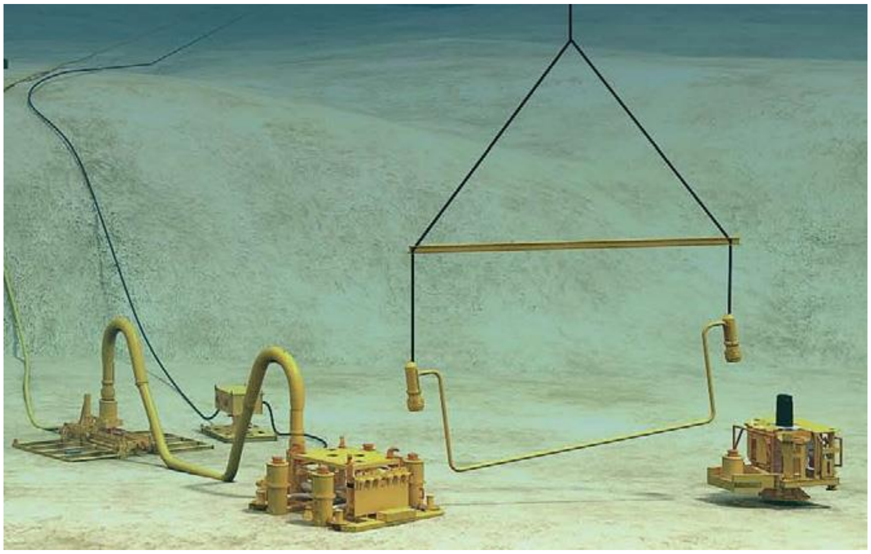

Figure 2-1 A typical jumper in a subsea field

If the involved pipe is rigid, one jumper is called a rigid jumper; otherwise, if the pipe is flexible, the jumper is called a flexible jumper [10]. Figure 2-2 shows some rigid jumper configurations [1]. For the rigid pipe jumpers, the M-shaped style and inverted U-shaped style are two commonly used styles. There is also the horizontal Z-shaped style among others. Jumper configurations are dictated by design parameters, interfaces with subsea equipment, and the different modes in which the jumper will operate.

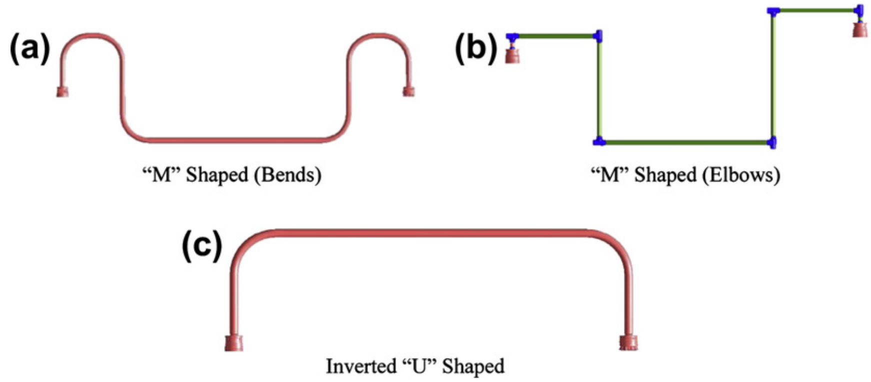

Figure 2-2 Rigid Jumper Configurations

Figure 2-3 shows a configuration for a subsea flexible jumper [1]. It consists of two end connectors and a flexible pipe between the two connectors. The subsea flexible jumpers are usually used for transporting fluid between two subsea components. They are also used to separate the rigid riser from the vessel to effectively isolate the riser from the fatigue due to the motions of the FPSO [8] [9].

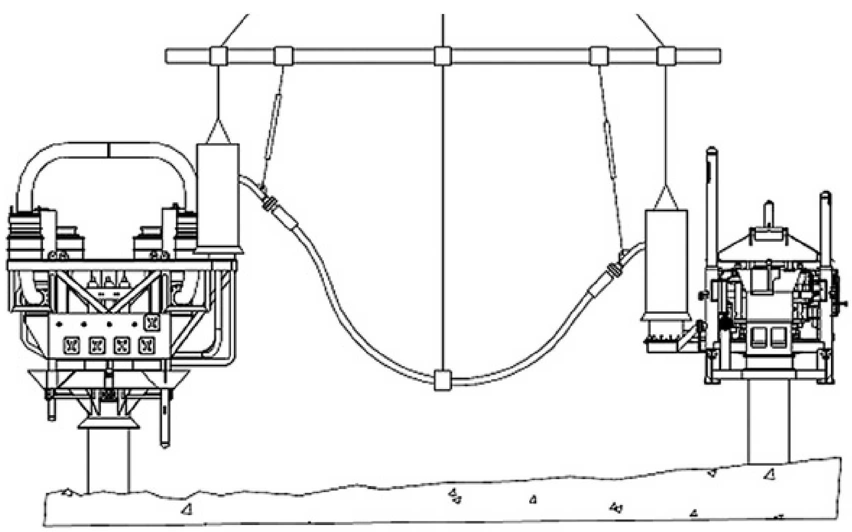

Figure 2-3 A Flexible Jumper between Tree and Manifold

2.3 Vortex Induced Vibrations (VIV)

However, when a fluid flows externally across a jumper, the flow separates, vortices are shed, and a periodic wake is formed [10] [11]. Each time a vortex is shed it alters the local pressure distribution, and the jumper experiences a time-varying force at the frequency of the vortex shedding [12]. Under resonant conditions, sustained oscillations can be excited and the jumper will oscillate at that frequency [13]. This oscillation will fatigue the jumper and eventually lead to catastrophic failure if the vibration is not properly mitigated.

These oscillations are normally in-line with the flow direction but can be transverse (cross-flow), depending on the current velocity and span length [14]. In-line oscillations are excited at flow velocities lower than the critical velocities for cross flow motion. However, the amplitude of the in-line motion is only 10% of those associated with cross-flow motion. Additionally, excitation in the cross-flow direction is potentially more dangerous than that in in-line since amplitudes of the response are much greater than those associated with in-line motion [10] [13] [15]. Thus, jumpers are subject to vortex induced vibration (VIV) when the vortex shedding frequency matches with one of the jumper's natural frequencies [16].

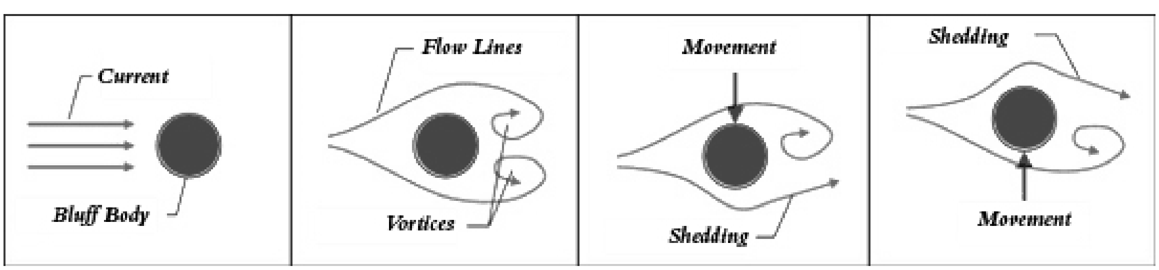

Figure 2-4 Vortex induced vibration motions

3 Study of Vibration and Vibration Control

3.1 Introduction to Vibration

Vibration is the repetitive or periodic behavior in a mechanical system [17]. Vibrations can occur naturally or may be forced through certain forms of excitation that may be either generated internally with the system or imparted through external sources [18]. When given an initial excitation and allowed to freely vibrate without any external forcing excitation, the system will tend to vibrate at a particular frequency and maintain a particular geometric shape. This frequency is termed the natural frequency of the system, and the corresponding shape or the motion ratio of the system is termed as mode shape [19]. In general, every vibrating system can be represented in terms of the natural frequency and mode shape.

The analysis of a vibrating system can be done either in the time domain or in the frequency domain. In the time domain, time is the independent variable of a vibration signal. In the frequency domain, the independent variable of a vibration signal is frequency. Even though the inputs and outputs are functions of time, they can also be represented as function of frequency through the Fourier transform. The resulting Fourier spectrum of a signal can be interpreted as the set of frequency components that the original signal contains. At the frequency domain, it will be helpful to highlight many salient characteristics of the signal and those of the corresponding dynamic system.

When the frequency of the forcing excitation coincides with that of the natural motion, the system will respond more vigorously with increased amplitude. This condition is known as resonance, and the associated frequency is called the resonant frequency.

Vibration may be simply termed as good or bad, the former referring a useful purposes and the latter having harmful or unpleasant effects. In most cases, vibration responses at a resonance would be undesirable and even may be destructive. Hence, suppression or elimination of undesirable vibrations and generation of desired forms and levels of desired vibration are the goals of vibration control.

3.2 Vibration Control

Vibration control, by definition, is to eliminate or reduce the undesirable effects of vibration [20]. In general, vibrations are a part of our environment and daily life. Many of them are useful and are needed for many purposes, with one of the best examples being the hearing system. However, vibrations are often undesirable and must be suppressed or absorbed, as they may be harmful to structures by generating damages or compromise the comfort of users through noise generated by mechanical bodies. The purpose of vibration control is to efficiently controlling the undesirable vibrations and limiting their effects so as to control its vibration responses in a desired manner [21].

For example, consider a simple spring-mass-damper system shown in the Figure 3-1 [22]. The vibration responses y is caused by the forcing excitation $f(\mathrm{t})$ to the mechanical system S. The objective of the vibration control is to suppress the vibration responses y to an acceptable level.

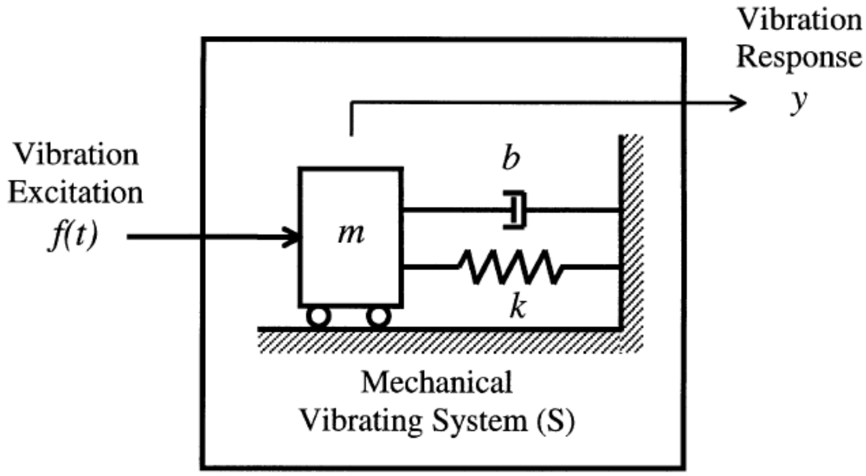

Figure 3-1 A vibrating mechanical system

A vibration control system can be classified as active control, passive control, or semi-active control depending on whether the external power required for the vibration control system to perform [23]. A passive vibration control consists of a resilient component (stiffness) and energy dissipater (damper) either to dissipate vibration energy as the function of the damper, or absorbs and stores the vibration energy where it is slowly dissipated as the case of a dynamic absorber [24]. A passive vibration control system may be attached or embedded to a primary structure, designed to modify the stiffness or the damping of the structure in an appropriate manner without requiring an external power source to operate, generating the control forces opposite to the motion of controlled primary structural system [25]. This type of vibration control system performs significantly within the frequency region of its highest sensitivity. For wideband excitation frequency, its performance can be enhanced considerably by optimizing the system parameters. However, this improvement is achieved at the cost of lowering narrowband suppression characteristics [26].

The passive vibration control has significant limitations in structural applications where broadband disturbances of highly uncertain nature are encountered. To compensate for these limitations, active vibration-control systems are utilized [27]. Typically, for an active control systems, vibration responses are explicitly sensed by the sensors and transducers, and the forces needed to counteract the vibrations are estimated by the controller. The corresponding forces or torques are applied to the system through one or more actuators. With an additional active force introduced as a part of absorber subsection, the system is controlled using different algorithms to make it more responsive to sources of disturbance. In most cases, an active solution should normally be considered only after all other passive means have been exhausted. Unlike passive systems, active systems can constantly supply energy into the system based on the change in the instantaneous operating conditions as measured by sensors [21] [28]. The key of the control system is the actuator that is typically made of pneumatic, hydraulic, electromagnetic, or intelligent materials, which can potentially affect the system in an intelligent manner [23]. However, the real problem of the active system is that their energy requirement is large for control actuators that apply forces to the structure in the required manner.

Semi-active (SA) vibration control systems are a combination of active and passive control systems [29]. The external energy requirements of the SA vibration control system are smaller than those of typical active controls. It is intended to reduce the amount of external power necessary to achieve the desired performance characteristics. Semi-active devices cannot add or remove energy to the structural system, but can control in real time parameters of the structures such as spring stiffness or coefficient of viscous damping [30].

The stability is guaranteed, in the sense that no instability can occur, because semi-active device utilize the motion of the structure to develop the force.

3.3 Tuned Mass Dampers

3.3.1 Introduction

The tuned mass damper (TMD) is a passive energy absorbing device consisting of a mass, a spring, and a damper that is attached to a primary structure in order to reduce the undesirable dynamic response of the structure [24]. This concept was first introduced by H. Frahm in 1909. Since then, much research have been carried out to investigate their effectiveness for different dynamic loading applications [25]. Den Hartog described the solution for determining the optimum tuning frequency and the optimum damping ratio of the TMD for undamped primary structures subjected to harmonic external force, thereby reducing the steady-state response of the primary systems to a minimum over a broad spectrum of forcing frequencies [22]. Warburton, based on Den Hartog's procedure, derived the optimum damper parameters for an undamped main structure subjected to harmonic support motion where the acceleration amplitude is fixed for all input frequencies and other kinds of harmonic excitation sources. Later work generalized the vibration control of the structures subjected to wide excitation and to the control of continuous systems [31].

The mechanism of suppressing structural vibrations by attaching a TMD to the structure is to transfer the vibration energy of the structure to the TMD, and dissipate the energy through the damper of the TMD [32]. To receive a significant portion of the vibration energy from the primary system in a TMD, it is important to tune the natural frequency of the TMD to the natural frequency of the primary structure and to select the appropriate capacity of the damper [22] [26] [33]. The resulting vibration of the TMD in effect applies an oscillatory force opposing the vibration of the primary system and thereby reduce vibration. In theory, the vibration of the primary structure can be completely removed while the TMD itself undergoes vibratory motion. This means that there exist optimum parameters of a TMD and the optimization of a TMD for different types of structural oscillations, such as free vibration, randomly forced vibration, and harmonically forced vibration, has been investigated by many researchers [34]. The explicit formulae for the optimum parameters of a TMD and its effectiveness are now available for the different types of vibrations.

However, there are some disadvantages in a TMD. It is effective only for the case where the frequency of the excitation was determined so that the TMD system could be designed with a natural frequency equal to the excitation frequency. The effectiveness of a TMD is decreased significantly by the mistuning or the off-optimum damping ratio of the TMD. Hence, the disadvantages is the sensitivity of the effectiveness of a TMD to a fluctuation in either the natural frequency of the structure or the damping ratio of the TMD.

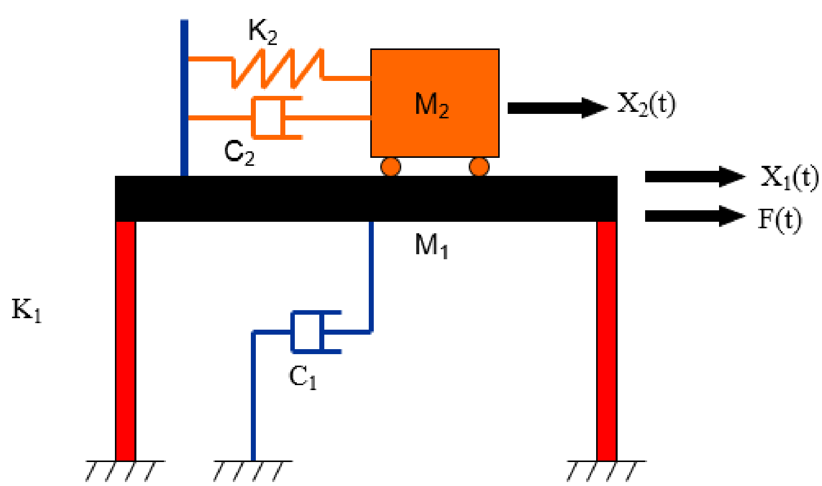

Figure 3-2 one-degree-of-freedom system mounted with TMD

3.3.2 Basic Theory

Consider a one-degree-of-freedom system fitted with a tuned mass damper, as shown in figure 2-2 [33]. The equation of the motion of the resultant two-degree-of-freedom system subject to an external excitation force F can be written in the form

$$m_1 \ddot{x}_1 + k_1 x_1 + c_1 \dot{x} + k_2 (x_1 - x_2) + c_2 (\dot{x}_1 - \dot{x}_2) = F \quad (3.1)$$

$$m_2 \ddot{x}_2 + k_2 (x_2 - x_1) + c_2 (\dot{x}_2 - \dot{x}_1) = 0 \quad (3.2)$$

where m_1, m_2 are mass of main system and mass of tuned mass damper, k_1, k_2 are the stiffness of main system and stiffness of tuned mass damper, c_1, c_2 are damping capacity of the main system and tuned mass damper, respectively. $F(t)$ is the force acting on main mass. In case of base excitation with acceleration $\ddot{x}_g(t)$,

$$F(t) = -m_1 \ddot{x}_g(t) \quad (3.3)$$

When the support excitation is harmonic and its displacement amplitude is fixed and independent of input frequencies, i.e. $u_g = He^{iwt}$, the steady-state response of the system can be solved from

$$\begin{bmatrix} \omega_1^2 + \gamma\omega_2^2 - \omega^2 + i2\omega(\xi_1\omega_1 + \gamma\xi_2\omega_2) & -\gamma\omega_2^{\,2} - i2\gamma\xi_2\omega_2\omega \\ -\gamma\omega_2^{\,2} - i2\gamma\xi_2\omega_2\omega & \gamma\omega_2^2 - \gamma\omega^2 + i2\gamma\xi_2\omega_2\omega \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \omega^2 He^{i\omega t} \begin{Bmatrix} 1 \\ \gamma \end{Bmatrix} \quad (3.4)$$

where $\omega_1 = \sqrt{k_1 / m_1}$ and $\omega_2 = \sqrt{k_2 / m_2}$ are the natural frequencies of the main mass and the tuned mass damper, respectively. $\xi_1 = c_1 / 2\omega_1 m_1$ and $\xi_2 = c_2 / 2\omega_2 m_2$ are their associated damping ratios. Another parameter, $\gamma = m_2 / m_1$, is the ratio of the damper mass to the main mass.

Den Hartog developed closed form expressions of optimum damper parameters f and ξ_2 which minimize the steady-state response of the main mass subjected to a harmonic excitation. The expressions for calculating optimum damper parameters are given as:

$$f_{opt} = \frac{1}{1+\mu} \quad , \tag{3.5}$$

$$\xi_{2opt} = \sqrt{\frac{3\mu}{8(1+\mu)}} \quad . \tag{3.6}$$

For the case when the structure is subjected to harmonic base excitation, the corresponding expressions can be easily found to be:

$$f_{opt} = \frac{1}{1+\mu}\left(\sqrt{\frac{2-\mu}{2}}\right) , \tag{3.7}$$

$$\xi_{2opt} = \sqrt{\frac{3\mu}{8(1+\mu)}}\left(\sqrt{\frac{2-\mu}{2}}\right) . \tag{3.8}$$

Using the values of f and ξ_2, the optimum values of damping c and stiffness k of the tuned mass damper can be calculated as

$$k_{opt} = f_{opt}^2 (\frac{k_1}{m_1})^2 m_2 \quad , \tag{3.9}$$

$$c_{opt} = 2\xi_{2opt} f_{opt} (\frac{k_1}{m_1}) \mathrm{m}_1 \quad . \tag{3.10}$$

4. Dynamic Testing of a Full-Scale Jumper

In this chapter, we will conduct experiments on the natural frequency of a real-scale jumper in field. And the experimental results of the natural frequencies of the jumper have been verified by the simulation results by the FEM. By determining the natural frequency of the jumper, it is beneficial for designing a particular vibrating system in the water to mimics the vibration motions of the jumper, and thus realizing the vibration control of the PTMD on the short-pipe in water for the next chapter.

4.1 Introduction

The complex shape of the jumper means that accurate estimation of the natural frequency is not possible with the direct calculation methods. The natural frequency of a jumper system is a function of many variables, i.e., the current profile, vortex shedding, hydrodynamic damping, span length, leg height, pipe diameter and thickness, buoyancy placement, buoyancy uplift, buoyancy OD, insulation thickness and contents of the jumper. Since many variables in the calculation of the natural frequency should be considered, an accurate estimation of the natural frequency of the jumper can become complicated.

Thus, this chapter will present a method to determine the natural frequency of a full-scale jumper in the field. We utilized a unit accelerometer sensor, a displacement sensor, and an ultrasonic sensor to monitor the vibration motions of the real-scale jumper in field. Through analyzing the vibration signals we can acquire the natural frequencies of the jumper and the results are also verified by the finite element analysis by the ABAQUS.

4.2 Experimental Setup

To measure the natural frequency of the jumper, a full-scale M-shaped rigid jumper consisting two end connecters was set up in the field as shown in the Figure 4-2. The main dimension of this jumper are as follows: the total length is 780 inches; the total height is 239 inches; the outer diameter is 12.75 inches; and the wall thickness is 0.843inches. Meanwhile, as shown in the Figure 4-3, Figure 4-4, and Figure 4-5, a unit accelerometer sensor, a displacement sensor, an ultrasonic sensor and a data acquisition system were utilized to monitor the vibration motions of the full-scale jumper.

In the experiment, we gave an initial excitation in X, Y, and Z directions respectively, as shown in the Figure 4-1, and was subsequently allowed to freely vibrate without any external forcing excitation. The jumper vibrated at its natural frequency, and the corresponding shape or the motion ratio of the jumper was the mode shape related to the natural frequency. The free vibration response of the jumper can be recorded or measured by the sensors in the time domain. Even though the inputs and outputs are functions of time, they can also be represented as function of frequency through the Fourier transformation.

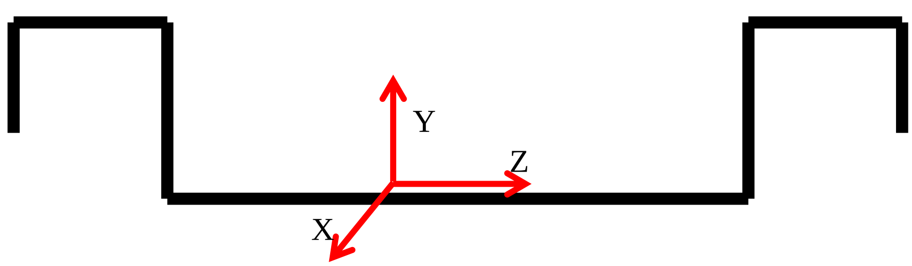

Figure 4-1 Free vibration of the jumper in three directions

Figure 4-2 A real-scale jumper with two end connecters in field

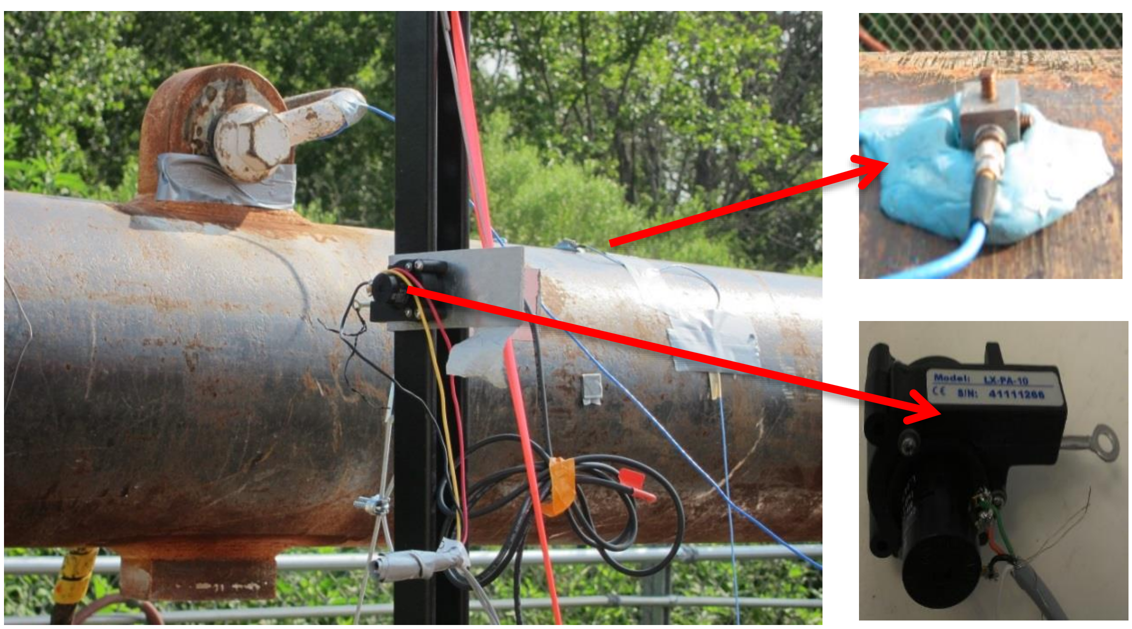

Figure 4-3 A displacement sensor and a PZT accelerometer sensor

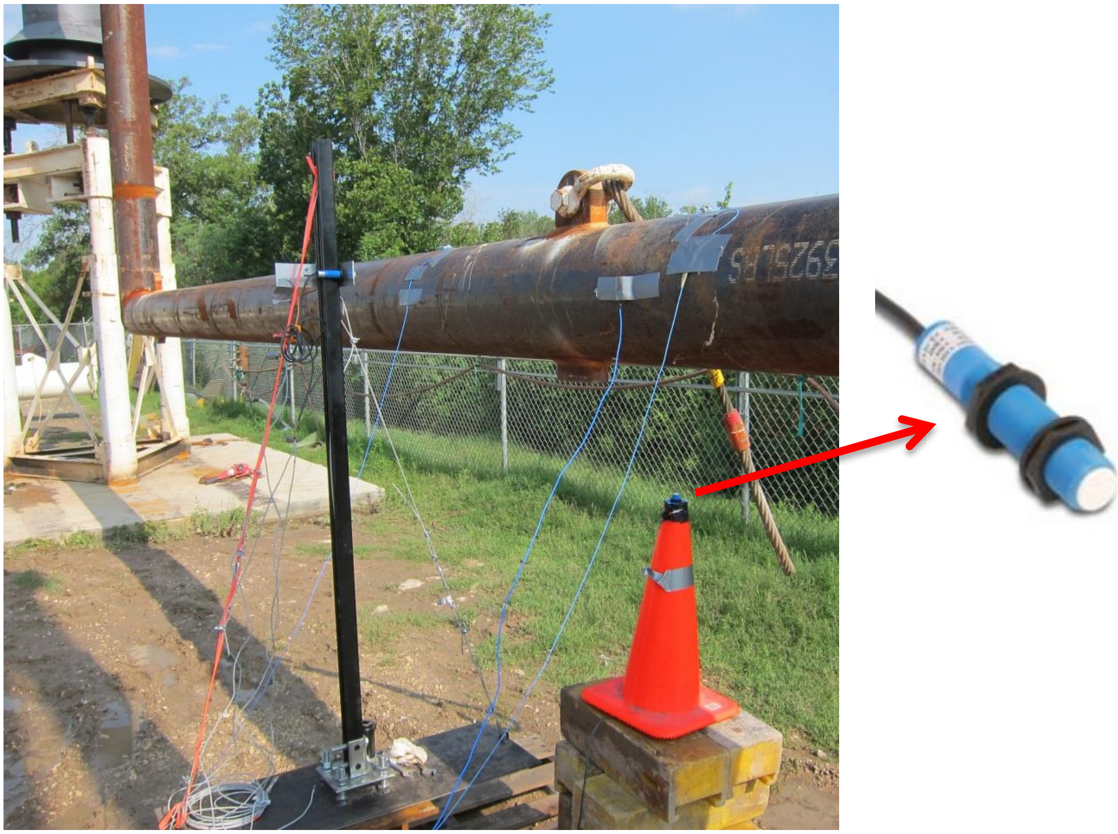

Figure 4-4 An ultrasonic sensor

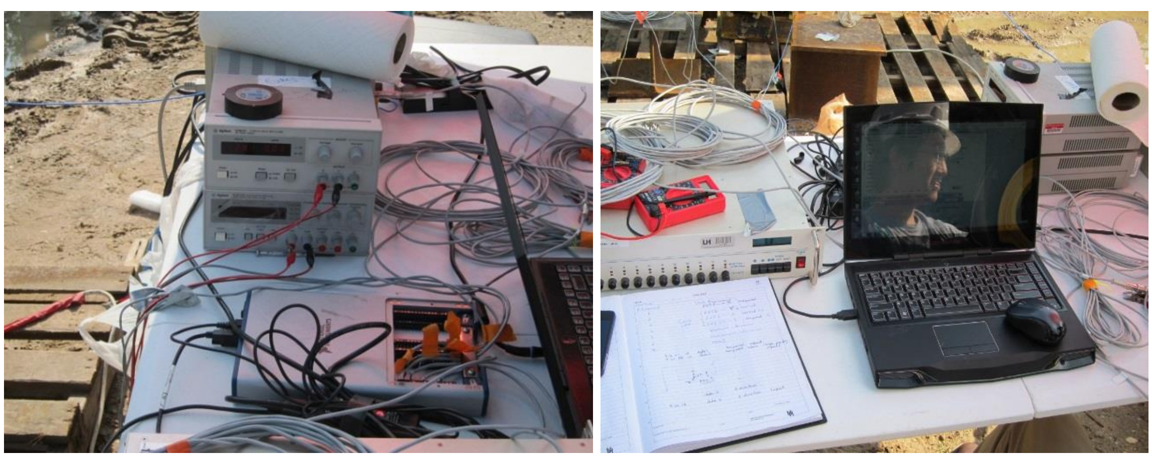

Figure 4-5 Data acquisition system

4.3 Experimental Results

4.3.1 Free Vibration of the Jumper in X Direction

Both Figure 4-6 and Figure 4-8 show the free vibration responses of the full-scale jumper measured by the displacement sensor and the accelerometer sensor, respectively, in the X direction. The frequency representation of the time domain data via Fourier transform is shown in Figure 4-7 and in Figure 4-9. The natural frequency of the jumper was 1.099Hz (Figure 4-7) and 1.172Hz (Figure 4-8) in the X direction. In addition, a peak at 10.4Hz demonstrates the natural frequency of a higher mode shape. That value was also encountered in the Y and Z directions. These values have been verified by the finite element analysis of the jumper.

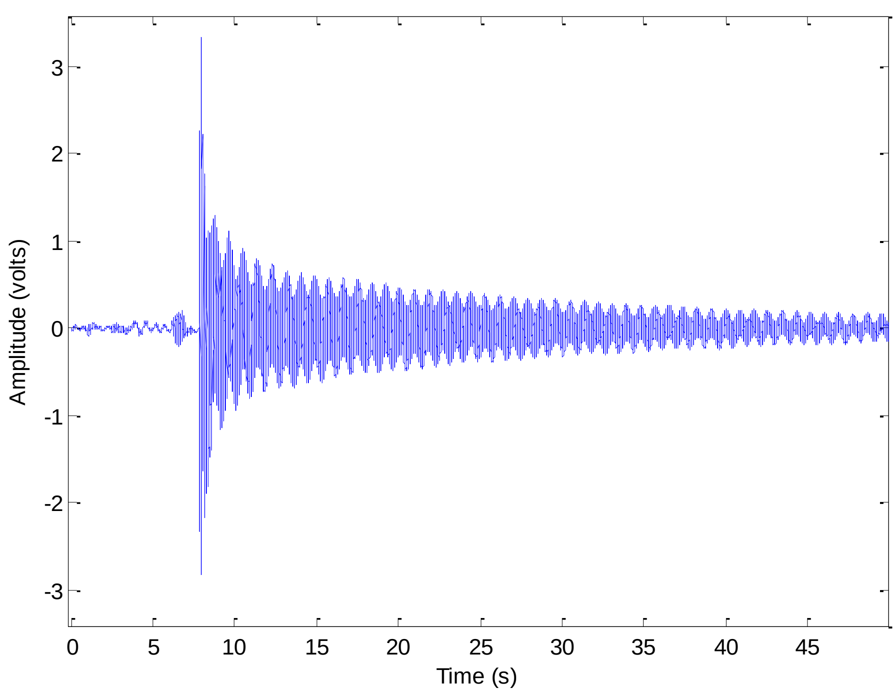

Figure 4-6 Free vibration response from the displacement sensor in X direction

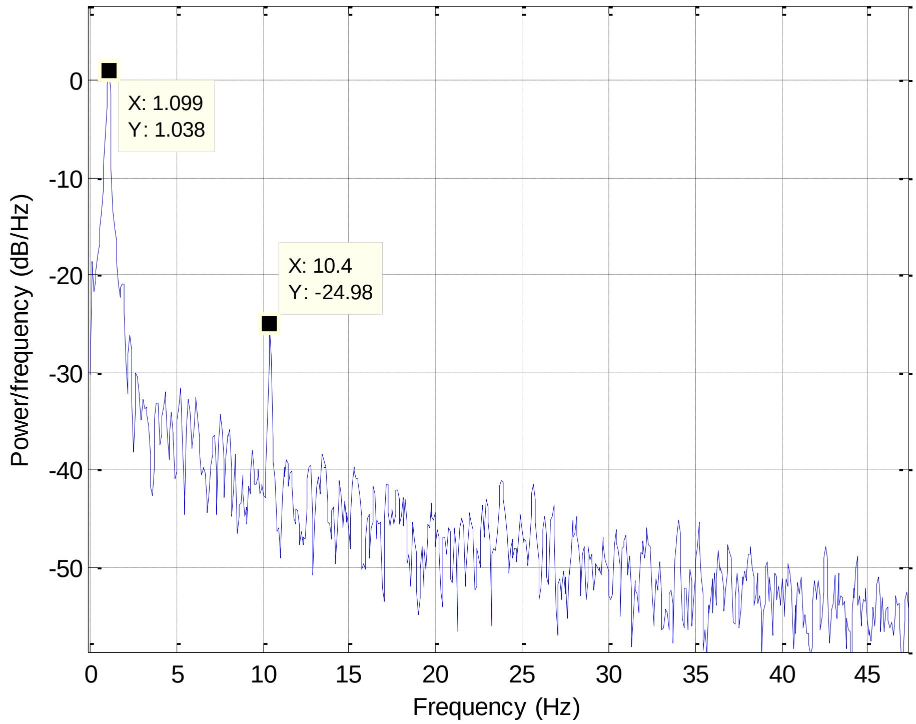

Figure 4-7 Frequency spectrum of the signal of the displacement sensor in X direction

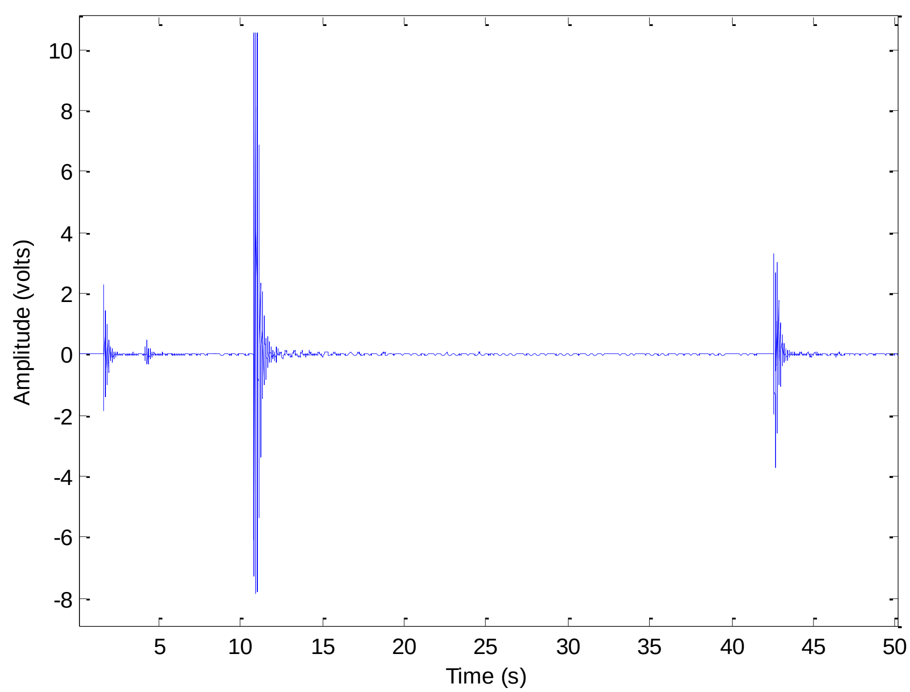

Figure 4-8 Free vibration response from the accelerometer sensor in X direction

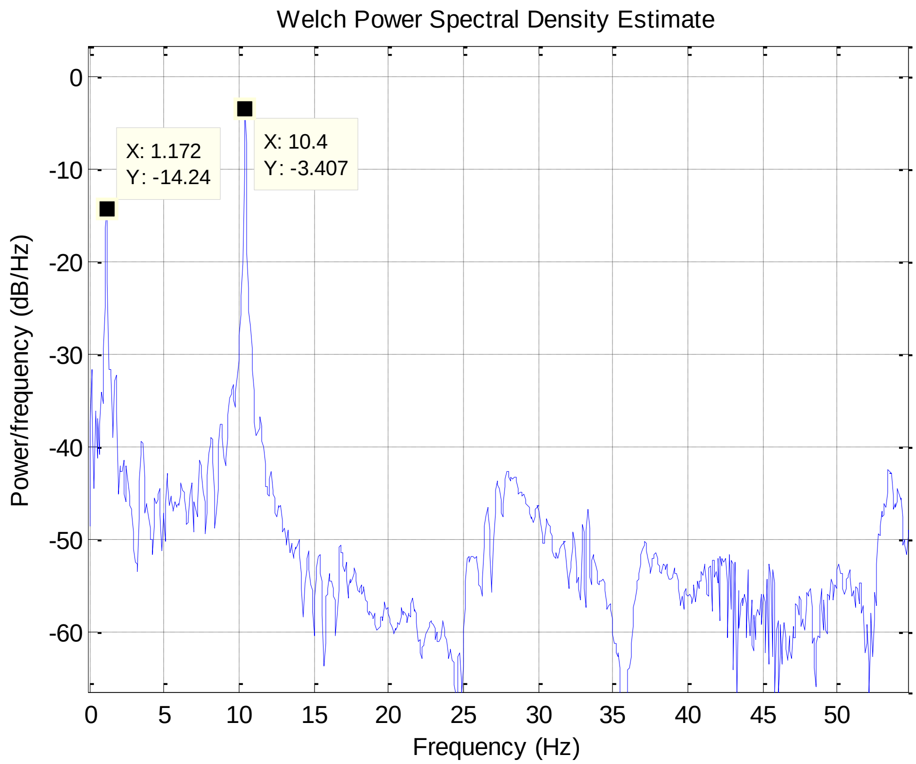

Figure 4-9 Frequency spectrum of the signal of the accelerometer sensor in X direction

4.3.2 Free Vibration of the Jumper in Y Direction

Figure 4-10 and Figure 4-12 show the free vibration of the jumper by the accelerometer sensor and the ultrasonic sensor, respectively. In the frequency domain of the accelerometer signal in the Y direction (Figure 4-11), the natural frequency of the jumper was measured at 1.978Hz, and the natural frequency of the higher mode shape was measured at 10.4Hz. Meanwhile, by the analysis of frequency domain of the signal from the ultrasonic sensor (Figure 4-13), the natural frequency of the jumper was 2.051Hz in the Y direction.

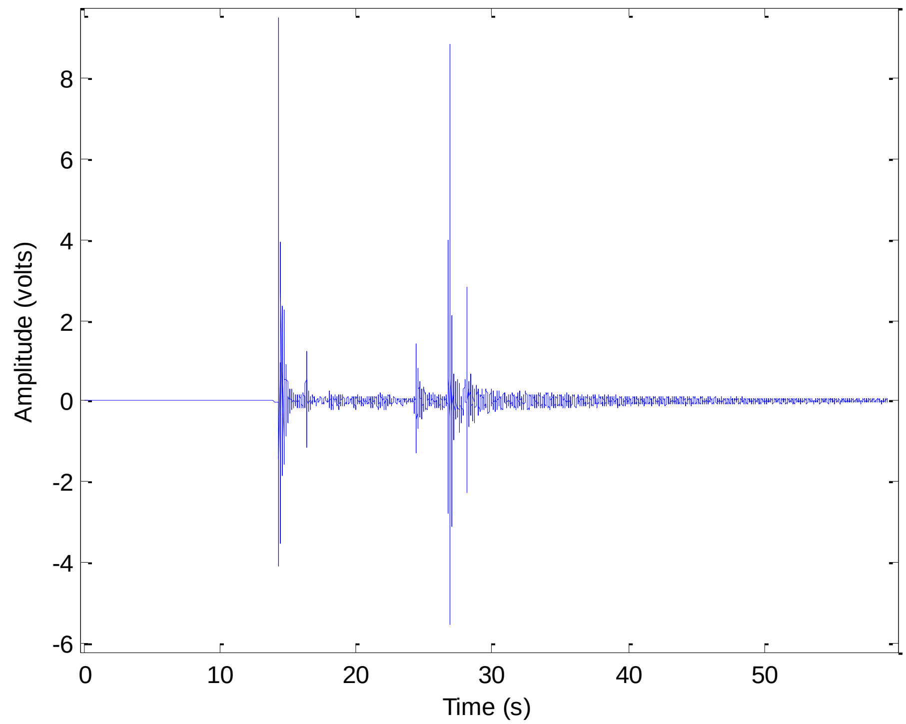

Figure 4-10 Free vibration response from the accelerometer sensor in Y direction

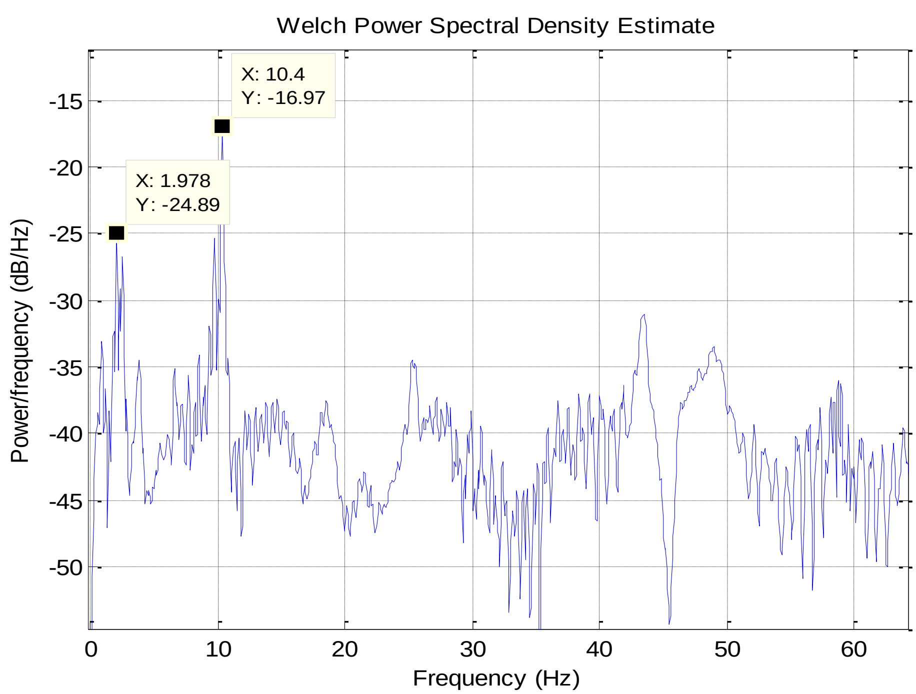

Figure 4-11 Frequency spectrum of the signal of the accelerometer sensor in Y direction

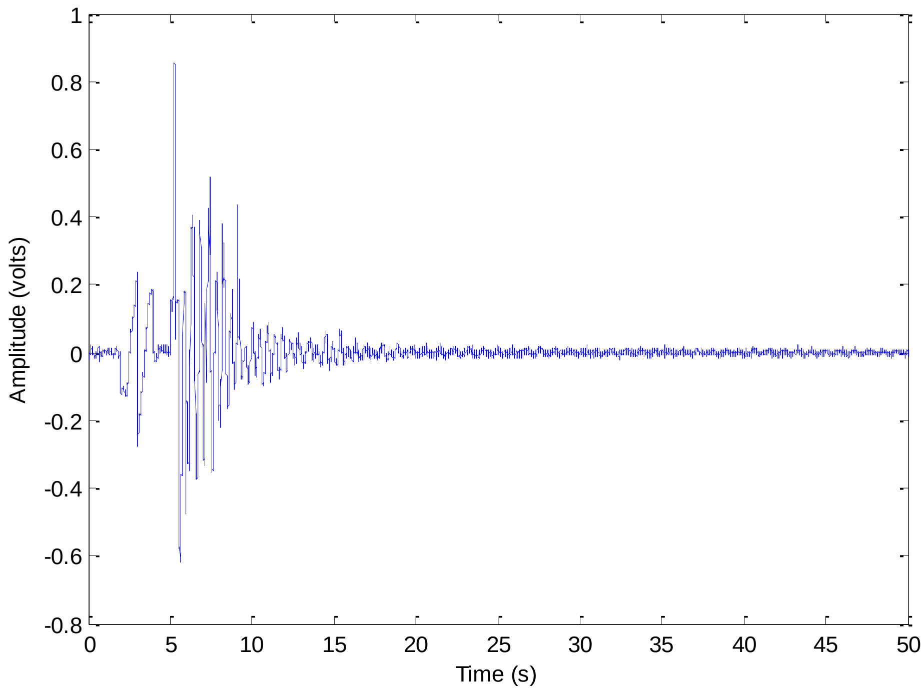

Figure 4-12 Free vibration response from the ultrasonic sensor in Y direction

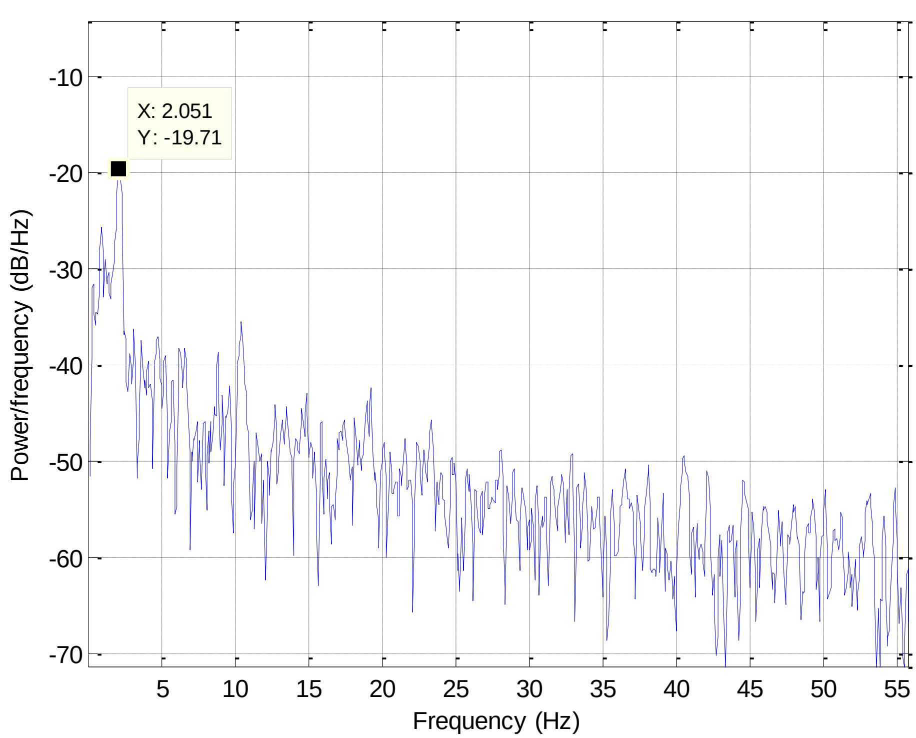

Figure 4-13 Frequency spectrum of the signal of the ultrasonic sensor in Y direction

4.3.3 Free Vibration of the Jumper in Z Direction

Figure 4-14 shows the free vibration response of the jumper by the accelerometer sensor in Z direction. In addition, Figure 4-15 demonstrates the corresponding function in the frequency domain, it obviously shows the natural frequencies of the jumper in the Z direction. The natural frequency of the 1st mode of vibration was 1.904Hz, and the natural frequency of the higher mode vibration was 10.4Hz. Figure 4-16 shows the free vibration responses of the jumper in the time domain in the Z direction. According to the frequency domain analysis, the natural frequency of the 1st mode was 1.978Hz, and the natural frequency of the higher mode was 10.33Hz, as shown in the Figure 4-17. The natural frequencies of the jumper in all three directions are summarized in table 4-1.

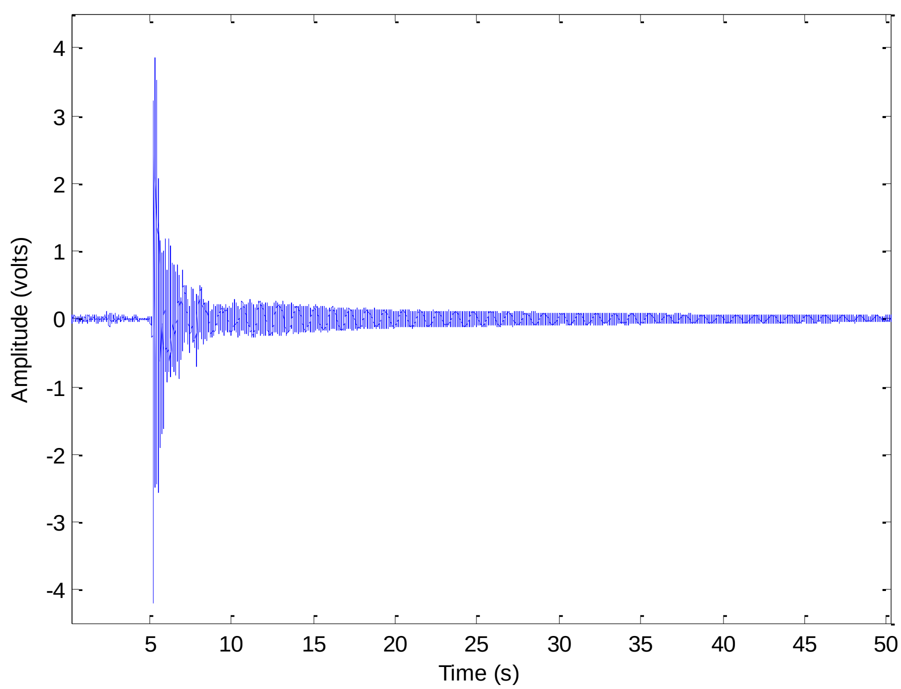

Figure 4-14 Free vibration response from the accelerometer sensor in Z direction

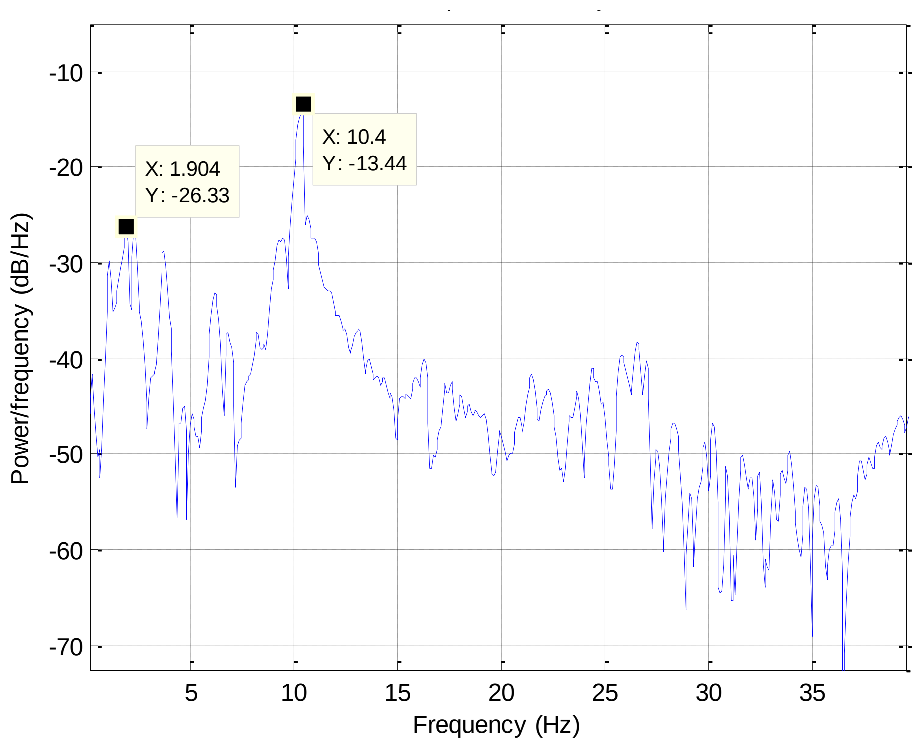

Figure 4-15 Frequency spectrum of the signal of the accelerometer sensor in Z direction

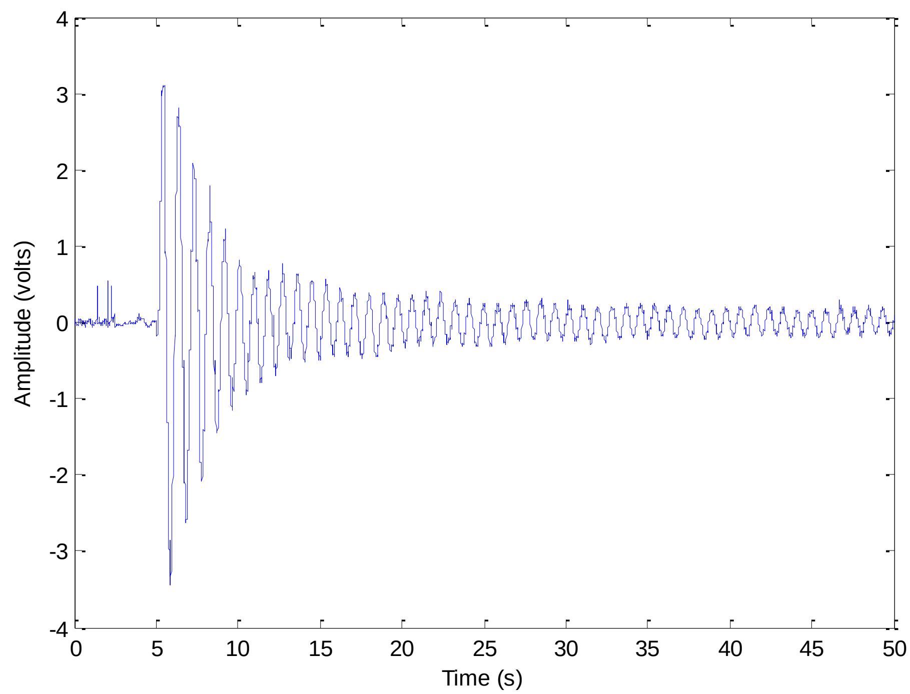

Figure 4-16 Free vibration response from the displacement sensor in Z direction

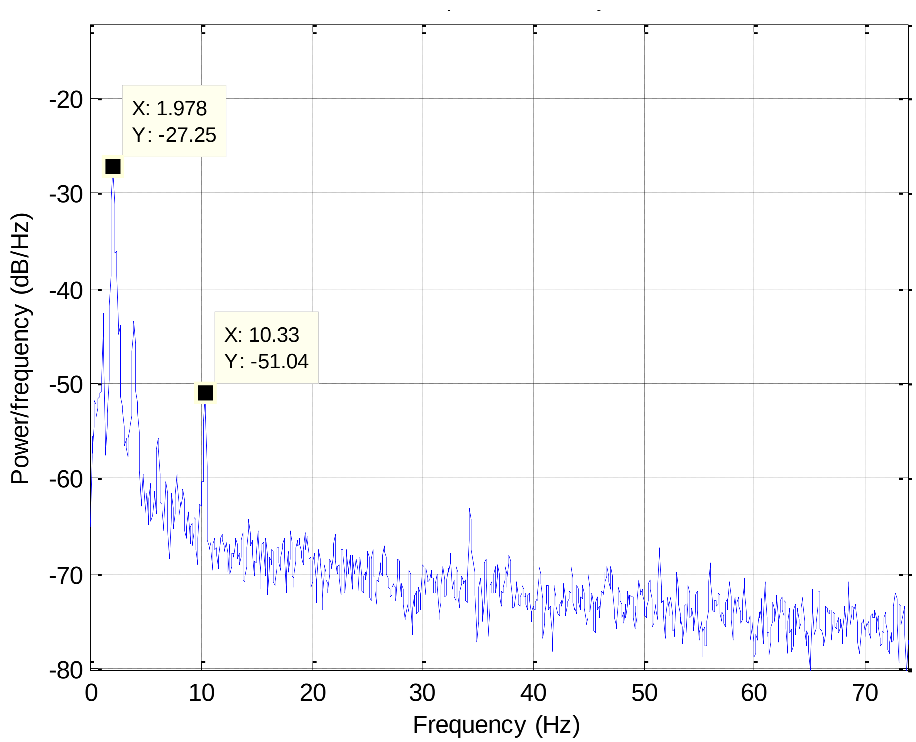

Figure 4-17 Frequency spectrum of the signal of the displacement sensor in Z direction

Table 4-1 Natural frequencies of the jumper from the measurement in field

No.	Direction	Description	1st Natural Frequency (Hz)	2nd Natural Frequency (Hz)
1	X	Displacement Sensor	1.099	10.4
		Accelerometer Sensor	1.172	10.4
2	Y	Accelerometer Sensor	1.978	10.4
		Ultrasonic Sensor	2.051	N/A
3	Z	Accelerometer Sensor	1.904	10.4
		Displacement Sensor	1.908	10.33

4.4 Simulation Results by Finite Element Method (FEM)

To simulate the vibration response of the real-scale jumper, a 3D model has been built, based on the detailed dimension, in ABAQUS. In general, the number of eigenvalues is the number of natural frequencies to be determined for a structure. There are multiple natural frequencies of a structure, and typically only the first four or five are physically relevant. Higher values for mode shapes exist mathematically, but do not significantly impact the physical behavior of the structure. The mode shape calculated by ABAQUS presents the deformed shape the structure will assume during vibration at the particular natural frequency.

By defining the dimensions and the material properties as shown in Table 4-2, we can calculate the multiple natural frequencies of the jumper utilizing the ABAQUS FEM software. Figure 4-18 demonstrates the simulation of vibration response of the jumper in the X direction; Figure 4-19 shows the simulation results in the Y direction; and Figure 4-20 represents the vibration response in the Z direction. In addition, Figure 4-21 shows the higher mode shape of the jumper which has been monitored by the measurement of natural frequencies of the jumper in field. Each vibration response of the simulation results represents a particular value of the natural frequency. The mode shape displayed by ABAQUS is not the true dimensions for the mode shape, as there are no true deformation scales in a natural frequency extraction, but is instead a relative deformation that shows how much one node deflects compared to the other nodes. The simulation results of the natural frequency of the jumper are listed in Table 4-3.

Table 4-2 A real-scale jumper section dimensions

No.	Description	Unit (inch)
1	Outer Diameter (OD)	12.75
2	Wall Thickness (WT)	0.843
3	Total Length	780
4	Total Height	239
5	Bottom Length	480

Table 4-3 Summary of simulation results of the natural frequencies of the real-scale jumper by Abaqus

Index	Description	Natural Frequency (Hz)
1	X	1.184
2	Y	1.956
3	Z	2.018
…		
9	Higher mode	10.305

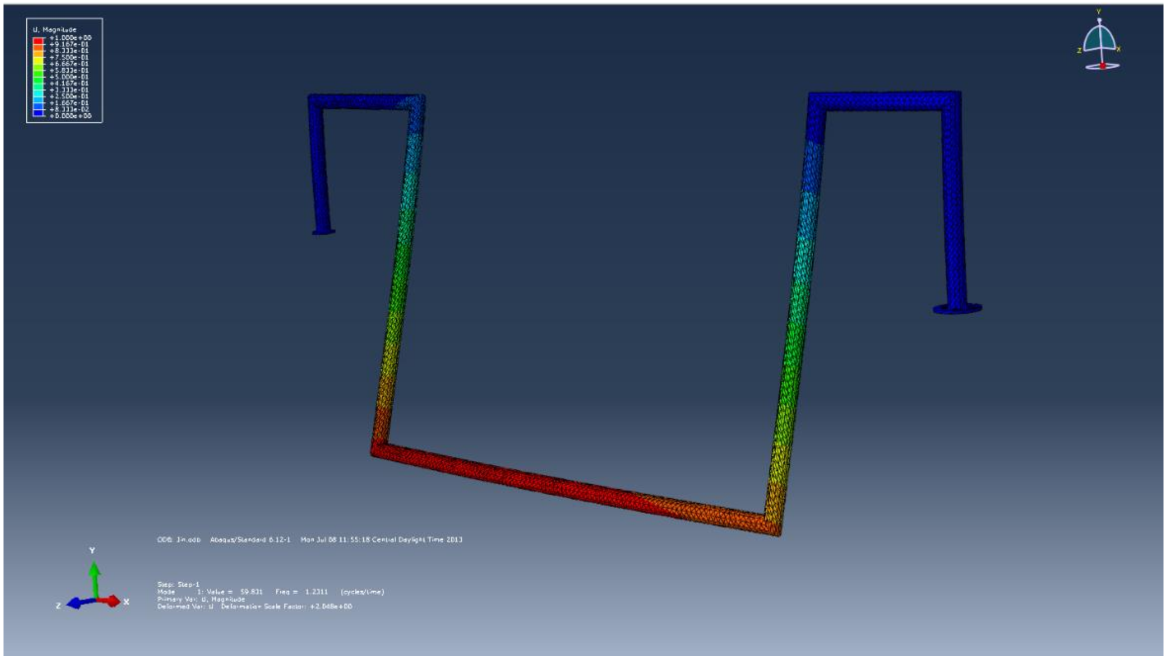

Figure 4-18 Vibration response in X direction from ABAQUS software

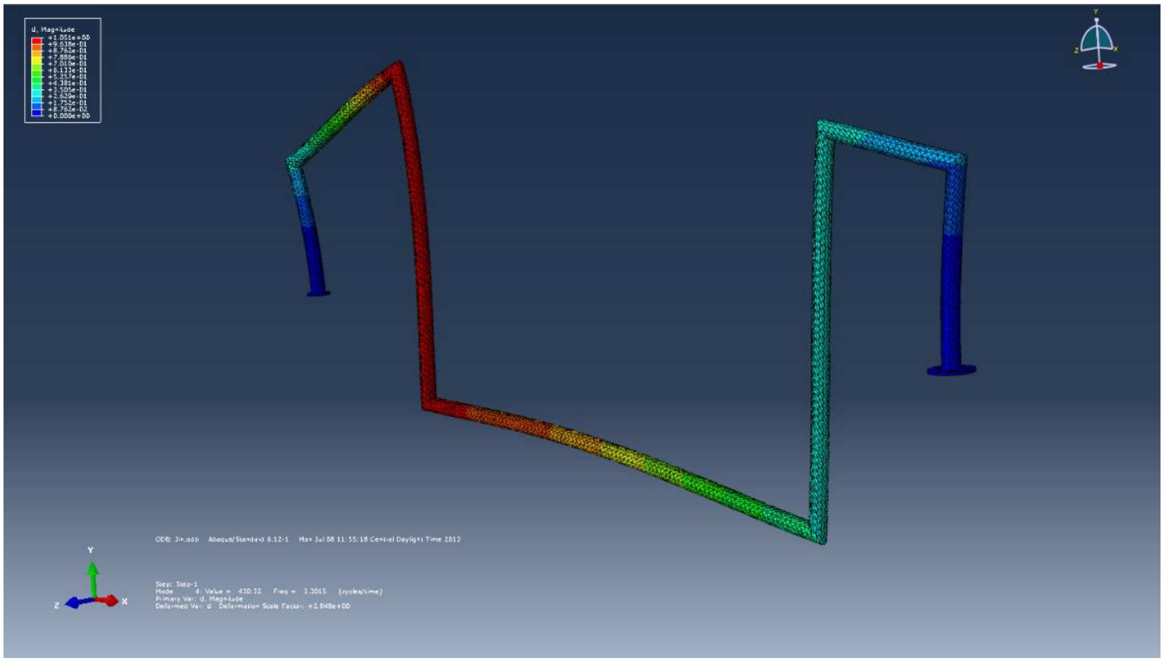

Figure 4-19 Vibration response in Y direction from ABAQUS software

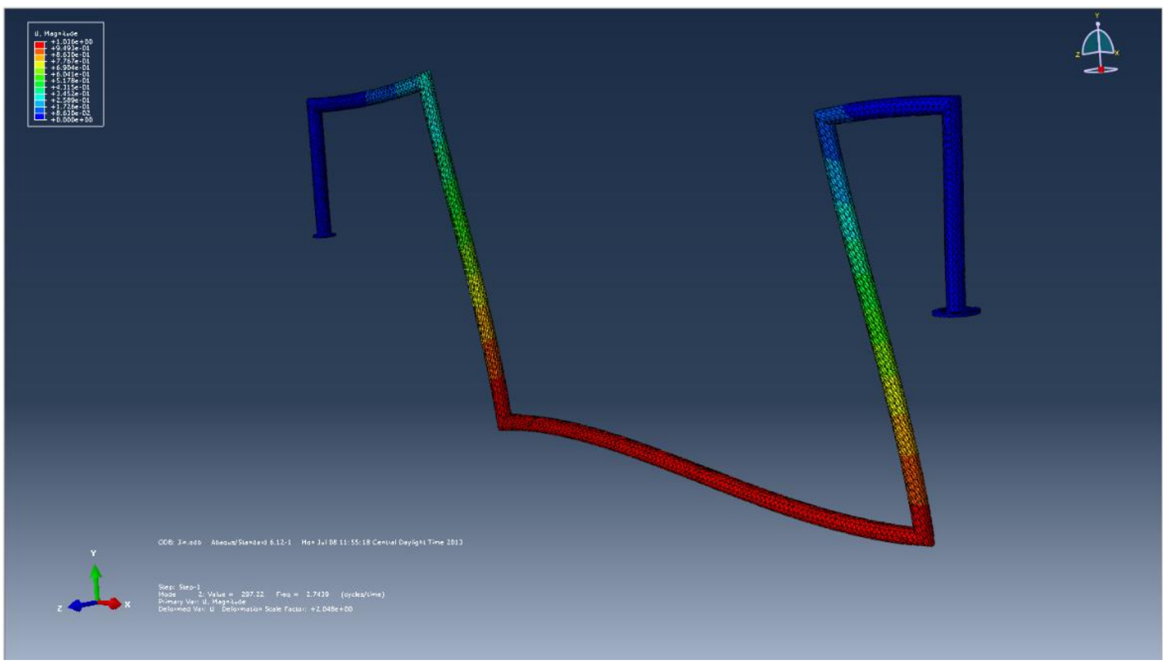

Figure 4-20 Vibration response in Z direction from ABAQUS software

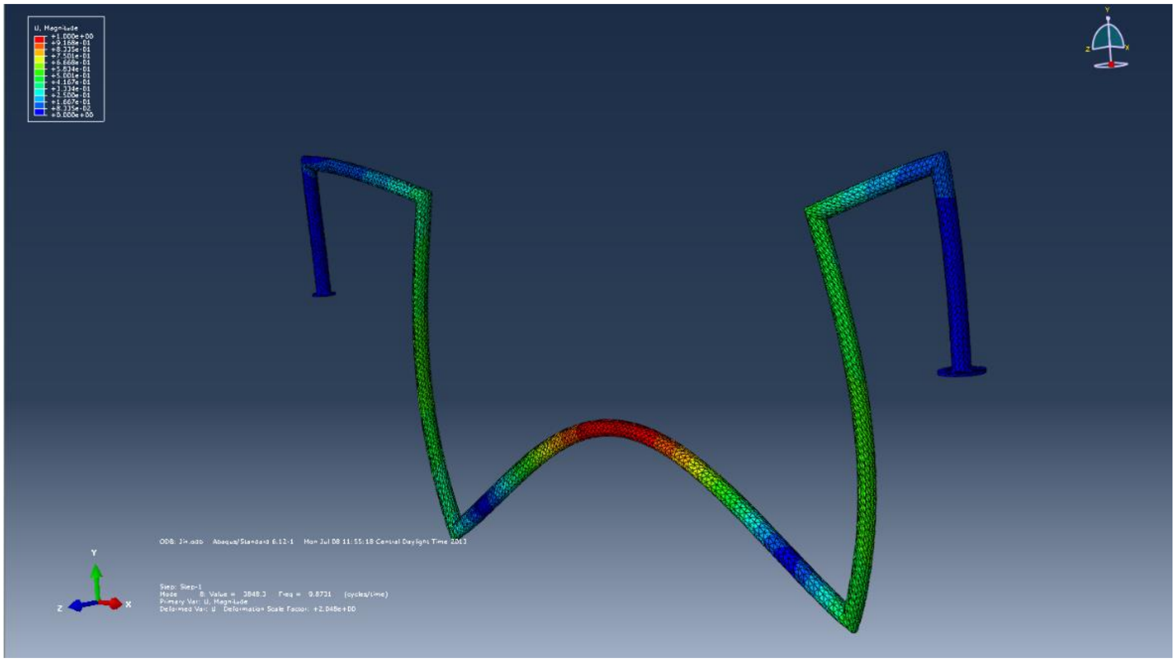

Figure 4-21 Vibration response with the higher mode shape from ABAQUS software

4.5 Summary

In this chapter, we introduced the concept of an M-shaped jumper and vortex induced vibration (VIV). The natural frequency of a jumper is an important parameter, and it is also one of the significant parameters for the design of the pounding tuned mass damper (PTMD) for the vibration control of the jumper. Therefore, we conducted an experiment to measure the natural frequency of the full-scale jumper in the field. The experimental results of the natural frequency of the jumper in field have been verified by the simulation results from the FEM. Determining the natural frequency of the jumper is the fundamental step in designing a particular vibration system that mimics the vibration motions of the jumper.

5. Direct Method for Determining Natural Frequencies of Structures in Water

The goal of the work described in this chapter is to accurately estimate the natural frequencies of the pipe and the PTMD system submerged in water based on the analysis of natural frequencies in air. The natural frequencies of the pipe and PTMD are the key parameters that should be tuned to help realize vibration control of the pipe in water. Therefore, a direct method for determining natural frequencies of vibration for the pipe and PTMD submerged in water is essential. Through a series of the experiments, we have devised a method for estimating natural frequencies both for the pipe and PTMD in water. A comparison of the experimental results and the estimated results shows that the estimated natural frequencies are close to the natural frequencies measured from the experiments. Hence, our method is particularly useful for determining natural frequencies of structures in water.

5.1 Introduction

Through the analysis of the natural frequencies of a submerged pipe, we found that the pipe natural frequency relies mainly on the stiffness of the spring and especially the effective mass of the system in water. The effective mass is the sum of total unit mass of the pipe, the unit mass of the pipe contents, and the unit mass of the displaced water (added mass). Note that we estimated the effect of the added mass of the surrounding water and found that the added mass significantly lowered the natural frequency of the short-pipe in water [35]. As demonstrated from the agreeing experimental results and numerical calculations, the effect of the added mass can help predict the natural frequency of the short pipe with high accuracy. Thus, the method provides a simple and straightforward way for calculating the natural frequency of the short-pipe submerged in water. The natural frequency of the short-pipe is given by Equation 5.1.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{4K}{M_e}}, \tag{5.1}$$

where f_n is the natural frequency of Pipe; M_e is the effective mass; K is the stiffness of the each spring. Therefore, the natural frequency of the underwater pipe may be predicted by the effect of the added mass on the natural frequency of the pipe in air. Thus, Equation 6 can be simplified as the following formula,

$$f_{\mathrm{p}\_underwater} = \sqrt{\frac{M_{\mathrm{p}\_e1}}{M_{\mathrm{p}\_e2}}} \bullet f_{\mathrm{p}\_air}, \tag{5.2}$$

where $f_{j\_underwater}$ is the natural frequency of the pipe underwater; f_{j_air} is the natural frequency of the pipe in air; M_{j_e1} is the effective mass of the pipe in air; and M_{j_e2} is the effective mass of the pipe in water.

The effective mass is the sum of total unit mass of the pipe, the unit mass of the pipe contents, and the unit mass of the displaced water (added mass) is obtained by

$$M_e = M_p + M_c + M_a, \tag{5.3}$$

where M_p is the unit mass of pipe including coatings; M_c is the unit mass of contents; M_a is the added unit mass in water; and M_a is the added mass coefficient. The added mass of the pipe in water can be calculated by the following equation [36]:

$$M_a = \frac{\pi}{4} \cdot D^2 \cdot \rho \cdot C_a. \tag{5.4}$$

The PTMD should be tuned to the natural frequency of the pipe in order to receive the highest amount of vibration energy from the primary system. Initially, we determine the natural frequency of PTMD in air by designing the dimensions of the L-shaped frame (e.g., length and height) via simulation in SAP 2000 software. The vibration forces generated through the pounding process depends on the mass of the PTMD system. From the simulation, we recommend a PTMD mass ratio of 2% in air. At that value, the PTMD system can effectively work for controlling suppression for the pipe.

During the experiments, we utilized both the finite element method (FEM) and experimental results to determine the exact natural frequency of the PTMD in air. Also, we also can approximate the natural frequency of the PTMD in air by Equation 5.5. The natural frequency of the PTMD underwater would be predicted by the effect of the added mass on the natural frequency of the PTMD in air (Equation 8). Thus, Equation 5.5 can be modified to yield the following formulas [37],

$$f_{p\_air} = \sqrt{\frac{6EI}{L_1^2(2L_1 + 3L_2)M_{e\_p1}}} , \tag{5.5}$$

$$f_{ptmd\_underwater} = \sqrt{\frac{M_{e\_ptmd1}}{M_{e\_ptmd2}}} \bullet f_{ptmd\_air} , \tag{5.6}$$

where $f_{\mathrm{ptmd}\_underwater}$, f_{ptmd_air} are the natural frequency of the PTMD in water and in in air, respectively. M_{ptmd_e1}, M_{ptmd_e2} are the effective mass of PTMD in air and in water, respectively. L_1, L_2 are the vertical height of the PTMD and the horizontal length of the PTMD, respectively.

5.2 Experimental Setup

To alter the natural frequencies of the structure, we changed its mass. For the pipe, we mounted a series of mass blocks on the surface of the pipe, and measured its free vibration both in air and in water, respectively. For the PTMD, we tested its free vibration from one mass to five masses on the same location of the L-shaped beam both in air and in water. Note that we considered the effect of the added mass of the surrounding water and found that the added mass significantly lowered the natural frequency of the short-pipe in water. Therefore, we utilize the method discussed in the last section to estimate the natural frequencies of the pipe and the PTMD in water and then compared the experimental results with the estimated results.

Figure 5-1 Masses installed on the L-beam

Table 5-1 Detailed dimension of a mass block

No.	Weight (lbs)	Outer Diameter (inch)	Thickness (inch)
1	2.5	10	0.5

5.3 Experimental Results

5.3.1 Analysis of Natural Frequency of the Pipe in Water

Figure 5-2 shows the natural frequencies measured in air. Meanwhile, we also calculated the corresponding values using the Equation 5-1 as shown in the Figure 5-2. It was obvious that the value of the natural frequency of the pipe in air decreased significantly as more masses were added. Additionally, the results from the experiments and calculation matched closely, as shown in the Figure 5-3. The natural frequency at 10 lbs showed a small deviation but was within acceptable error. Comparing with the experimental results and estimated results of the natural frequencies of the pipe in air, the accuracy was up to 99.6%. Hence, this method was verified to be feasible predicting the natural frequencies if the pipe in air.

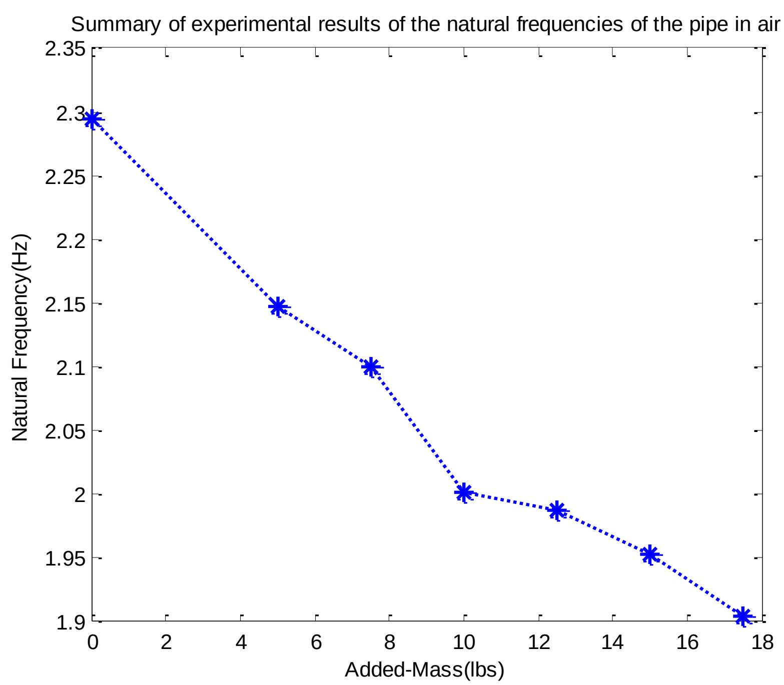

Figure 5-2 Experimental results of the natural frequencies of the pipe with being mounted a series of mass in air

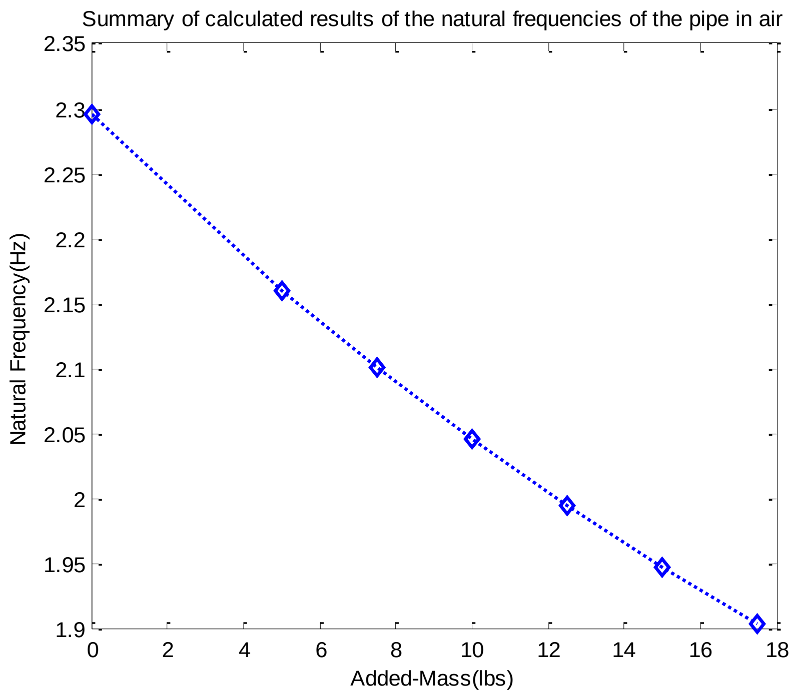

Figure 5-3 Estimated results of the natural frequencies of the pipe with being mounted a series of mass in air

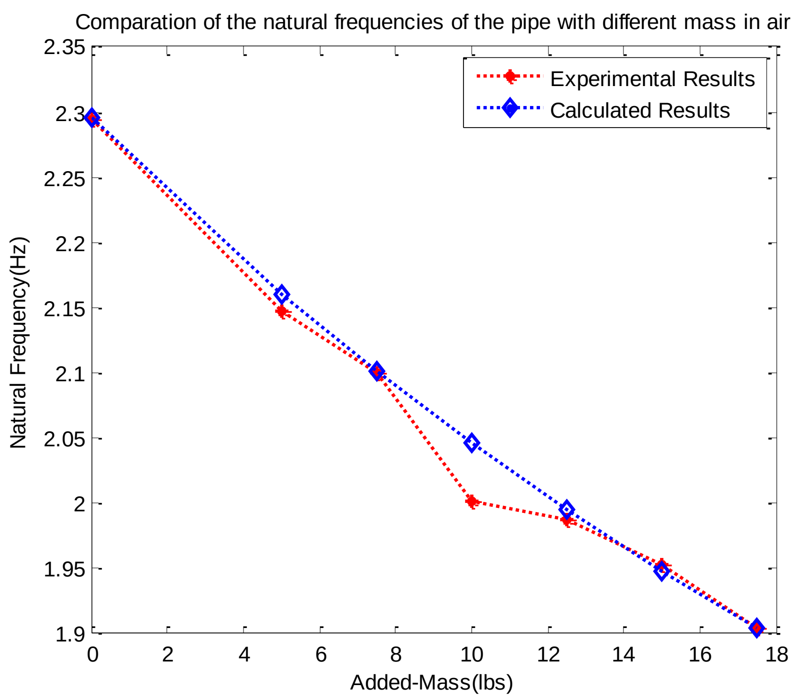

Figure 5-4 Comparison of the experimental results and estimated on the natural frequencies of the pipe

Figure 5-5 shows the natural frequencies of the pipe mounted with a series of masses in water. We used the same method as in air to estimate the natural frequency in water while also considering the effective mass in water, especially the added mass, as shown in the Figure 5-6. A comparison of the experimental and calculated values in Figure 5-7, clearly demonstrates that the results follow each other closely albeit with a small biasing difference. This phenomenon may be explained by the tolerance in the estimation of the effective mass of the pipe. Even with the tolerance, the accuracy of this method for predicting the natural frequencies of the pipe was up to 99.1%. In addition, we considered the effect of the added mass of the surrounding water of the pipe and found that the added mass significantly lowered the natural frequency of the short-pipe in water, as shown in the Figure 5-8.

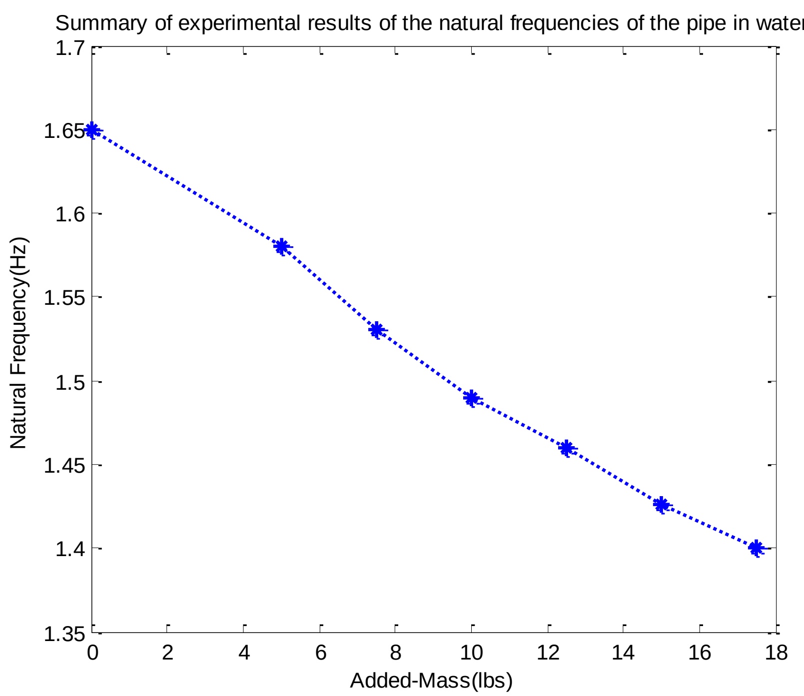

Figure 5-5 Experimental results of the natural frequencies of the pipe with being mounted a series of mass blocks in water

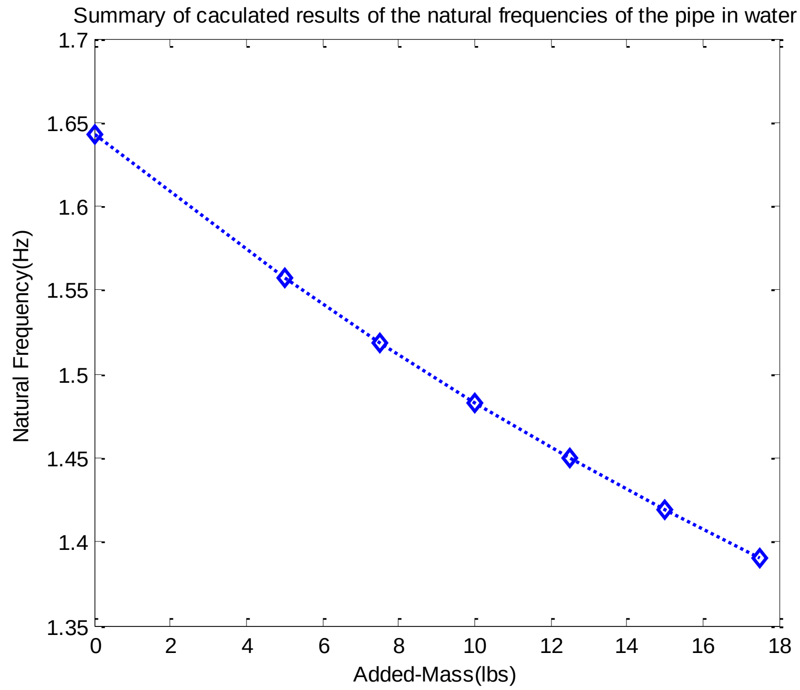

Figure 5-6 Estimated results of the natural frequencies of the pipe with being mounted a series of mass blocks in water

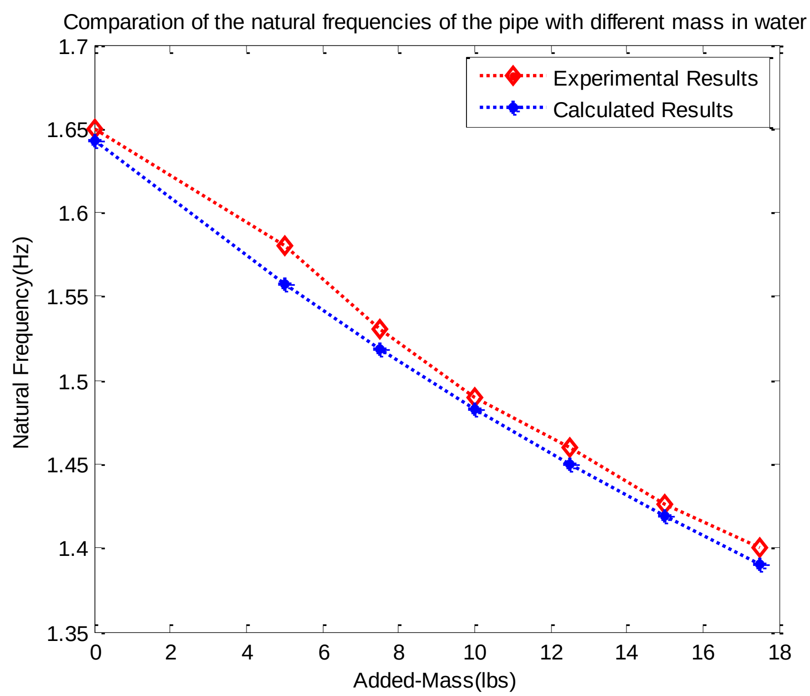

Figure 5-7 Comparison of the experimental results and estimated on the natural frequencies of the pipe in water

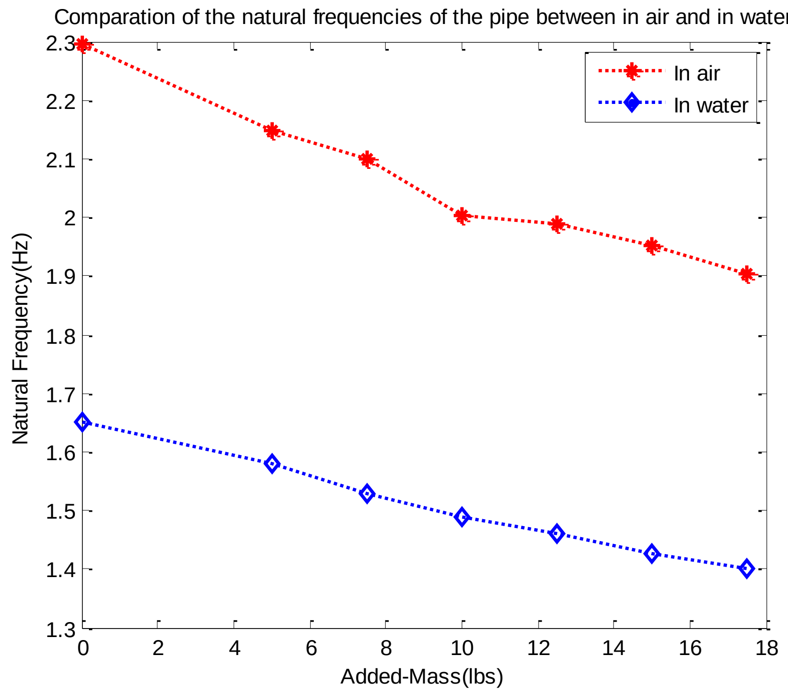

Figure 5-8 Comparison of the experimental results of the natural frequencies of the pipe between in air and in water

The values of the weight of the pure pipe, mass added to the pipe, added mass from water, and effective mass of the pipe in water, are summarized in Table 5-2. The corresponding natural frequencies of the pipe from the experiments and calculation are shown in the Table 5-3.

Table 5-2 Summary of the weight of pipe, mass added to pipe, added mass from the water, and the effective mass of the pipe in water

No.	Weight of Pipe (lbs)	Add-Mass (lbs) In Air	Added Mass (lbs) by Water	Effective Mass (lbs) In Water
1		0	38.955	75.346
2		5	40.563	83.198
3		7.5	41.367	86.051
4	38.537	10	42.171	88.904
5		12.5	42.975	91.757
6		15	43.779	94.609
7		17.5	44.583	97.462

Table 5-3 Natural frequencies of the pipe in air and in water via the experiments and calculations

No.	Experimental Natural Frequency (Hz) In Air	Estimated Natural Frequency (Hz) In Air	Experimental Natural Frequency (Hz) In water	Estimated Natural Frequency (Hz) In Water
1	2.283	2.296	1.650	1.643
2	2.155	2.16	1.560	1.557
3	2.089	2.101	1.530	1.519
4	2.020	2.046	1.500	1.483
5	1.987	1.995	1.470	1.450
6	1.934	1.948	1.440	1.419
7	1.900	1.904	1.400	1.390

5.3.2 Analysis of Natural Frequency of PTMD in water

Figure 5-9 shows the natural frequencies of the PTMD with a series of the masses, from 1 to 5 pieces (2.5 lbs each), in air. The downward slope in Figure 5-10 shows that the values of the natural frequency decreases as more masses were added to the L-shaped beam. The trend shown in the Figure can be approximated by Equation 5.5. The corresponding natural frequencies of the underwater PTMD is presented in Figure 5-10. As discussed in the previous sections, the added mass in water would lower the natural frequency of the PTMD in water. A comparison of the natural frequencies of the PTMD in air and in water shown in Figure 5-11 indicates that the natural frequency of the PTMD was slightly reduced by the added mass of the surrounding water. Because the volume of a mass itself is small, the added masses of the PTMD are too small to significantly affect the natural frequency of PTMD in water. Figure 5-12 compares the natural frequencies of the underwater PTMD between the experiments and calculations, and shows that the estimated values were close to the experimental results.

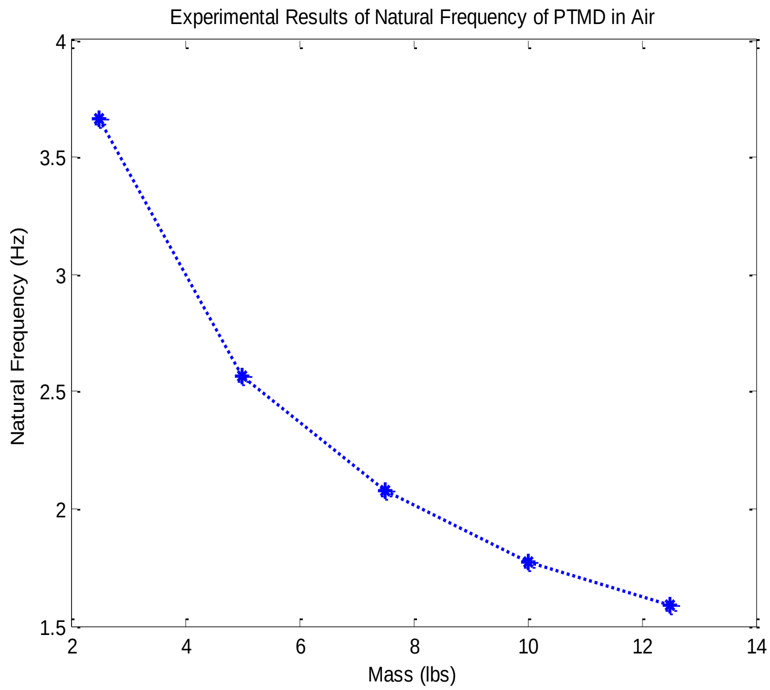

Figure 5-9 Experimental results of the natural frequencies of PTMD with a series of mass blocks in air

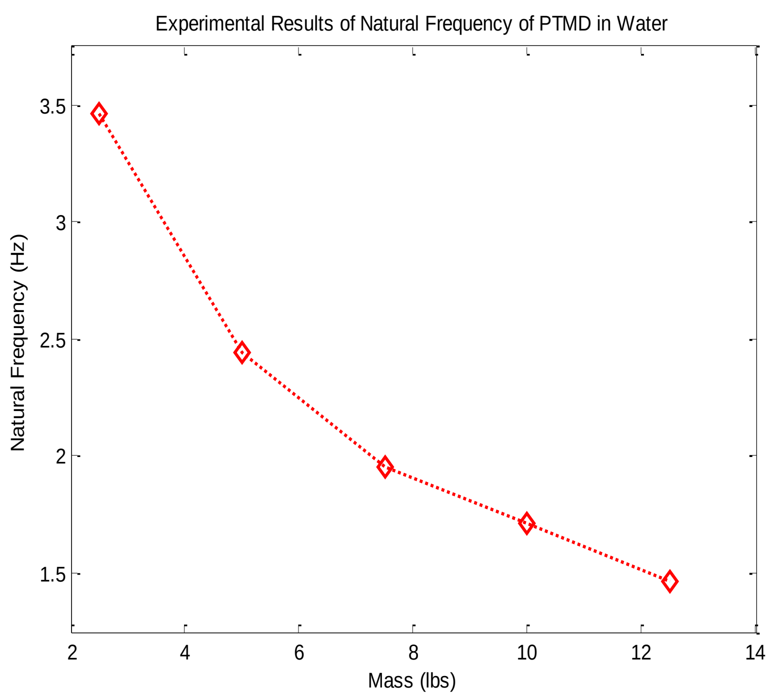

Figure 5-10 Experimental results of the natural frequencies of PTMD with a series of mass blocks in water

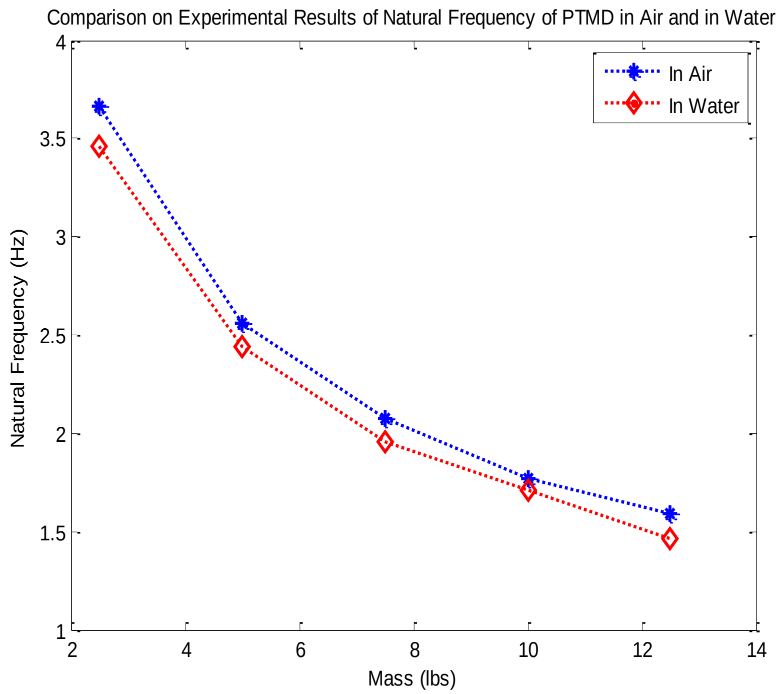

Figure 5-11 Comparison of the experimental results of the natural frequencies of the PTMD between in air and in water

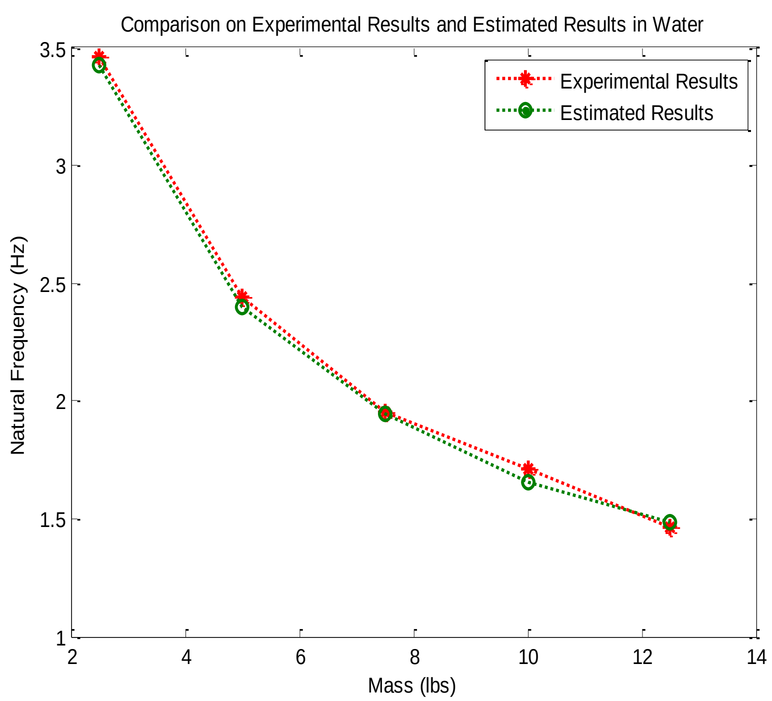

Figure 5-12 Comparison of the experimental results and estimated on the natural frequencies of the PTMD in water

The detailed values regarding the weight of a series of masses, the added mass from water, and the effective mass of the PTMD in water are summarized in Table 5-4. The corresponding natural frequencies of the PTMD from the experiments and calculations are shown in the Table 5-5.

Table 5-4 Summary of a series of mass added to PTMD, added mass from water, and the effective mass of the PTMD in water

No.	Mass (lbs) In Air	Added Mass (lbs)	Effective Mass (lbs) In Water
1	2.5	0.353	2.853
2	5	0.705	5.705
3	7.5	1.058	8.558
4	10	1.411	11.411
5	12.5	1.764	14.264

Table 5-5 Natural frequencies of the PTMD in air and in water via the experiments and calculations

No.	Experimental Natural Frequency (Hz) In Air	Experimental Natural Frequency (Hz) In Water	Estimated Natural Frequency (Hz) In Water
1	3.662	3.462	3.428
2	2.563	2.441	2.399
3	2.075	1.953	1.942
4	1.77	1.709	1.657
5	1.587	1.465	1.486

5.4 Summary

In this chapter, a direct method for determining natural frequencies of vibration for the pipe and PTMD submerged in water was presented. Conducting a series of experiments, and comparing the experimental results and the estimated results showed that the estimated natural frequencies were close to the natural frequencies measured from the experiments. Hence, this method is particularly useful for determining natural frequencies of structures in water.

6. Vibration Control of a Submerged Pipe by PTMD

In this chapter, a PTMD was developed and applied for the vibration suppression of a pipe submerged in water. A new vibrating system consisting of a short-pipe in the water tank and a new PTMD device were designed, based on the experimental results from the measurement of the natural frequency of the full-scale jumper in the field. The vibration response of the short pipe mimics the vibration motions of the subsea jumper so that the natural frequency of the short-pipe and the jumper are the same in the cross-flow direction (Y direction). During the experiment, vibration motions of the pipe submerged in the water were monitored. The experiments verified the effectiveness of the PTMD by the comparing the vibration amplitude of the pipe with and without PTMD control both in the air and in water.

6.1 Pounding Tuned Massed Damper (PTMD)

A PTMD is an innovative damping device that consists of a mass, L-shaped beam, and a ring with viscoelastic damping material as shown in Figure 6-1. The shape and stiffness of the L-beam is designed by the specific frequency of the device, especially taking into consideration the effect of the mass mounted on the L-beam. A metal ring with viscoelastic (VE) materials is a delimiter, which limits the motion of the TMD and dissipates the vibration energy for the structure through the collision or pounding process. A PTMD device is typically able to control the vibration motions both in horizontal and vertical directions. Hence, this is a significant advantage over a traditional TMD device.

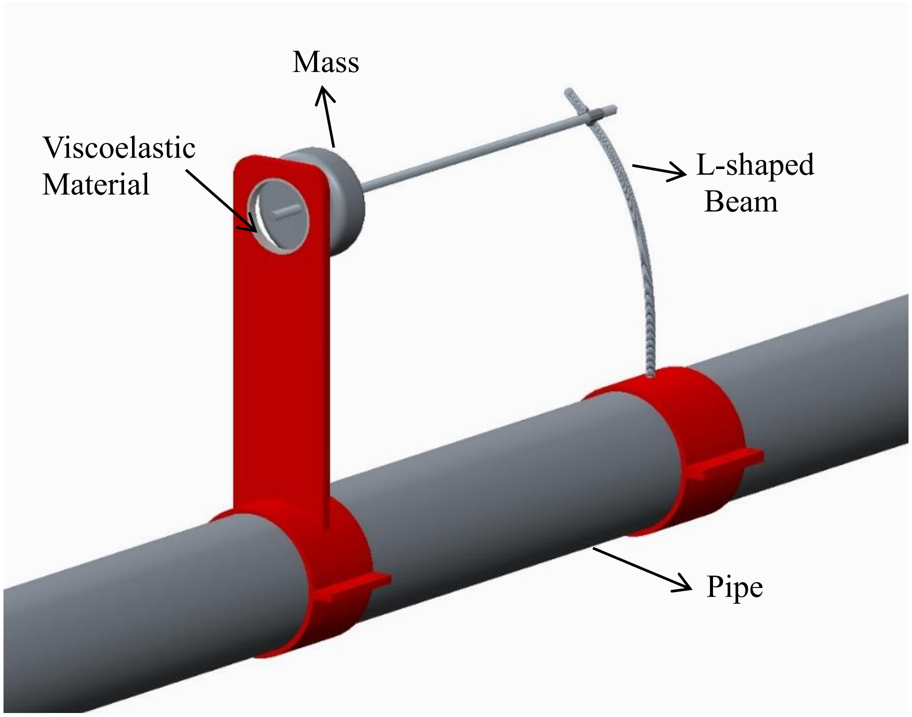

Figure 6-1 Pounding tuned mass damper (PTMD)

The vibration energy of the primary structure can be dissipated by the PTMD system through impacting or pounding of the mass with a ring equipped with a layer of viscoelastic material on the inner surface. To ensure that the PTMD receives a significant portion of the vibration energy from the primary system, the natural frequency of PTMD should be tuned to that of the primary structural system. At that frequency, the pounding process functions by applying a force to the primary system in the opposite direction to the excitation force, thereby suppressing vibrations of the primary structure. The vibration forces generated by the pounding process depends largely on the mass of the PTMD system, and plays a key role in the damping ability for vibration control. Furthermore, the viscoelastic materials in the ring dissipate the energy of the impacts into thermal energy that will be released to the surroundings.

Figure 6-2 A typical 3M viscoelastic tape

6.1.1 Basic Theory of PTMD

A numerical pounding force model is required to analyze the response of a jumper mounted with a PTMD. In the recent decades, several models have been investigated to study the mutual impact of adjacent buildings in severe earthquakes. The Linear spring model was firstly used to represent the force during impact [38] [39]. However it fails to account for energy loss during collision. The Kelvin model combined the linear model with a damper to introduce energy dissipation [40]. However, both those two models cannot describe pounding as a highly non-linear phenomenon. Alternatively, a non-linear spring based on the Hertz contact law, can be used to model impact [41]. The Hertz contact law is representative of elastic impacts but fails to include the energy dissipation during impact. A non-linear viscoelastic model based on the Hertz contact law in conjunction with a damper that is active only during the approach period of impact has been used to analyze structural pounding and this thesis adapts this non-linear model to simulate the pounding force [42]:

$$F = \begin{cases} \beta(x_1 - x_2 - g_p)^{3/2} + c(\dot{x}_1 - \dot{x}_2) & x_1 - x_2 - g_p > 0 \text{ and } \dot{x}_1 - \dot{x}_2 > 0 \\ \beta(x_1 - x_2 - g_p)^{3/2} & x_1 - x_2 - g_p > 0 \text{ and } \dot{x}_1 - \dot{x}_2 < 0 \\ 0 & x_1 - x_2 - g_p < 0 \end{cases} \quad (6.1)$$

where x_1 and x_2 are the displacements of the device and pounding layers, g_p is the distance between the device and pounding layers. $x_1 - x_2 - g_p$ is the relative pounding displacement and $\dot{x}_1 - \dot{x}_2$ is the pounding velocity. β is the pounding stiffness coefficient that depends on material properties and the geometry of the colliding bodies. c is the impact damping, which at any instant of time can be obtained from the Equation 4.2:

$$c = 2\xi\sqrt{\beta\sqrt{x_1 - x_2 - g_p}\,\frac{m_1 m_2}{m_1 + m_2}}, \quad (6.2)$$

$$\xi = \frac{9\sqrt{5}}{2}\frac{1-e^2}{e(e(9\pi - 16) + 16)}, \quad (6.3)$$

where m_1 and m_2 are the masses of the two colliding bodies, and ξ is the impact damping ratio correlated with the coefficient of restitution e, which is defined as the relation between the post-impact (final) relative velocity, $\dot{x}_1^f - \dot{x}_2^f$ and the prior-impact (initial) relative velocity, $\dot{x}_1^0 - \dot{x}_2^0$, of two colliding bodies

$$e = \frac{\left|\dot{x}_1^f - \dot{x}_2^f\right|}{\dot{x}_1^0 - \dot{x}_2^0}. \quad (6.4)$$

Its values can be easily determined experimentally by dropping a sphere on a massive plane plate and observing the rebound height:

$$e = \sqrt{h^f / h^0}, \quad (6.5)$$

where h^0 and h^f are the initial height and rebound height, respectively. The case when $e = 1$ represents a fully elastic collision, whereas $e = 0$ stands for a perfectly plastic impact. After assessing the value of ξ, the impact stiffness parameter, β can be determined numerically through iterative simulations, which tend to fit the experimentally obtained pounding force time histories.

6.1.2 Viscoelastic Material

A viscoelastic material is characterized as possessing both viscous and elastic behavior [43]. For a purely elastic material, all the energy stored in the material during loading is returned when the load is removed. For a purely viscous material, it does not return any of the energy stored during loading. All the energy is lost as “pure damping” once the load is removed. In this case, viscoelastic materials do not store all the energy under deformation, but actually lose or dissipate some of this energy. This dissipation is also known as hysteresis [44]. Hysteresis explicitly requires that the loading portion of the stress strain curve must be higher than the unloading curve, as shown in the Figure 6-3. Recently, viscoelastic materials are used to reduce noise transmission, vibration transfer and vibration related stress [45].

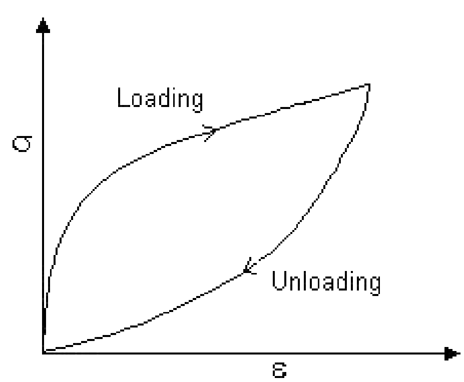

Figure 6-3 Typical hysteresis loop for a viscoelastic material

6.2 Experimental Setup

In this project, our objective was to verify the effectiveness of PTMD for the vibration suppression of a short pipe in water. Utilizing the determined natural frequency of the real jumper, a vibrating system using four springs and a short-pipe were designed as shown in Figure 6-4. To simulate the vibration response of the jumper, we used a similar outer diameter and the material properties of the pipe as shown in Figure 6-6. The detailed information of the dimensions and material property of the pipe as shown in Table 6-1. In addition, in order to design the natural frequency of the short pipe to match the natural frequency of the real jumper in the vertical direction in air, the particular stiffness of the spring was chosen as shown in Table 6-2. The four springs were connected with a short pipe in parallel, thereby the whole stiffness of the vibrating system is the sum of the stiffness of the each spring.

A new vibrating system in the water tank was built as shown in Figure 6-5. The U-shaped bolt at the top of the tank which was able to hold the pipe and to adjust the height of the pipe as well. Meanwhile, a new submerged PTMD was designed. During the experiment, we monitored the vibration response of the pipe through a displacement sensor, an accelerometer, and a data acquisition system, as shown in Figure 6-9. The natural frequencies of the primary structure and the PTMD device were acquired through the analysis of the power spectrum of its free vibration, and verified the effectiveness of the PTMD for the vibration control by comparing the vibration amplitude of the short-pipe with and without PTMD control in the time domain.

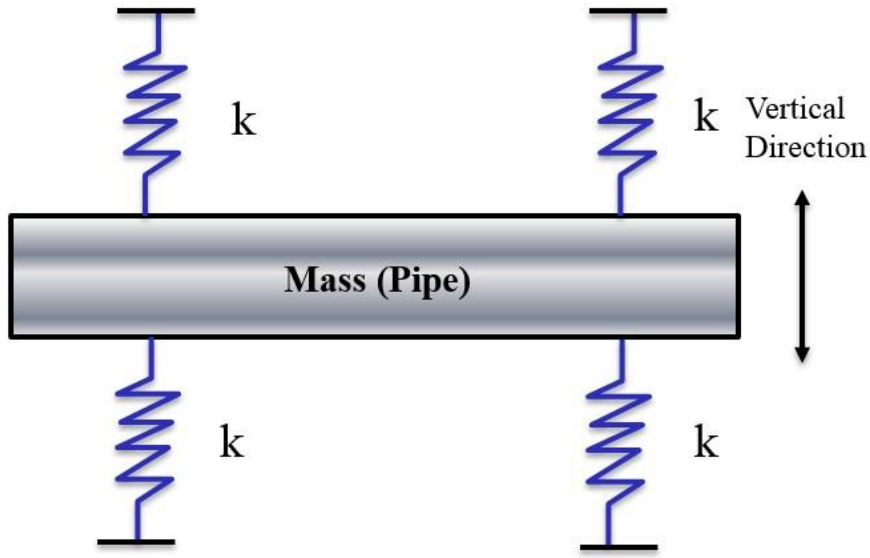

Figure 6-4 A schematic diagram of the new vibrating system by four spring and a pipe

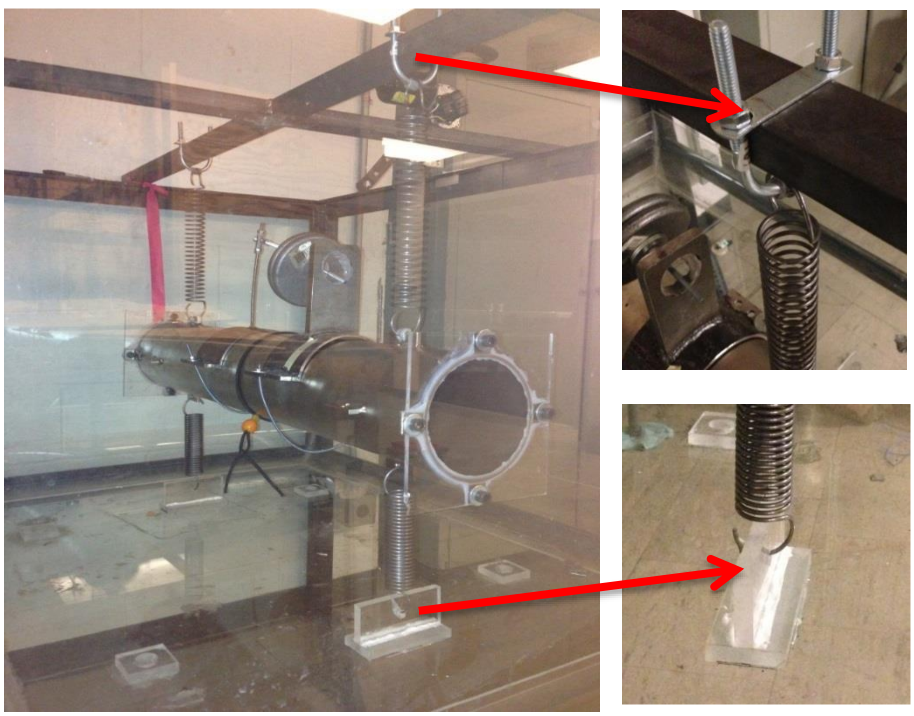

Figure 6-5 A new designed vibrating system in the water tank

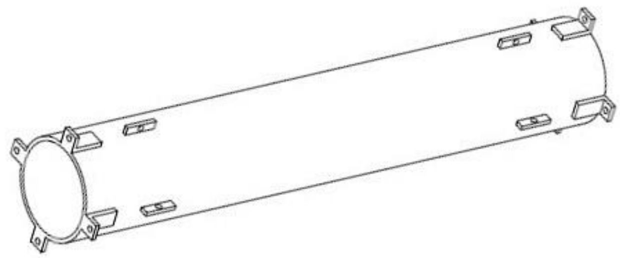

Figure 6-6 A schematic diagram of the short-pipe

Table 6-1 Basic dimensions and material property of the short-pipe

No.	Description	Value
1	Outer Diameter (in)	6
2	Wall Thickness (in)	0.25
3	Total Length (in)	36
4	Empty Weight (kg)	16
5	Material	Steel

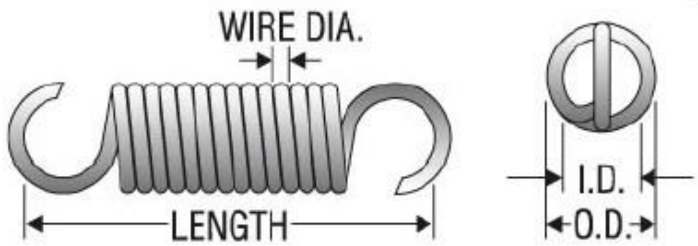

Figure 6-7 A schematic diagram of the spring

Table 6-2 Basic dimensions and material property of the spring

No.	Description	Value
1	Outer Diameter (in)	1.5
2	Total Length (in)	6
3	Rate (lbs/in)	5.2
4	Initial Tension (lbs)	4
5	Sugg. Max Defl. (in)	9
6	Sugg. Max Load. (lbs)	51
7	Wire Diameter (in)	0.125
8	Material	Music Wire

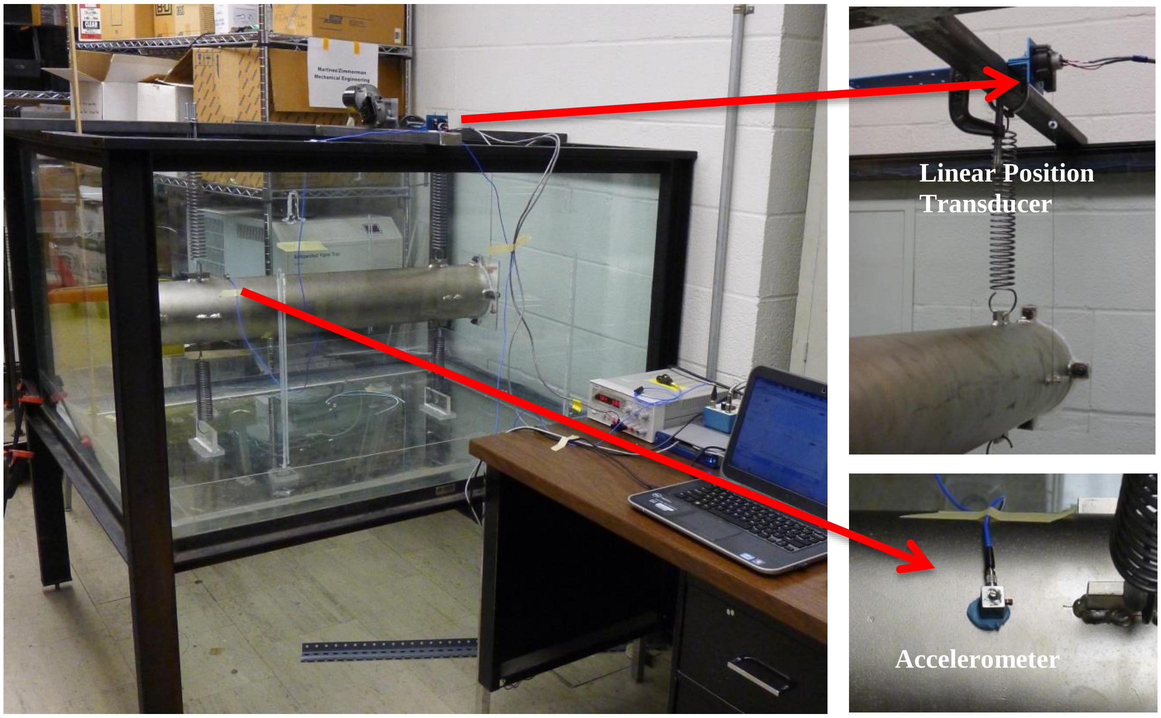

Figure 6-8 Vibrating system in the water tank in air

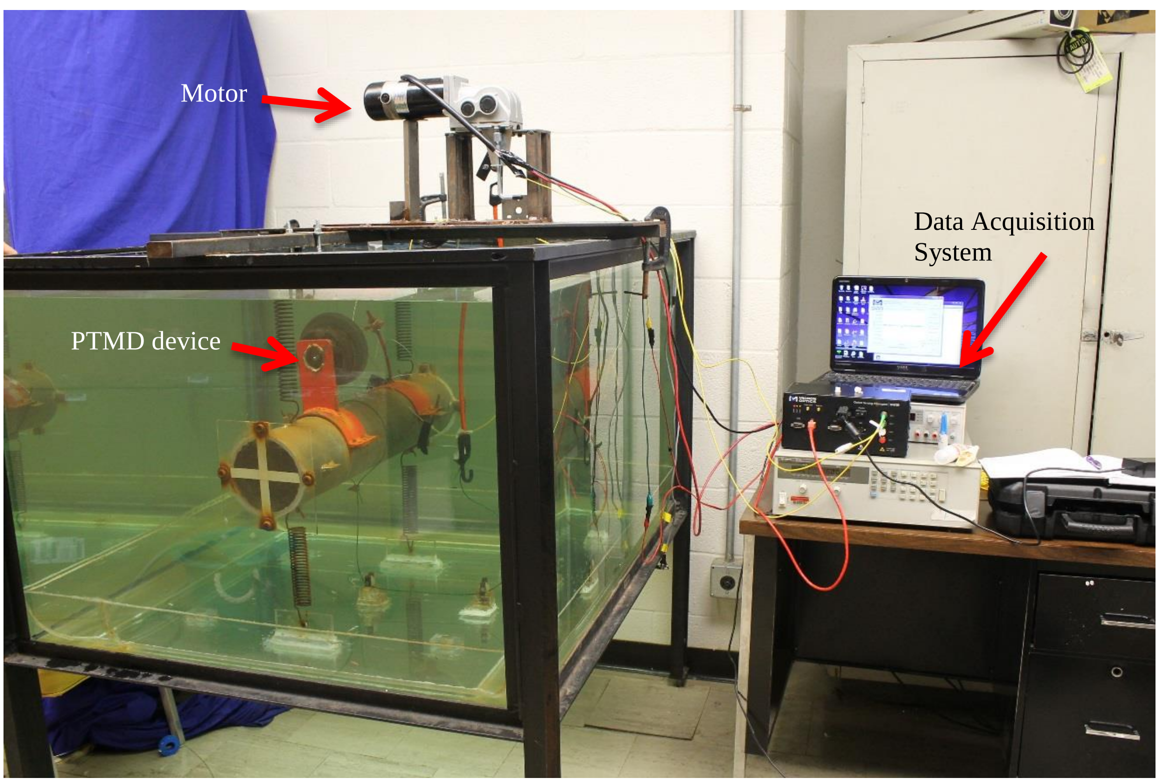

Figure 6-9 Vibration system of the short-pipe in the tank in water

6.3 Experimental Results

To verify the effectiveness of the PTMD for vibration control of the short-pipe submerged in water, the experiment was initially conducted to ensure that the PTMD system can effectively suppress the vibrations of the pipe in air. And then, the short pipe and the PTMD were submerged in water. During the experiment, the natural frequencies of the short pipe and the PTMD device were measured. Note that the added mass from water, to some extent, lowered the natural frequency of the short-pipe and the PTMD device. To ensure that the PTMD receives a significant portion of the vibration energy from the primary system, the natural frequency of PTMD should be tuned to that of the primary structural system. Thus, the effectiveness of the PTMD for vibration suppression can be successfully realized even when it is submerged in water.

6.3.1 Vibration Control of the PTMD in Air

Figure 6-10 shows the free vibration of the single pipe without being mounted to the PTMD device, and the Figure 6-11 shows the corresponding frequency spectrum. The free vibration of PTMD mounted on the pipe is shown in Figure 6-12, while the corresponding frequency spectrum is shown in Figure 6-13. The data indicates that the added mass of the PTMD device reduced the natural frequency of the primary structure from 2.393Hz to 2.1 Hz. To realize the effectiveness of the PTMD for vibration control of the pipe, the natural frequency of PTMD should be exactly tuned to that of the pipe. Thus, a specific PTMD device was designed and its natural frequency can be tuned to be 2.1Hz. The vibration control of the PTMD would be conducted during both the free vibration and the forced vibration.

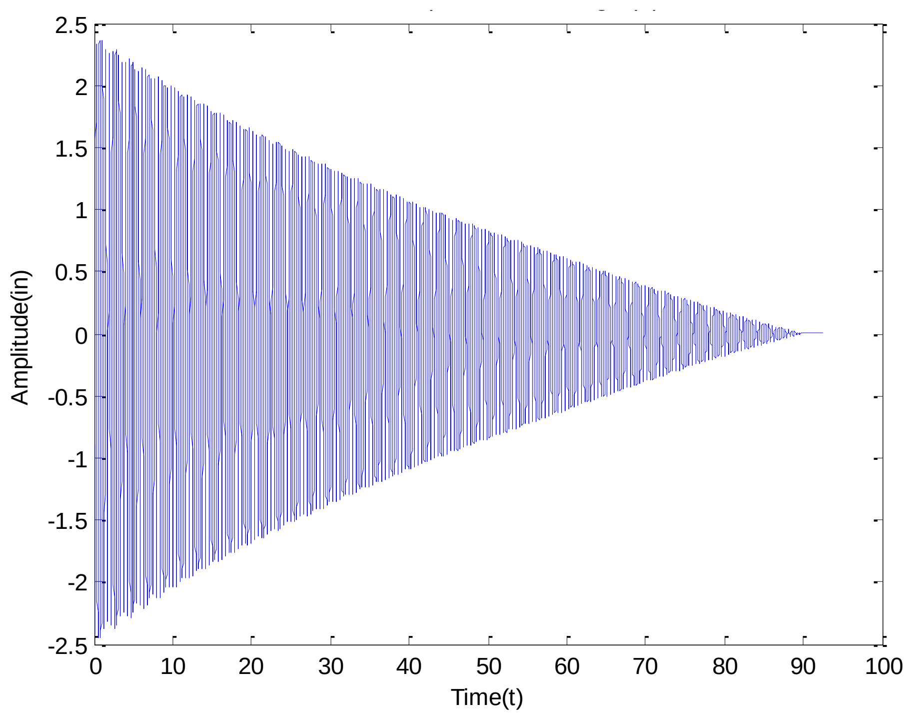

Figure 6-10 Free vibration of the single pipe without being mounted PTMD device

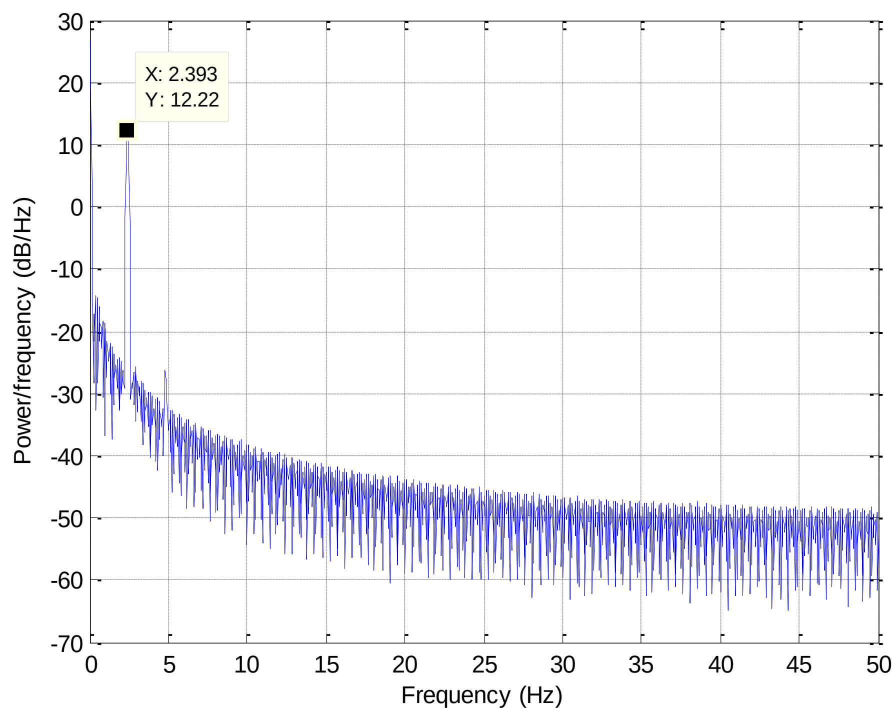

Figure 6-11 Frequency spectrum of the free vibration of the single pipe

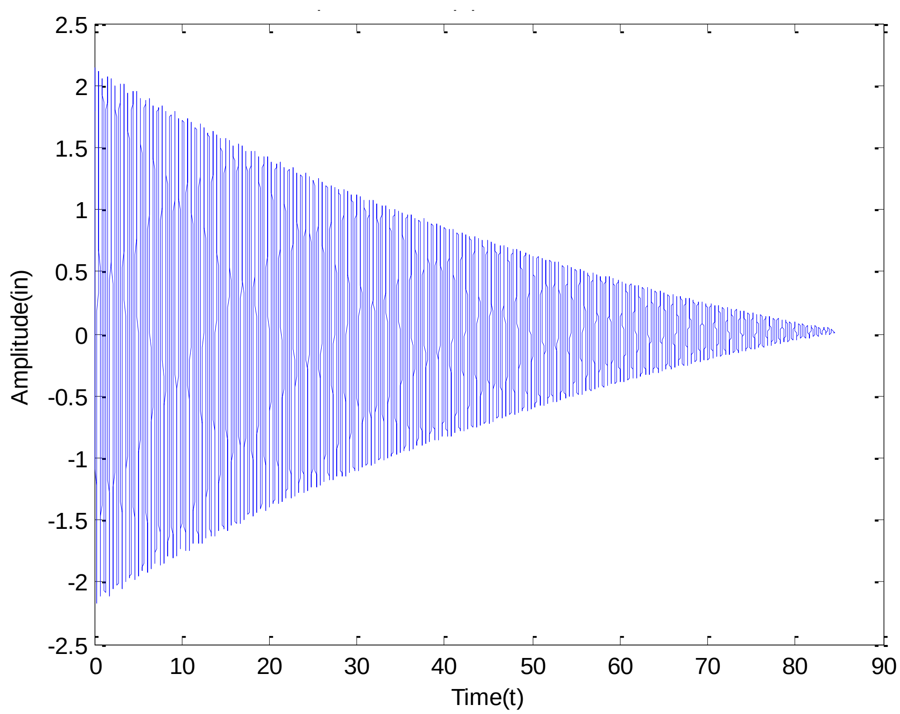

Figure 6-12 Free vibration of the pipe with being mounted PTMD device

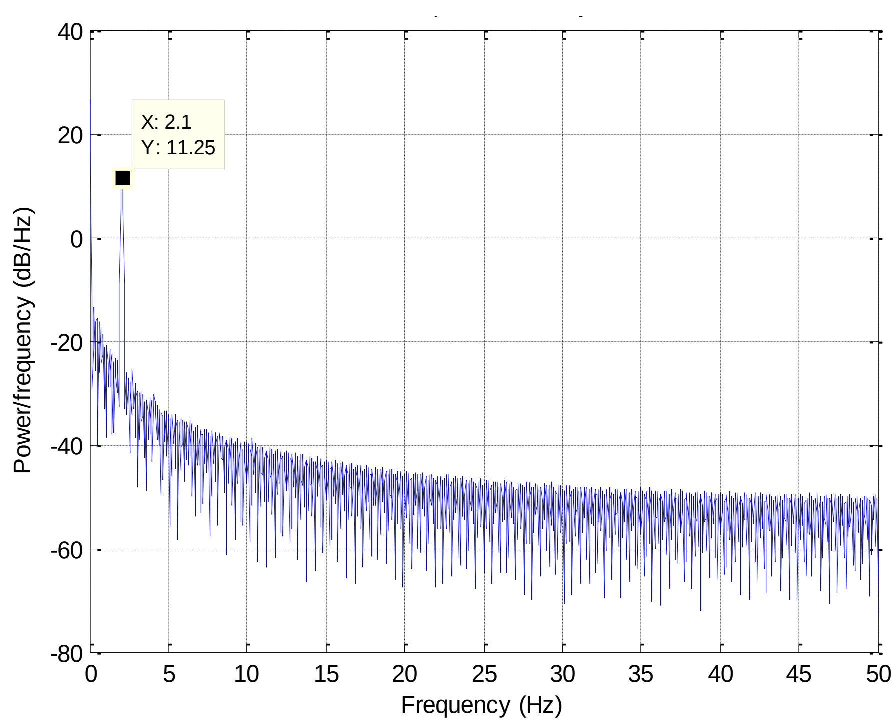

Figure 6-13 Frequency spectrum of free vibration of the pipe mounted with PTMD

The main goal of the PTMD is to realize the vibration suppression on the short pipe. The experiments for verifying the effectiveness of the PTMD for the short pipe were conducted during both free vibration and forced vibration in the air. Figure 6-14 clearly shows the vibration motion when we released the PTMD as the short pipe was excited by the motor. We compared the vibration performance of the pipe between with and without PTMD control in the Figure 6-15. The zoomed in experimental comparison is shown Figure 6-16. It clearly shows that the significant improvement of the PTMD control compared to that of the single short-pipe itself. When comparing the duration time and amplitude of the vibration of the pipe in the time domain, the vibration response was dramatically reduced by 87.5% with the control of the PTMD during free vibration in air. In addition, it only took 10 seconds for PTMD on suppressing the vibration of the pipe, whereas the pipe itself took more than 80 seconds.

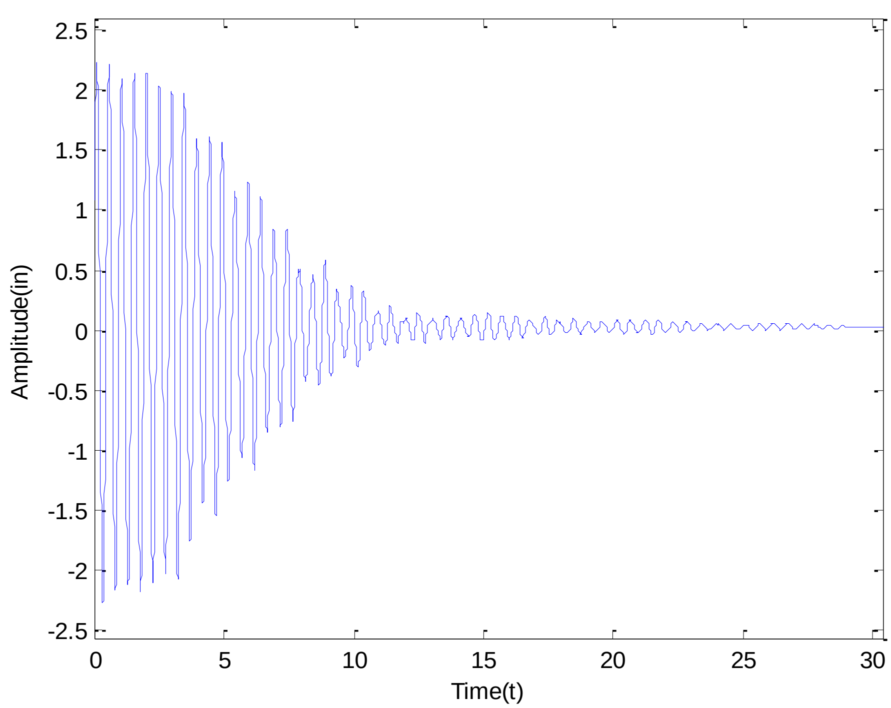

Figure 6-14 Vibration response of the pipe with PTMD control during free vibration

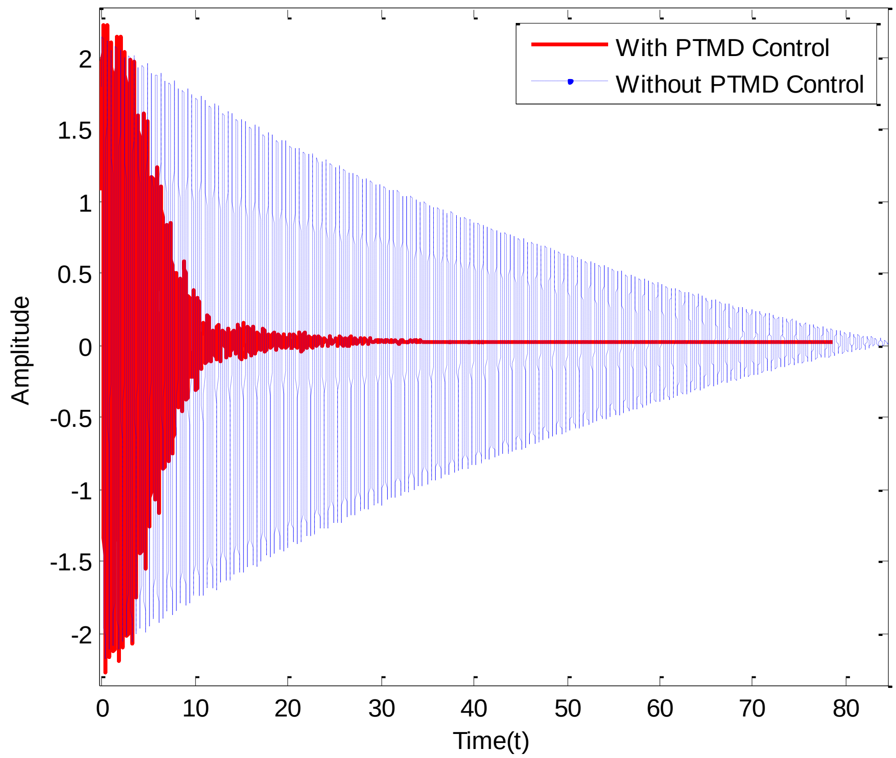

Figure 6-15 Vibration performance comparison of the pipe between with and without PTMD control during free vibration in air

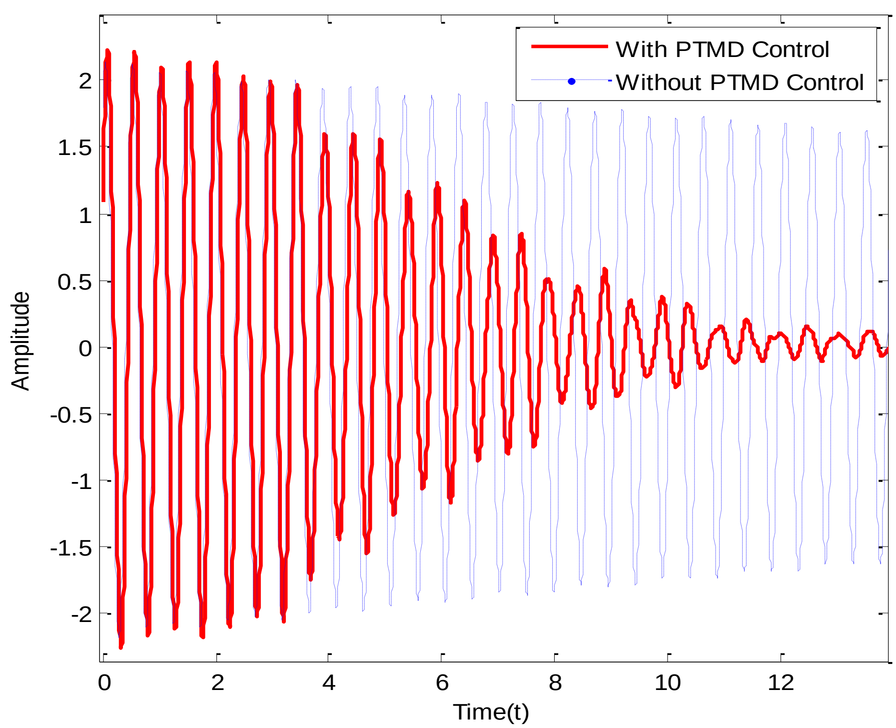

Figure 6-16 Zoom-in the vibration performance comparison during free vibration in air

Free vibration is a basic and simple model for vibrations. It occurs typically when the system is set off with an initial excitation and then allowed to execute free vibrations without any external forcing excitation, the system will tend to vibrate at a particular as well. Whereas forced vibration is characterized by a time-varying disturbance that is applied to the system. This disturbance may be a periodic, steady-state, transient, or random input. This phenomenon is common in vortex induced vibrations. Therefore, we also examined the vibration control of the PTMD for the forced in air.

As shown in Figure 6-17, a dramatic improvement of the vibration response of the short-pipe under the control of PTMD. The blue line represents the vibration motion of the pipe without the control of PTMD. The maximum vibration amplitude was 4.12 inches. In contrast, the red line shows vibration response of the pipe with the control of PTMD, and the amplitude was 0.35 inches. The maximum amplitude of the short-pipe was significantly reduced by 91.5% through the control efforts of the PTMD. Therefore, the effectiveness of the PTMD has been verified on the short-pipe during the free vibration and forced vibration in air.

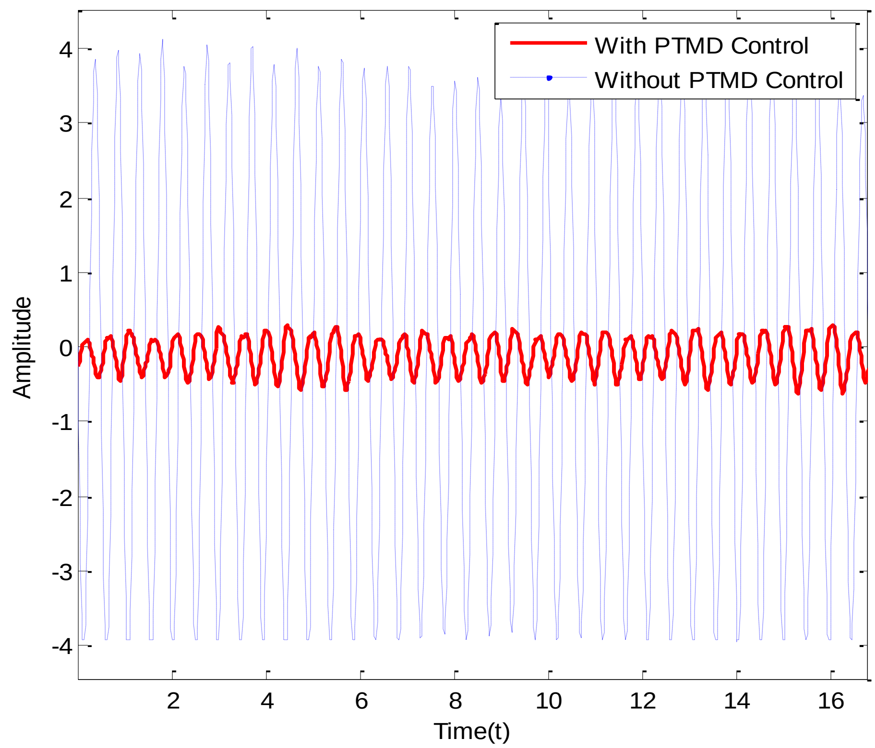

Figure 6-17 Vibration response comparison of the pipe between with and without PTMD control during forced vibration in air

6.3.2 Vibration Control of the PTMD in Water

After determining the effectiveness of the PTMD in air, the experiment of the vibration control of the PTMD submerged in water was conducted. However, it was expected that the pipe and PTMD will behave differently when submerged in water. In particular, the natural frequencies would be changed by the added mass. In this section, we designed a new PTMD device that was able to match the natural frequency of the short-pipe and realize the vibration control for the pipe in water. Figure 6-18 shows the free vibration of the pipe in water, and the corresponding frequency spectrum as shown in the Figure 6-19. We can clearly observe that the natural frequency of the pipe was 1.563Hz in water. As we observed in the last section, the natural frequency of the short-pipe was 2.1Hz as shown in the Figure 6-13. This value was dramatically reduced by the added mass that is defined by the unit mass of the displaced water around the pipe. Note that we estimated the effect of the added mass and found that the added mass significantly lowered the natural frequency of the short-pipe as well as the PTMD as mentioned in last chapter.

We utilized Fiber Bragg Grating (FBG) sensors to monitor the strain conditions of the PTMD when it was vibrating in water, as shown in the Figure 6-20. Fiber optics-based sensors have a unique resistance to water-related damage, in contrast to most commercially available electronic sensors, which may be vulnerable to circuit shorting due to contact with liquid moisture. It is easily observed that the natural frequency of the PTMD was 1.563Hz in water as shown in Figure 6-21. Hence, we tuned the natural frequency of the PTMD to the natural frequency of the short pipe in water.

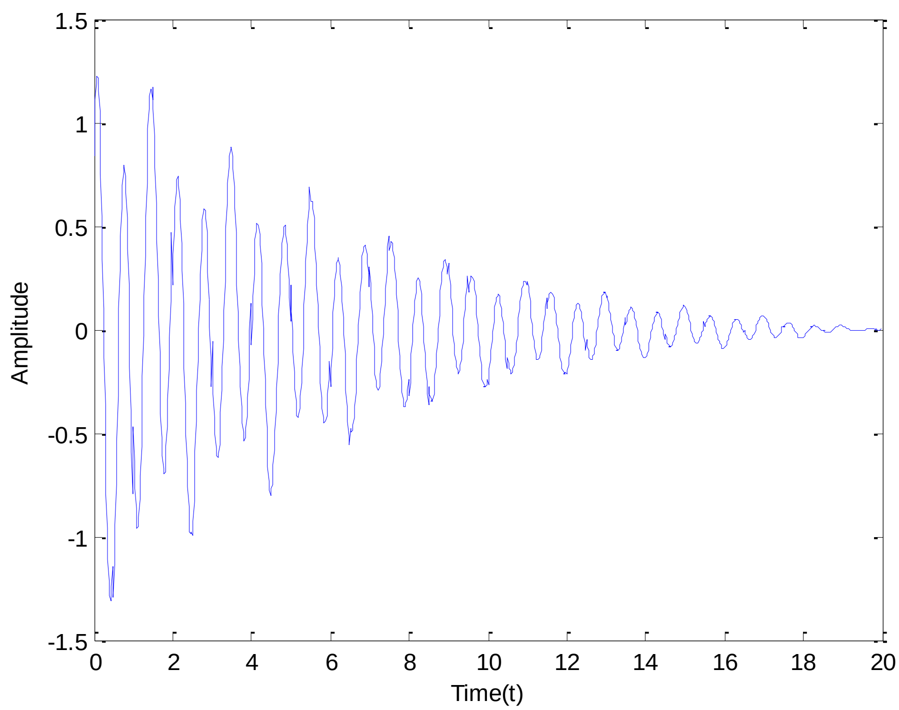

Figure 6-18 Free vibration of the pipe in water

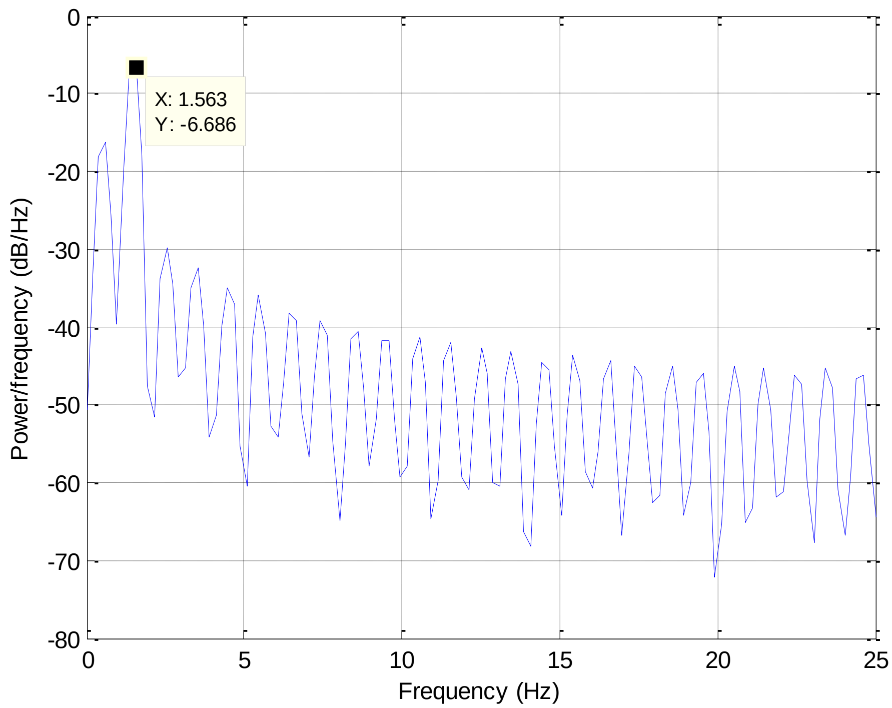

Figure 6-19 Frequency spectrum of vibration of the pipe in water

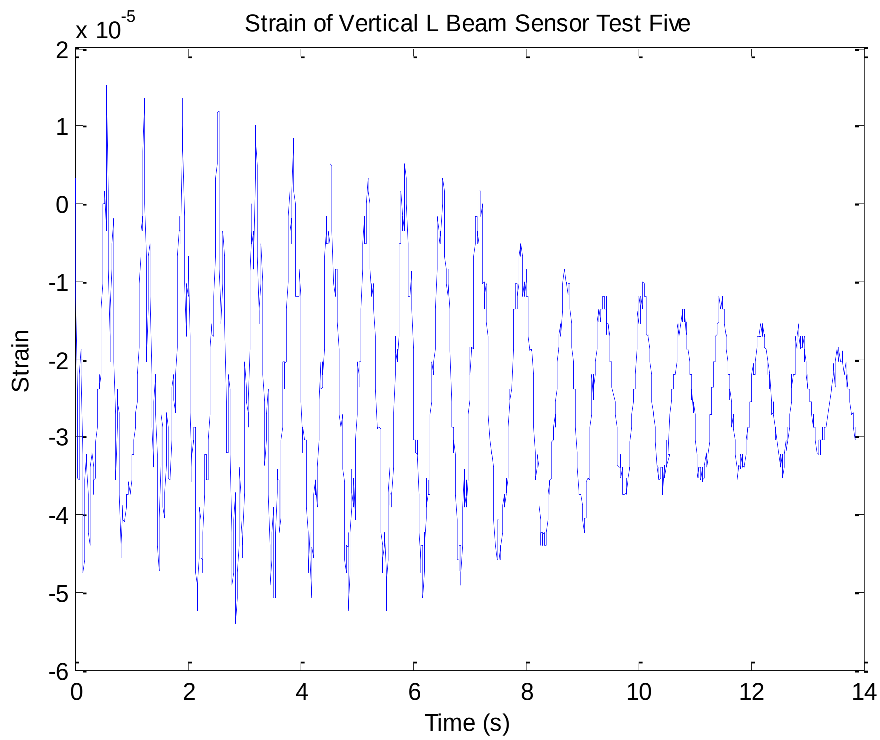

Figure 6-20 Strain of the vertical L-shaped beam of the PTMD in water

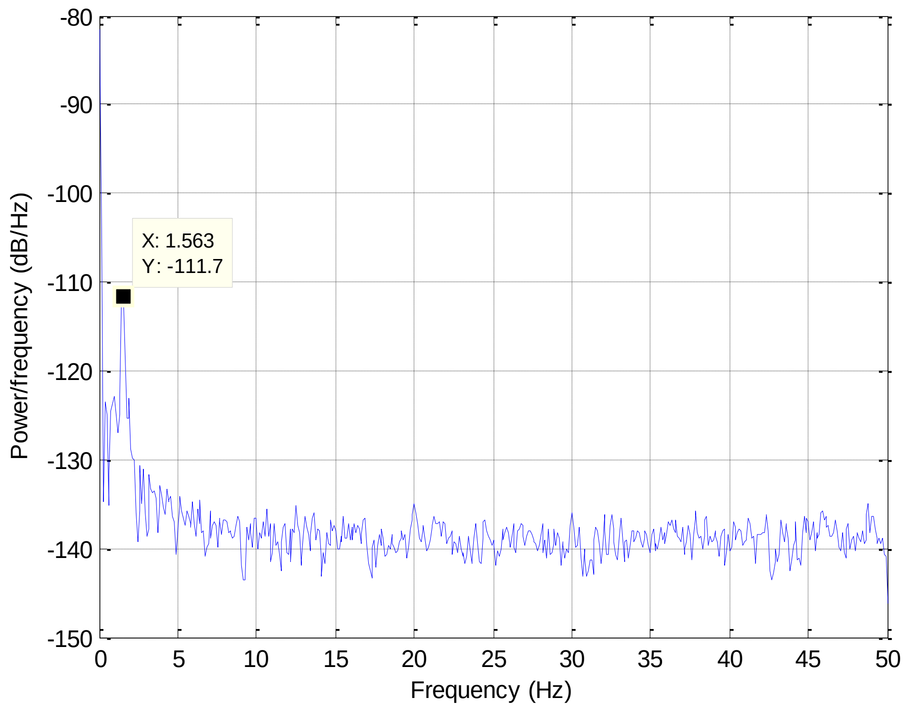

Figure 6-21 Frequency spectrum of vibration of the PTMD in water

The natural frequency plays a key role in the vibration control of the PTMD not only in air, but also in water. At that frequency, PTMD is able to receive a significant portion of the vibration energy from the primary system. In addition, the pounding process functions by applying a force to the primary system in the opposite direction to the excitation force, thereby suppressing vibrations of the primary structure. The vibration forces generated by the pounding process mainly depends on the mass of the PTMD system. Furthermore, viscoelastic materials in the ring dissipate the energy of the impacts as thermal energy that will be released to the surroundings. To some extent, the effectiveness of the PTMD is strongly dependent on the tuned frequencies between the PTMD and the primary structure.

In water, the damping of PTMD was greatly increased which served to dissipate the vibrating energy. This increased damping contributed to the effectiveness of vibration control for the short-pipe by the control of the PTMD submerged in water than that when in air. The next progression of the experiment was to investigate the vibration control capabilities of the PTMD in water when its natural frequency matches that of the pipe.

Figure 6-22 shows the vibration of the short-pipe with the control of the PTMD in water. We observed the forced vibration motion of the short-pipe without the control of the PTMD before the 6^{th} second. The maximum amplitude of the forced vibrations was 1.51 inches. After the 6^{th} second, we released the PTMD and the vibration of the short-pipe was dramatically reduced within 7 seconds. In contrast, the duration time, as we discussed in the last section, of the vibration control for the short-pipe by the control of PTMD was approximately 12 seconds in air. Therefore, the raised damping capability of the PTMD definitely enhanced the efficiency of vibration control for the pipe in water. Figure 6-23 shows the vibration comparison of the pipe between with and without the control of PTMD in water. Through comparing the maximum amplitude of the experimental results between the vibrations of the pipe between with and without the control of PTMD, we observed that the amplitude of the uncontrolled vibrations was 1.52 inches and that the amplitude of the controlled vibrations was 0.15 inches. In other words, the PTMD suppressed vibrations of the pipe by up to 90% in water.

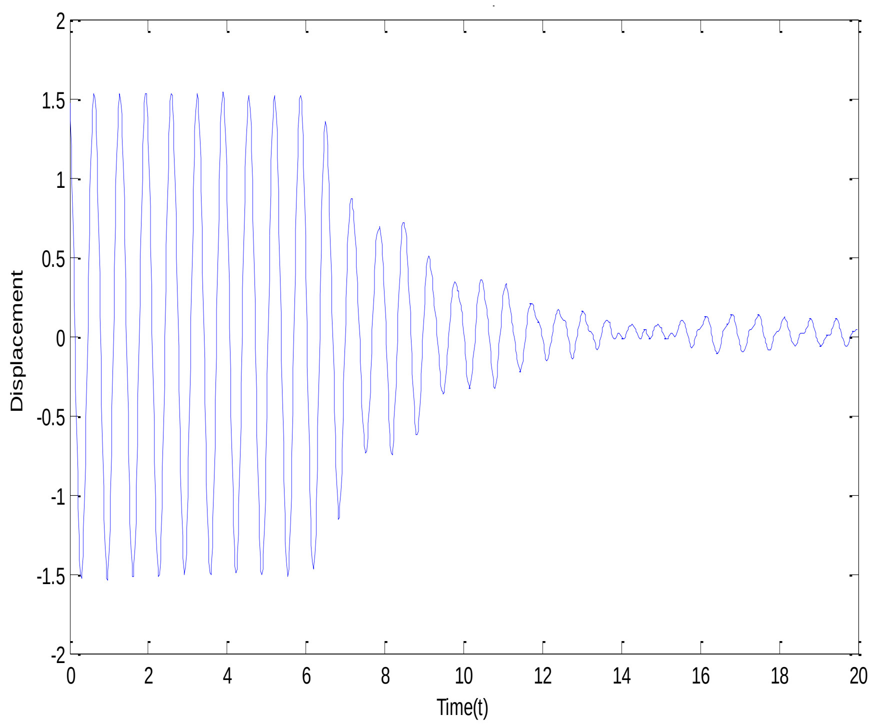

Figure 6-22 Experimental vibration of the pipe with the control of PTMD in water

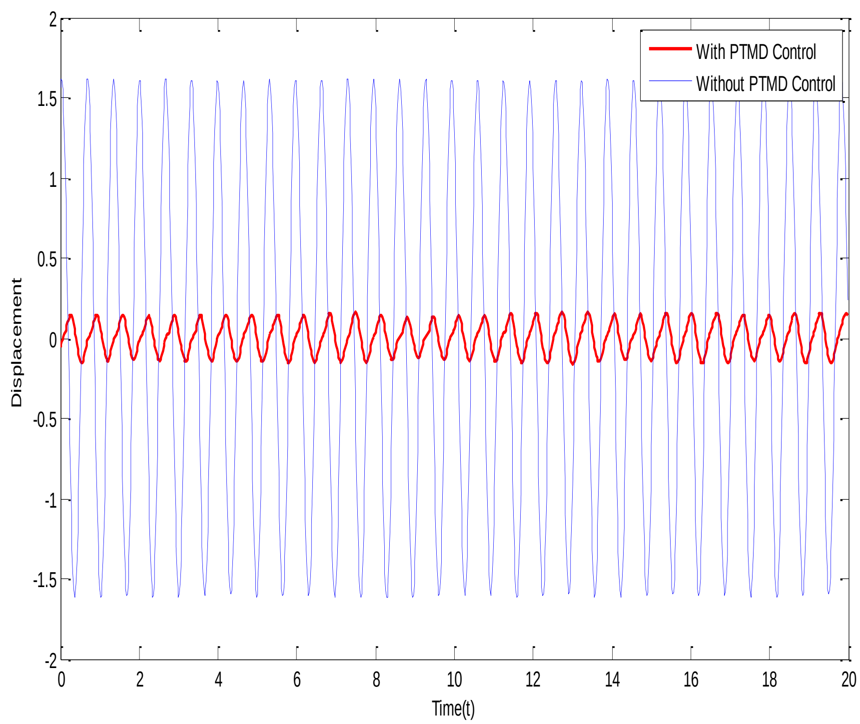

Figure 6-23 Experimental vibration comparison of the pipe between with and without the control of PTMD in water

6.4 Summary

In this chapter, we introduced a novel passive vibration control device, Pounding Tuned Mass Damper (PTMD), and its working mechanism. The main goal of the PTMD is suppress vibrations of a pipe submerged in water. A new vibrating system consisting of a short pipe and a new especially tuned PTMD device were designed. The natural frequency of the short-pipe was closely matched to the natural frequency of the full-scale jumper we measured previously in the field. During the experiment, we monitored the vibrations of the pipe through a displacement sensor, an accelerometer, and a data acquisition system in the water tank. In addition, we installed FBG sensors on the L-shaped beam to monitor the strain of the PTMD and finally to determine its natural frequency. As observed in the experimental results, the effectiveness of the PTMD for the vibration control was successfully realized on the short-pipe submerged in water. Moreover, we observed that the PTMD submerged in water worked much better than in air due to assistance from the damping properties of water.

7. Robustness Analysis of the Effectiveness of the Submerged PTMD

In this chapter, the effect of mistuning the resonant frequency on the performance of the vibration control of PTMD for the underwater pipe is studied. We will summarize the results to enhance the understanding of PTMD behavior and its robustness in water. During the experiments, we maintained the natural frequency of the pipe in water, but changed the natural frequency of the PTMD from 1.404Hz to 1.953Hz. The corresponding experimental results show a notable difference in the performance for vibration control of the PTMD in water. Furthermore, we examined a wide frequency range to verify the relative robustness of the PTMD for the vibration control of the pipe in water.

7.1 Introduction

As introduced in chapter 3, the resonant frequency between the primary structure and PTMD plays a key role in the performance of the PTMD for the vibration control. For a typical PTMD, to receive a significant portion of the vibration energy from the primary system, the natural frequency of PTMD should be tuned to that of the primary structure. The vibrating energy of the primary structure can be dissipated by the PTMD system through impacting or pounding of the mass with a ring equipped with a layer of viscoelastic material on the inner surface. The viscoelastic materials in the ring dissipate the energy of the impacts as thermal energy that will be released to the surroundings.

However, in practice, it can be difficult to tune the natural frequency of the PTMD exactly to that of subsea jumpers in deep-water. In general, the effectiveness of the PTMD is sensitive to a fluctuation in the natural frequency of the structure. If the natural frequency of the PTMD shifts away from the resonant frequency value, its performance in the vibration control for the pipe is expected to degrade. Hence, it is essential to study its performance during a particular frequency range.

7.2 Experimental Setup

In the experiments, we maintained the frequency of the harmonic excitation to the pipe at 1.587Hz. The forcing frequency was also the resonant frequency between the pipe and the PTMD in water to optimize the performance of the vibration control of the PTMD. Meanwhile we altered the PTMD's distance to the end of the L-shaped beam from 0.25 inches to 4.00 inches as shown in Figure 7-1, and the corresponding natural frequency of the PTMD ranged from 1.404 Hz to 1.953 Hz as shown in Table 7-1.

Table 7-1 A detailed values of the frequency range of PTMD in water

No.	Mass of PTMD (lbs)	Distance to the End of L Beam (inch)	Natural Frequency (Hz) In Water
1		0.25	1.404
2		0.5	1.466
3		1.0	1.526
4		1.5	1.587
5	10.00	2.0	1.648
6		2.5	1.709
7		3.0	1.770
8		3.5	1.831
9		4.0	1.953

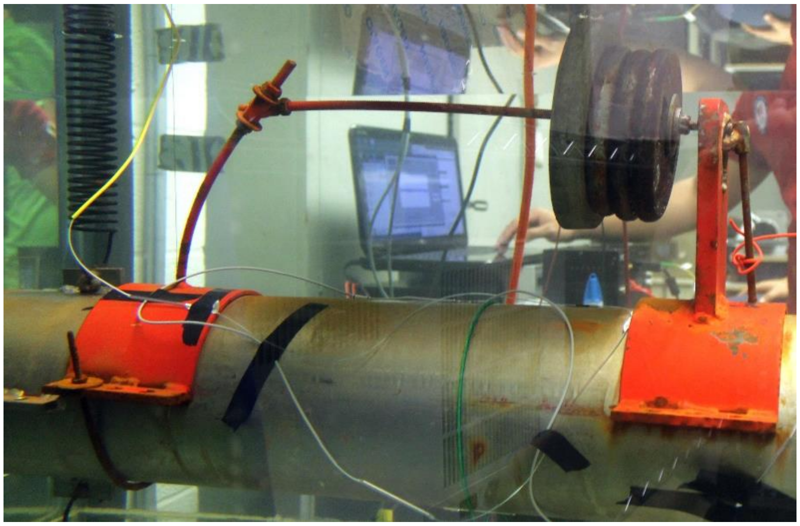

Figure 7-1 A PTMD mounting on the pipe in water

7.3 Experimental Results

The resonant frequency of the short pipe and PTMD was 1.587 Hz in water. That means the performance of the short pipe by the vibration control of PTMD should be at optimum and the corresponding vibration amplitude should be the minimum. In contrast, the effectiveness of a PTMD was decreased significantly by the mistuned frequency. Thus, the frequency range was divided into two parts, one part was from1.587 Hz to 1.953 Hz, and the other one was from 1.587 Hz to 1.404 Hz. Figure 7-2 shows the vibration responses of the short-pipe by the vibration control of PTMD in a frequency range from 1.587 Hz to 1.953 Hz in water. The response shows that the effectiveness of the PTMD is reduced when the frequency of the PTMD is increased too much. Figure 7-3 shows the vibration responses of the short-pipe by the vibration control of PTMD in a frequency range from 1.404 Hz to 1.587 Hz in water. The performance of the PTMD is improved when the frequency of the PTMD is close to the resonant frequency. Hence, the effectiveness of PTMD largely relies on the frequency range in water.

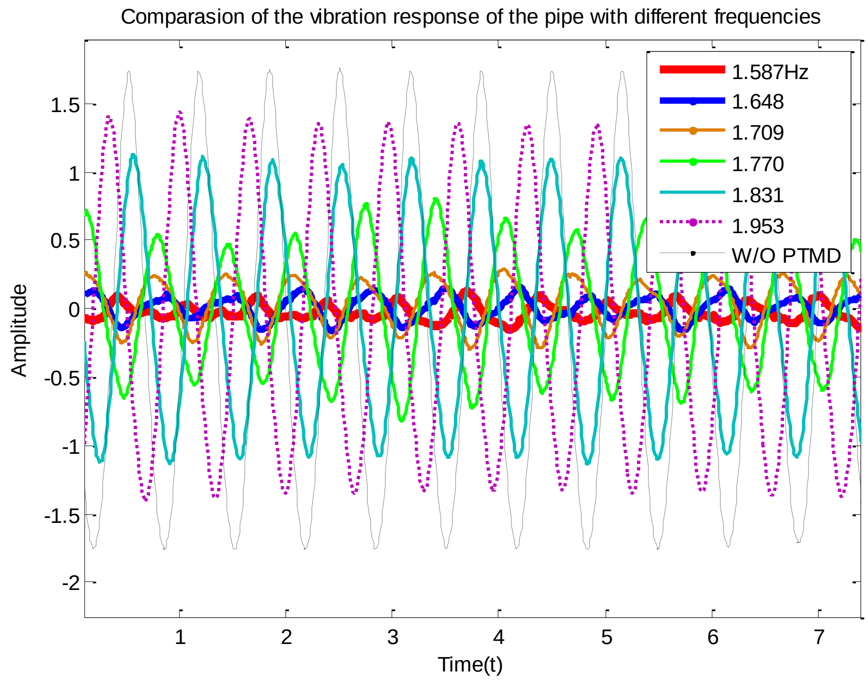

Figure 7-2 Vibration responses of the short-pipe by the vibration control of PTMD in a frequency range (1.587 Hz-1.953 Hz) in water

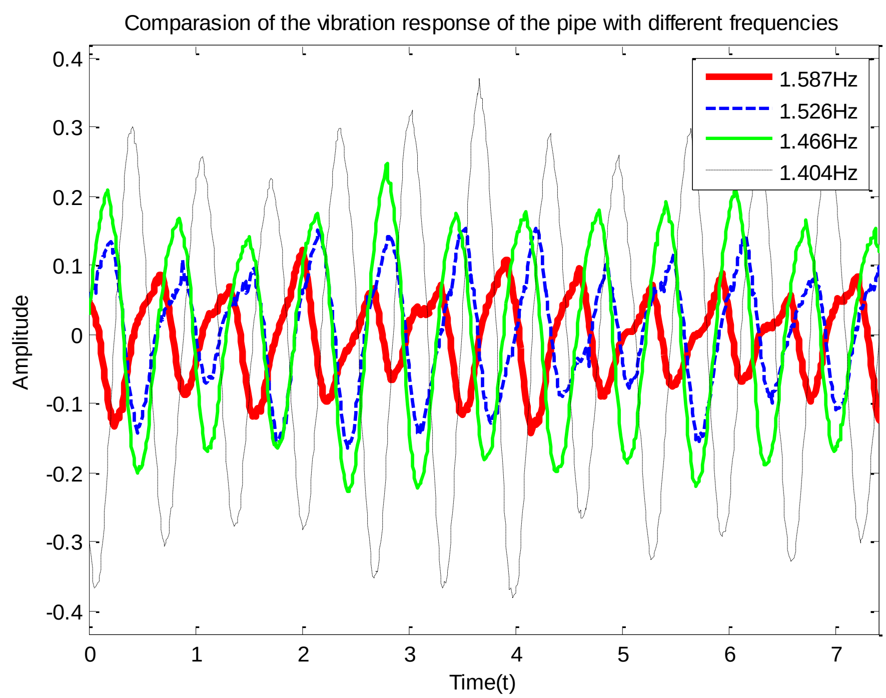

Figure 7-3 Vibration responses of the short-pipe by the vibration control of PTMD in a frequency range (1.404 Hz-1.587 Hz) in water

Figure 7-4 shows the peak amplitude of the pipe by the vibration control of the PTMD for the particular frequency range in water. In addition, the detailed values of the amplitude of the pipe are listed in Table 7-2. Note that the minimum natural frequency of the PTMD, 1.404Hz, can be reached by altering the distance of the mass blocks on the L-shaped beam. The resonant frequency between the short-pipe and the PTMD is 1.587Hz, where the performance of the vibration control is optimum. The red dotted line represents the level at which the PTMD dramatically reduced the vibration motions of the pipe by 70%, as shown in Figure 7-4. The associated frequency is 1.76Hz. Therefore, it is assumed that there is an optimum value of the frequency range for the significant vibration control of PTMD at least from 1.404Hz to 1.76Hz in water, thus verifying the relative robustness of the PTMD in water.

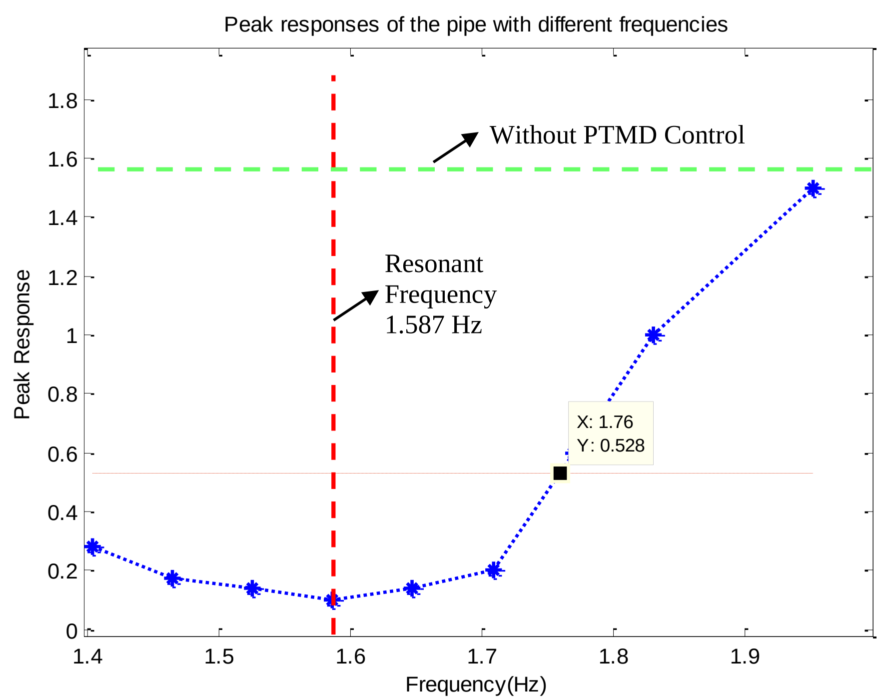

Figure 7-4 The performance of PTMD for vibration control of short-pipe in water in a particular frequency range

Table 7-2 Summary of the amplitudes of the pipe under a frequency range in water

No.	Natural Frequency (Hz) In Water	Average Amplitude of Pipe in Water (inch)
1	1.404	0.300
2	1.466	0.250
3	1.526	0.175
4	1.587	0.009
5	1.648	0.140
6	1.709	0.239
7	1.770	0.600
8	1.831	1.000
9	1.953	1.500

7.4 Summary

In this chapter, we investigated the effect of mistuning of the natural frequency on the performance of the vibration control of PTMD for the pipe in water. During the experiments, we maintained the natural frequency of the pipe and altered the natural frequency of the PTMD from 1.404Hz to 1.953Hz by changing the distance of the mass blocks of PTMD on the L-shaped beam. We summarized the results of the performance of vibration control of the PTMD in water during a particular frequency range. The corresponding experimental results show a notable difference in the performance for vibration control of the PTMD in water. As observed, we may conclude a proper frequency range to verify the relative robustness of the PTMD for the vibration control of the pipe in water, which will be of benefit to better understand the PTMD behavior in water.

8. Conclusion and Future Work

This chapter summarizes the main results of the thesis and recommends directions for future work.

8.1 Conclusion

The concept of subsea field development was first suggested in the early 1970s through placing the wellhead and production equipment on the seabed. A subsea jumper is typically subject to vortex induced vibration (VIV), which may reduce its fatigue life and eventually lead to catastrophic failure in the subsea applications if the vibration is not mitigated. The remoteness of the jumper can complicate inspection and servicing, which means that a passive vibration control device will be more feasible in realizing vibration control of the subsea jumpers without requiring any external power sources. This thesis continues the research of a novel passive vibration control device, Pounding Tuned Mass Damper (PTMD), which was recently invented in the Smart Materials and Structure Laboratory (SMSL) at the University of Houston. In the past, extensive studies of the PTMD and its applications to vibration control were carried out at the SMSL; however, none of the studies experimentally involved a submerged structure. This thesis describes the experimental studies for the vibration control of a submerged structure using the PTMD technology. Extensive experiments were carried out and the results clearly demonstrated the effectiveness of the PTMD in vibration mitigation of a submerged structure. To ensure the proper design of a PTMD, a direct method for estimating the analysis of the natural frequencies of the submerged pipe and PTMD has been proposed and experimentally verified. Furthermore, the robustness of the PTMD in water was experimentally investigated, which is expected to contribute to a better understanding of the PTMD characteristics and its application for vibration control of subsea structures.

In chapter two, the background of the subsea field, the concepts of the subsea jumper and vertex-induced-vibration (VIV) were presented. In chapter three, a basic concept of vibration was introduced. The main methods to realize the vibration control in engineering were presented. A vibration control system mainly includes active control, passive control, or semi-active control depending on whether the external power required for the vibration control system to perform. Furthermore, the working mechanism of a tuned mass damper (TMD) was mainly discussed.

In chapter four, we introduced an experiment for measuring the natural frequency of the full-scale jumper in field. The experimental results of the natural frequency of the jumper in field have been verified by the simulation results by the FEM on the software of ABAQUS. It is beneficial for designing a specific vibrating system (through determination of the natural frequencies) in the water tank to realize the vibration control of the PTMD on the short-pipe.

In chapter five, a direct method for determining natural frequencies of vibration for the pipe and PTMD submerged in water was presented. A series of the experiments was conducted, and a comparison between the experimental results and the estimated results showed that the estimated natural frequencies were close to the natural frequencies measured from the experiments. Hence, this method is particularly useful for determining natural frequencies of structures in water

In chapter six, the vibration suppression of the PTMD on the pipe submerged in water was investigated. A new vibrating system and a new PTMD device were designed, especially the natural frequency of the short-pipe was designed to match the natural frequency of the full-scale jumper we measured in the field. As observed in the experimental results, the effectiveness of the PTMD for the vibration control was successfully realized on the short-pipe submerged in water. Moreover, we observed that the PTMD submerged in water worked much better than in air due to the damping effect of water on the PTMD, which may serve to dissipate the significant portion of vibrating energy from the short pipe in water tank.

In chapter seven, we investigated the effect of mistuning frequencies on the performance of the vibration control of PTMD for the pipe in water. We summarized the results of the performance of vibration control of the PTMD in water during a particular frequency range. The corresponding experimental results have been definitely investigated to show the difference in the performance for vibration control of the PTMD in water. As observed, we may conclude a proper and wide functional frequency range to verify the relative robustness of the PTMD for the vibration of the pipe in water, which will be of benefit to better understand the PTMD behavior in water.

8.2 Future Work

In this thesis, we analyzed the effect of a specific frequency range of the PTMD for its robustness in water. However, the effect of the damping ratio and mass ratio in PTMD parameters are also essential for understanding its behavior in water. For the current experimental setup in the lab, when we changed the mass of the PTMD, the natural frequency would be altered as consequence, thus a difficulty arises when attempting to change the mass ratio while at the same time maintaining the same frequency of the PTMD. In the future, to solve this problem, we may need an optimum design of PTMD relying on the particular masses.

Furthermore, we focused on the application of the single PTMD in this thesis, and it has been verified to be effective in the vibration control of the short-pipe in the water tank. However, a system consisting of multiple PTMDs with different dynamic characteristics should be conducted in the future. Since each PTMD will be tuned to a different natural frequency of the structure, it may be more effective and robust in vibration control than a single PTMD [46].

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