# OTC 8698

# Research Challenges in the Vortex-Induced Vibration Prediction of Marine Risers

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## Abstract

Current research in the prediction of vortex-induced vibration of marine risers is driven by problems in several areas. These include understanding the physics of the fluid-structure interaction, developing adequate structural dynamic modelling tools, acquiring high quality full-scale response data, finding appropriate techniques for analysis of response data, calibrating response prediction programs, and inventing cost effective suppression and protection methodologies. Problems in each of these areas are defined and key limiting issues are described.

## Introduction

Exploration drilling and production in many new locations around the world are being confronted by significant current hazards. Challenges posed by greater water depths, more hostile conditions and greater field development costs are resulting in many new design concepts, such as catenary risers, slim risers and novel vibration protection strategies. In some cases our ability to predict response amplitudes, stress levels and fatigue damage rates of risers due to vortex-induced vibration (VIV) is not adequate. This paper addresses five of the problem areas confronting us. They are (1) the understanding of the fluid mechanics of the fluid-structure interaction, (2) the development of better structural dynamic modelling techniques, (3) the acquisition of high quality, full-scale, riser response data for the purpose of calibration of response prediction models, (4) the analysis of multi-channel field data for comparison to response prediction models, and (5) the development of VIV suppression and protection technologies. No doubt there are many other important issues, but these five are highlighted here.

## Fluid-Structure Interaction

There are many issues involving the hydrodynamics of VIV that we do not fully understand. Those that are the most important to the fatigue life prediction of risers have to do with our ability to model the lift force and damping on the cylinder as a function of the local fluid velocity and the cylinders own motion. It is a non-linear interaction that is sensitive to Reynolds number, roughness, velocity, and turbulence, as well as the interaction between the amplitude and frequency content of the cylinder motion. There are many aspects that are not well understood. One is examined here by means of an hypothetical example.

A uniform straight riser as shown in Figure 1. It is exposed to a current profile consisting of two regions, each occupying one-half of the length of the riser. The upper region has velocity $U_1$ and the lower region has a lesser velocity $U_2$. The hypothetical problem is to predict the response of this riser to the lift forces that result from the vortex shedding from these two regions. To further simplify the problem, assume that the Strouhal number is the same in both regions and is estimated to be 0.2. This allows us to say that the expected frequency of the lift force in each region is given by the simple relationship,

$$F(Hz) = S_tU/D, \text{ where } St = 0.2. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Further assume that this riser has natural frequencies that are coincident with the expected lift force frequency in each of the two zones of the riser. A completely linear solution would say that the total response of the riser would be the sum of the responses to the excitation in each of the two different flow regions. However, since the problem is not linear, it is possible that the response in the lower region due to the higher frequency input in the upper region will influence the excitation in the lower region and vice-versa.

In fact, it has often been observed in sheared flows that response from one region may dominate the total response of the riser, apparently by disrupting the excitation process in competitor regions. Such phenomena are not well understood and present prediction models do not reliably take them into account. The state of the art of modelling local lift forces, including the effect local motion is quite primitive. We need relevant experimental data and better simultion methods.

Some interesting research work is presently being conducted by Kyrre Vikestad at NTNU in Trondheim, Norway. His doctoral dissertation is on the subject of the interaction between externally imposed structural vibration and local vortex shedding processes. Some preliminary results of this experimental work may be found in Ref. 1.

## Structural Dynamic Models

The preliminary design of fatigue resistant risers requires relatively easy-to-use structural dynamic models, which have the capability to estimate dynamic stress levels in the riser as a function of the properties of the structure and imposed velocity profiles. The programs must lend themselves to easy parametric variations of current profiles, tension and structural properties. The user must understand the assumptions and program limitations. The most widely used program at the present time is the MIT program SHEAR7.

SHEAR7 combines easy to run features with a reasonabley sophisticated, but invisible to the user, non-linear, fluid-structure, interaction model. The interaction model allows for the local lift coefficient and local hydrodynamic damping coefficient to depend on the response amplitude. SHEAR7 does not, as yet, include a means of allowing the response of one mode to influence the excitation of other modes, which is the issue posed in the previous section.

SHEAR7 is based on mode-superposition and therefore has a practical limit of about one hundred participating modes. The program was initially written to model straight risers with constant diameter but with spatially varying tension. It has been extended to model structures such as catenaries, by hybrid techniques in conjunction with finite element models. As with all existing VIV design programs for risers, SHEAR7 requires calibration with measured data.

The relative sparsity of data at super-critical Reynolds numbers limits the absolute accuracy of all programs currently available. In many straight riser scenarios in sheared currents, common to the industry today, the likely error in the response amplitude prediction may be as high as a factor of two. Much of the reason for this lack of accuracy is to be found in our poor ability to model the hydrodynamics and in the lack of calibration data at high Reynolds numbers. The hydrodynamics issues were mentioned in the previous section and the calibration issue is addressed in a later section on field data. For the remainder of this section the focus is on the limitations of current structural dynamic, modeling methods.

As mentioned before, SHEAR7 is based on the mode superposition method, which has practical limitations when the number of excited modes becomes large. SHEAR7 is limited to 100 participating modes. Many deepwater production risers will require modeling of dynamic properties that may be best described as typical of structures that behave as if infinite in length. For example, vortex shedding in high velocity surface currents may produce travelling waves at the top of the riser which are damped out before reaching the bottom end. Mode superposition models are poorly suited for such scenarios.

Conventional finite element models are not very attractive, because of the large numbers of elements required. One alternative method is described here. It is called a wave-based, finite element model. This method is the topic of current research at MIT, by two doctoral students, Mark Hayner and Enrique Gonzalez. The method is introduced here by means of an example and uses figures taken from the students' work.

About one year ago the author and student Mark Hayner provided preliminary design estimates of the natural frequencies and modes shapes of the riser model described in a paper in this session by Erling Huse[2]. The riser model was to be constructed of a .030m diameter steel pipe with an inside diameter of 0.026m and a mass per unit length of 2.313 kg/m. The pipe was to be 90m long, connecting to a light-weight synthetic cable, 5m to 40m in length with a mass/length of approximately 0.8 kg/m. The system was designed so as to isolate the transverse vibration response of the pipe from that of the cable by means of a large rigid body inserted between the two sections. One of the challenges was to optimize the size of the rigid body. It was desireable to have it be as small as possible to facilitate handling, yet have the necessary dynamic properties. The eventual design selected was a water flooded aluminum cylinder, 1.0 m long, 0.5m in diameter and having a mass of 231 kg, when filled with water. A diagram is shown in Figure 2. The tension in the system used in the design calculations was 2000 N.

A dynamic model was needed which could simulate the effect of the rigid body motion as well as the vibration of the pipe and the cable. The wave-based, finite element method was selected. A wave-based finite element is one in which the full wave propagation solution is applied inside of the individual beam elements. Everywhere inside of the element the wave solution is known and may be evaluated explicitly. It is not necessary to have many elements per wavelength. Thus one advantage of the method is that models valid at high frequency may require relatively few elements, as shown in the following example. The power of the method becomes even more apparent when significant levels of damping are present.

In all cases shown here the model had only five elements, three for the upper pipe, one for the rigid body and one for the short cable. First consider the effect that the lumped mass had on the mode shapes under lightly damped conditions. Figures 3 and 4 simply show the mode shapes of two particular modes and illustrate how effectively an appropriately sized rigid body may be used to isolate the dynamic response of two different structural regions.

Figure 3 shows the lightly damped mode shape at 1.23 Hz, which corresponds to the seventh mode of the pipe. Figure 4 shows the lightly damped mode shape of the first mode of the cable section at 8.96 Hz. The motion of the one-meter long cylinder is included in the figures at the junction between the two structural regions. However, the cylinder is quite short at this scale and is not highlighted. In each case the vibration response is confined to one of the two regions. In the actual system that was deployed the lower cable was to have variable length. This design allowed the natural frequencies and mode shapes of the pipe to be independent of the length of the cable.

Figure 5 is the response of the system to a sinusoidal force at 4.71 Hz. The force is applied one-third of the distance down from the top of the riser. Significant damping has been imposed on the system. Dramatic spatial attenuation is evident. It would be difficult to model such a system by mode superposition. There are many common features between this example and a lazy-wave riser. The lazy-wave riser will have buoyancy modules, significant hydrodynamic damping, and complex, wave propagation behavior, which is not easily modelled by conventional finite element or mode superposition models.

## Full Scale Riser Instrumentation Projects

The next issue is the need for high quality, full-scale calibration data. All existing riser response prediction programs require calibration with full-scale data, especially at super-critical Reynolds numbers. There have been some recent efforts to provide such data. These early efforts have revealed problems, which need to be overcome, so as to provide high quality calibration data.

The first problem is that the number of sensors is often inadequate for the purpose of resolving mode shapes. A typical installation may have from five to seven accelerometer packages attached to the riser. This is just adequate to resolve the lowest modes of risers in shallow water. In deeper water the number of excited modes will increase, increasing the required number of sensors. The use of more sensors of course drives up the cost, which discourages instrumentation programs.

The second problem is measurement simultaneity. Most instrumentation projects attach independent battery operated instrument cannisters to the riser at varous depths and leave them in the water for the duration of the drilling, which may be as long as eighty days. The clocks of the individual instruments do not stay synchronized, presenting a correction problem for the person faced with analyzing the data. The alternative is to hardwire every sensor to the surface, usually a much more expensive alternative.

A third problem is with the use of accelerometers. Some riser natural frequencies of interest are at less than 0.1 Hz. Accelerometers that function at such low frequencies are sensitive to gravity. The tilt of the riser associated with curvature of the mode shapes and with surface vessel motion introduces a gravitational error component into the signal. It is sometimes possible to work around this problem with clever data reduction. There are alternatives to accelerometers. Strain gages, for example, are not sensitive to gravity. However, they are more difficult and expensive to work into the riser plan, especially in large numbers.

The author's ideal instrument would be a spoolable, tough instrumented cable, which could be deployed with and strapped to the riser. The cable would contain multiple strain or acceleration sensors and the signals would be recorded on a single deck mounted acquisition system.

One final instrumentation issue is the lack of riser tension data, recorded with and at the frequency of the riser vibration data. The importance of dynamic variations in tensions may be much greater than previously thought, especially in regulating the mode shapes and therefore the fatigue life of curved risers.

Good current data is also a must. The availability of acoustic doppler profilers has made it possible to acquire data of good quality when the instrument program is well planned and executed.

## Data Reduction and Analysis

A primary purpose of riser instrumentation systems is to gather data, which may be used to improve response prediction models. Current efforts have focused on extraction of mode participation factors from the measurements of the motions at a small number of discrete points. These measurements may be contaminated by tilt and may not be exactly synchronous. However, even noise free, synchronous data may have yet another hidden problem. The extraction of mode participation factors usually requires the knowledge of the mode shapes. If the mode shapes are known exactly the extraction of mode participation factors is relatively straightforward. A typical analysis might proceed as follows.

Let $\vec{w}(x,t)$ be a vector representing the dynamic response of the riser at all measurement points. At every instant in time the response may be considered to be a weighted sum of the mode shapes of the system. The weighting factors are the mode participation factors, which are represented by a vector $\vec{A}(t)$. The superposition formula is simply,

$$\vec{w}(x,t) = [\Phi]\vec{A}(t),\ \text{where } [\Phi] \text{ is the mode shape matrix.} \quad \text{.........................................(2)}$$

If the number of modes in the mode shape matrix is equal to the number of sensors, and if these mode shape vectors are orthogonal normal modes, then it is a simple matter to solve for the mode participation factors. At every time step one must multiply both sides of Equation 2 by the inverse of the mode shape matrix, as follows.

$$\vec{A}(t) = [\Phi]^{-1}\vec{w}(x,t) \quad \text{.....................................(3)}$$

This method works as long as the assumed normal mode shapes are the correct ones. The problem is that our knowledge of the mode shapes is usually based upon our structural dynamic computer model and not on real measured mode shapes. If the mode shapes used in Equation 3 are not the correct ones the resulting estimates for the mode participation factors will be in error. This problem is currently plaguing efforts to interpret results from full-scale riser instrumentation projects in the North Sea and even from simple model tests.

There are two reasons that assumed and measured mode shapes may be different. The first is that damping, mass and stiffness properties of the actual structure may differ from that used in the computer model. The second is that the instrumentation has errors in calibration or installation (such as position or orientation).

One simple example is given in Figure 6. This shows two possible mode shape estimates for the first mode of vibration of the constant tension riser model, which was tested in the rotating rig apparatus operated by Marintek. This model test is described in OTC paper 8700, by H. Lie, K. Mo and J.K. Vandiver[3]. The purpose of the model test was to investigate the influence of staggered buoyancy modules on VIV response. A baseline set of data was acquired for the uniform diameter, bare riser. There were nine functioning cross-flow accelerometers. Since the cylinder was close to constant tension, and had uniform mass distribution, a reasonable estimate of the mode shape for the first mode of vibration would be a half sine wave, which is the theoretical solution for the first mode of a beam under tension with pinned ends. Mathematically this nine element mode shape vector may be expressed as,

$$\vec{w}_1(x_i, t) = \sin(\pi x_i / L) \text{ for } i = 1 \text{ to } 9. \qquad (4)$$

The $x_I$ are the nine accelerometer locations. These nine points of the theoretical mode shape are plotted on the figure and connected by straight lines, making it a rather rough looking sine wave curve. The second curve is a mode shape estimated from actual measured data, with no a priori assumptions as to the mode shape. The details of the particular measurement and experimental conditions were as follows.

L = 11.48 m, length
D = 0.02 m, diameter
M = 0.462 kg/m, mass per unit length in air
T = 713 N, tension
U = 0.216 m/s, uniform flow velocity
$F_1$ = 1.5 to 1.7 Hz, estimated first mode natural frequency
$F_s$ = 1.8 Hz, observed vibration frequency

This is a rather ideal situation in which only the first mode had significant response. However, the ideal theoretical mode shape and the estimated one are quite different. The estimated mode shape was obtained by Dr. Caterina Stamoulis, a post-doctoral research associate working with the author at MIT. The estimation technique was a state-space-based system identification method, which is being evaluated as a possible tool for extracting natural, frequencies, mode shapes and mode participation factors from multi-channel experimental data.

It is our experience that idealized mode shapes estimated from structural dynamic computer models may be inadequate for estimation of modal participation factors from measured laboratory or full-scale experiments, when several modes are participatiing in the response. This problem has been encountered during attempts to analyze full-scale data from drilling risers in the North Sea. The problem is expected to become even greater when analyzing data from very long, low-tension risers, such as SCR's or lazy-wave risers. New data reduction and analysis tools and techniques need to be developed.

## VIV Suppression and Protection Technologies

There are many locations in the world with current conditions that require protection of the riser from vortex-induced vibration. For example, West of the Shetland Islands some drilling risers have required complete coverage with airfoil shaped fairings. In the Gulf of Mexico helical strakes have been used to protect catenary production risers from VIV. In milder conditions installation of buoyancy modules in an alternating pattern on the riser has been shown to be somewhat effective at reducing VIV. However, reliable data on the efficacy of all of these techniques at super-critical Reynolds numbers do not exist in the literature. Some proprietary data exists for short lengths.

Experiments on models at sub-critical Reynolds' numbers are useful in giving qualitative insight as to the likely behavior under super-critical conditions and are often useful in refining plans and designs for full scale tests. One example is described below, regarding the likely behavior of a riser, partially covered with fairings. Figures 7 and 8 are from experiments conducted at the Offshore Technology Research Center in College Station, Texas. This was a collaborative experiment conducted by the author and Prof. John Niedzwecki of Texas A&M and his student, Scott Chitwood. These figures are from Scott Chitwood's master's thesis work[4]. The research was primarily investigating the VIV behavior of a long tensioned cylinder in combined waves and current. However, a moment of opportunity presented itself, allowing a brief test of a riser with full and partial fairing coverage.

The riser model was 97 feet long, and was made of a single piece of composite coil tubing, 1.5 inches in diameter. The cylinder had an in air weight per unit length of 0.718 lb/ft. Six biaxial pairs of accelerometers were positioned inside of the tube, which was stretched horizontally under the moveable bridge of the wave basin. Figure 7 shows a diagram of the horizontal cylinder, as it was positioned about two feet beneath the surface. The tension was 512 pounds. The bridge was motor driven, providing controlled simulated current speeds.

Figures 8 shows the response of the cylinder to vortex shedding at one particular towing speed, 0.8 ft/s. With no fairing protection this speed resulted in lock-in behavior of the second mode of the cylinder, which had a natural frequency of approximately 1.09Hz. The fairings were of very simple construction. Rectangular, 12 inch by 15 inch, pieces of ABS plastic, $1/16^{th}$ inches thick, were heated softened with an hot air gun and then draped around a cylindrical mandrel forming an airfoil shape. Each section was twelve inches in span and had a chord length of approximately 6.5 inches. The maximum thickness of the airfoil shape was approximately 1.7 inches, slightly greater than the diameter of the test cylinder. The fairings were free to rotate with the flow. At very slow speed the fairings oriented with the flow and completely suppressed VIV when covering the entire span.

This experiment was motivated by the speculative question, "How much fairing coverage is really required to suppress the vibration of a drilling riser?" Lacking relevant data from other sources, this simple test had the promise of providing some much-needed insight, particularly because this type of system is difficult to simulate numerically, and because the effective added mass and damping of the fairings are unknown.

Figure 8 shows the cross-flow response of the riser for various fractions of fairing coverage. For this model, 70% or greater coverage was effective at suppressing significant response. With progressively less coverage response steadily increased. At about 40 % coverage response reached levels comparable to the model with zero coverage. Although these results are not directly translatable to a riser operating at super-critical Reynolds numbers, they inform us as to the general expected behavior.

On one final issue regarding needs for experimental work, not much is really known about the VIV of catenary risers and other shapes with significant curvature. Furthermore, we do not know much about the optimum design of suppression devices for risers operating at various angles of inclination to the flow.

## Conclusions

There are many exciting research and development activities, related to VIV, going on in the offshore petroleum industry at the present time. In the authors twenty three years of working in the field of flow-induced vibration, the interest and activity level have never been greater. At any one time there are typically three or four model or full scale tests being conducted. Several of these activities are described in the other papers in this session. Many of the issues discussed in this paper are the subject of current research projects, which will hopefully yield interesting results in the relatively near future.

## Nomenclature

*F = frequency in Hz*
$S_t$ = *Strouhal number*
*U = flow velocity, ft/s or m/s*
*D = cylinder diameter, inches, feet or meters*
*L = cylinder length, feet or meters*

$\vec{w}(x,t) =$ displacement vector

$\vec{A}(t)$ = modal amplitude vector

$[\Phi]$ = mode shape matrix

## References


1. Vikestad, K., Larsen, C.M., & Vandiver, J. K., "Experimental study of excited circular cylinder in current", Proc. Of the International Offshore Mechanics and Artic Engineering Conference, OMAE, Yoohama 1997.

2. Huse, E,. Kleiven, G., Nielsen, F.G., "Large scale model testing of deep sea risers"' Proc. 1998 Offshore Technology Conference, Paper number 8701, Houston, May 1998.

3. Lie, H., Mo, K., & Vandiver,J.K., "VIV model test of a bare and staggered buoyancy riser in a rotating rig"', Proc. Of the 1998 Offshore Technology Conference, Paper Number 8700, Houston, May 1998.

4. Chitwood, S., Soon to be completed Master's thesis, Dept. of Civil Engineering, Texas A&M Univ. , College Station, Texas. 1998.


## SI Metric Conversion Factors

Ft x 0.3048 = m
Ft/s x 0.3048 = m/s
Inches x 2.54 = cm
Lbf x 4.448 = N

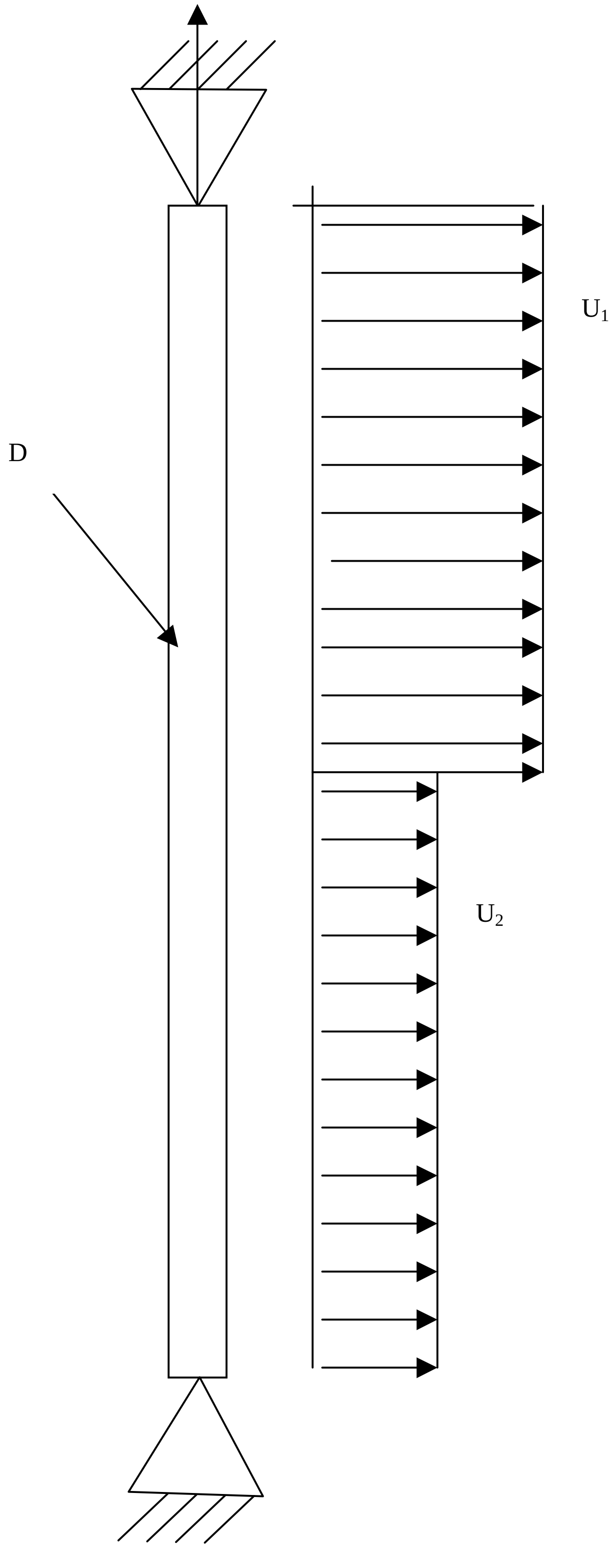


**Figure 1. Two flow speed sample problem**

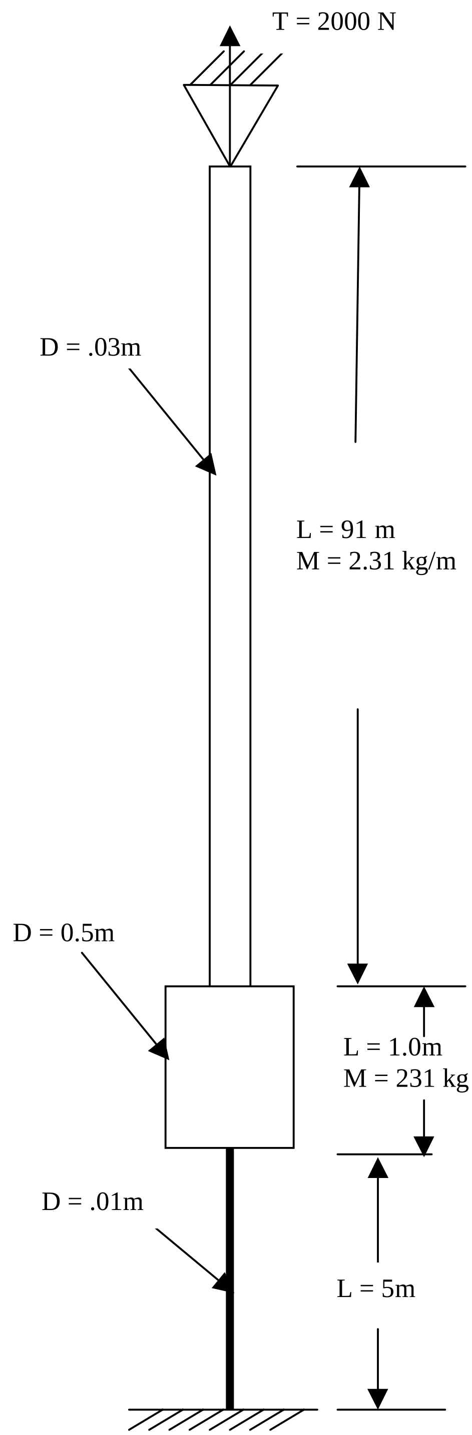


**Figure 2. Pipe, cable and rigid body model**

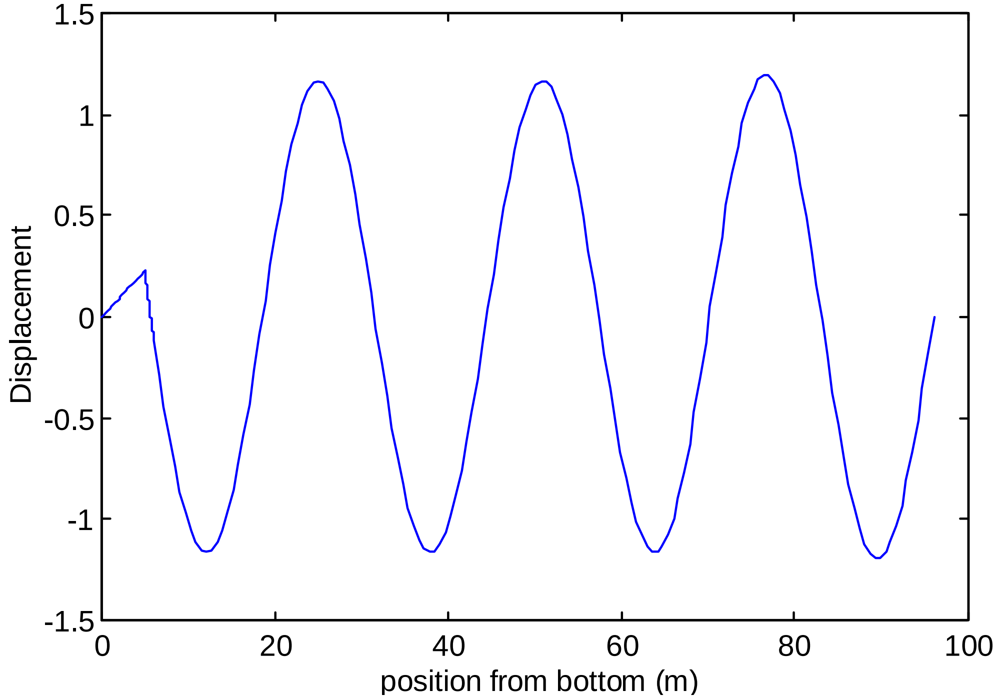


**Figure 3. Pipe 7th mode at 1.23 Hz.**

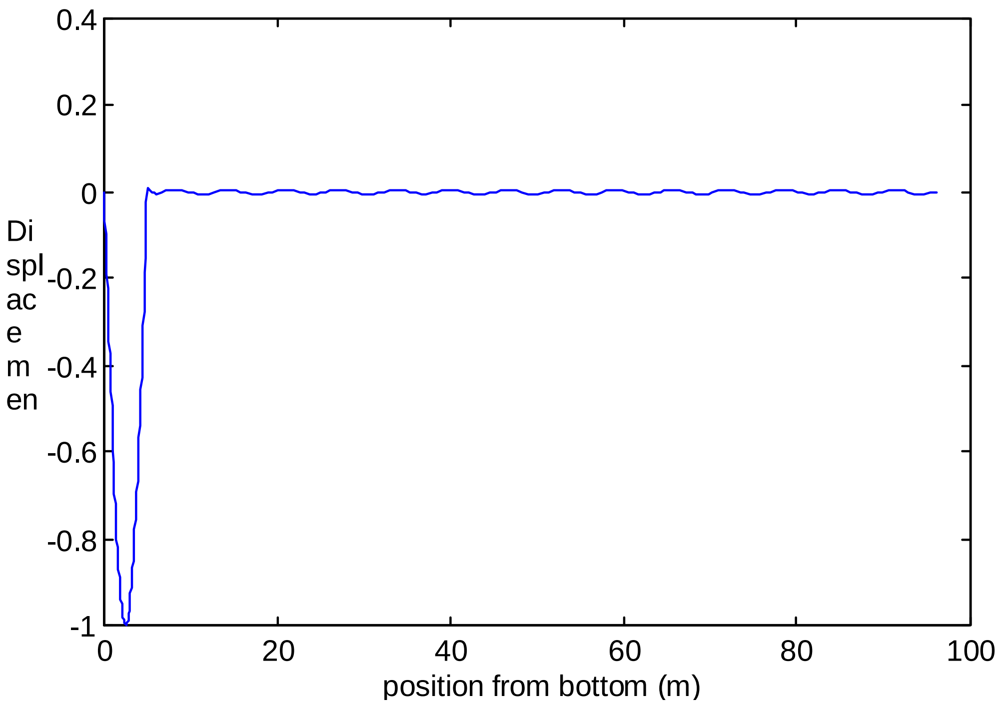


**Figure 4. Cable 1st mode at 8.96 Hz.**

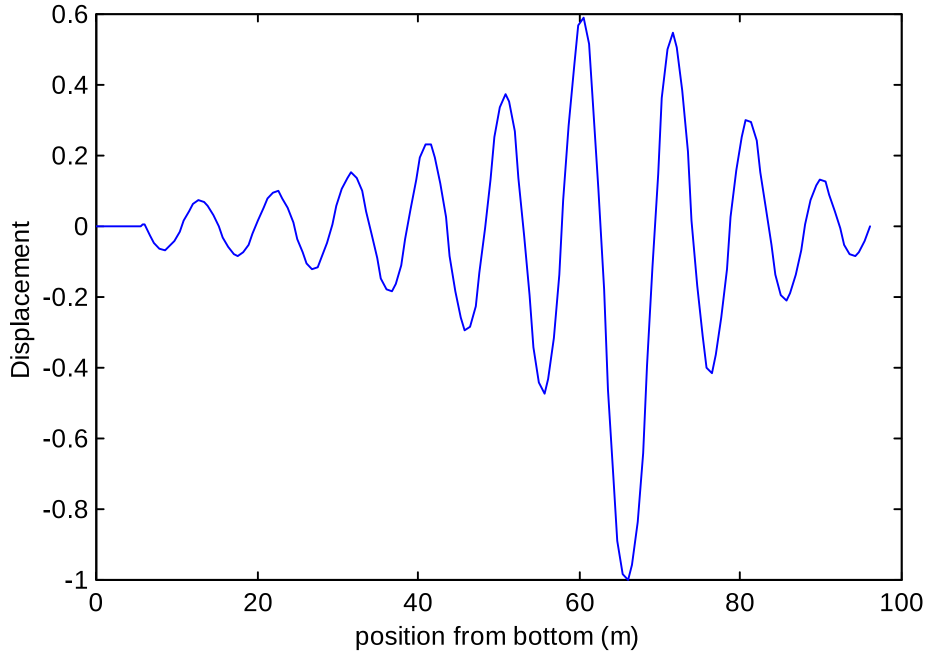


**Figure 5. Damped response to excitation at 4.71 Hz.**

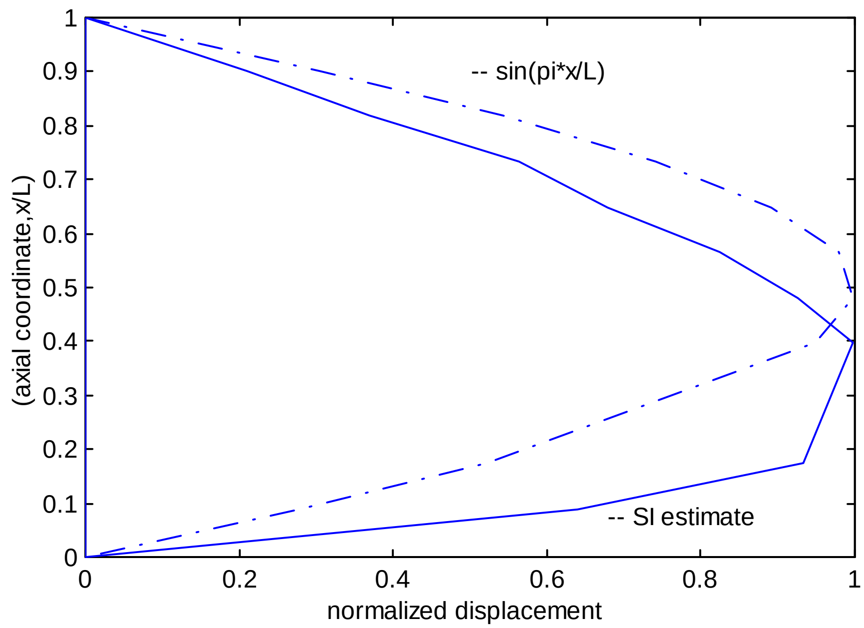


**Figure 6. System ID first mode shape and theoretical sine wave**

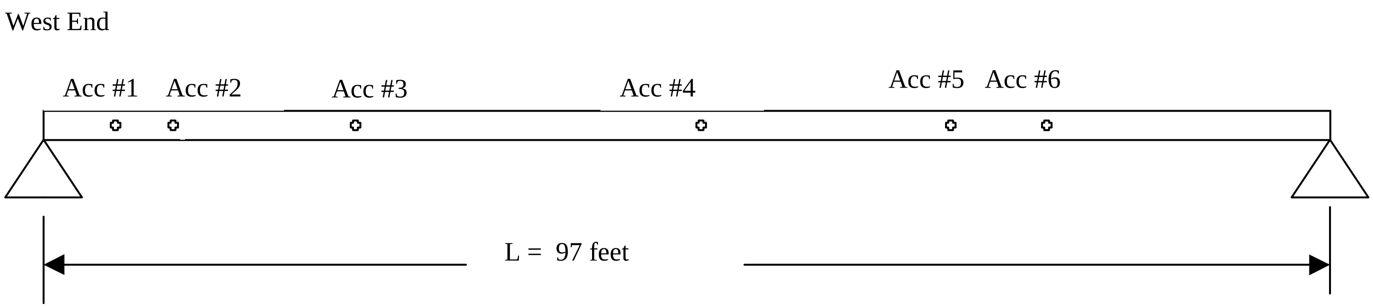


**Figure 7. OTRC test cylinder , Accelerometer distances from the west end.**

**#1 @13.35 ft, #2 @17.33 ft, #3 @25.25 ft, #4 @49.0 ft, #5 @66.78 ft, #6 @78.69 ft**

**Figure 8. OTRC cylinder, cross-flow vibration response versus Per cent fairing coverage.**