Bending Vibration Analysis of Pipes and Shafts Arranged in Fluid Filled Tubular Spaces Using FEM

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ABSTRACT

The interaction between fluid and cylindrical structures has been received extensive research focus over the past decades, and an enormous effort has been paid to investigate various aspects of this complex multi-physics phenomenon. This is not surprising at all since circular cylindrical shells and shafts are one of the most commonly used construction members in a wide variety of engineering structures.

An important step in vibration analysis of fluid-cylindrical structures interaction is the evaluation of their vibration modal characteristics, such as natural frequencies, mode shapes, and added mass. This modal information plays a key role in the design and vibration suppression of these structures when subjected to dynamic excitations. This thesis aims to investigate the bending vibration characteristics of two engineering problems namely, first: shaft surrounded by fluid (oil) confined concentrically by outer cylindrical tube and immersed in infinite fluid (e.g., stern tube) and second: cylindrical pipe filled with fluid and confined in cylindrical fluid medium (sea water), confined concentrically by cylindrical outer tube surrounded by infinite fluid (e.g., Overboard discharge line).

The study has been divided into two as PART-1, bending vibration analysis of stern tube both 2D and 3D acoustic fluid structure finite element model with ANSYS has been employed, and PART-2 bending vibration of an overboard discharge line arranged in a tabular caisson partly filled with sea water has been analyzed with 3D acoustic fluid structure finite element analysis.

To validate the acoustic fluid structure interaction vibration analysis using finite element (ANSYS), a well calculated fluid cylindrical structure interaction problems such as, bending vibration of shaft inside infinite fluid, bending vibration of shaft inside fluid filled rigid tabular space have been re-analyzed using ANSYS and validated with available theoretical results. Using the same trend further complicated arrangements (stern tube, Overboard discharge line) vibration characteristics, added mass coefficients have been analyzed using acoustic fluid structure interaction finite element model (ANSYS). Furthermore mesh adaptation and parametric study has been determined for PART-1 and the corresponding empirical formula to determine added mass of shaft and tube for stern tube has been suggested over specific dimensions. The analysis has been also examined with different density of fluids and found that added mass much depend on fluid density. For PART-2, bending vibration characteristic of system parts have been analyzed before proceeding to the whole system, which helps to know individual vibration characteristic and the assembled system vibration characteristic has been studied including the effect of ballast water level change.

Harmonically forced vibration analysis has been also performed and result reveal that due to transmission of vibration via fluid, both shaft and tube vibrate together at their resonance frequencies and same characteristic has been observed for PART-2 as well.
The investigation of vibration characteristic, added mass, added mass coefficient, mesh adaptation, and parametric studies of the mentioned fluid cylindrical structure interaction has vital role in design and vibration suppression of the system.

Keywords: *Fluid cylindrical structure interaction, bending vibration, added mass, natural frequency, modal analaysis, harmonic analysis.*

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Declaration of Authorship

I declare that this thesis and the work presented in it are my own and have been generated by me as the result of my own original research.

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CHAPTER 1

1. INTRODUCTION

1.1 Motivation, FSI

Solid structures are often in contact with at least one fluid. Therefore, the motion of the fluid and that of the solid are not independent from each other but constrained by a few kinematical and dynamical conditions which model the contact. As a corollary, the fluid and the structure, considered as a whole, behave as a dynamically coupled system [1]. The system could be splitted into fluctuating and permanent motion components, when there is no permanent motion or absence of any permanent flow, the fluid-structure coupled system is always dynamically stable and called fluid-structure interaction (FSI). Whereas incase permanent flow exists and various dynamical instabilities can occur which may have disastrous consequences on the mechanical integrity of the vibrating structures and referred as flow-induced vibration problems. This distinction is extremely useful, as it has profound implications concerning the physical behavior and mathematical modeling of the coupled system. Practical relevance of fluid-structure interaction to engineering is nowadays asserted by a host of problems which are currently addressed to design structural components against excessive vibrations and noise in most industrial fields. It is also convincing that fluid-structure interaction problems are fascinating and challenging which makes the study very appealing.

Some engineering application areas require consideration of an elastic structure surrounded by or conveying a fluid.

- Technical devices- membrane pumps, heat exchangers, pipe-systems, stirring techniques, turbomachinery, airbags, jet engines.
- Aeroelasticity - airfoil flutter
- Civil engineering wind-induced oscillations of high buildings and bridges
- Hydroelasticity - water penetration of off-shore structures, submarines, stern tube, overboard discharge lines, etc
- Biology - the blood circulation in human body, modeling of the heart valves

In fluid structure interaction problems fluid flow depends on the fluid-structure interface deformation and structural displacements are caused by the fluid forces at the interface, and resulted in the interaction between the fluid and the structural fields which is non-linear.

Therefore, analytical solution of whole coupled problem is not possible at all in nearly all cases. As a result, the only possibility is to solve the FSI task numerically. Obviously, the numerical solution of the coupled FSI problem includes the numerical solutions of the fluid and the structural subtasks.

In the past decades the computational fluid dynamics (CFD) has developed many efficient methods for the numerical solution of various fluid dynamics problems. Thanks to this progress lots of commercial programs have been created and successfully applied to diverse complex fluid dynamics problems. Traditionally, the governing equations have been written using Eulerian (spatial) coordinates. From a numerical point of view the finite volume discretisation has been preferred for Eulerian formulation because of its conservative properties.

On the other hand the computational structural dynamics (CSD) has also achieved a great advance independently from the CFD. Numbers of structural dynamics solvers have been developed to solve various structural dynamics tasks. The modelling of a wide range of material laws and structural properties has been made possible by creating special finite elements holding desired features. Contrarily to the fluid dynamics, the Lagrangian (material) coordinates have been selected for the description of the governing equations.

In many applications the structural effect on the fluid can be neglected and only the fluid dynamics part is enough to be modelled, in which only a fluid solver is needed. To the contrary, the fluid forces are very small compared to other external forces and hence, the structural problem may be solved with the existing structural codes without taking into account the flow response. However, in many processes neither the fluid forces nor the structural deformations can be neglected and special programs for the combined solution of fluid and structure dynamics problems are required as it is schematically shown in Fig. 1.1.

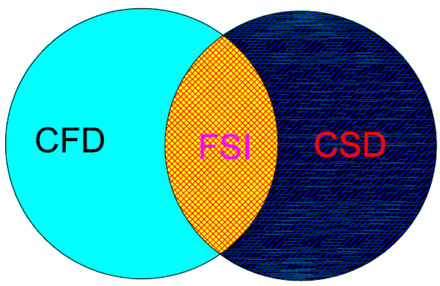


Figure 1.1 FSI problem-subset of fluid (CFD) and structural (CSD) dynamics problems

1.2 Main Thesis Contribution

From the research point of view, study of the interaction between fluid and cylindrical structure is always challenging, both theoretically and experimentally. Since structural deformation and fluid complex motion has to be incorporated. This complexity makes experiments very technically demanding and costly, and the alternative to this problem is mathematical and computational modeling. Adding a strong potential applications and more demanding requirement of cylindrical structures this area of study remains a topic of active research interest to this day.

Circular shafts and tabular cylindrical pipes are commonly used as primary structural components in aerospace, naval and offshore structures. These components are typically either surrounded by fluid or immersed in fluid for different engineering applications. When these structural components vibrate in a fluid, the presence of the fluid gives rise to a fluid reaction force which can be interpreted as an added mass and a damping contribution to the dynamic response of the component. Added mass and damping are known to be dependent on fluid properties (in particular, fluid density and viscosity) as well as to be functions of component geometry and adjacent boundaries, whether rigid or elastic.

For concerned engineering components of this thesis, the excitation from oscillating or rotating machinery that is diesel engine or propeller, random pressure fluctuations, fluid elastic instabilities, and resonant vibration associated with a coincidence between component natural frequencies and vortex-shedding or other flow-related characteristic frequencies has great importance in design of the components. In analyzing the vibration response to these excitations, added mass and damping of the components are important considerations. In general, the added mass will decrease the component natural frequencies; it thus can have a significant effect on the response and has a potential to create large amplitude motion caused by a resonance or instability. While damping is generally less important for "off-resonance" excitation by harmonic driving forces, it is important in predicting component response to broad-band random excitation in which energy is contained over a wide frequency range and damping controls the response amplitude. Therefore, this project is intended to improve design margins and ensure safety and satisfactory operating performance of these structural components, by making known the vibration characteristic and added mass of the components which is the important design parameters for complex fluid-strcuture interaction problems in which structural motion creates fluid flow that in turn influence the structural motions.

1.3 Thesis Objective and Potential Applications

As part of this work, different fluid cylindrical structure arrangements and dynamic conditions will be concerned. Since the studies will be computationally dependent on finite element analysis of fluid cylindrical structure interactions, the well analyzed (both theoretically and numerically) fluid cylindrical structure interaction will be considered as an initial point for further studies. The well calculated fluid cylindrical structure interaction problems such as, bending vibration of shaft inside infinite fluid, bending vibration of shaft inside fluid filled rigid tabular space will be re-analyzed using ANSYS and validated with available theoretical results. Using the same trend, further complicated arrangements and fluid structure interaction problems will be analyzed using ANSYS. This problem includes:

- Shaft surrounded by fluid (oil) confined concentrically by outer cylindrical tube and immersed in infinite fluid (e.g., stern tube).
- Cylindrical tube filled with fluid and confined in cylindrical fluid medium (sea water), confined concentrically by cylindrical outer tube surrounded by infinite fluid (e.g., Overboard discharge line).

This project has two important engineering applications namely:

- Bending vibration of a ship propeller shaft in oil filled steel tube.
- Bending vibration of an overboard discharge line arranged in a tubular caisson partly filled with sea water.

For the mentioned engineering problems, systematic and parametric vibration analysis will be investigated using finite element analysis, which includes,

- Determination of natural frequencies, mode shapes and validation with theoretical results.
- Determination of added mass and its effect.
- Vibration analysis, parametric study and mesh adaptation.

Determination of these vibration characteristics and parametric study has a profound relevance on vibration suppression of the mentioned structural components working in a such environments. Which inturn, improve design margins and ensure safety and satisfactory operating performance of these structural components.

For better convenience, the study will be divided into two: PART-1 and PART-2: which corresponds to the two application areas. For PART-1, bending vibration analysis of stern tube, both 2D and 3D finite element analysis with ANSYS will be employed, and PART-2:

bending vibration of an overboard discharge line arranged in a tabular caisson partly filled with sea water which will be analyzed using 3D finite element analysis as it is not possible with 2D model (partially filled).

In the upcoming chapters, some governing equations related to structural and fluid dynamics will be discussed with their coupling and their boundary conditions. Continuing, the literature review related to this study will be discussed. Finally the full study with a result, discussion and conclusion will be presented consequetly.

CHAPTER 2

2. THEORETICAL BACKGROUND: FSI

The increasing accuracy requirements in many of today's simulation tasks in science and engineering more and more often entail the need to take into account more than one physical effect. Among the most important and, with respect to both modelling and computational issues, most challenging of such 'multiphysics' problems are fluid-structure interactions (FSI), i.e. interactions of some movable or deformable elastic structure with an internal or surrounding fluid flow. The variety of FSI occurrences is abundant and ranges from huge tent-roofs to tiny micro pumps, from parachutes and airbags to blood flow in arteries. From which, vibration of pipes and shafts in fluid and arranged in fluid filled tabular spaces is a typical example. The theoretical investigation of fluid structure interaction problems is complicated by the need of a mixed description. The dynamic equations of solid structures are formulated within the frame work of continuum mechanics using Lagrangian coordinate system which is natural for structural dynamics and the physical quantities used to describe the motion are related to the material points (particles). Whereas the dynamic equations of fluids are also formulated within the framework of continuum mechanics theory, however, they are derived by using the Eulerian coordinate system in which properties such as density, pressure, etc are defined as a fluctuating fields referenced to the space but not the fluid particles.

In the case of their combination some kind of mixed description (usually referred to as the Arbitrary Lagrangian-Eulerian description or ALE) has to be used which brings additional nonlinearity into the resulting equations [2].

A numerical solution of the resulting equations of the fluid structure interaction problem poses great challenges since it includes the features of solid deformation, fluid dynamics and their coupling. The easiest solution strategy, mostly used in the available software packages, is to decouple the problem into the fluid part and solid part, for each of those parts using some well established method of solution; then the interaction process is introduced as external boundary conditions in each of the sub problems. This has the advantage that there are many well tested numerical methods for both separate problems of fluid flow and elastic deformation, while on the other hand the treatment of the interface and the interaction is problematic. In contrast, there are approaches also to treat the problem as a single continuum with the coupling automatically taken care of as internal interface [2]. More detail discussion on fluid structure coupling will be discussed in later sections.

Though, the coupled system of equations are major interest in this study, the separate solid and fluid parts governing dynamic equations should be formulated separately with boundary conditions accordingly. As part of this work, the structural and fluid dynamics governing equations will be formulated in cylindrical coordinate system as this study will be concerned tubes or shaft surrounded by fluid. More over the general fluid structure interaction formulation will be extended to acoustic fluid structure interaction. The possible coupling methodology and solution techniques are also discussed in proceeding section. Use of finite element method for fluid structure interaction and possible ways to modelize acoustic fluid structure interaction using ANSYS will be discussed as well. In general, this chapter will be concerned with theoretical aspects concerning the dynamics system equation formulation, coupling, finite element discretization and solution techniques

2.1 Structural Dynamics

In system equations formulation of solid structural dynamics, the undeformed volume will be denoted as reference state and due to force acting in the volume or on the surface of the volume the structure will experience some deformation. This deformation can either be a real dynamic process yielding a time-depending deformed volume or it can result in a new equilibrium after some initial dynamic deformation. In structural dynamics, deformation fields (e.g. displacement) are a major interest for every material-point of initial undeformed reference volume [3].

In this work, the structural dynamic governing equation of solid structure has been formulated from state of stress in the structure body. As concerned structural components are cylindrical, the dynamic structural equilibrium equations have been developed in cylindrical coordinate using Newton's second law. For formulation of dynamic system equations, the following state of stresses of an infinitesimal element will be used as presented graphically in Fig. 2.1. Fig. 2.2, represents the direct and shear stresses in the radial, transversal and longitudinal directions and the variation of direct and shear stresses in the respective directions.

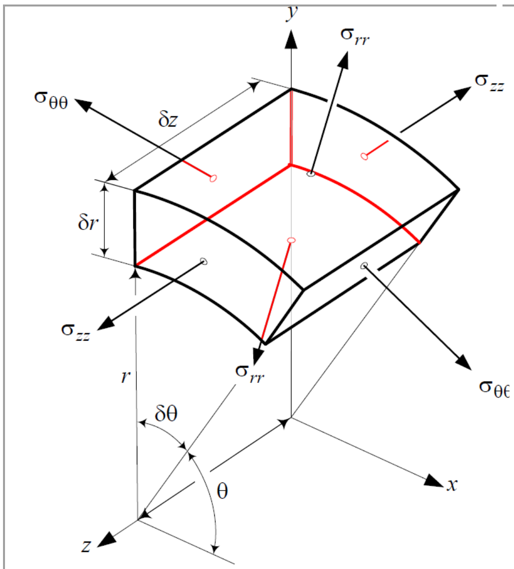


Figure 2.1 Direct Stresses in cylindrical Coordinates, adopted from [4]

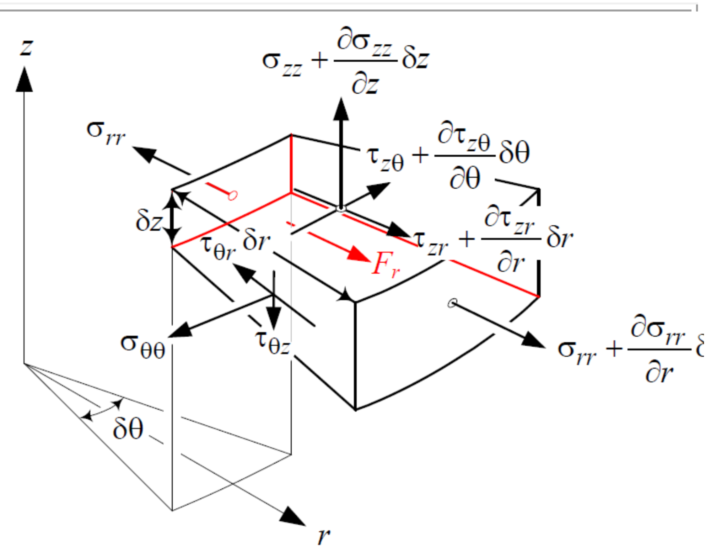


Figure 2.2 Stresses in the plane perpendicular to r and z direction, adopted from [4]

The state equilibrium equations for infinitesimal element in a cylindrical coordinates has been derived in both r, θ, z directions, and the appropriate terms from both sides of the equations were cancelled and after simplifying, it yields:

$$\frac{\partial \sigma_{rr}}{\partial r} + \frac{1}{r}\frac{\partial \tau_{r\theta}}{\partial \theta} + \frac{\partial \tau_{rz}}{\partial z} + \frac{\sigma_{rr} - \sigma_{\theta\theta}}{r} + F_r = 0 \tag{2.1}$$

$$\frac{\partial \tau_{r\theta}}{\partial r} + \frac{1}{r}\frac{\partial \sigma_{\theta\theta}}{\partial \theta} + \frac{\partial \tau_{\theta z}}{\partial z} + \frac{2\tau_{r\theta}}{r} + F_\theta = 0 \tag{2.2}$$

$$\frac{\partial \tau_{rz}}{\partial r} + \frac{1}{r}\frac{\partial \tau_{\theta z}}{\partial \theta} + \frac{\partial \sigma_{zz}}{\partial z} + \frac{\tau_{rz}}{r} + F_z = 0 \tag{2.3}$$

Due to very small angle of $\delta\theta$, $\cos\frac{\delta\theta}{2} \approx 1$ *and* $\sin\frac{\delta\theta}{2} \approx \frac{\delta\theta}{2}$, has been considered and the body forces *Fr , Fθ, and Fz* acts throughout the body in respective directions. Due to the cancellation of the moments about each of the three perpendicular axes, the relations among the six shear stress components are presented by the following three equations:

$\tau_{r\theta} = \tau_{\theta r}$ $\quad\quad$ $\tau_{\theta z} = \tau_{z\theta}$ $\quad\quad$ $\tau_{zr} = \tau_{rz}$

Therefore, the stress at any point in the cylinder may be accurately described by three direct stresses and three shear stresses.

Assume homogeneous and isotropic material (same properties in all directions) the constitutive relation between stresses and strains can be expressed by Hooke's law. As shown in the following Table 2.1, the Stress-Strains, Strain-Displacement, Stress-Displacement Relationships, have been developed to define system equations of motion interms of displacement function.

Table: 2.1 Stress, strain, displacement relationships

Stress-strain	Strain-Displacement
$\sigma_{rr} = \lambda\varepsilon + 2Ge_{rr}$ $\sigma_{\theta\theta} = \lambda\varepsilon + 2Ge_{\theta\theta}$ $\sigma_{zz} = \lambda\varepsilon + 2Ge_{zz}$ $\tau_{r\theta} = Ge_{r\theta}, \tau_{rz} = Ge_{rz}, \tau_{\theta z} = Ge_{\theta z}$	$e_{rr} = \frac{\partial u_r}{\partial r}, \; e_{zz} = \frac{\partial u_z}{\partial r}$ $e_{\theta\theta} = \frac{\partial u_\theta}{r\partial\theta} + \frac{u_r}{r}$
$G = \frac{E}{2(1+\nu)}$ $\lambda = \frac{\nu E}{(1-2\nu)(1+\nu)}$ $\varepsilon = e_{rr} + e_{\theta\theta} + e_{zz}$	$e_{r\theta} = \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r} + \frac{\partial u_\theta}{r\partial\theta}$ $e_{\theta z} = \frac{\partial u_\theta}{\partial z} + \frac{\partial u_z}{r\partial\theta}$ $e_{rz} = \frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r}$

Where $e_{rr}, e_{\theta\theta}, e_{zz}$ are direct strain in respective directions, E-Young's modulus of elasticity, ν - Poisson's ratio, G- shear modulus (Modulus of rigidity), ε - volumetric strain, λ - Lame's elastic constant, $e_{r\theta}, e_{\theta z}, e_{rz}$ shear strain in respective directions.

The stress-strain and strain-displacement relations are developed based up on Hooke's law as shown in Table 2.1, and using these relations, the stress-displacement relations has been developed, more detail formulation could be obtained [4].

Using above relations, the most tractable governing equation in terms of a displacement vector could be developed as follow,

$$u = u_r i_r + u_\theta i_\theta + u_z i_z$$

where i_r , i_θ, and i_z denote unit vectors directed along the (r, θ, and z) axes, respectively. Substituting Hooke's law equations into the dynamic equilibrium equations and introducing strain-displacement relationships yield the governing equations of motion:

$$\mu\nabla^2 u_r + (\lambda + \mu)\frac{\partial\varepsilon}{\partial r} = \rho\frac{\partial^2 u_r}{\partial t^2} \quad (2.4)$$

$$\mu\nabla^2 u_\theta + (\lambda + \mu)\frac{\partial\varepsilon}{r\partial\theta} = \rho\frac{\partial^2 u_\theta}{\partial t^2} \quad (2.5)$$

$$\mu\nabla^2 u_z + (\lambda + \mu)\frac{\partial\varepsilon}{\partial z} = \rho\frac{\partial^2 u_z}{\partial t^2} \quad (2.6)$$

Where μ is the same as shear modulus G, ∇^2 is the three dimensional Laplacian operator in cylindrical coordinates and defined by,

$$\nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{\partial}{r\partial r} + \frac{\partial^2}{r^2\partial\theta^2} + \frac{\partial^2}{\partial z^2} \qquad (2.7)$$

The vector form of the equation could be written as

$$\mu\nabla^2 u + (\lambda + \mu)\nabla(\nabla.u) = \rho\frac{\partial^2 u}{\partial t^2} \qquad (2.8)$$

Where $\nabla = \left(\frac{\partial}{\partial r} + \frac{1}{r}\right)\hat{i}_r + \frac{\partial}{r\partial\theta}\hat{i}_\theta + \frac{\partial}{\partial z}\hat{i}_z$

2.2 Fluid Dynamics

Representation of real fluid motion and solving the realistic governing equations is not simple task yet. Often engineers made some assumptions to represent the real fluid with ideal fluids to study the motion of fluid and their effect. In this work, the fluid medium is assumed as continuous, homogeneous, Newtonian, and without thermal effect, the mass and momentum conservation laws provide the compressible Navier-Stokes equations which governs fluid dynamics. The Navier-Stokes equations can be derived by using an infinitesimal control volume either fixed in space with the fluid moving through it or moving along a streamline with a velocity vector equal to the flow velocity at each point.

In this work, the fluid flow will be modeled by describing its properties in points x in a volume V. This will be done assuming a volume of fluid V and space centered position x.

This space-centered focus is the classical Eulerian view-point. In every point *x* the velocity *v* and density ρ_f will be modeled.

In continuum mechanics, the physical properties of single particles or the interaction of one particle with another is not an interest, instead, the behavior of densities in a continuum is major interest. Using the same interest area, one assumption has to be made that, the observed space is completely filled with the substance and the fact, that matter is made of discrete atoms on a very fine scale has to be ignored.

Let *V* (*t*) be a (moving) volume of fluid. The material in *V* (*t*) has certain distributed properties like the *mass-density* ρ_f , and the momentum ρ_f *v*. The basic equations of continuum mechanics are derived from the following three important physical conservation principles.

- *Conservation of mass*, mass is not added nor removed
- *Conservation of momentum*, change in momentum is equivalent to the acting force
- *Conservation of energy,* energy is not added nor removed

2.2.1 Conservation of Mass

From physical evidence, mass is neither created nor destroyed. Therefore, it could be assumed that for each volume V it holds,

$$\frac{d(m(V(t)))}{dt} = \frac{d}{dt}\int_{V(t)} \rho_f(x,t)dx = 0 \tag{2.9}$$

From Reynolds Transport Theorem,

$$\frac{d}{dt}\int_{V(t)} \rho_f(x,t)dx = \int_{V(t)} \left\{\frac{\partial}{\partial t}\rho_f + div(\rho_f v)\right\}dx \tag{2.10}$$

implies

$$\int_{V(t)} \left\{\frac{\partial}{\partial t}\rho_f + \nabla\cdot(\rho_f v)\right\}dx = 0 \tag{2.11}$$

This holds for every Volume V and assumed that the integrand is continuous and concluded in every spatial point x the equation of mass conservation.

$$\frac{\partial}{\partial t}\rho_f + \nabla\cdot(\rho_f u_f) = 0 \tag{2.12}$$

2.2.2 Conservation of Momentum

Let V be a volume. The forces acting on V can be split into volume forces in V and surface forces on ∂V . Volume forces will be denoted by a distributed function f. Surface forces only act with contact on the surface of the body. They are described by a tensor σ_f . The force acting on a surface with normal n is given by n. σ_f . Together, the total force F(V) acting on V is

$$F(V) = \int_V \rho_f f dx + \int_{\partial V} n.\sigma_f dx = \int_V \{\rho_f f + \nabla\cdot\sigma_f\}dx \tag{2.13}$$

The momentum M(V) of V is given by

$$M(V) = \int_V \rho_f v dx \tag{2.14}$$

Newton's law tells that the change in momentum is equal to the acting forces, equating Eq. 2.13 and Eq. 2.14.

$$\frac{d}{dt}M(V) = F(V) \Rightarrow \frac{d}{dt}\int_V \rho_f v dx = \int_V \{\rho_f f + \nabla\cdot\sigma_f\}dx \tag{2.15}$$

Applying Reynolds transport theorem for the scalar values to get

$$\frac{d}{dt}\int_V \rho_f v dx = \int_V \left\{ \frac{\partial}{\partial t}(\rho_f v) + \nabla \cdot (\rho_f v v) \right\} dx \tag{2.16}$$

This yields the point-wise equation, by equating Eq. 2.15 and Eq. 2.16,

$$\frac{\partial}{\partial t}(\rho_f v) + \nabla \cdot (\rho_f v \otimes v) = \rho_f f + \nabla \cdot \sigma_f \tag{2.17}$$

Where $v \otimes v = (v_i v_j)_{i,j=1}^3$ is the dyadic product. With the conservation of mass using the identity,

$$\frac{\partial}{\partial t}(\rho_f v) = \frac{\partial}{\partial t}\rho_f v + \rho_f \frac{\partial}{\partial t} v = \rho_f \frac{\partial}{\partial t} v - v\nabla \cdot (\rho_f v) \tag{2.18}$$

Furthermore, with

$$\nabla \cdot (\rho_f v \otimes v) = v\nabla \cdot (\rho_f v) + \rho_f v \cdot \nabla v \tag{2.19}$$

Substituting Eq.2.18 and Eq. 2.19 into Eq. 2.17, the equation of momentum conservation will be obtained as,

$$\rho_f \frac{\partial}{\partial t} v + \rho_f v \cdot \nabla v = \rho_f f + \nabla \cdot \sigma_f \tag{2.20}$$

The conservation equations are derived from very basic principles and basically hold for all different materials. The set of equations includes 4 equations (1 continuity and 3 momentum conservation) for the 10 physical quantities density (1), velocity (3) and symmetric tensor σ_f (6). To close this gap, the constitutive laws for the dependence of the stress tensor σ_f on the other variables will be derived. These laws have to be understood as modeling.

Stokes fluid has the property, that the stress tensor σ_f is spherically symmetric if the fluid is at rest $v = 0$, implies the stress-tensor for a fluid at rest is diagonal

$$\sigma\big|_{u=0} = -pI \tag{2.21}$$

where p is the scalar hydrostatic pressure. For a general moving fluid, shear stress tensor τ_f should be introduced to capture the remaining parts

$$\sigma_f = -pI + \tau_f \tag{2.22}$$

The shear stress tensor must be symmetric. Further, the trace of shear stress tensor must be zero. Otherwise it would be “hidden” in the pressure-part “pI”. Linking the shear tensor to the strain rate tensor $\dot{\varepsilon}$ via a general material law,

$$\tau_f = F(\dot{\varepsilon}) \tag{2.23}$$

For Stokes flow the symmetric tensor and isotropic property of the fluid has been considered, and further assumed that the material law is linear and the fluid is Newtonian and the stress tensor will have the form.

$$\sigma_f = -pI + F(\dot{\varepsilon}) = -pI + 2\mu_f \varepsilon + \lambda_f (\dot{\varepsilon})^T I \tag{2.24}$$

with the two material constants μ_f (fluid shear viscosity) and λ_f (fluid volume viscosity). These two parameters usually depend on the temperature and the density of the fluid. For isothermal fluids, the shear and volume viscosity are constant in space and time, thus

$$div\sigma_f = -\nabla p + div\tau_f \qquad \tau_f = \mu_f \left\{\nabla v_f + \nabla v_f{}^T\right\} + \lambda_f div v_f I \tag{2.25}$$

Therefore, the divergence of τ_f is given by,

$$\nabla \cdot \tau_f \equiv (\mu + \lambda)\nabla(\nabla \cdot v) + \mu \nabla^2 v \tag{2.26}$$

Using the identity, $\nabla^2 v = \nabla(\nabla \cdot v) - \nabla \times \nabla \times v$ and

$$\nabla \cdot \tau_f \equiv (2\mu + \lambda)\nabla(\nabla \cdot v) - \mu \nabla \times \nabla \times v$$

Substituting the identities in equation Eq. 2.26, then using the relation between Eq. 2.25 and Eq. 2.26 and substitute in Eq. 2.20, so finally the most simplified conservation equation of Newtonian Stokes fluid could be expressed as follow:

$$\frac{\partial}{\partial t}\rho_f + \nabla \cdot (\rho_f v) = 0 \tag{2.27}$$

$$\rho_f \frac{\partial}{\partial t} v + \rho_f v \cdot \nabla v - (2\mu + \lambda)\nabla(\nabla \cdot v) + \mu \nabla \times \nabla \times v + \nabla p = \rho_f f \tag{2.28}$$

The above formulation of conservation equations work for any Newtonian stokes fluids, as the acoustic fluid will be the primary concern in this study, this equations will be more simplified based up on acoustic theory.

2.3 Acoustic Fluid Theory

Acoustics is the study of the generation, propagation, absorption, and reflection of sound pressure waves in a fluid medium.The propagation of sound waves in a fluid (such as air, water, oil, etc) can be modeled by an equation of motion (conservation of momentum) and an equation of continuity (conservation of mass).

2.3.1 Derivation of the Governing Equations

Using conservation of mass and conservation of momentum equations derived above for any Newtonian Stokes fluid, the corresponding equation for acoustic fluid will be formulated as follow, under certain properties and assumption related to acoustic fluid.

The following assumption related acoustic fluid will be considered,

- The fluid is compressible (density changes due to pressure variations).
- The fluid is inviscid (no viscous dissipation).
- There is no mean flow of the fluid.
- The mean density and pressure are uniform throughout the fluid.

2.3.2 Conservation of Momentum

Taking conservation of momentum equation derived earlier Eq. 2.28 the corresponding acoustic momentum equation will be derived under the following assumptions,

Assumption 1: Irrotational flow

For most acoustics problems we assume that the flow is irrotational, that is, the vorticity is zero.

$$\nabla \times v = 0 \tag{2.29}$$

Assumption 2: No body forces

Another frequently made assumption is that effect of body forces on the fluid medium is negligible.

$$\rho_f f = 0 \tag{2.30}$$

Assumption 3: No viscous forces

Additionally, if we assume that there are no viscous forces in the medium (the bulk λ_f and shear viscosities μ_f are zero).

Taking the above three assumption into account, the momentum equation is simplified to:

$$\rho_f \frac{\partial}{\partial t} v + \rho_f v \cdot \nabla v + \nabla p = 0 \tag{2.31}$$

Assumption 4: Small disturbances

An important simplifying assumption for acoustic waves is that the amplitude of the disturbance of the field quantities is small. This assumption leads to the linear or small signal acoustic wave equation. Then we can express the variables as the sum of the (time averaged) mean field $(\langle \bullet \rangle)$that varies in space and a small fluctuating field ($(\tilde{\bullet})$) that varies in space and time. That is

$$p = \langle p \rangle + \tilde{p}, \quad \rho_f = \langle \rho_f \rangle + \tilde{\rho}_f \quad v = \langle v \rangle + \tilde{v} \tag{2.32}$$

and $\frac{\partial \langle p \rangle}{\partial t} = 0 \quad \frac{\partial \langle \rho_f \rangle}{\partial t} = 0 \quad \frac{\partial \langle v \rangle}{\partial t} = 0$, no variation in time,

Substituting this assumption into Eq. 2.31 and since the fluctuations are assumed to be small, products of the fluctuation terms will be neglected. Then the momentum equation is reduced to,

$$\langle \rho_f \rangle \frac{\partial \tilde{v}}{\partial t} + \left[\langle \rho_f \rangle + \tilde{\rho}_f\right] [\langle v \rangle \cdot \nabla \langle v \rangle] + \langle \rho_f \rangle [\langle v \rangle \cdot \nabla \tilde{v} + \tilde{v} \cdot \nabla \langle v \rangle] = -\nabla [\langle p \rangle + \tilde{p}] \tag{2.33}$$

Assumption 5: Homogeneous medium

Next we assume that the medium is homogeneous; in the sense that the time averaged variables $\langle p \rangle$ and $\langle \rho_f \rangle$ have zero gradients, i.e.,

$$\nabla \langle p \rangle = 0, \quad \nabla \langle \rho_f \rangle = 0 \tag{2.34}$$

Assumption 6: Medium at rest

At this stage we assume that the medium is at rest which implies that the mean velocity is zero, i.e. $\langle v \rangle = 0$.

Substituting all above assumption in momentum Eq. 2.33, and drop the tildes using $\langle \rho_f \rangle = \rho_f$

The acoustic momentum equation will be as follow

$$\rho_f \frac{\partial v}{\partial t} = -\nabla p \tag{2.35}$$

2.3.3 Conservation of Mass

The equation for the conservation of mass for an acoustic medium can also be derived in a manner similar to that used for the conservation of momentum as follow.

Using similar assumption made for conservation of momentum case, such as small disturbance, homogeneous medium, medium at rest, and the mass conservation equation takes the following form,

$$\frac{\partial \widetilde{\rho}_f}{\partial t} + \langle \rho_f \rangle \nabla \cdot \widetilde{v} = 0 \tag{2.36}$$

The additional assumptions for this case are ideal gas, adiabatic, reversible fluid. In order to reach the system of equations assumed (equation of state for the pressure). To do that, the medium is assumed an ideal gas and all acoustic waves compress the medium in an adiabatic and reversible manner. The equation of state can then be expressed in the form of the differential equation:

$$\frac{dp}{d\rho_f} = \frac{\gamma p}{\rho_f}, \qquad \gamma = \frac{c_p}{c_v}, \qquad c^2 = \frac{\gamma p}{\rho_f}, \tag{2.37}$$

where c_p is the specific heat at constant pressure, c_v is the specific heat at constant volume, and c is the wave speed.

For small disturbances

$$\frac{dp}{d\rho_f} \approx \frac{\widetilde{p}}{\widetilde{\rho}_f}, \qquad \frac{p}{\rho_f} \approx \frac{\langle p \rangle}{\langle \rho_f \rangle}, \qquad c^2 \approx c_0^2 = \frac{\gamma \langle p \rangle}{\langle \rho_f \rangle}, \tag{2.38}$$

where c_0 is the speed of sound in the medium.

Therefore,

$$\frac{\widetilde{p}}{\widetilde{\rho}_f} = \gamma \frac{\langle p \rangle}{\langle \rho_f \rangle} = c_0^2 \Rightarrow \frac{\partial \widetilde{p}}{\partial t} = c_0^2 \frac{\partial \widetilde{\rho}_f}{\partial t} \tag{2.39}$$

Dropping the tildes and defining $\langle \rho_f \rangle = \rho_0$ gives us the commonly used expression for the balance of mass in an acoustic medium:

$$\frac{\partial p}{\partial t} + \rho_0 c_0^2 \nabla \cdot v = 0 \tag{2.40}$$

2.3.4 Governing Equations in Cylindrical Coordinates

As the project work will be developed in cylindrical coordinate system (r, θ, z) with basis vectors e_r, e_θ, e_z, then the gradient of pressure (p) and the divergence of velocity (v) are given by

$$\nabla p = \frac{\partial p}{\partial r} e_r + \frac{1}{r} \frac{\partial p}{\partial \theta} e_\theta + \frac{\partial p}{\partial z} e_z \tag{2.41}$$

$$\nabla \cdot v = \frac{\partial v_r}{\partial r} + \frac{1}{r}\left(\frac{\partial v_\theta}{\partial \theta} + v_r \right) + \frac{\partial v_z}{\partial z} \tag{2.42}$$

where the velocity has been expressed as $v = v_r e_r + v_\theta e_\theta + v_z e_z$

The equations for the conservation of momentum may then be written as

$$\rho_0\left[\frac{\partial v_r}{\partial t}e_r+\frac{\partial v_\theta}{\partial t}e_\theta+\frac{\partial v_z}{\partial t}e_z\right]+\frac{\partial p}{\partial r}e_r+\frac{1}{r}\frac{\partial p}{\partial \theta}e_\theta+\frac{\partial p}{\partial z}e_z=0 \tag{2.43}$$

The equation for the conservation of mass can similarly be written in cylindrical coordinates as

$$\frac{\partial p}{\partial t}+\rho_0 c_0^2\left[\frac{\partial v_r}{\partial r}+\frac{1}{r}\left(\frac{\partial v_\theta}{\partial \theta}+v_r\right)+\frac{\partial v_z}{\partial z}\right]=0 \tag{2.44}$$

2.4 Fluid-Structure Interface

In fluid structure interaction, the motion of fluid and structure are not independent, so they are constrained by few kinematical and dynamical conditions at the interface. There fore the fluid structure interaction problem includes the system equations from fluid dynamics and structural dynamics with coupling conditions at the interface of the fluid and structure, which intended to satisfy the geometrical compatibility and equilibrium conditions.

The conditions are given by the following two general cases:

- On the interface, the fluid and the solid have the same motion, because the fluid adheres to the wall.
- The fluid and structural stresses exerted on interfaces are exactly balanced, because interfaces must be in local dynamical equilibrium.

More precisely the interface boundary conditions will be explained in cylindrical coordinate as follow.

2.4.1 Fluid Part

In the FSI problem the fluid domain movement is prescribed by the structural displacements. Denoting the time-dependent coordinate vectors of the fluid and structural grid points with r_f (t) and $r_s(t)$, respectively. Therefore, the flow domain position should be such that the condition

$$r_f(t) = r_s(t) \tag{2.45}$$

is fulfilled on the interface boundary.

The boundary flow velocity v_f should be also the same as the structural velocity v_s on the interface, i.e.

$$v_f(t) = v_s(t) \tag{2.46}$$

Additionally, non-slip boundary conditions are applied on the moving walls. For viscous fluids they are simply:

$$v(t) = v_g(t) \tag{2.47}$$

Eq.2.46 guarantees that the convective fluxes through the fluid-structure interface are zeros.

2.4.2 Structural Part

To consider the action of the flow on the structure, the fluid forces have to be taken into account when solving the structural subproblem.

There are two fluid forces acting on the surface of the structure - the pressure and the shear forces. Considering structures (for example elastic pipes) conveying fluid or surrounded by it, additional external forces may also exist. Hence, all forces acting on the structure within the fluid-structure interaction have to be taken into account. So, the total force acting on the structure contains pressure and shear force (viscous case) and any externally applied force.

Therefore, the system of equations necessary for the complete description of a fluid structure interaction problem consists of dynamic equations from structure and fluid with the above interface boundary condition.

2.4.3 Acoustic Fluid Structure Interaction Coupling

Taking acoustic fluid momentum equation Eq. 2.35 and continuity equation Eq. 2.40, multiply both by $\frac{\partial}{\partial t}$ and equate together to find the acoustic wave equations

$$\frac{\partial^2 v}{\partial t^2} - c_0^2 \nabla^2 v = 0 \ \text{ or } \ \frac{\partial^2 p}{\partial t^2} - c_0^2 \nabla^2 p = 0 \tag{2.48}$$

Where

$c_0 -$ speed of sound ($\sqrt{\frac{k}{\rho_f}}$ in fluid medium)

ρ_f - mean fluid density

k - bulk modulus of fluid

p - acoustic pressure ($p(r, \theta, z, t)$)

t- time

Since the viscous dissipation has been neglected, Eq. 2.48 is referred to as the lossless wave equation for propagation of sound in fluids. The discretized structural form of Eq. 2.4- 2.6 and the lossless wave Eq. 2.48 has to be considered simultaneously in fluid-structure interaction problems. The discretized wave equation followed by consideration of damping matrix to account for dissipation of energy at fluid structure interface will be presented as followed. In addition to this, the discretized form of structural dynamics including fluid pressure acting on the structure at fluid structure interface will also be presented and finally both equations are assembled together to form coupling matrix which is most tractable format. The detail discretization process is not included in this report; detail derivation could be obtained from [5]. In this report only the major assumptions and description of coupling matrix equation will be addressed.

To more simplify wave equation, Eq. 2.48, assume that pressure varies harmonically, i.e.

$$p = p_e e^{j\varpi t} \qquad (2.49)$$

Where:

p_e - amplitude of the pressure

$\varpi = 2\pi f$

f - frequency of oscillations of the pressure

$j = \sqrt{-1}$

Substitute Eq. 2.49 into Eq. 2.48, the lossless wave equation will be reduced to,

$$\frac{\varpi^2 p_e}{c_0^{\ 2}} + \nabla^2 p_e = 0 \qquad (2.50)$$

The lossless wave equation expressed above is discretized considering the interaction with structure which resulted in equation with fluid pressure and the struruture displacements as the dependent variables to solve. Therefore, the wave equation could take a matrix form:

$$\left[M_e^p\right]\left\{\ddot{P}_e\right\} + \left[K_e^{\ p}\right]\left\{P_e\right\} + \rho_0\left[R_e\right]^T\left\{\ddot{U}_e\right\} = \{0\} \qquad (2.51)$$

Where

$\left[M_e^p\right]$- fluid mass matrix

$\left[K_e^{\ p}\right]$- fluid stiffness matrix

$\rho_0\left[R_e\right]$- coupling mass matrix (fluid structure interaction)

U_e - nodal displacement components

P_e - nodal pressure components

In order to account for dissipation of energy due to damping, if any, present at the fluid a dissipation term is added to the lossless Eq. 2.48.

The discretized wave equation accounting loses at the interface is given by,

$$\left[M_e^p\right]\left\{\ddot{P}_e\right\}+\left[C_e^{\,p}\right]\left\{\dot{P}_e\right\}+\left[K_e^{\,p}\right]\left\{P_e\right\}+\rho_0\left[R_e\right]^T\left\{\ddot{U}_e\right\}=\left\{0\right\} \tag{2.52}$$

Where

$C_e^{\,p}$ - fluid damping matrix, which could be considered in ANSYS by defining MU in material property command [5]. But not considered in the study of this project as the interest is not energy dissipation.

Likewise the structural system equation of motion in matrix form could be derived from equation of motion derived earlier eq.2.4 to 2.6 and takes the form:

$$[M_e]\{\ddot{U}_e\} + [C_e]\{\dot{U}_e\} + [K_e]\{U_e\} = \{F_e\} \tag{2.53}$$

Where,

$\left[M_e\right]$- structure mass matrix

$\left[K_e\right]$- structure stiffness matrix

$\left[C_e\right]$- structural damping matrix

U_e - nodal displacement components

F_e - Excitation force

In order to completely describe the fluid-structure interaction problem, the fluid pressure load acting on the structure at the interface is now added to Eq. 2.53. This effect is included in FLUID29 and FLUID30 only KEYOPT (2) =0 [5]. So, the structural equation will take the form,

$$[M_e]\{\ddot{U}_e\} + [C_e]\{\dot{U}_e\} + [K_e]\{U_e\} = \{F_e\} + \{F_e^{pr}\}$$

The fluid pressure load vector $\left\{F_e^{\,pr}\right\}$ at the interface could be obtained by integrating the pressure over area of the surface and could be expressed as

$$\left\{F_e^{\,pr}\right\}=[R_e]\left\{P_e\right\}$$

So, the discretized coupled complete finite element equations for fluid structure interaction problem could be expressed as follow in assembled form

$$\begin{bmatrix}[M_e] & [0]\\ [M^{fs}] & [M_e^p]\end{bmatrix}\begin{Bmatrix}\{\ddot{U}_e\}\\ \{\ddot{P}_e\}\end{Bmatrix}+\begin{bmatrix}[C_e] & [0]\\ [0] & [C_e^p]\end{bmatrix}\begin{Bmatrix}\{\dot{U}_e\}\\ \{\dot{P}_e\}\end{Bmatrix}+\begin{bmatrix}[K_e] & [K^{fs}]\\ [0] & [K_e^p]\end{bmatrix}\begin{Bmatrix}\{U_e\}\\ \{P_e\}\end{Bmatrix}=\begin{Bmatrix}\{F_e\}\\ \{0\}\end{Bmatrix} \tag{2.54}$$

Where,

$$\left[M^{fs}\right]=\rho_0\left[R_e\right]^T$$

$$\left[K^{fs}\right]=-\left[R_e\right]$$

This is typical dynamic system of equations under excitation force and most tractable format for finite element analysis.

2.4.4 Determination of Added Mass

The concept of added mass is sometimes misunderstood to be a finite amount of water which oscillates rigidly connected to the body. This is not true. The whole fluid will oscillate and with different fluid particles amplitudes throughout the fluid. In three-dimensional flow the amplitudes will always decay far away and become negligible. The added mass concept should be understood in terms of hydrodynamic pressure induced forces. The forced motion of the structure generates outgoing waves. The forced motion results in oscillating fluid pressures on the body surface. Integration of the fluid pressure forces over the body surface gives resulting forces and moments on the body [6]. Nevertheless, mass is physically added to the structure if water is trapped in compartments or even between sidewalls, e.g. inside open-ended cylinders: in this case the enclosed water mass is forced to oscillate with the structure in directions perpendicular to the surrounding walls, but has no influence on axial motions [7].

Added mass coefficients is a function of body form, frequency of oscillation and the forward speed. Other factors like finite water depth and restricted water area will also influence the coefficients. since it depends on the shape of the component and its direction of relative motion, hydrodynamic mass has great significance for the dynamics of a structures, and can be deliberately included as a decisive design variable parameter. As part of this work, acoustic fluid structure interaction added mass formulation will be discussed as follow.

As described before energy dissipation is not the main concern in this study therefore the damping term could be omitted from Eq. (2.54) and yields:

$$\begin{bmatrix} [M_e] & [0] \\ [M^{fs}] & [M_e^p] \end{bmatrix} \begin{Bmatrix} \{\ddot{U}_e\} \\ \{\ddot{P}_e\} \end{Bmatrix} + \begin{bmatrix} [K_e] & [K^{fs}] \\ [0] & [K_e^p] \end{bmatrix} \begin{Bmatrix} \{U_e\} \\ \{P_e\} \end{Bmatrix} = \begin{Bmatrix} \{F_e\} \\ \{0\} \end{Bmatrix} \tag{2.55}$$

Substituting $[M^{fs}] = -\rho_0 [K^{fs}]^T$ the combined system equations describe the complete finite element discretized equations for the fluid structure interaction problem and can be written as follow,

$$[M_e]\{\ddot{U}_e\} + [K_e]\{U_e\} + [K^{fs}]\{P_e\} = \{F_e\} \tag{2.56}$$

$$-\rho_0 [K^{fs}]^T \{\ddot{U}_e\} + [M_e^p]\{\ddot{P}_e\} + [K_e^{\ p}]\{P_e\} = \{0\} \tag{2.57}$$

For incompressible fluid the speed of sound tends very high which implies the fluid mass matrix tend to zero, as it is inverse proporsional to square of speed of sound in a fluid.

$$\left[M_e^p\right]=[0] \tag{2.58}$$

Then the Eq. (2.57) will be simplified to,

$$\rho_0\left[K^{fs}\right]^T\left\{\ddot{U}_e\right\}=\left[K_e{}^p\right]\left\{P_e\right\} \tag{2.59}$$

and implies

$$\left\{P_e\right\}=\rho_0\left[K^{fs}\right]^T\left[K_e{}^p\right]\left\{\ddot{U}_e\right\} \tag{2.60}$$

Substitute Eq.(2.60) into Eq. (2.56),

$$\left([M_e]+\rho_0[K^{fs}]\left[K_e^p\right]^{-1}[K^{fs}]^T\right)\left\{\ddot{U}_e\right\}+[K_e]\{U_e\}=\{F_e\} \tag{2.61}$$

In equation Eq. (2.61), the term $\rho_0[K^{fs}]\left[K_e^p\right]^{-1}[K^{fs}]^T$ the added mass matrix,

Therefore, the added mass matrix of acoustic fluid structure interaction could be expressed as,

$$[M_a]=\rho_0[K^{fs}]\left[K_e^p\right]^{-1}[K^{fs}]^T \tag{2.62}$$

In this study the added mass will be determined from the decrement of structure natural frequency which resulted from added mass. Therefore, modal analysis will be performed to determined the natural frequencies of dry and wetted structural elements considered in this project work.

2.5 Methodology to Solve FSI Problems

Fluid-structure interaction problems or multiphysics problems in general are often too complex to solve analytically and so they need to be analyzed by means of experiments or numerical simulation. Research in the fields of computational fluid dynamics and computational structural dynamics is still ongoing but the maturity of these fields enables numerical simulation of fluid-structure interaction [1].

Fluid-structure interaction problems are heterogeneous mechanical components physically or computationally. With regards to their dynamic interactions, they can be categorized as two-way or one-way coupled system. The coupled system is called one-way, if there is no feedback between the subsystems and two-way, if there is feedback between the subsystems. The concept of coupled systems can be generalized to multi-coupled systems with subsystems. This can be multi-structure-fluid interaction, fluid-structure-fluid-interaction (e.g. water-boat-air). This fully coupled system could be called multi-way coupled.

Basically all fluid structure interaction problems are time-dependent and need to be solved at each time step t_1, t_2, … etc for a subsystem (s(t)). The solution S_{n+1} depends on the state of both subproblems at time t_n as well as on the interaction between both subproblems as shown in Fig. 2.3.

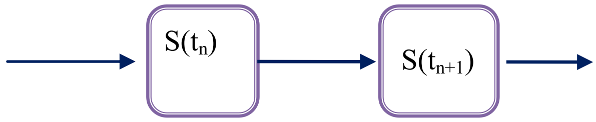


Figure 2.3 Time-approximation of the coupled problem

Roughly there are two simulation approaches for fluid-structure interaction problems,

- Monolithic approach: the equations governing the flow and the displacement of the structure are solved simultaneously at the same time, with a single solver as shown in Fig. 2.4.

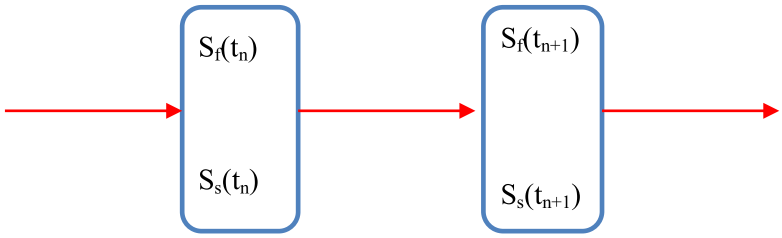


Figure 2.4 Monolithic solution of the coupled system

For this approach, one combined fluid and structure equation has to be formulated.

- Partitioned approach: the equations governing the flow and the displacement of the structure are solved separately, with two distinct solvers.

In every time-step t_n - t_{n+1} both problems are solved separately. The flow problem S_f at time t_{n+1} depends on the flow and on the structure problem at time t_n, but the interaction

at time t_{n+1} is not taken into account as shown in schematic diagram, Fig. 2.5. For the structural problem the same approach is used. Fluid and structural dynamics itself is considered in an implicit fashion, the interaction between both problems is included in an explicit way giving rise to stability problems and asking for small time steps. An advantage of the partitioned approach is that different solvers can be used for the different subproblems. The coupling between fluid and structure comes into the problem by means of boundary conditions on the interface between both subdomains.

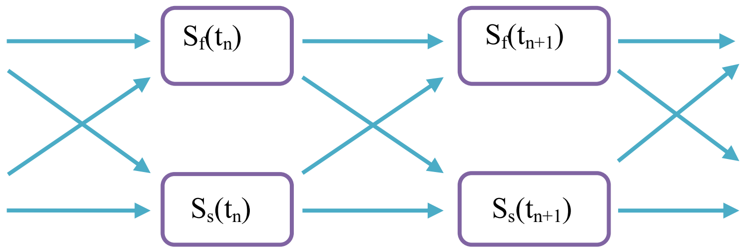


Figure 2.5 Partitioned solution of the coupled system.

In general, monolithic approach requires a code developed for this particular combination of physical problems whereas the partitioned approach preserves software modularity because an existing flow solver and structural solver are coupled. Moreover, the partitioned approach facilitates solution of the flow equations and the structural equations with different, possibly more efficient techniques which have been developed specifically for either flow equations or structural equations. On the other hand, development of stable and accurate coupling algorithm is required in partitioned simulations.

Many researchers used both or either methods in different application areas and detail review will be discussed in next chapter.

Regarding numerical simulation techniques both the Newton-Raphson method and fixed point iteration can be used to solve FSI problems. Methods based on Newton-Raphson iteration can be used in both the monolithic [8][9][10] and the partitioned [11][12]. Detail information about Numerical simulation could be found on mentioned references.

2.6 Finite Element Method

This project work will be entirely dependent on finite element model computation of FSI problem. Therefore it sounds to discuss FEM from the very definition to specific project concern. So, finite element Method (FEM) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as integral equations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then numerically integrated using standard numerical techniques such as Euler's method, Runge-Kutta, etc. The Finite Element Method is a good choice for solving partial differential equations over complicated domains (like cars and oil pipelines), when the domain changes (as during a solid state reaction with a moving boundary), when the desired precision varies over the entire domain, or when the solution lacks smoothness. This powerful design tool has significantly improved both the standard of engineering designs and the methodology of the design process in many industrial applications. In summary, benefits of FEM include increased accuracy, enhanced design and better insight into critical design parameters, virtual prototyping, fewer hardware prototypes, a faster and less expensive design cycle, increased productivity, and increased revenue.

2.7 Basic Steps in Finite Element Methods

Finite element analysis (FEA) consist three key simulation steps namely: idealization, discretization and solution. Idealization (Mathematical modeling), is a process by which an engineer or scientist passes from the actual physical system under study, to a mathematical model of the system. The mathematical model of physical system is not easily tractable, as often involve coupled partial differential equations in space and time subject to boundary and/or interface conditions. Such models have an infinite number of degrees of freedom. To solve this mathematical model, either with analytical or numerical techniques, most physical system mathematical models cannot be treated with analytical method, therefore one has to go for numerical method where finite element method enter the scene. The multilevel decomposition of mathematical model either in space dimensions or in time spaces to reduce the degree of freedom to finite number is a discretization process. There are number of numerical methods to solve depending on type of problems towards discrete solution. Here it has to be noted that each steps in FEM is a source of errors as shown in following Fig. 2.6.

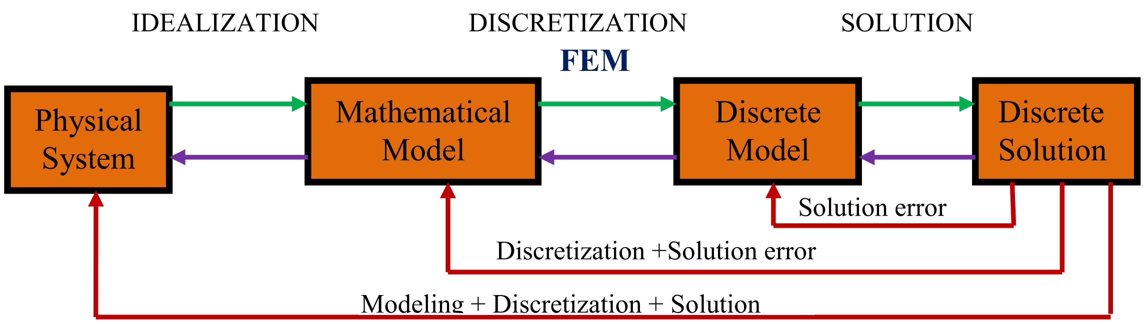


Figure 2.6 A simplified view of physical simulation process

To simulate physical models using finite element methods, different softwares has been developed. In any numerical method softwares there are basic steps to followed to obtain result. In this work, one of the most well known commercial software (ANSYS) will be used.

2.8 Finite Element Method for FSI Problems

The approximation for the FSI problem depends basically on the coupling of the fluid and structure equations. Based on this coupling FSI problems may be divided into problems with weak interaction and problems with strong interaction. The later are found when elastic deformation of the solid takes place. The weak interpolation case happens when large rigid displacements are present [13].

Many researchers have developed different finite element techniques towards problems related to FSI. Still now, it is an active research area to enhance the methodology, effectiveness, stability of fluid structure simulations using finite element method.

The movement of solids in fluids is usually analyzed with the finite element method (FEM) [14] using the so called arbitrary Lagrangian-Eulerian (ALE) formulation [15]. In the ALE approach the movement of the fluid particles is decoupled from that of the mesh nodes. Hence the relative velocity between mesh nodes and particles is used as the convective velocity in the momentum equations.

Typical difficulties of FSI analysis using the FEM with both the Eulerian and ALE formulation include the treatment of the convective terms and the incompressibility constraint in the fluid equations, the modelling and tracking of the free surface in the fluid, the transfer of information between the fluid and solid domains via the contact interfaces, the modelling of wave splashing, the possibility to deal with large rigid body motions of the structure within the fluid domain, the efficient updating of the finite element meshes for both the structure and the fluid, etc [16].

Most of these problems disappear if a Lagrangian description is used to formulate the governing equations of both the solid and the fluid domain. In the Lagrangian formulation the motion of the individual particles are followed and, consequently, nodes in a finite element mesh can be viewed as moving "particles". Eugenio et. al. [16], developed Lagrangian formulation to solve problems involving the interaction between fluids and solids in a unified manner. The method, called the particle finite element method (PFEM), treats the mesh nodes in the fluid and solid domains as particles which can freely move and even separate from the main fluid domain representing.

Ofcourse this simulation method mostly adapted for big deformation or motion of structure and fluid element. For analyses such as small deformation, the later methods are good enough to obtain reasonable simulation results.

2.9 Finite Element Analysis Using ANSYS

As described earlier, any physical system modeling using finite element software (ANSYS) consists of three basic steps:

- Build the model.
- Apply boundary conditions and loads and obtain the solution.
- Review the results.

Below the three steps are described in simplified way,

2.9.1 Build the Model

Building a model is a time consuming steps of finite element analysis, and consists of definition of element type, real constant, material properties, geometry of physical system and meshing which is performed using PREP7.

2.9.2 Apply Boundary Conditions, Loads and Obtain the Solution

This step can be performed either with SOLUTION processor or PREP7 (Loads) and used to define analysis type (e.g. Harmonic, Modal, etc), analysis option depend on analysis type (for Modal, e.g. Block Lanczos, Reduced, Unsymmetric). Load application can be forces, body load, surface load, inertia load, coupled field loads, boundary condition, and load step options are options that can be changed from load step to load step, such as number of substeps, time at the end of a load step, and output controls. After all above steps accomplished successfully, it is ready to initiate the solution which is step to start calculating the result.

2.9.3 Review the Results

After the solution has been calculated, ANSYS postprocessors (POST1 and POST26) could be used to review the result. POST1 is a general postprocessor to review results at one substep (time step) over the entire model or selected portion of the model. POST26 is the time history postprocessor to review results at specific points in the model over all time steps.

2.10 ANSYS for Fluid Structure Interaction Simulation

Fluid-Structure Interaction (FSI) analysis is an example of a multiphysics problem where the interaction between two different analyses is taken into account. The FSI simulation involves performing a structural or thermal analysis taking the interaction with the corresponding fluid into account or CFD analysis which could be performed in ANSYS CFX or ANSYS multiphysics and mechanical for acoustic fluid. The interaction between the two analyses typically takes place at the boundary of the model - the fluid-structure interface, where the results of one analysis is passed to the other analysis as a load.

ANSYS fluid structure interaction simulation could support both one way and two way fluid structure interactions [5]. ANSYS CFX, which is typical to solve conventional fluid structure interaction, will not be discussed in this work. Because, as described in thesis objective, the main target of thesis will be to simulate the vibration characteristic of fluid structure interaction problems, which could be simulated with simpler and time saving acoustic fluid structure interaction technique.

The ANSYS element library includes two different types of fluid elements that can be applied to acoustic fluid-structure interaction analysis. The elements are based on either a pressure or a displacement formulation. The FLUID29 and FLIUD30 elements are based on a pressure formulation in which only a single degree of freedom (DOF), the acoustic pressure, is assigned to each node. The FLUID 29 element is for two-dimensional analysis, while FLUID30 element is for three-dimensional analysis.

Both ANSYS FLUID29 and FLUID30 elements available in the ANSYS Mechanical and ANSYS Multiphysics products. These elements have their origin in acoustic applications; typically, they are used for simulating sound radiation. Their elasto-acoustic and hydro-elastic capabilities, however, are very helpful in solving FSI vibration problems by providing straightforward fluid– structural coupling in a given range of vibration analyses in which ANSYS acoustic elements accomplish the required fluid–structural coupling because they have four degrees of freedom (DOF): one for the sound pressure and three optional displacement DOFs. Thus, a consistent matrix coupling is set up between structural and fluid elements in which strongly coupled physics cause no convergence or performance problems [5].

Coupling structural elements to acoustic elements in this manner allows for transient analysis and, more importantly, for modal and harmonic analysis in the frequency domain. Consequently, for the latter, simulation of the desired stationary peak response within one single frequency step is performed very efficiently. This fluid element will be used for bending vibration of analyses of concerned problem of this thesis.

CHAPTER 3

3. LITERATURE REVIEW

Historically, the early studies of fluid-structure interaction were mostly focused on structural aspects, aiming to provide naval architects and ocean engineers with the most critical estimates needed at the initial stages of ship and submarine design, such as maximum stresses and deflections of the structures. The earliest work dealt with analysis of a few lowest modes and/or early-time approximations [17, 18], and several years later the first papers where complete time-histories of the displacements and strains and/or surface pressure were analyzed and published [19, 20], however most of them were still considered two-dimensional simplifications of the interaction. By the early 1970s, rather realistic three-dimensional structural analysis became possible [21, 22], which continued to advance through the decade, with in the limitations imposed by the computational technologies of the time [23].

With the development of numerical methods, a major shift in the modeling of shell-and structure-fluid wave interaction occurred from analytical approaches to numerical ones sometime in the mid 1980s. An increasingly large number of numerical studies addressing various aspects of the interaction were published in the late 1980s and through the 1990s, eventhough it seems that the primary focus of those studies was on rigid structures, not elastic shells [24–28]. The elastic cylinder subjected to an external shock wave was considered, and the structure-radiated waves were analyzed numerically; an elastic sphere was investigated experimentally [29, 30].

Then, the late 1990s and the first few years of the new millennium saw an unprecedented growth the number and sophistication of studies published on coupled interaction between shells and shock waves and/or non- stationary acoustic pulses. Sandusky, et. al. [31] used numerical simulations to design a series of experiments aimed at detailed stress-strain analysis of a fluid-filled shell responding to an external explosion. Chambers et al. [32] studied the pressure on the inner surface of a fluid-filled shell induced by an internal explosion. Mair [33, 34] compiled an extensive and highly informative review of the computational approaches to the interaction between structures and under water explosions, and discussed the existing analytical and experimental benchmarks [23]. In the study of fluid structure interaction (FSI) with regards to vibration the identification of analysis method and determination of modal characteristics are the most important tasks.

3.1 Determination of FSI Modal Characteristics: Review

A general method for calculating the natural frequencies of such a coupled structure/shell-fluid system has been published by Chen [35] and earlier by Chen et al. [36] for infinitely long shells. They revealed two fundamental modes of vibration for the thin fluid-coupled coaxial shells: in-phase and out-of-phase modes. The lowest natural frequencies of the system are always associated with the out-of-phase modes. Many author investigated the effect of shell geometry and fluid viscosity on the natural frequencies of coupled shell-fluid systems. Yeh et al. [37] studied the effect of fluid viscosity on natural frequency theoretically and found that the influence of the viscosity on the system's natural frequencies was negligible in the cases studied (steel shells with liquid sodium). However, the damping of the out-of-phase modes was found to be noticeably higher than for in-phase modes. Chu et al. [38], investigated experimental study of vibration of water-coupled coaxial acrylic cylinders and demonstrated qualitative agreement with the theory. They found that the natural frequencies of the out-of-phase modes dramatically decreased to a very low limiting value as the annular gap size decreased. The damping ratio of the cylinder system was found to be several times higher for out-of- phase modes than for in-phase modes, and the damping increased significantly as the width of the annular gap decreased. Similar experiments were reported for a related system by Chung et al. [39] who studied the single thin-walled steel cylindrical shell vibrating in a cylindrical water-region confined by the other concentric outer thick-walled cylinder made of concrete. The results for natural frequencies and damping were qualitatively close to those mentioned above. Horficek et al. [40] investigated experimental and numerical study of vibratory modal characteristics of a vertical thin-walled cylindrical shell containing water, or oil, in an adjacent coaxial region. In their study a system of two fluid-coupled cylindrical shells and the effect of increasing liquid level in the shell, the effects of the thickness of liquid layer in the annular region between the vibrating outer shell and another coaxial rigid cylinder, or between two thin coaxial cylindrical shells are studied. The computation of natural frequencies and mode shapes of the coupled fluid-structural vibrations were carried out using finite element method and compared with experimental results. They studied the influence of viscosity of the fluid in the narrow annulus upon the dynamics behavior of the shell.

3.2 Different FSI Analysis Methods: Review

The main approaches employed in the solution of FSI problems are the simultaneous (direct or monotholic) and the partitioned (iterative) solution procedures as well described in chapter two. It should be pointed out that a full coupling between the media is achieved by using any of the two procedures. Usually, partitioned procedures are preferred when the interaction between the fluid and the structure is weak and the simultaneous solution procedure is employed when there is a strong coupling [41]. The vibration analysis of fluid filled cylindrical shell has been one of the great interests of fluid structure interaction. In literature, fluid-filled cylindrical shells have been analyzed for their vibratory behavior using different approaches for instance, Haroun [42] has carried out earthquake analysis by using Bessel function approach for fluid domain and finite element method for structure. However, his studies were limited to axial and first circumferential modes. Ramasamy et al. [43] have carried out studies on fluid-filled visco-elastic shells by using semi-analytical method. Amabili [44] has studied fluid-filled shells by experimental results and along with closed-form solutions. In his method, both structure and fluid domain are treated by using boundary solution technique. Such an approach needs to identify set of trail functions, which satisfies wave equation as well as boundary conditions. Amabili [45] studied the free flexural vibrations of a partially fluid-loaded simply supported circular cylindrical shell for various wet angles. The fluid is assumed to be inviscid along with a free surface parallel to the shell axis. The presence of either external or internal fluid is studied for both compressible and incompressible cases using the virtual added mass approach. Krishna et al. [46], used semi-analytical finite element approach for shell structure and polynomial function for fluid domain in contrast to the usual Bessel function approach, in which they carried out conventional shells and visco-elastic shells as well.

As fluid coupled with cylindrical structure is used extensively in many engineering applications, many researchers were/are interested to make broad investigation about this coupled system. So, detail revision of all available research works could be impossible with this work. However, some more related articles are reviewed with more explanations below.

Leblond et al. [47] formulated time-domain method for a clamped-free elastic circular cylinder, confined in a cylindrical fluid domain initially at rest and subjected to small displacement transient motion along radial line. To cover different fluid-structure interaction regimes, they considered three fluid models, namely potential, viscous and acoustic. The fluid model derived from the general compressible Navier-Stokes equation by formal perturbation method so as to underline their links and ranges of validity a priori, were linear owing to the small-amplitude-displacement hypothesis. For simplicity, they considered the cylinder model as elastic flexure beam. For their general model, they used semi-analytical approach which is based on the methods of Laplace transform in time, in vacuo eigen vector expansion with time-dependent coefficients for the transverse beam displacement and separation of variables for the fluid. The three models were written in convolution products and described through the analysis of their kernels functions, which involve both the wave propagation phenomena in fluid domain and the beam elasticity. Finally they found their work permits to cover broad range of motion and very important to mimic shock loading. In the same paper they pointed out some limitation of their model that it do not consider large-relative-displacement beam motions are not considered, so that boundary-layer separation issues are not handled. Furthermore, the viscous flow is assumed stable, and thus centrifugal and transient hydrodynamic instabilities are not taken into account. Another restriction comes from the single-phase assumption, which makes the models incapable of taking cavitations phenomena into account. However, their investigation could converge three-dimensional fluid–structure solutions that can be used to estimate shock- loaded structural displacement in the first step of design studies and, if required, for selecting a meaningful fluid model for further investigations. They are also believed to be suitable for validating more evolved numerical codes that can solve fluid–structure interaction problems in more complex geometries.

Jeong et al. [48], presented a theoretical study of free vibration of horizontally guided, circular cylindrical shell eccentrically submerged in a fluid-filled rigid vessel. In the analysis ideal fluid has been considered filled in the interior and annulus cavity of the shells. For theoretical formuation, Donnell-Mushtari's shell equations and the velocity potential for fluid motion has been used and Beltrami's theorem has been used to translate the modified Bessel functions in shifted coordinates which used to introduce the eccentricity between shell and the vessel axes. After formulation, they found that eccentricity reduces the coupled natural frequencies for all axial and circumferential modes. In addition, non-dimensional added virtual mass incremental factor that reflects the increase of kinetic energy due to the fluid motion is analytically obtained as a function of eccentricity and modes of vibration. Some other authors also investigate similar work before Jeong et al. with small differences. Chiba [49] and Chiba, et al. [50] carried out a theoretical and experimental study on free vibration of a clamped-free cylindrical shell, which is partially and concentrically submerged in a liquid-filled rigid container using Galerkin method.

Tomomi [51] performed numerical analysis of added mass and damping of a circular cylinder, which oscillates in an air–water bubbly mixture enclosed by a concentric shell. The mixture is assumed to be incompressible. This is because the oscillation frequency of the cylinder is low in his study, and accordingly the pressure change around the cylinder is not so large. An incompressible two-fluid model is solved by the finite element method, to calculate the bubbly flow around the oscillating cylinder. The analysis reveals the effects of the diameter ratio of the cylinder to the shell, the air volumetric fraction and the bubble diameter. It is also clarified that the increase of damping ratio in the bubbly mixture is attributable to the phase lag of the drag force acting on the cylinder behind the cylinder displacement.

Zhou, et al., [52], combined finite element with boundary element methods to calculate the elastic vibration and acoustic field radiated from an underwater structure. FEM is employed for computation of structural vibration and an uncoupled boundary element method, based on the potential decomposition technique, is used to determine the acoustic added mass and damping coefficients that result due to fluid loading effects. Results obtained from their study suggest that the natural frequencies of underwater structures are only weakly dependent on the acoustic frequency if the acoustic wavelength is roughly twice as large as the maximum structural dimension.

Lin, et al. [53] solved linear two-dimensional acoustic wave equation for multiple circular cylinders vibrating harmonically in an infinite compressible fluid. To satisfy the interface boundary condition of a particular cylinder, all cylindrical wave functions are transformed to the coordinates associated with that cylinder. The resulting equations are a system of equations for the undetermined coefficients, which are used to obtain the velocity potential, pressure, and force acting in each cylinder in terms of these coefficients. The added mass matrix is found symmetrical and dependent on the wave number, the cylinder radius, and the distance and orientation between cylinders. On the surface of each cylinder the pressure field also depends on those parameters as well as the orientation of point on that surface. Numerical values of added mass matrix and pressure distribution are obtained for many cases.

As reviewed from different articles, many attempts have been made to improve design margins and ensure safety and to secure satisfactory operating performance of cylindrical machine parts interact with fluids. This is not surprising at all since circular cylindrical components are one of the most commonly used construction members in a wide variety of engineering structures. Specifically, the offshore, naval, aerospace, nuclear, chemical and petroleum industries all extensively employ fluid-interacting cylindrical shell structures. From the research point of view, study of the interaction between fluid and cylindrical structure is always challenging, both theoretically and experimentally. Since structural deformation and fluid complex motion has to be incorporated. This complexity makes experiments very technically demanding and costly, and the alternative to this problem is mathematical modeling. Adding strong potential applications and more demanding requirement of cylindrical structures, this area of study remains a topic of active research interest to this day.

An important step in vibration analysis of fluid-cylindrical structures is the evaluation of their vibration modal characteristics, such as natural frequencies, and mode shapes. This modal information plays a key role in the design and vibration suppression of these structures when subjected to dynamic excitations [4]. These modal characteristics are most affected by added mass and damping of the fluid.

As mentioned above, different fluid cylindrical structure arrangements have been considered for analysis by different researchers. For instance, cylindrical objects such as shafts/pipes with infinite fluid or confined fluid are one of the typical as they have usual industrial construction members. Free or forced vibration natural frequency and hydrodynamic mass of pipes/shafts surrounded by infinite fluid has been calculated well. In addition the shaft immersed in fluid filled rigid tabular space has been also studied by different researchers as described above. However, for shafts/tubes immersed in fluid filled elastic tube situated in infinite fluid, has not been studied theoretically, numerically and experimentally to determine the natural frequencies, mode shapes and added mass of such arrangement. In addition there are no parametric studies performed related to boundary limitations for fluid space or gap between shaft and tube to the writer's knowledge. As a consequence, in this project free and forced bending vibration of pipes and shafts arranged in fluid filled tabular spaces will be investigated using finite element method to determine modal vibration characteristics including parametric study.

CHAPTER 4

4. BENDING VIBRATION ANALYSIS OF SHAFT AND TUBE COUPLED WITH FLUIDS

4.1 Introduction

As described in previous different sections, this thesis intended to deal with bending vibration analysis of tubes and shafts immersed in fluid filled tabular spaces using finite element method (ANSYS). Depending on application areas, the thesis is categorized into two parts for better convenience. PART-1 deals with bending vibration analysis of shaft surrounded by fluid (oil) confined concentrically by outer cylindrical tube and immersed in infinite fluid (e.g., stern tube), and PART-2 deals with bending vibration of cylindrical tube filled with fluid and surrounded by confined partially filled fluid medium (sea water), confined concentrically by cylindrical outer tube surrounded by infinite fluid (e.g., Overboard discharge line).

In both parts, as the main objectives were to determine the vibration characteristics and hydrodynamic mass, acoustic fluid structure interaction using finite element method (ANSYS) has been used to determine all vibration characteristics. As far as both fluid and structural elements are concerned, fluid structure interaction analysis should be properly applied to find useable result. The fluid part could be considered as viscous, non viscous (potential flow), or acoustic depending on the objective and visibility of the problem. As the main objective of this work is to determine the vibration characteristics and their effect, acoustic fluid has been selected to treat the problem which is ofcourse, beneficial interms of simplicity and less computational time.

4.1.1 Finite Element Model Development

The fluid structure interaction bending vibration finite element (ANSYS) analysis of both parts have been developed using structural element (PLANE45 for 2D, SOLID95 for 3D) and acoustic fluid element (FLUID29 for 2D, FLUID30 for 3D).The acoustic elements require density (DENS) and speed of sound (SONC) as material properties. The structural elements require to specify the Young's modulus (EX), density (DENS), and Poisson's ratio (PRXY or NUXY). For 2D analysis, the structural element thickness and fluid reference pressure has to be defined as real constant. The following characteristics of the structure and fluid have been considered in this thesis work as shown in Tables 4.1-4.2.

Table: 4.1 Material property of structural element of models

Structural Material	DENS (kg/m^3)	EX (N/m^2)	NUXY	Thickness (m)
PLANE45	8160	2.08e+11	0.3	0.001
SOLID95	8160	2.08e+11	0.3	-

Table: 4.2 Material property of fluid element of models

Fluid Material	DENS (kg/m3)	SONC (m/s)	Reference pressure (N/m2)
FLUID29	1030	1460	0.1
FLUID30	1030	1460	0.1

Note that: For 2D finite element analysis, the spring element has been defined using COMBIN14 which is ofcourse different for all cases, and will be defined in specific sections. Likewise, the geometry and mesh size will also be described in respective cases and parts of the thesis work, as they are different in each case and parts.

4.1.2 Boundary Condition and Interface Definition

The boundary conditions were applied in accordance with the type of analysis and problem. For PART-1, the bending vibration of simply supported cylindrical beam and for PART-2 rigidly supported cylindrical beam has been considered. Inorder to simulate infinite fluid and free surfaces the pressure at the interface/extreme has been introduced with zero pressure to avoid the reflection of pressure wave at the extreme.

In principle, the infinite acoustic elements absorb the pressure waves generated at the interface, and the pressure wave disappear as the distance goes away from the source, so here the aim is to simulate the outgoing effects of a domain that extends to infinity beyond the FLUID29 and FLUID30 elements. Since modeling infinite fluid is impossible, some auxiliary boundary condition should be defined. In ANSYS there are two different methods to approximate the simulation of infinite domain fluid: setting pressure zero at the boundary and use fluid element (FLUID129 and FLUID130). FLUID129 and FLUID130 provide a second-order absorbing boundary condition so that an outgoing pressure wave reaching the boundary of the model is absorbed with minimal reflections back into the fluid domain. FLUID129 is used to model the boundary of 2-D fluid regions and as such is a line element. FLUID130 is used to model the boundary of 3-D fluid regions and as such is a plane surface element.

For this work, the fluid is assumed as extending only to a finite radius r_4 where the pressure P_o is zero and r_4 is approximated from 2 to 3 times of structural element radius. Taking r_4 twice of structural radius should result in an error of less than 1% compared to the frequency for an unbounded fluid (r_4 = infinity) according to the inverstigation made in this work and ANSYS help. This aims ofcourse to set the pressure wave reaching the bound to zero, or in other word to inhabit the reflection of pressure wave. Generally, setting PRES to zero at an acoustic boundary will define a free surface. If no PRES or fluid loads are applied to a boundary, then it behaves as a symmetry plane with complete deflection of a pressure wave.

In order to consider the fluid structure interaction coupling, the interface should be defined in special way. At the interface between fluid and structure, the fluid in contact with elastic solid element were introduced with displacement (U_x, U_y, U_z) and pressure degree of freedom where as other fluid domain has only pressure degree of freedom as shown in Fig. 4.1.

For acoustic elements that are in contact with the solid, the KEYOPT(2)=0, which is the default setting to define fluid-structure interaction. This results in unsymmetric element matrices with U_x, U_y, U_z, and PRES as the degrees of freedom. For all other acoustic elements, the KEYOPT(2)=1, which results in symmetric element matrices with the PRES degree of freedom. The reasons behind using symmetric matrices are to save space and time as it requires much less storage and computation time.

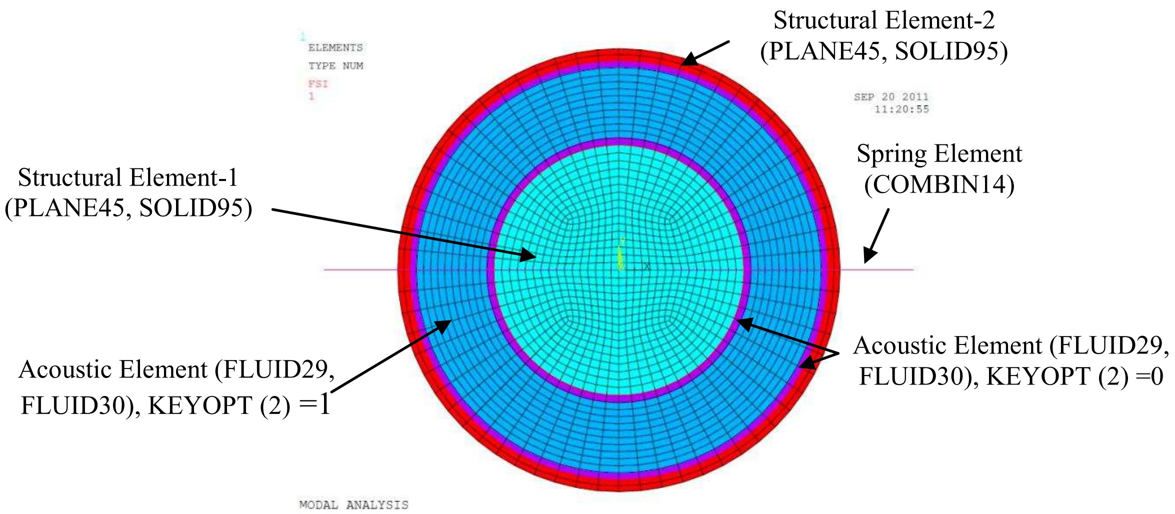


Figure 4.1 Acoustic fluid structure interaction finite element model

4.1.3 Analysis

Simply or rigidly supported bending vibration of cylindrical structural elements has been analyzed for modal and harmonic analysis to determine the vibration characteristic of the structure immersed in a fluid totally (PART-1) or partially (PART-2). The analysis were performed using ANSYS acoustic fluid structure coupled, as a consequence, Block Lanczos solver has been used for dry structural element, and Unsymmetric solver (which is suitable for acoustic fluid structure element) for coupled acoustic fluid structural coupled vibration analysis. Mostly a frequency range of 0 to 200Hz has been used to extract mode shapes and natural frequencies of the system.

With regards to harmonic analysis there are three response analysis methods available in ANSYS: full, reduced, and mode superposition. From which full method is applicable for acoustic problems allowing unsymmetrical matrices. Mode superposition doesn't work with unsymmetric solver.

4.2 PART-1

PART-1 deals with bending vibration analysis of shaft and tube surrounded by fluid (oil) confined concentrically by outer cylindrical tube and immersed in infinite fluid (e.g., stern tube). The analysis will be made in four separate cases (CASE-1, 2, 3, 4), which will attempt to study the analysis of vibrations characteristic from basic solid structural vibration to advanced acoustic fluid structure interactions. In addition, successive analysis of already well investigated problems has been re-analyzed (except CASE-4) to see how much acoustic fluid structure interaction finite element model with ANSYS can be accurately investigate vibration characteristic and added mass of the components. For all cases, the real data from navy ship stern tube has been taken as a reference values, (r_1=0.2125m) for shaft, (r_2=0.3555m) and (r_3=0.3755m) for tube and length (L=8.5m).

4.2.1 CASE-1 Bending Vibration of Solid Elastic Shaft and Elastic Tube in Air

4.2.1.1 Model Development and Analysis

In this case, the natural frequencies and mode shapes of elastic shaft and tube were determined using theoretical formula, 2D and 3D finite element model. For any simply supported beam, the bending natural frequency can be calculated using,

$$f = \frac{1}{2\pi}\left(\frac{\pi}{L}\right)^2 \sqrt{\frac{EI}{M}} \qquad (4.1)$$

where

L is length of the beam/shaft (m)

E is Young's modulus of beam/shaft material (N/m^2)

I is Area moment of inertia (m^4)

M is mass per length of beam/shaft (kg/m) (M_s for shaft, M_t for tube)

f is frequency of the beam/shaft (Hz)

Knowing the natural frequency of shaft or tube, it could be possible to determine the stiffness of the shaft, to develop a 2D lumped parameter model and to further analysis vibration characteristic of the shaft/tube in a fluid. From

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \tag{4.2a}$$

where K (K_s for shaft, K_t for tube) is stiffness of the shaft/tube (N/m), mass M of the shaft/tube (M_s for shaft and M_t for tube).

Using the same formula, the stiffness of the shaft and the tube for different geometry will be determined using

$$K = M(2\pi f)^2 \tag{4.2b}$$

To determine the natural frequencies and mode shapes using FEM, 2D and 3D model have been developed.

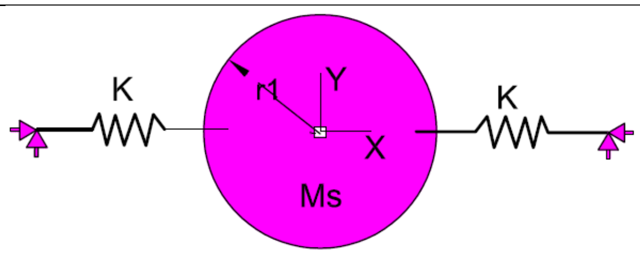


Figure 4.2 Schematic drawing of shaft lumped parameter 2D model

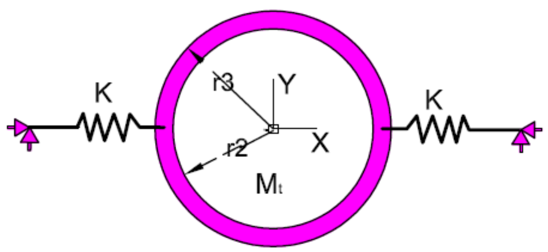


Figure 4.3 Schematic drawing of tube lumped parameter 2D model

The theoretical natural frequency has been determined using Eq. 4.1 and 2D finite element model has been developed using PLANE45 element using a thickness of 0.001m. The corresponding stiffness and mass density has been calculated to apply a lumped parameter model, as shown in Figs. 4.2-4.3. 3D model has been simply developed to determine the natural frequency of the shaft and the tube as shown in Fig. 4.5 and Fig. 4.7.

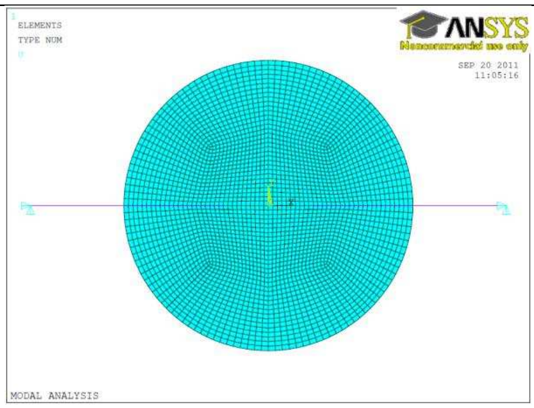


Figure 4.4 2D lumped parameter finite element model of dry shaft

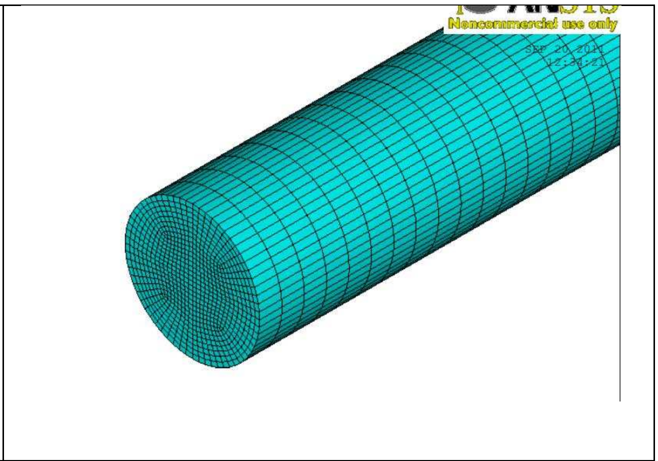


Figure 4.5 3D finite element model of dry shaft

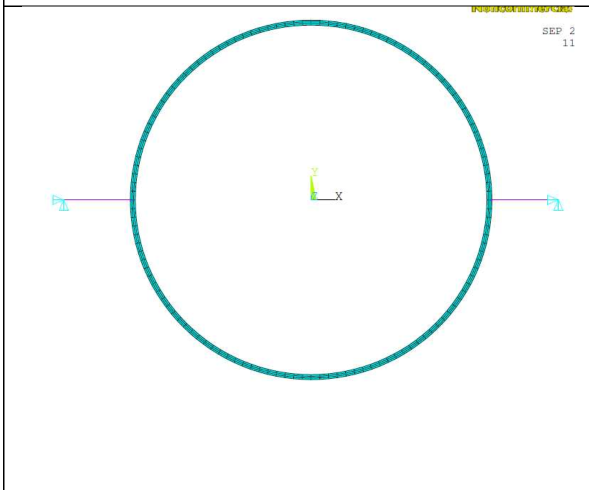


Figure 4.6 2D lumped parameter finite element model of dry tube

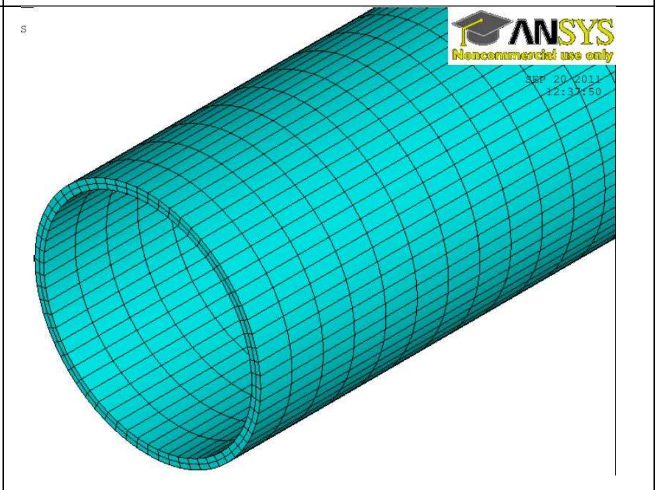


Figure 4.7 3D finite element model of tube shaft

The results from both methods (theoretical, 2D-ANSYS, 3D-ANSYS) have been determined as tabulated in Tables 4.3 - 4.4 for different shaft radius and tube tubes radius respectively.

Table: 4.3 Shaft geometry and corresponding natural frequency in air

r_1(m)	Area (m2)	Mass Ms(kg/m)	Moment of inertia I(m4)	Theoretical result f(Hz)	Stiffness K(N/m3)	2D-ANSYS result f(Hz)	3D-ANSYS result f(Hz)
0.100	3.142E-02	2.564E+02	7.854E-05	5.49	304843.55	5.487	5.470
0.125	4.909E-02	4.006E+02	1.917E-04	6.86	744246.94	6.8544	6.835
0.175	9.621E-02	7.851E+02	7.366E-04	9.60	2859099.04	9.5672	9.561
0.2125	1.419E-01	1.158E+03	1.601E-03	11.66	6216024.85	11.562	11.599
0.275	2.376E-01	1.939E+03	4.492E-03	15.09	17434431.06	14.731	14.985
0.325	3.318E-01	2.708E+03	8.762E-03	17.84	34010298.85	17.027	17.680
0.356	3.970E-01	3.240E+03	1.254E-02	19.51	48689512.31	18.276	19.316
0.425	5.675E-01	4.630E+03	2.562E-02	23.33	99456397.56	20.566	23.023

Table: 4.4 Tube geometry and corresponding natural frequency in air

r_2(m)	r_3 (m)	Area (m^2)	Mass Mt (kg/m)	Moment of inertia I(m^4)	Theoretical result f(Hz)	Stiffness K(N/m^3)	2D-ANSYS result f(Hz)	3D-ANSYS result f(Hz)
0.1800	0.200	2.388E-02	1.948E+02	4.322E-04	14.77	1677371.13	14.62	14.63
0.2680	0.288	3.493E-02	2.851E+02	1.352E-03	21.59	5246418.88	20.67	21.24
0.3180	0.338	4.122E-02	3.363E+02	2.219E-03	25.47	8613745.07	23.49	24.92
0.3555	0.376	4.593E-02	3.748E+02	3.070E-03	28.38	11916647.91	25.18	27.63
0.4180	0.438	5.378E-02	4.389E+02	4.929E-03	33.23	19130896.17	27.02	32.06
0.4680	0.488	6.007E-02	4.901E+02	6.865E-03	37.11	26646533.33	27.62	35.49
0.4985	0.519	6.390E-02	5.214E+02	8.265E-03	39.48	32078037.39	27.65	37.54
0.5680	0.588	7.263E-02	5.927E+02	1.214E-02	44.87	47106461.51	27.02	42.06

4.2.1.2 Results

At this step, the shaft natural frequency in air (dry shaft) has been determined and validated with theoretical result as shown in Fig. 4.8. The result from theoretical, and 3D-ANSYS have shown good agreement. But, 2D lumped parameter spring-mass model, fails to simulate the natural frequency correctly as the mass of the object (shaft/tube) dominate the stiffness. This has been reflected specially in lumped parameter model of tube as the diameter increases as shown in Fig. 4.9. This problem could be clearly demonstrated in mode shape as shown in Figs. 4.10-4.12.

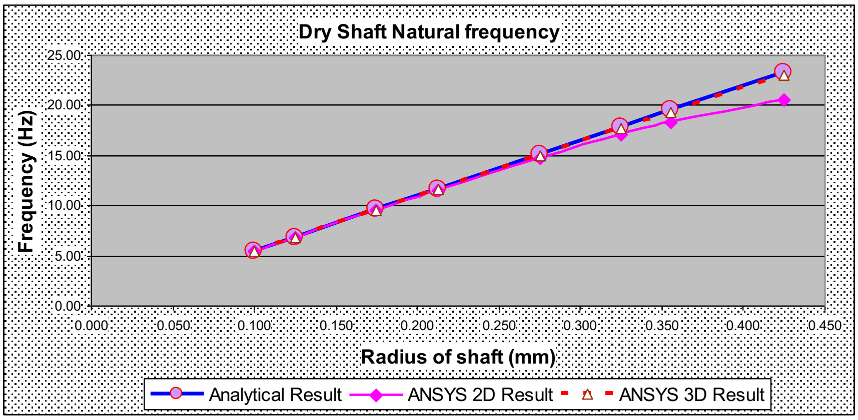


Figure 4.8 Shaft natural frequency in air with different radius (r_1)

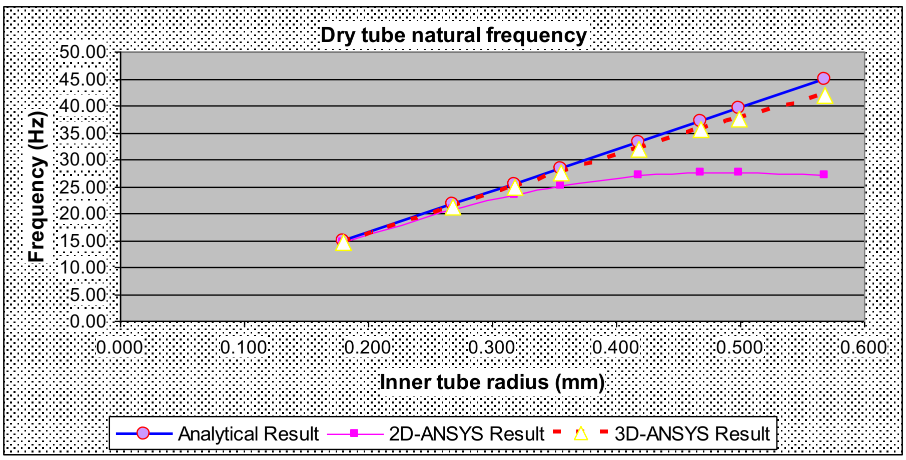


Figure 4.9 Tube natural frequency in air with different radius (r_2 and r_3)

Fig. 4.9 reveal that 2D lumped parameter model should be checked before using for further analysis as it fails to simulate properly as the mass of the component dominate the stiffness. This result shows also that, 2D model can be used for small diameter of shaft and tube.

As shown in Figs. 4.8-4.9, the natural frequency increases linearly as the diameter of the shaft/tube increases,

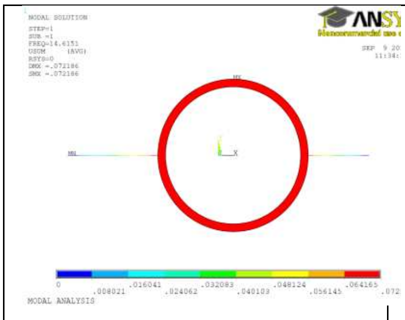
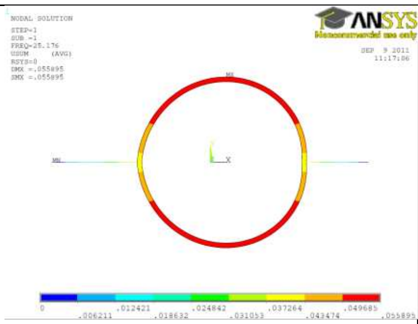
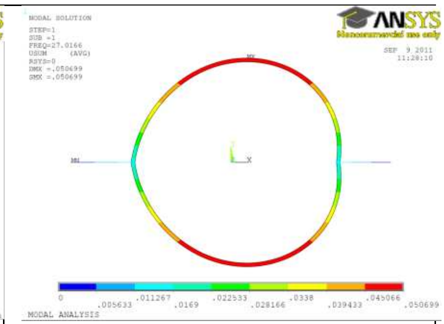
		

Figure 4.10 2D lumped parameter model of tube, mode shape and displacement for (r2=0.180)	Figure 4.11 2D lumped parameter model of tube, mode shape and displacement for (r2=0.3555)	Figure 4.12 2D lumped parameter model of tube, mode shape and displacement for (r2=0.568)

4.2.2 CASE-2 Bending Vibration of Solid Elastic Shaft in Infinite Fluid

In this case, simply supported beam has been placed in infinite fluid domain as shown in Fig. 4.13. The main target of this study is to determine natural frequency of wetted shaft, hydrodynamic mass (M_a), validation of ANSYS result with analytical result, identification of proper outer fluid extreme (r_4) to simulate infinite fluid domain, and identification of suitable mesh size. 2D and 3D finite element method has been implemented to analyze the vibration characteristics of the shaft as shown in Fig. 4.14.

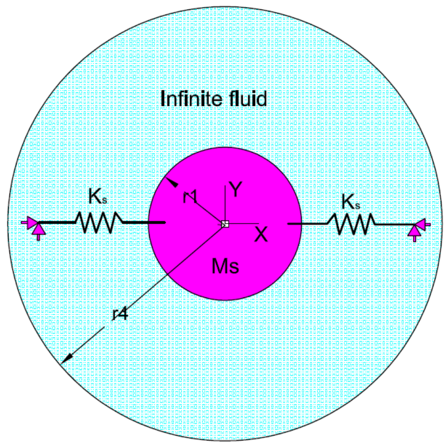


Figure 4.13 2D lumped parameter model of shaft in infinite fluid

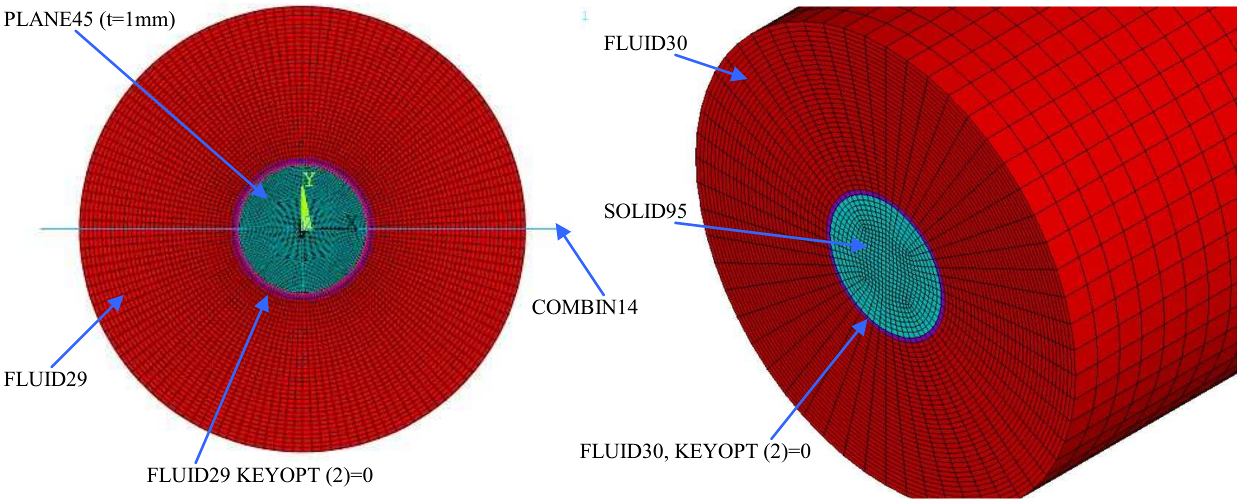


Figure 4.14 Finite element (2D and 3D) model of shaft inside infinite fluid

4.2.2.1 Determination of Outer Fluid Bound Radius for Infinite Fluid Simulation

As discussed in previous sections, finite element model of infinite geometry is impossible, so in this project, zero pressure has been defined at the outer boundary (r_4). For this study, the reference value of the shaft radius r_1=212.5mm has been taken and kept constant throughout the analysis and the outer bound extreme radius (r_4) has been kept changing to find the radius with minimum error as shown in Fig. 4.15. Ideally infinite radius leads to best result (minimum error), but there must be finite radius that could simulate infinite radius with minimum error, with zero pressure at the outer bound. The results obtain reveal that, 2 to 3 times of shaft radius could simulate infinite fluid, which match with ANSYS help recommendation. So, the value with strip in Table 4.5 (r_4=0.725) has been selected for further study or simulation of this part. The mesh size (division) of the model has been kept constant during this simulation.

Table: 4.5 Determination of outer bound radius (r_4) with small error

r_4(m) (mesh size)	Theoretical result f(Hz)	2D-ANSYS results f(Hz)	3D ANSYS results f(Hz)	%error in 2D	%error in 3D
0.2325 (4)	10.99	11.492	11.533	4.57	4.73
0.2625(8)	10.99	11.405	11.453	3.78	4.06
0.3225(12)	10.99	11.277	11.327	2.61	2.99
0.3825(16)	10.99	11.188	11.24	1.80	2.24
0.425(20)	10.99	11.141	11.194	1.38	1.83
0.525(24)	10.99	11.064	11.12	0.68	1.18
0.6375(28)	10.99	11.013	11.075	0.21	0.77
0.725(32)	10.99	10.987	11.045	0.02	0.50
0.825(36)	10.99	10.966	11.025	-0.22	0.32

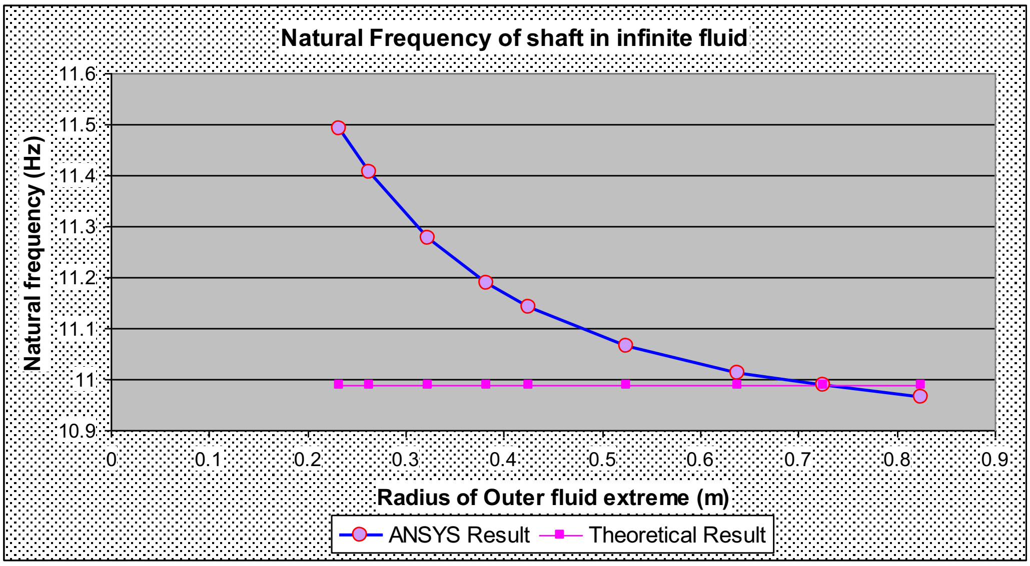


Figure 4.15 Natural frequency versus outer bound radius (r_4) of infinite fluid

4.2.2.2 Identification of Suitable Mesh Size

To properly simulate fluid structure interaction, the mesh size which compromise good result and computation time has to be determined properly, next to determination of outer bound radius of the fluid. In this case, the reference values (r_1 and r_4) kept constant, the mesh size has been varied as shown in Table 4.6 and the corresponding result has been compared with theoretical result.

Table: 4.6 Determination of proper mesh size for fluid part

Division	3D-ANSYS-result f(Hz)	Theoretical result f(Hz)	% error
4	11.204	10.99	1.950
8	11.062	10.99	0.658
16	11.017	10.99	0.249
36	10.987	10.99	0.024
64	10.978	10.99	0.106
90	10.970	10.99	0.179
112	10.960	10.99	0.270

Finally the radial division 36 has been selected considering the result and computation time as shown in Fig. 4.16.

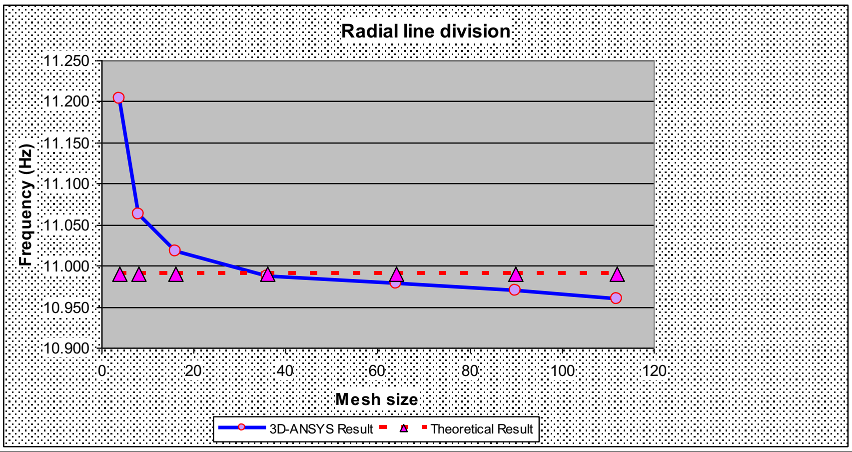


Figure 4.16 Frequency of shaft versus radial line division of fluid domain

4.2.2.3 Determination of Hydrodynamic Mass

Theoretically the hydrodynamic mass of circular shaft will be given by

$$M_a = \rho_f \pi r_1^2 \tag{4.3}$$

where ρ_f fluid density, and r_1 is radius of shaft.

The hydrodynamic mass of any cylindrical body can be written as

$$M_a = \rho_f C_m A \tag{4.4}$$

In which, A is the cross sectional area of the body ($A = \pi r_1^2$) and C_m is hydrodynamic mass coefficient of the shaft. Hydrodynamic mass coefficient for circular shaft is unity. In this section hydrodynamic mass (added mass) of circular shaft for different radius have been determined using both theoretical and 3D ANSYS and were compared. As shown in Fig. 4.17, the result from 3D ANSYS model has shown good agreement with theoretically calculated results.

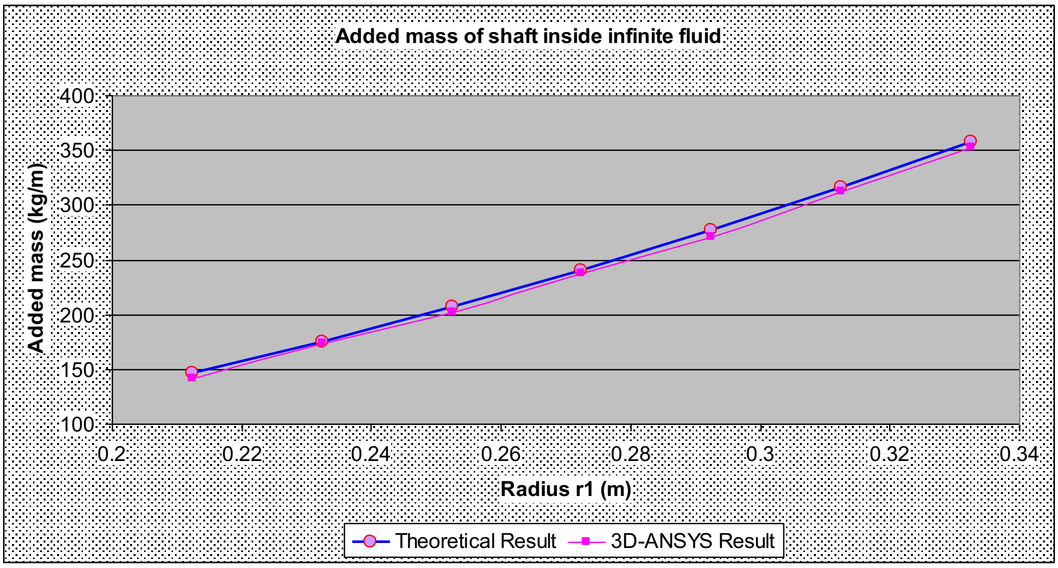


Figure 4.17 Hydrodynamic mass versus radius of shaft (r_1)

4.2.3 CASE-3 Bending Vibration of Solid Elastic Shaft in Fluid Filled Rigid Tube

The main objective of this case will be to perform modal analysis of solid elastic shaft in fluid filled rigid tube. The shaft is considered simply supported and the tube is considered rigid. The shaft natural frequency in fluid surrounded by rigid tube, the corresponding added mass, validation with available theoretical result, and parametric study will be the major task in this particular case.

4.2.3.1 Finite Element Model Development

Both 2D and 3D finite element model have been developed to determine the natural frequency of wetted shaft in ANSYS. The lumped parameter model has been developed in ANSYS as shown schematically in Fig. 4.18.

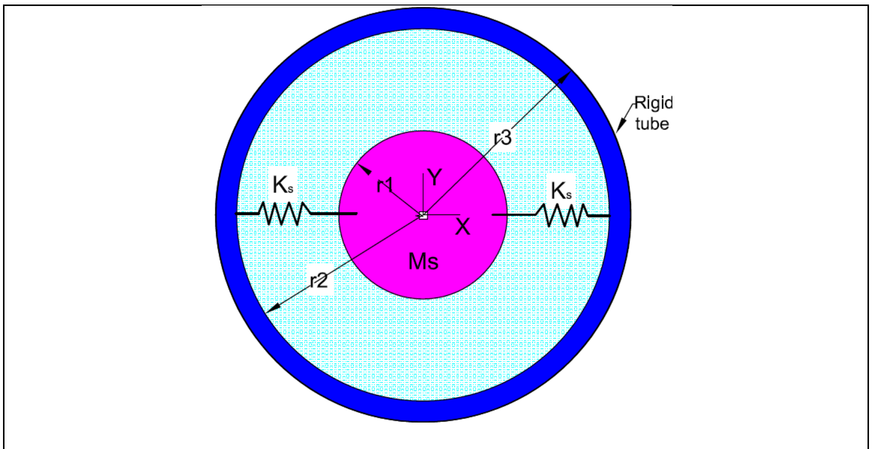


Figure 4.18 Schematic 2D model of shaft in fluid filled rigid tube

Using same principle shown in systematic drawing, 2D lumped parameter and 3D finite element have been developed as shown in Fig. 4.19.

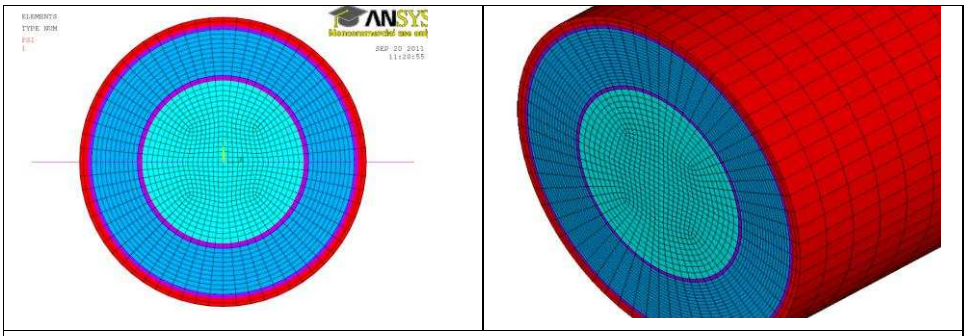


Figure 4.19 Finite element (2D and 3D) model of shaft inside infinite fluid

The added mass M_a of solid elastic shaft in fluid filled rigid tube could be determined using Eq. 4.4, where C_m is hydrodynamic coefficient of the shaft, and can be obtained from Fig.

4.20. The figure comprises radius ratio $\frac{r_2}{r_1}$ and βr_1 Where $\beta = \frac{2\pi}{\lambda_a}$, and where $\lambda_a = 2L$ for this specific vibration analysis (first bending mode shape).

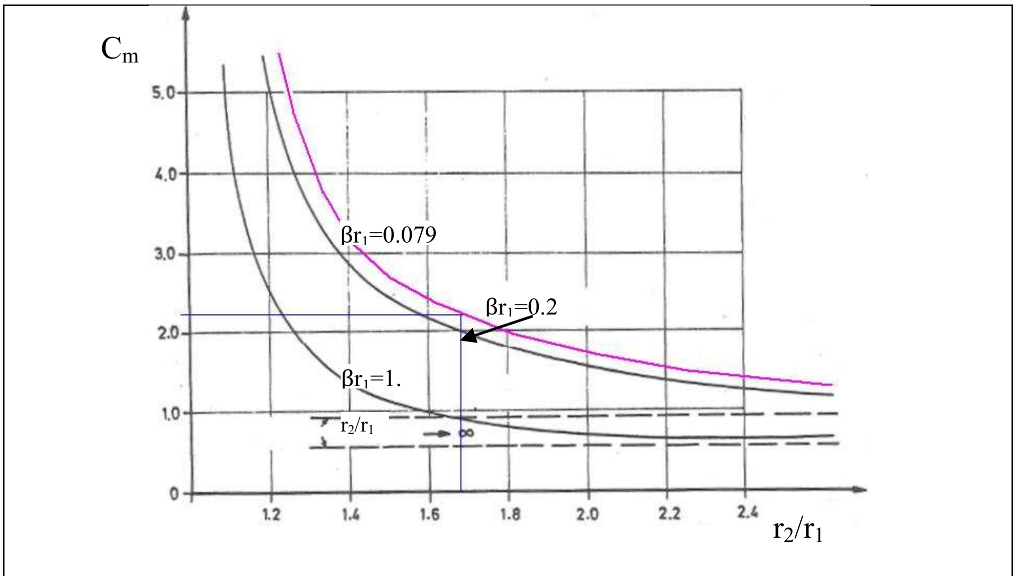


Figure 4.20 Hydrodynamic mass coefficient of rod/shaft in fluid filled rigid tube, adopted from [54]

Table: 4.7 Added mass as radius of shaft increases (technical data)

r_1	r_2	r_3	Shaft cross sectional area (m^2)	Shaft volume (m^3)	Moment of inertia (m^4)	Mass M_s(kg/m)	r_2/r_1	kr_1	C_m	Added mass M_a(kg/m)
0.2125	0.3555	0.3755	1.419E-01	1.206	1.601E-03	1157.60	1.67	0.079	2.1	306.85
0.2325	0.3555	0.3755	1.698E-01	1.443	2.295E-03	1385.75	1.53	0.086	2.6	454.79
0.2525	0.3555	0.3755	2.003E-01	1.703	3.193E-03	1634.42	1.41	0.093	2.9	598.28
0.2725	0.3555	0.3755	2.333E-01	1.983	4.331E-03	1903.59	1.30	0.101	3.7	889.04
0.2925	0.3555	0.3755	2.688E-01	2.285	5.749E-03	2193.27	1.22	0.108	4.5	1245.81
0.3125	0.3555	0.3755	3.068E-01	2.608	7.490E-03	2503.46	1.14	0.115	6.7	2117.20
0.3325	0.3555	0.3755	3.473E-01	2.952	9.600E-03	2834.15	1.07	0.123	11.8	4221.36

The natural frequency of shaft decreases because of added mass and can be determined using the following expression,

$$f_m = \frac{1}{2\pi}\sqrt{\frac{K_s}{M_s + M_a}} \qquad (4.5)$$

where,

f_m - is frequency of shaft in fluid filled rigid tube,

K_s - Stiffness of shaft

M_s -mass of shaft

M_a -hydrodynamic mass (added mass)

4.2.3.2 Parametric Study

To identify the effect of geometric parameters change on added mass, parametric study has been performed taking the reference value (r_1=0.2125m, r_2=0.3555m, r_3=0.3755m) as an initial. To see the effect clearly, two separate studies has been carried out. The first study kept tube geometry (r_2, r_3) constant and the shaft radius increased as a consequence the gap between the shaft and tube decrease step by step as shown in Table 4.7. The second study attempts to study by decreasing tube radius step by step while the shaft radius remains same. For both cases finite element modal analysis were performed and the corresponding natural frequency were registered as shown in Figs. 4.21-4.22.

These two different studies were very important to identify that, natural frequency does not only depend on the ratio of radius (r_2/r_1) but also on the absolute values of the both radii. The other very important finding were, changing shaft radius have more influence on frequency magnitude compared to change in tube radius as shown in Figs. 4.21-4.22.

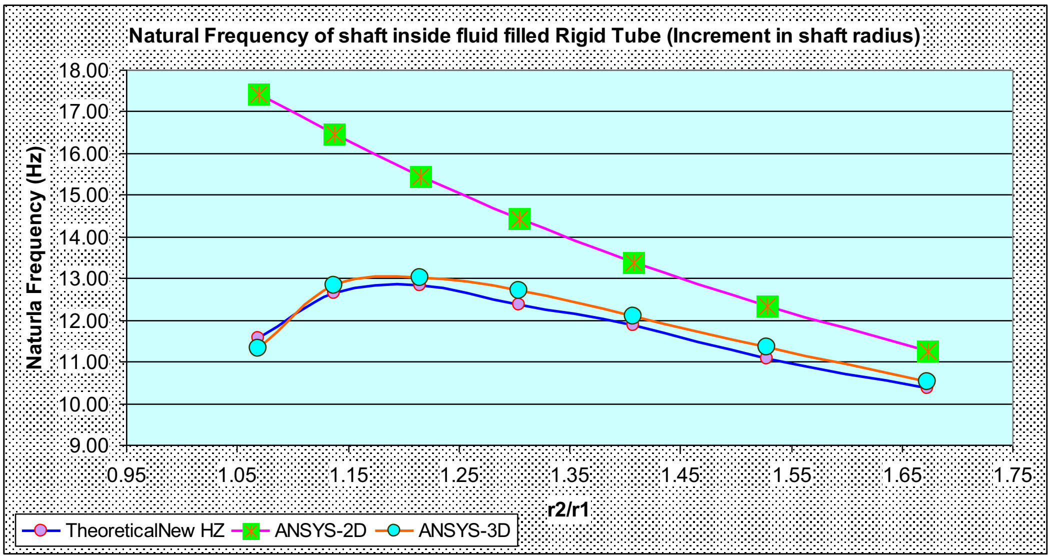


Figure 4.21 Natural frequency of shaft inside fluid filled rigid tube as shaft radius increases

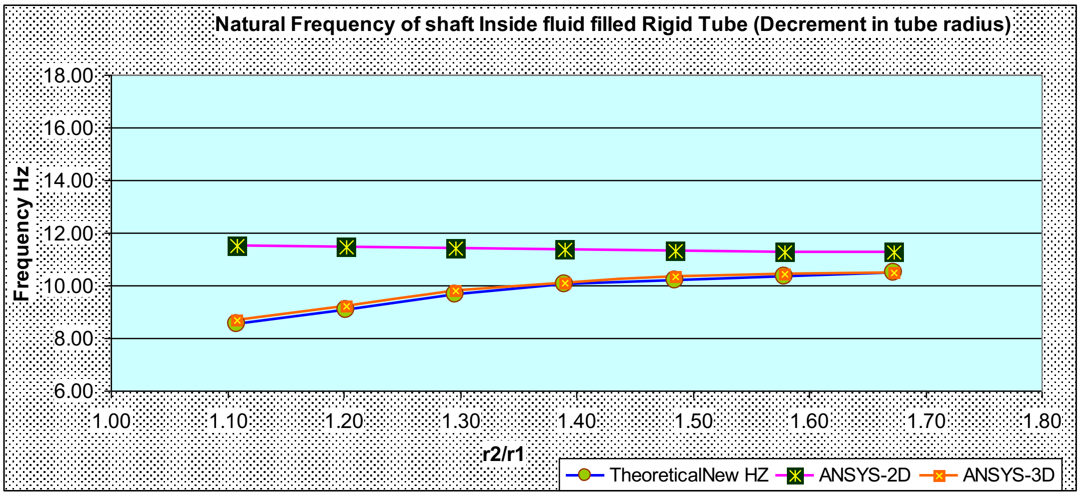


Figure 4.22 Natural frequency of shaft inside fluid filled rigid tube as tube radius decreases

In both cases, the result from ANSYS-2D, ANSYS-3D and theoretically calculated wetted frequency (new) have been compared.

4.2.3.3 Determination of Hydrodynamics Mass Coefficient

Hydrodynamic mass coefficients for both parametric studies (decrement of tube radius, and increment of shaft radius) were calculated theoretically and with ANSYS 3D model were compared. The result shows good agreement in both cases, as shown in Figs. 4.23-4.24.

The theoretical hydrodynamic mass coefficient was calculated using the expression derived from Eq. 4.4, $C_m = \frac{M_a}{M_{sd}}$, where M_{sd} is mass of volume displaced.

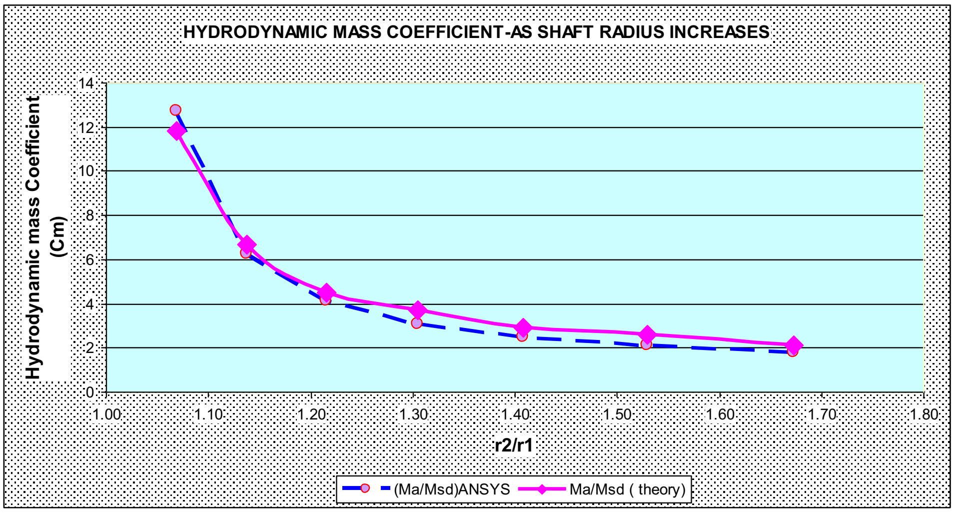


Figure 4.23 Hydrodynamic mass coefficient of shaft versus r2/r1 as shaft radius increases (case-3)

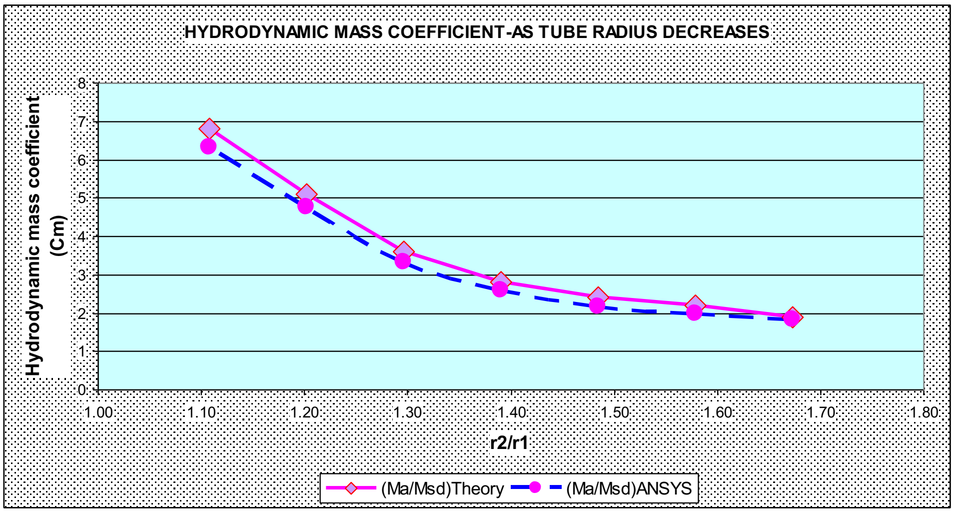


Figure 4.24 Hydrodynamic mass coefficient of shaft versus r2/r1 as tube radius decreases (case-3)

4.2.4 CASE-4 Bending Vibration of Solid Elastic Shaft in Fluid Filled Elastic Flexible Tube Immersed in Infinite Fluid

For all previous CASEs (1, 2 and 3) analytical formulae exists to determine the hydrodynamic mass. The primary purpose of re-analyzing these cases were to check whether vibro-acoustic fluid structure interaction works properly for this kind of analysis or not! As discussed in chapter 2, acoustic fluid structure interaction vibration analysis, have meaningful advantage interms of simplicity and lesser computation time especially for modal analysis. Another most important aim in this project was to extend the same trend of FEM model used in later cases for more complicated fluid structure interaction problem like bending vibration of solid elastic shaft in fluid filled elastic flexible tube surrounded by infinite fluid (e.g stern tube). As there is no theoretical formulation for this specific problem, to our knowledge. Therefore, same modeling techniques used in previous cases were extended for CASE-4.

4.2.4.1 Finite Element Model Development

Both 3D and 2D finite element model have been developed to determine shaft and tube natural frequency and the corresponding added mass in such arrangement. As described above same modeling technique has been used for this case.

The following basic steps have been followed to achieve the final target,

- The natural frequency of dry shaft and tube has been determined using Eq. 4.1.
- Using the determined natural frequency, the stiffness (K_s for shaft, and K_t for tube) have been calculated using Eq. 4.2b.

- Using PLANE45, with thickness 1mm, 2D, 1DOF lumped parameter model has been developed as shown in Fig. 4.27.
- 3D finite element model was developed assuming the arrangement simply supported at the two ends as shown in Fig. 4.28.
- Then modal analysis over a frequency range (0 to 200Hz) has been analyzed and the first mode was registered for each run.
- The added mass was determined from wetted frequency of shaft and tube, and the corresponding hydrodynamic mass coefficient was calculated using Eq. 4.4.

Similar analysis to CASE-1 was performed to determine the shaft and tube natural frequency in air. 3D ANSYS model and theoretical formulation used in previous cases has been applied to determined natural frequency of dry shaft and tube, as shown in Figs. 4.25-4.26.

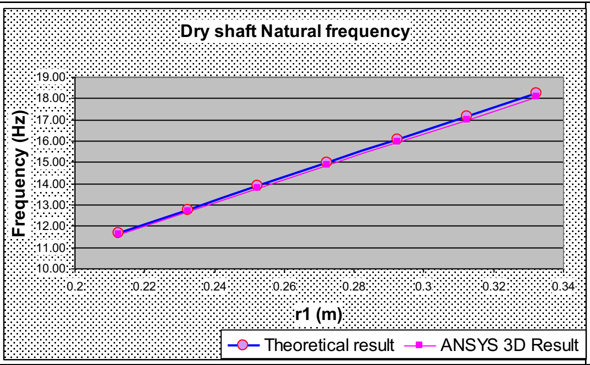


Figure 4.25 Bending natural frequency of dry shaft

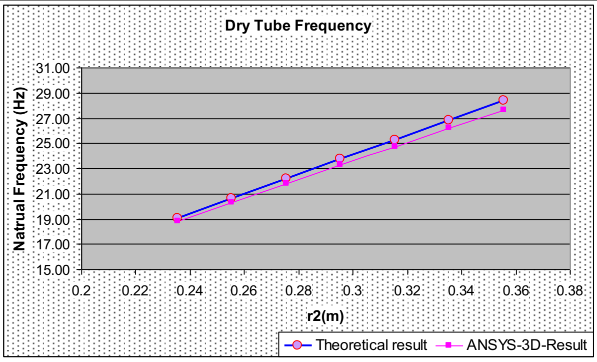


Figure 4.26 Bending natural frequency of dry tube

Once the natural frequency has been known, the corresponding stiffness of shaft as well as tube can be determined, as described above.

Therefore 2D finite element model can be implemented for further investigation as systematically shown in Fig. 4.27.

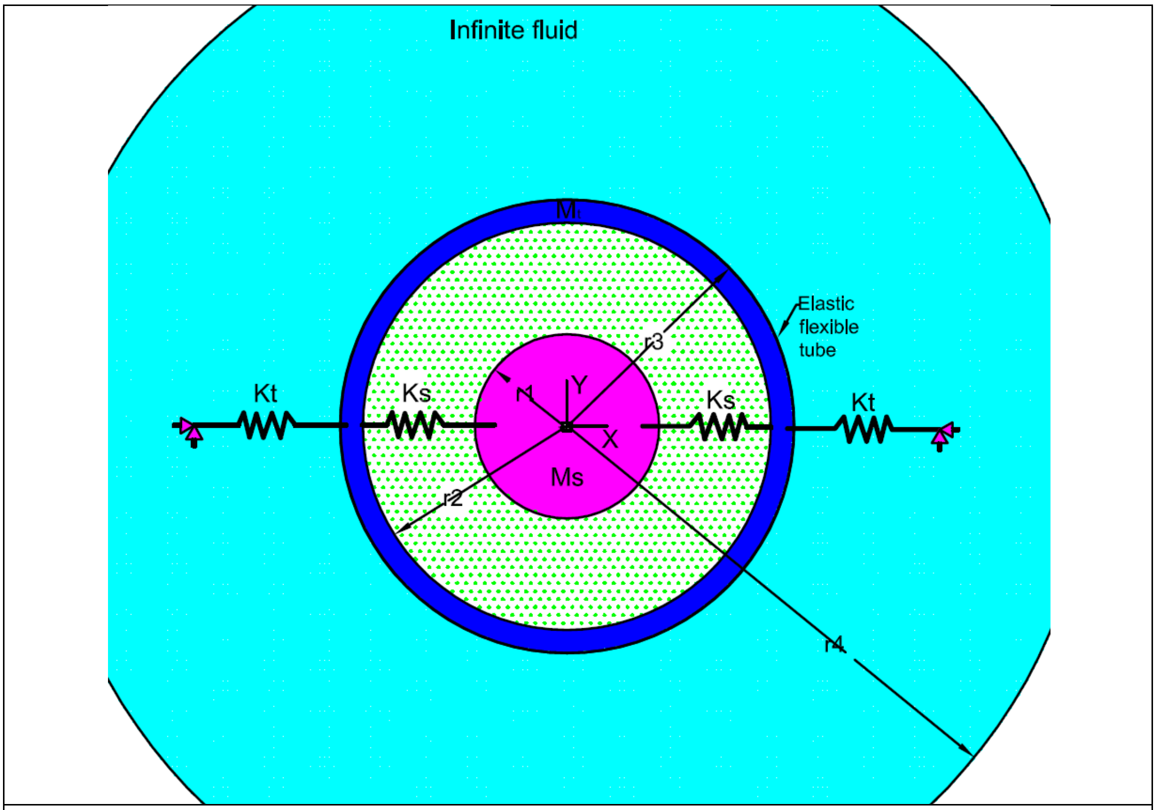


Figure 4.27 Schematic drawing of solid elastic shaft in fluid filled elastic flexible tube immersed in infinite fluid

Detail analysis of CASE-4 will be investigated using 2D and 3D finite element model as shown in Fig. 4.28. The same material property and element type used in previous studies such as (CASE-1, 2, and 3) for both 2D and 3D model will be used for this case as well.

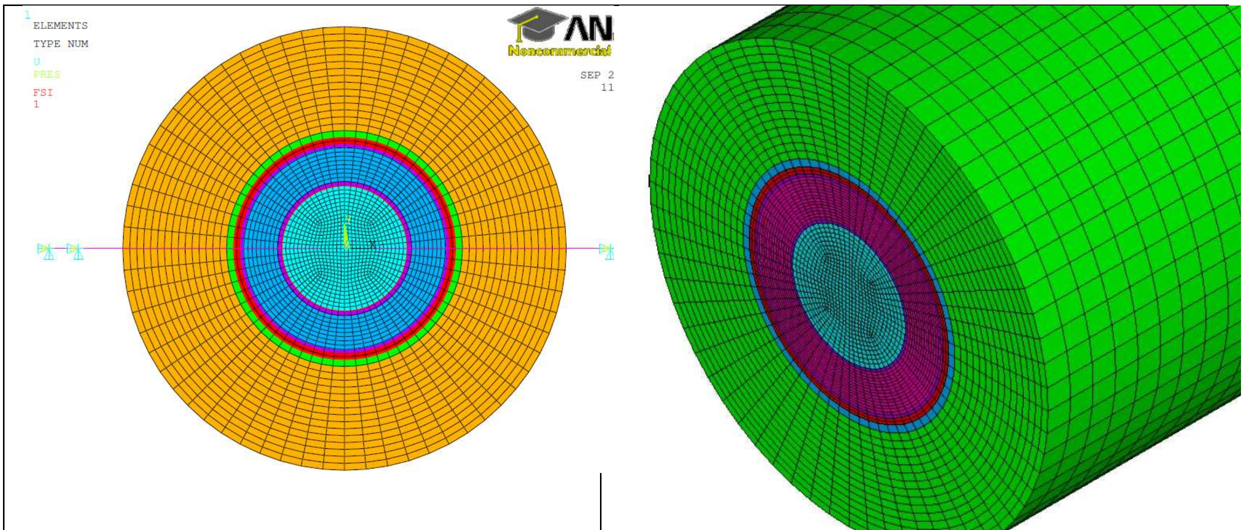


Figure 4.28 2D and 3D finite element model of CASE-4

4.2.4.2 Parametric Study

Parametric study has been performed to investigate the effect of geometric change on wetted natural frequency of shaft and tube. The studies related to the gap between shaft and tube have been made in two different ways, like in CASE-3. Ship stern tube real value obtained from an existing design of a navy supply vessel have been taken has a reference value (r_1=212.5mm, r_2=355.5mm, r_3=375.5mm and ofcourse the estimated r_4) for the study. Based on these reference values, two parametric studies have been investigated.

The first study aims to examine the outcome of modal vibration characteristic and added mass, as the gap between shaft and tube decreases by increasing the shaft radius (r_1) keeping the tube geometry unchanged. The results obtained divulge that the natural frequency of shaft decrease as the gap decreases by increasing r_1 as shown in Fig. 4.29, which is ofcourse resulted from increment of added mass as the gap decreases. The corresponding hydrodynamic mass coefficient were calculated and drawn as a function of tube inner radius and shaft radius (r_2/r_1) as shown in Figs. 4.31-4.32. The result shows that, hydrodynamic mass coefficient of shaft decreases as the radius ratio (r_2/r_1) increases and increases for tube as (r_2/r_1) increases. There are also significant change in frequency of tube as the gap decreases, the result reveal that natural frequency of tube increases a little bit as the gap increase as shown in Fig. 4.30, but in general much reduction because of fluid around.

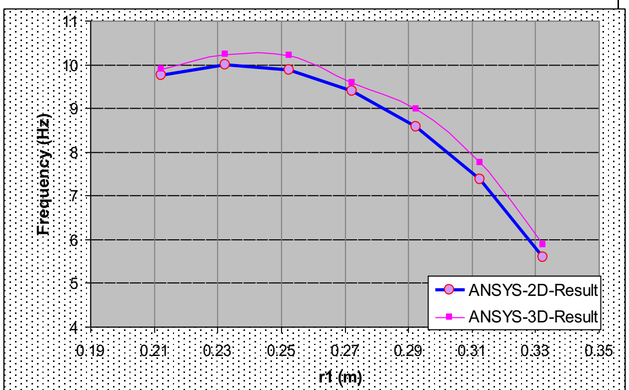


Figure 4.29 Natural frequency of shaft for CASE-4 as radius of shaft increases

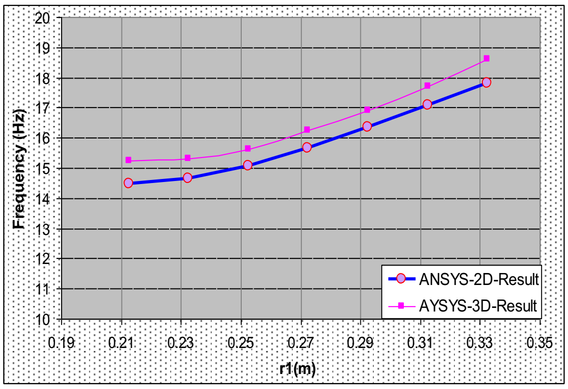


Figure 4.30 Natural frequency of tube for CASE-4 as radius of shaft increases

The hydrodynamic mass coefficient of tube also increased a little bit as the gap decrease to some extent and goes down as shown in Fig. 4.32.

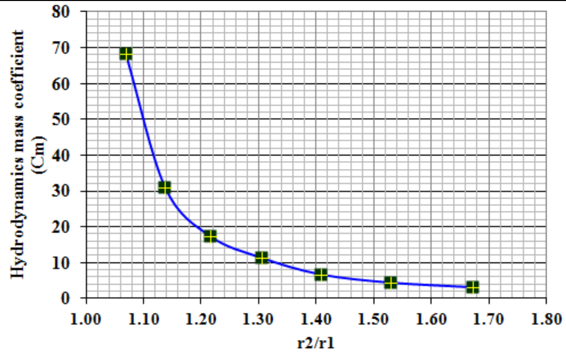


Figure 4.31 Hydrodynamic mass coefficient of shaft for CASE-4 as radius of shaft increases (ANSYS-3D-Result)

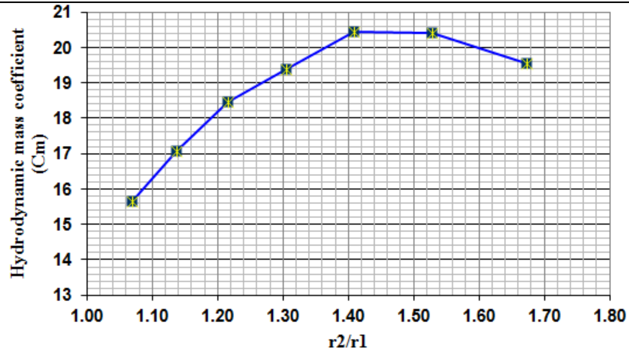


Figure 4.32 Hydrodynamic mass coefficient of tube for CASE-4 as radius of shaft increases (ANSYS-3D-Result)

The second study was conducted in reverse way, the tube geometry (r_2, r_3) reduced step by step while the shaft radius kept constant. For each decrement of tube radius, the modal analysis has been performed and the subsequent frequencies have been registered for first mode of vibration as shown in Figs. 4.33-4.34. Using the frequencies obtained from ANSYS 2D and 3D model, the added masses for each case was calculated. To express the hydrodynamic mass as a non dimensional coefficient, the hydrodynamic mass coefficient (C_m) were calculated and drawn with radius ratio (r_2/r_1) as shown in Figs. 4.35-4.36. the graphs from study one and two behaves the same, but big difference in magnitude.

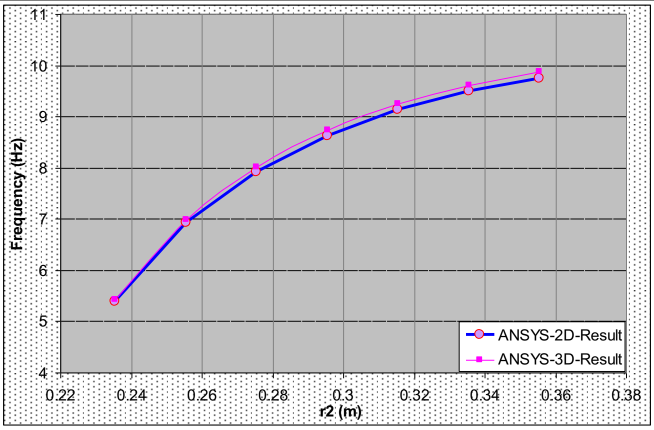


Figure 4.33 Natural frequency of shaft for CASE-4 as radius of tube decreases

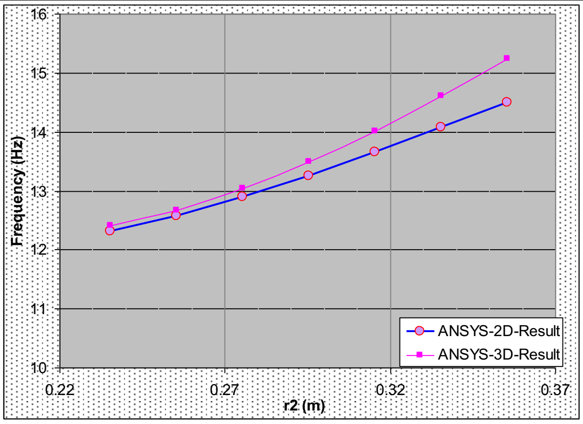


Figure 4.34 Natural frequency of tube for CASE-4 as radius of tube decreases

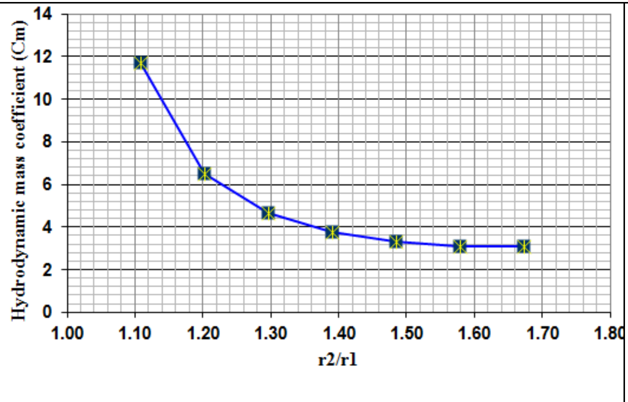


Figure 4.35 Hydrodynamic mass coefficient of shaft for CASE-4 as radius of tube decreases (ANSYS-3D-Result)

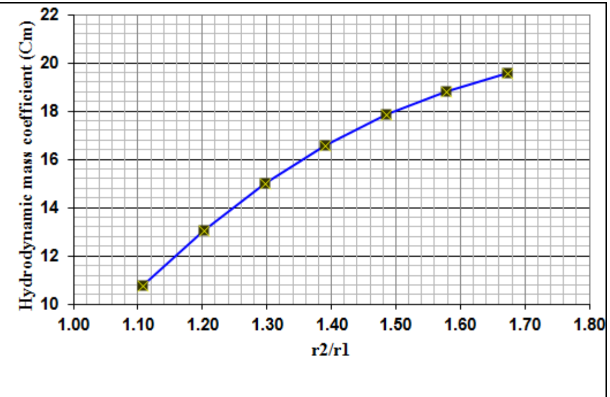


Figure 4.36 Hydrodynamic mass coefficient of tube for CASE-4 as radius of tube decreases (ANSYS-3D-Result)

4.2.4.3 Mode Shape, Displacement and Pressure Distribution

As described in previous sections, the structural element and the fluid element at the vicinity of the structure have displacement degree of freedom. So, the displacement of the shaft and the tube under wetted natural frequency behaves as shown in Figs. 4.37-4.38 respectively. It is clearly visible that the maximum displacement in shaft or tube occurs at their respective wetted natural frequencies.

From theoretical point of view, the sound pressure wave propagates through fluid with exponential function. So, it will decrease as it goes away from the source. The sound pressure has the following distribution as shown in Figs. 4.39-4.40.

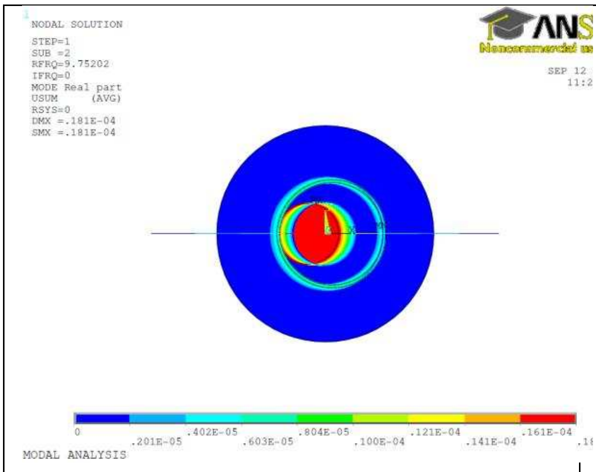


Figure 4.37 First mode shape and the respective displacement of coupled system (CASE-4)-shaft resonance

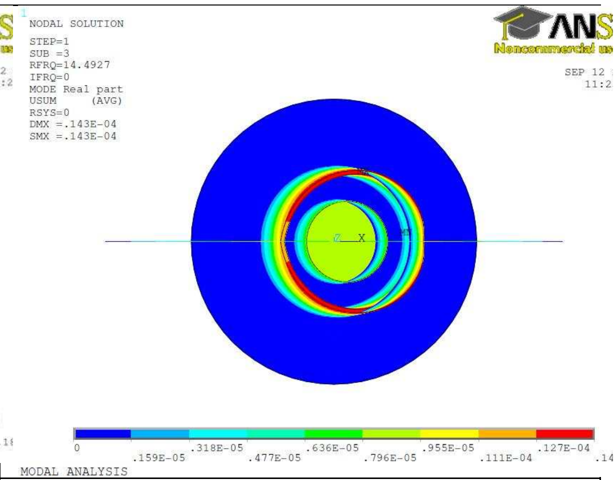


Figure 4.38 First mode shape and the respective displacement of coupled system (CASE-4)- tube resonance

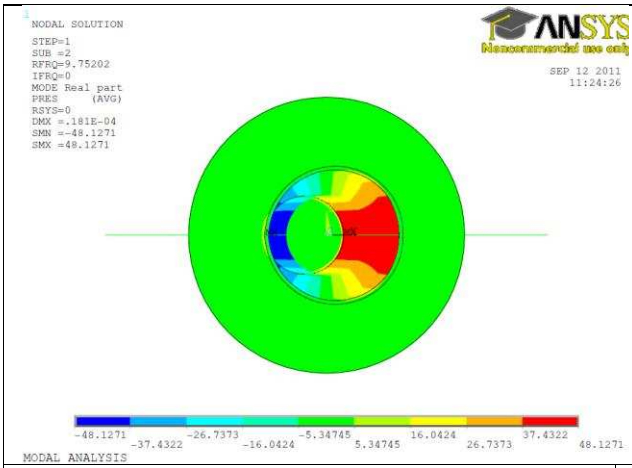
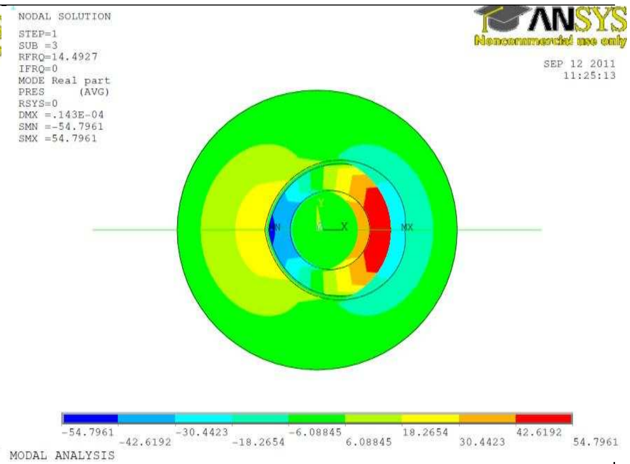
	

Figure 4.39 First mode shape and the respective pressure distribution of coupled system (CASE-4)- shaft resonance	Figure 4.40 First mode shape and the respective pressure distribution of coupled system (CASE-4)- tube resonance

4.2.5 Comparison of Different CASES

CASE-2, 3 and 4 have compared with regards to percentage decrement in natural frequency of shaft as shown in the following table 4.8. In other words natural frequency of dry shaft and wetted shaft have been compared to see the effect coming from fluid (added mass).

Table: 4.8 Percentage decrement of frequency for case-2, 3 and 4

r1(m)	CASE-1	CASE-2	CASE-3	CASE-4	% decr. for CASE-2	% decr. for-CASE-3	% decr. for-CASE-4
0.2125	11.60	11.01	10.51	9.89	5.08	9.37	14.77
0.2325	12.69	12.03	11.35	10.22	5.16	10.56	19.42
0.2525	13.77	13.08	12.09	10.20	5.03	12.20	25.94
0.2725	14.85	14.10	12.69	9.58	5.03	14.58	35.46
0.2925	15.93	15.14	13.01	8.97	4.93	18.34	43.67
0.3125	17.01	16.17	12.82	7.74	4.92	24.64	54.48
0.3325	18.08	17.21	11.30	5.89	4.83	37.51	67.45

Very big percentage decrement has been registered for CASE-4, which even exceeded 50% as the gap between shaft and the tube decreases. CASE-3 also has a big effect on shaft frequency, but seems to have 50% less effect as compared to CASE-4. CASE-2 found to have less effect as compared to CASE-3 and CASE-4 as shown in Fig. 4.41.

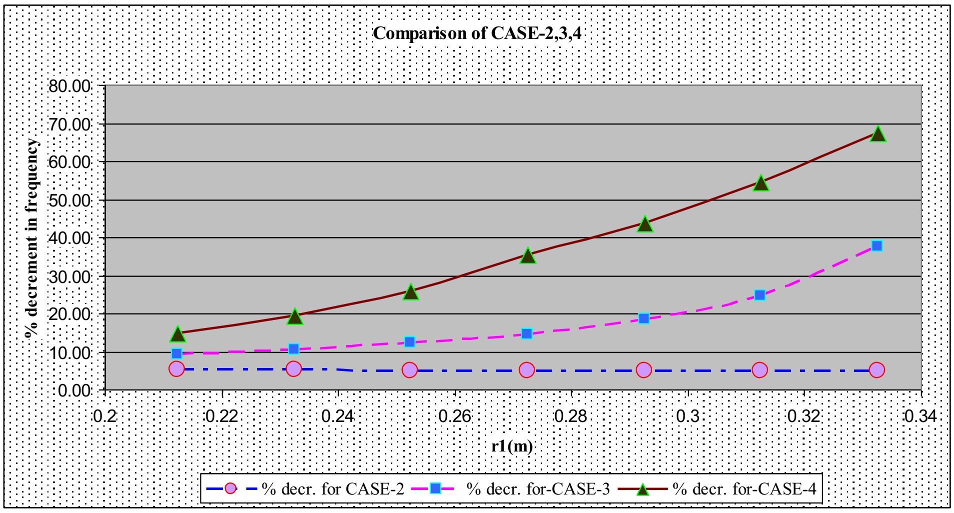


Figure 4.41 Percentage decrement in frequency with different cases as r_1 increases.

Likewise, the hydrodynamic mass coefficients of shaft have been also compared for CASE-2, CASE-3 and CASE-4 as shown in Fig. 4.42. The result reveal that, hydrodynamic mass coefficient increases has the gap between the shaft and the tube decreases, special in CASE-4 it shows radical change as compared to other cases. Here it could be concluded that CASE-4 has big effect interms of hydrodynamics mass effect on the structure.

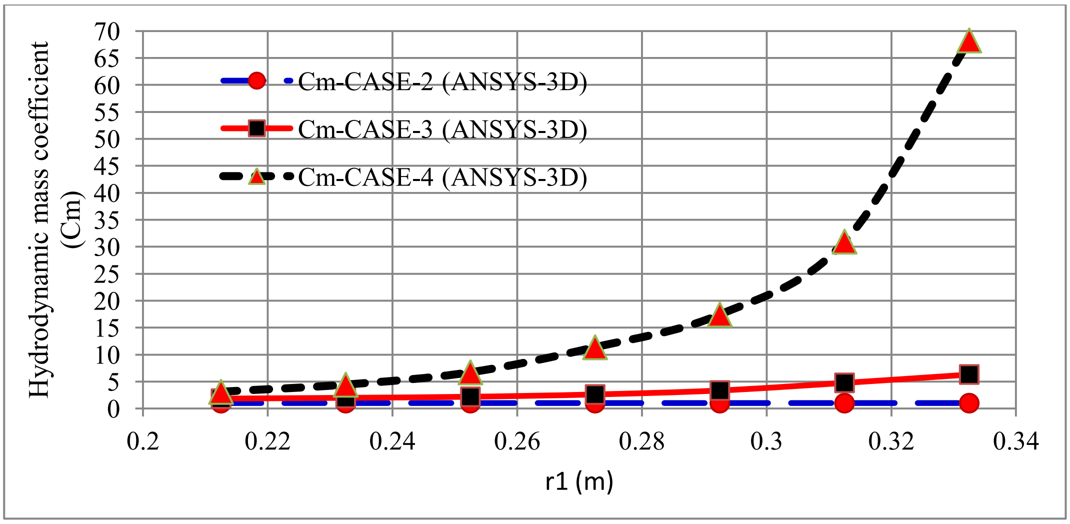


Figure 4.42 Hydrodynamic mass coefficient of shaft for different cases (ANSYS-3D result)

4.2.6 Comparison of Percentage Decrement in Shaft and Tube Natural Frequency

Percentage decrement in natural frequency of shaft and tube for CASE-4 has been compared as shown in Figs. 4.43-4.44.

Fig. 4. 43 shows that percentage decrement in shaft natural frequency increase as the gap between shaft and tube decrease by increasing shaft radius and the percentage decrement of tube behaves the reverse for same case.

Fig. 4.44 demonstrate that percentage decrement in shaft increase as gap between shaft and tube decreases by decreasing tube radius, but tube natural frequency decrease with small magnitude.

For this comparison, it could be examined that shaft natural frequency is more sensitive to parametric change than tube natural frequency. In addition, change in shaft radius has more effect on change in percentage decrement of tube natural frequency as compared to change in tube geometry.

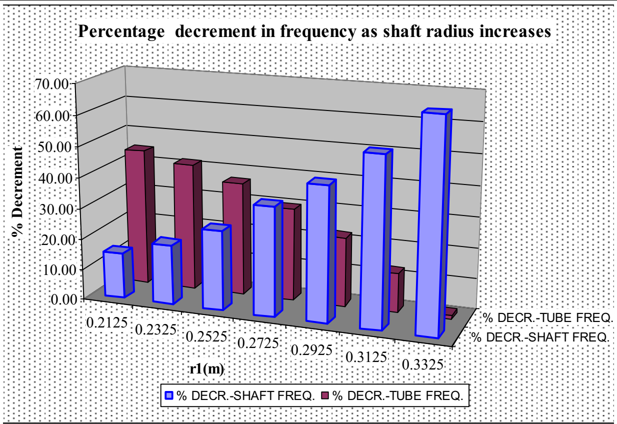


Figure 4.43 Percentage decrement of frequency as r_1 increases

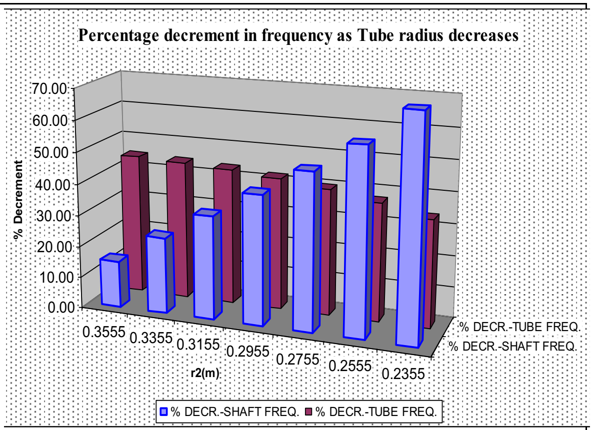


Figure 4.44 Percentage decrement of frequency as r_2 decreases.

4.2.6 *Effect of Fluid Density*

Density have significant effect in fluid structure interaction problems, with regards to acoustic fluid structure interaction analysis density is one of the governing material property for finite element analysis in addition to sound speed which intern depends on density. Therefore, in this section the effect of the fluid between the shaft for CASE-3 and CASE-4 has been studied considering different lubricating oil as shown in Table 4.9.

Table: 4.9 Lubrication oil characteristics

Lubricating oil	name	Density(kg/m3)	Speed of sound (m/s)	Bulk modulus (GPa)
Polyalphaolefin oil	PAO32	827	1451	1.74
paraffinic mineral oil	PBSN	855.2	1565	2.15
Traction oil	CVTF	957.9	1493	2.11
Water	Sea Water	1030	1460	2.196
Phosphate ester	Phosphate ester	1176.6	1361	1.66
Glycerol	Glycerol	1263.1	1872	4.4
perfluoropolyether oil(PEPE)	Krytox	1910	761	1.09

Finite element modal analysis has been performed to determine the natural frequency and corresponding added mass for each fluid. For CASE-4 only the fluid between the shaft and tube is changed while the other remains sea water. Obviously, the frequency decreases and added mass increases as fluid density increases as shown Fig. 4.45. Because, as the fluid density increases, the force required to deform the fluid element increases, as a result the pressure force which is the function of fluid density, comes from the fluid on moving structure increases.

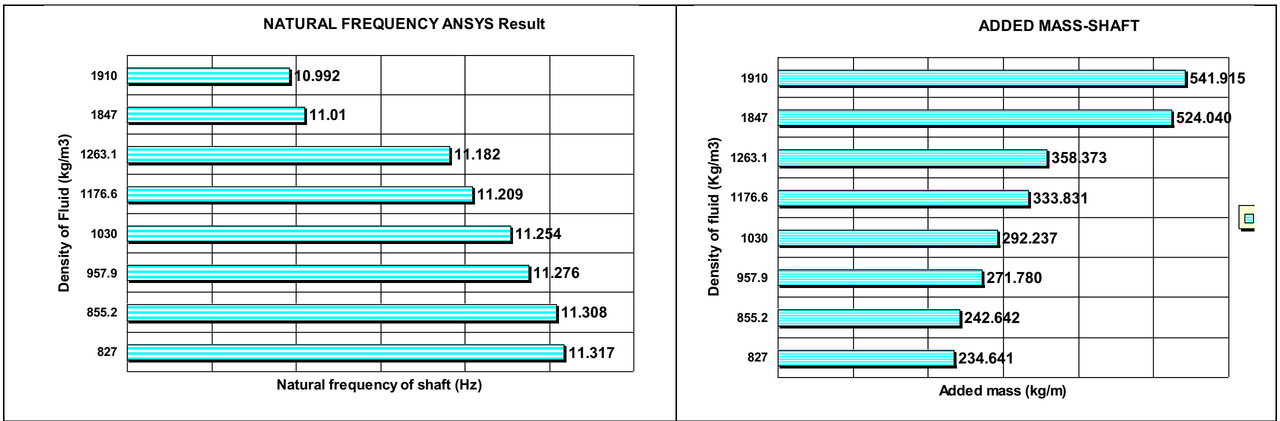


Figure 4.45 Natural frequency and added mass of CASE-3 under different density of fluid

For both case the reference value geometry has been used. The effect of fluid density on tube and shaft has been analyzed for CASE-4 as shown in Fig. 4.46 for shaft and Fig. 4.47 for tube.

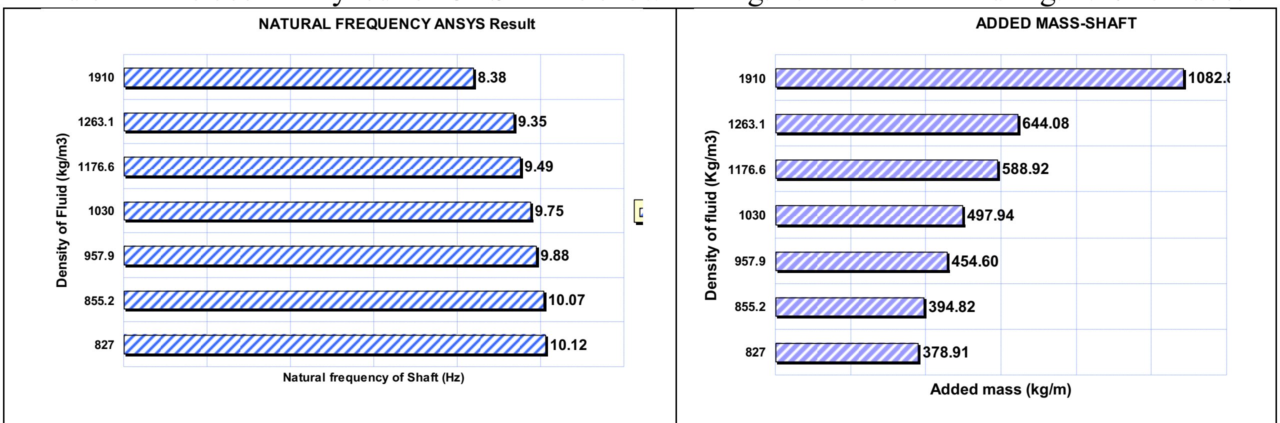


Figure 4.46 Natural frequency and added mass of CASE-4 (Shaft) under different density of fluid

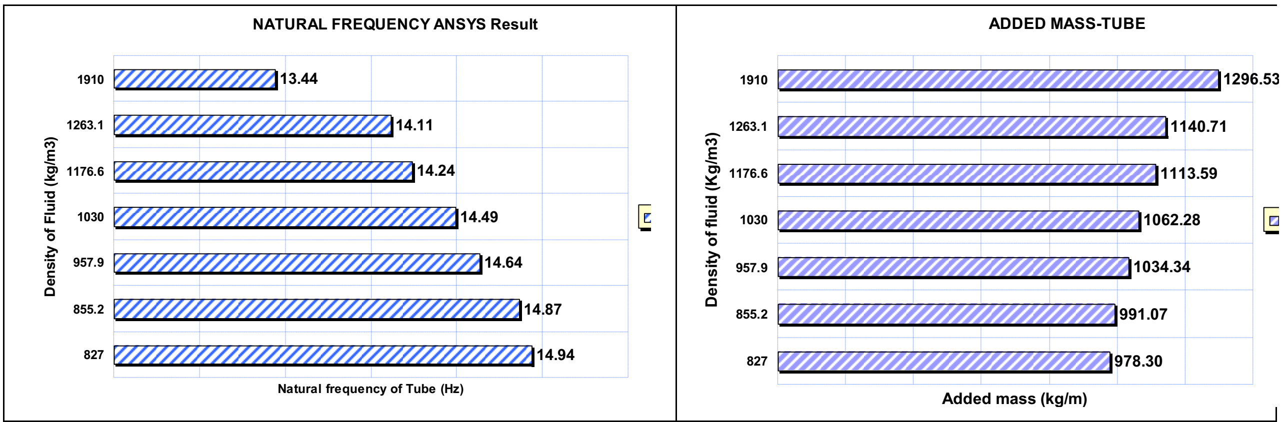


Figure 4.47 Natural frequency and added mass of CASE-4 (Tube) under different density of fluid

4.2.7 Harmonic Analysis

Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time. The idea is to calculate the structure's response at several frequencies and obtain a graph of some response quantity (usually displacements) versus frequency. Peak responses are then identified on the graph and stresses reviewed at those peak frequencies. Harmonic response analysis gives the ability to predict the sustained dynamic behavior of the structures, thus enable to verify whether or not the designs will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations. While, modal analysis used to determine the natural frequencies and mode shapes of a structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions.

This section attempts to perform harmonic finite element analyses of CASE-2 and CASE-4 using ANSYS 3D models. The propeller shaft is mainly excited with the forces induced from the propeller. To roughly imitate similar condition that could happen in shaft line, a F_h=2kN harmonic force has been applied at the center of simply supported shaft as shown in Fig. 4.48.

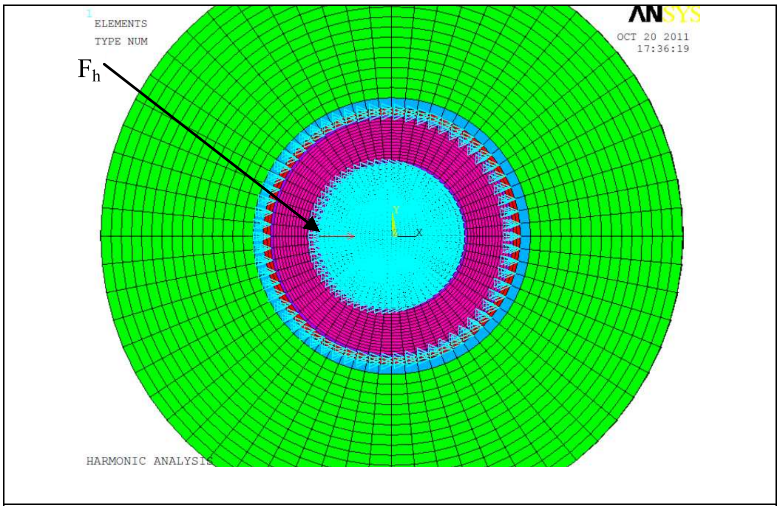


Figure 4.48 FEM of CASE-4 under harmonic force (F_h)

For the shaft vibrating inside infinite fluid (CASE-2), the displacement (DISP) versus frequency response of harmonic force has been plotted as shown in Fig. 4.49 and the result reveal that the first mode peak natural frequency occurs at the same value determined using modal analysis.

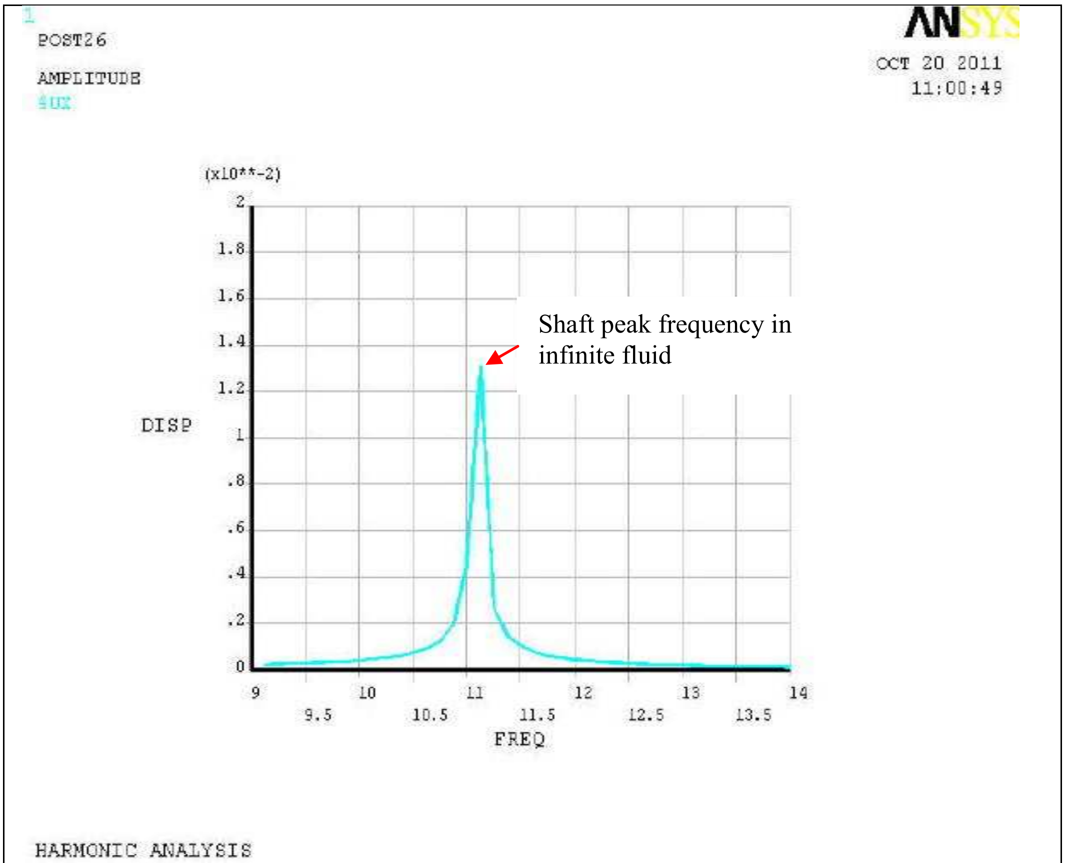


Figure 4.49 Harmonic response of CASE-2

Likewise, using same magnitude of force at the same position, harmonic analysis of CASE-4 has been also performed using finite element method (ANSYS). Fig. 4.50, illustrate the harmonic response of solid elastic shaft in fluid filled elastic flexible tube immersed in infinite fluid (CASE-4). Like in harmonic response of shaft in infinite fluid (CASE-2), the response obtained in this case has similar peak value (frequency) with modal analysis. Here one important note is that, since it is a coupled system, both shaft and tube vibrate together, and the response shows that there are peaks at both natural frequency of shaft and tube. But it is clearly visible that, shaft and tube resonance are at different places.

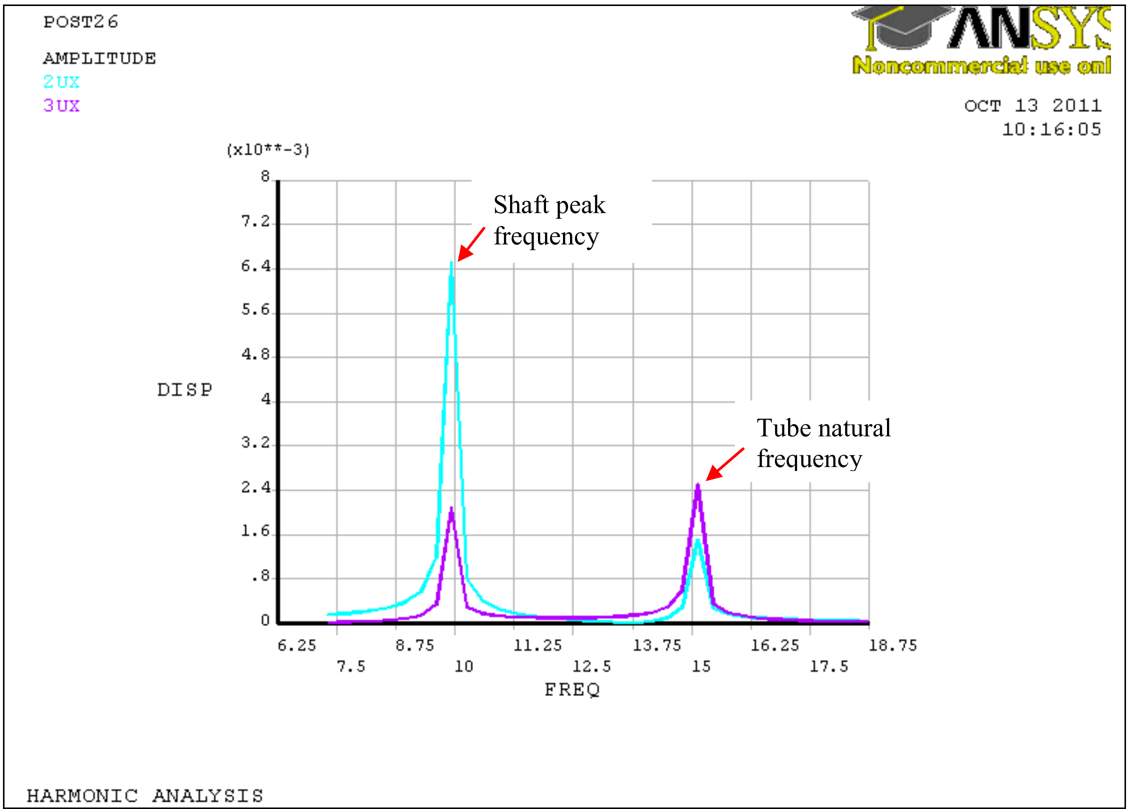
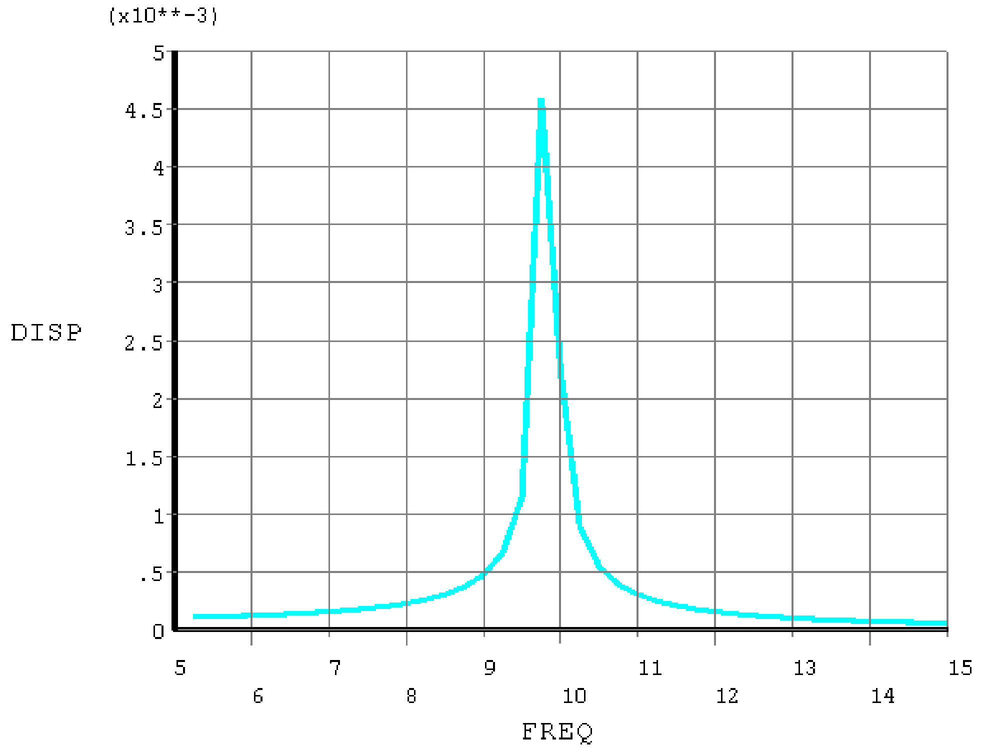


Figure 4.50 Harmonic response of CASE-4, green-for shaft, violet-for tube

It reveal that vibration of shaft transmit through flow element cause the tube to vibrate at resonance frequency of shaft, likewise, the shaft vibrated because of tube resonance as shown in Fig. 4.50.

4.2.8 Harmonic Analysis for increased density

In this analysis the added mass for reference value has been considered to perform macro analysis. The added mass obtained from CASE-4 (shaft immersed in fluid filled tabular space surrounded by infinite fluid) has been added to the original mass of the shaft, to analyse dry structural shaft element using finite element method (ANSYS). The density of the shaft has been increased keeping the shape and volume of the shaft same. This analysis is very essential in saving the compuational time and space of forced anaysis of similar work after determing the added mass using modal analysis. The system has been undamped and all the calculation has been employed neglecting any damping effect which resulted in sensitivity of displacement magnitude on resonance frequency.



4.51 Harmonic anaysis of shaft with increased density for CASE-4

4.3 PART-2:

In this part of the project, finite element bending vibration analysis of seawater overboard discharge (OVBD) system will be performed using FEM to determine the effect of surrounding fluid on natural frequency of pipe and caisson. Consequently, the corresponding added mass of the pipe and caisson will be estimated based on the computation of natural frequency. This part has vertical pipe and caisson arrangement with partially filled fluid in which it is quite different from PART-1.

4.3.1 Description of the Problem

As shown in schematic drawing Fig. 4.52, the system consists of inner pipe and caisson structural element arranged vertically and have the geometrical and material properties as shown in Table 4.10. The inner pipe is used to transport seawater, and filled with sea water and surrounded by seawater till the draft of the ship/structure and concentric with caisson.
The caisson which is made up of steel is surrounded by ballast water compartment and rigidly supported at four positions(base line, deck-3, deck-2 and main deck). But the pipe only supported at the bottom base line and main deck only. The gap between caisson and pipe is open to sea water and the height depends on the draft of floating structure. More over the caisson is surrounded by ballast water, which depend on loading condition of the floater. The side hull 2.6m far from the caisson. In vibration finite element analysis the wave reflection from hull wall has been neglected, assuming very small displacement of overboard discharge system as consequence small wave and small deflection which will not much affect the vibration characteristic of the OVBD system. So, it is considered as infinite fluid.

This seawater overboard discharge system has been damaged under fatigue loading resulted from vibration as presented from the client to Germanischer LIoyd. The long run study will be to know the source of vibration leading to this damage of the system. As there is permanent flow in a pipe, the vibration might be a flow-induced vibration. In solving this kind of real problem, the characteristics of system vibration has to be determined, prior to determination of sources; some vibration characteristic of the system should be identified. Therefore, as an initial study this project part aims to determine the bending vibration characteristics under different working condition of the system. Therefore, this problem can be considered as fluid structure interaction as the flow of the fluid is disregarded.

Table: 4.10 Geometry and material properties of seawater OVBD discharge system

Material	Modulus of Elasticity (GPa)	Poissons ratio	Density (kg/m3)	Outer radius (mm)	Inner radius (mm)	Length (mm)
Fiber reinforced pipe	11.1	0.37	1790	R2=381	R1=357	32000
Steel (Caisson)	2.08	0.3	8160	R4=508	R3=482.5	32000

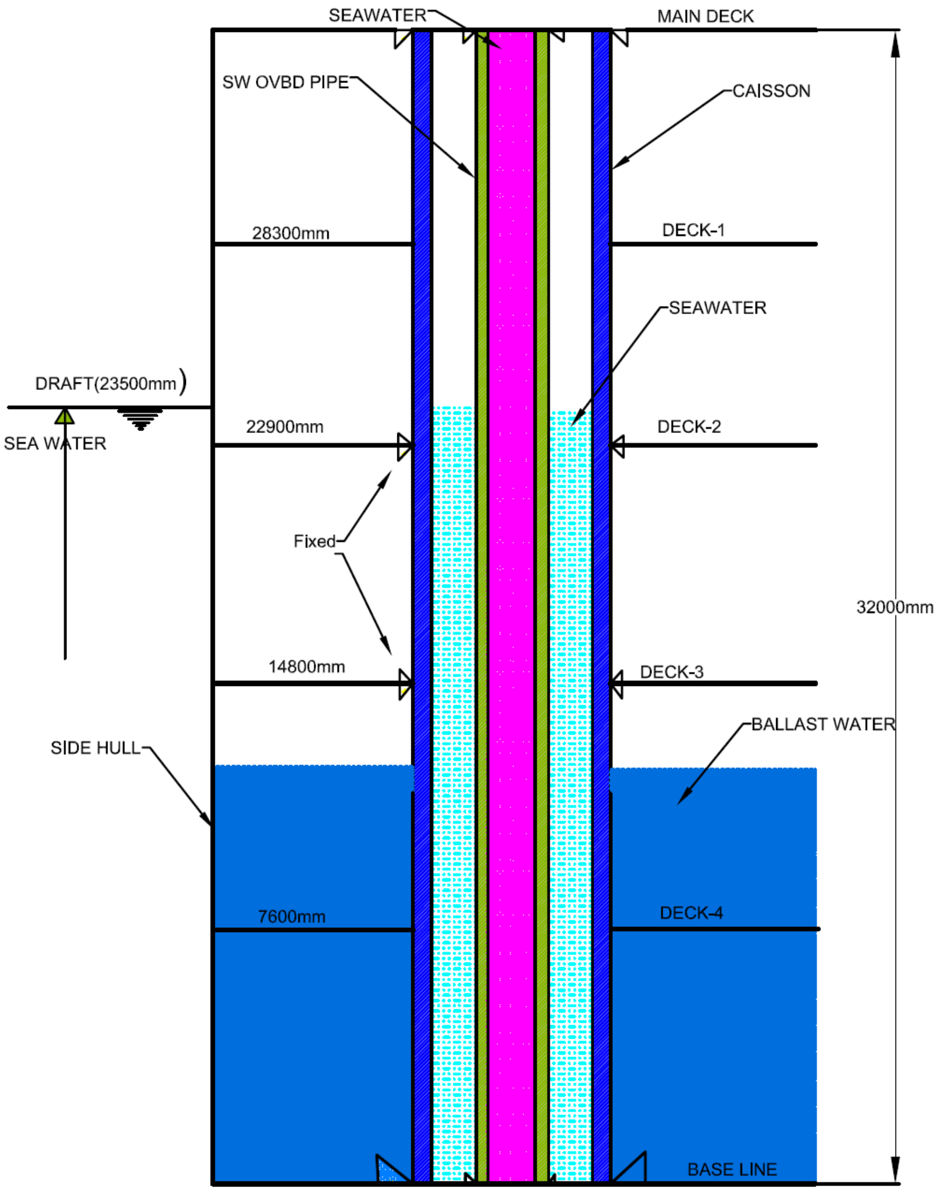


Figure 4.52 Seawater overboard discharge (OVBD) system schematic drawing

4.3.2 Finite Element Model

The same method and procedure used in PART-1 has been used to develop the finite element model of seawater OVBD system as well. The pipe and caisson element made of material property described in Table 4.10 have been modeled using solid element (SOLID95) and the fluid element with FLUID30. The fluid element in pipe, between pipe and caisson and the ballast fluid were considered by seawater as shown in Fig. 4.52. The fluid element at the vicinity of the structural element has been defined with displacements (Ux, Uy, Uz) and pressure degree of freedom, while the fluid elements beyond the neighborhood of structure element defined with only pressure degree of freedom for simplicity and to decrease the computational time and space of the model as shown Fig. 4.54. The outer extreme radius (R_5=1.6m) of the fluid has been limited to three times of caisson radius as suggested in PART-1 study and zero pressure has been defined at the outer boundary. The free surface of ballast water and the seawater between the caisson and pipe has been formulated by defining zero pressure at the free surface which used to avoid complete deflection of the fluid at free surface.

In Fig. 4.53, Fluid-1, Fluid-2, and Fluid-3 stands for fluid flowing through pipe sea, seawater, and ballast water respectively. The ballast water were considered as infinite fluid in finite element model for simplification and to minimize computational time and space with sufficient accuracy of result. Another very important assumption in this work, is that all the fluid parts are assumed non flowing initially.

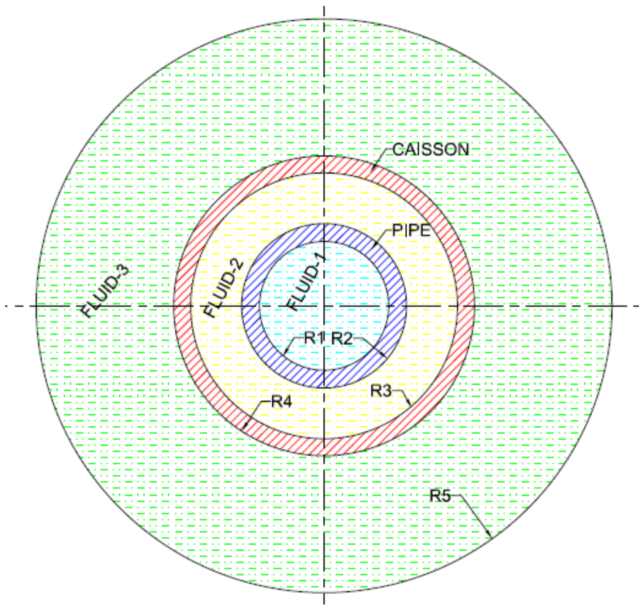


Figure 4.53 Cross section of seawater OVBD system (schematic drawing)

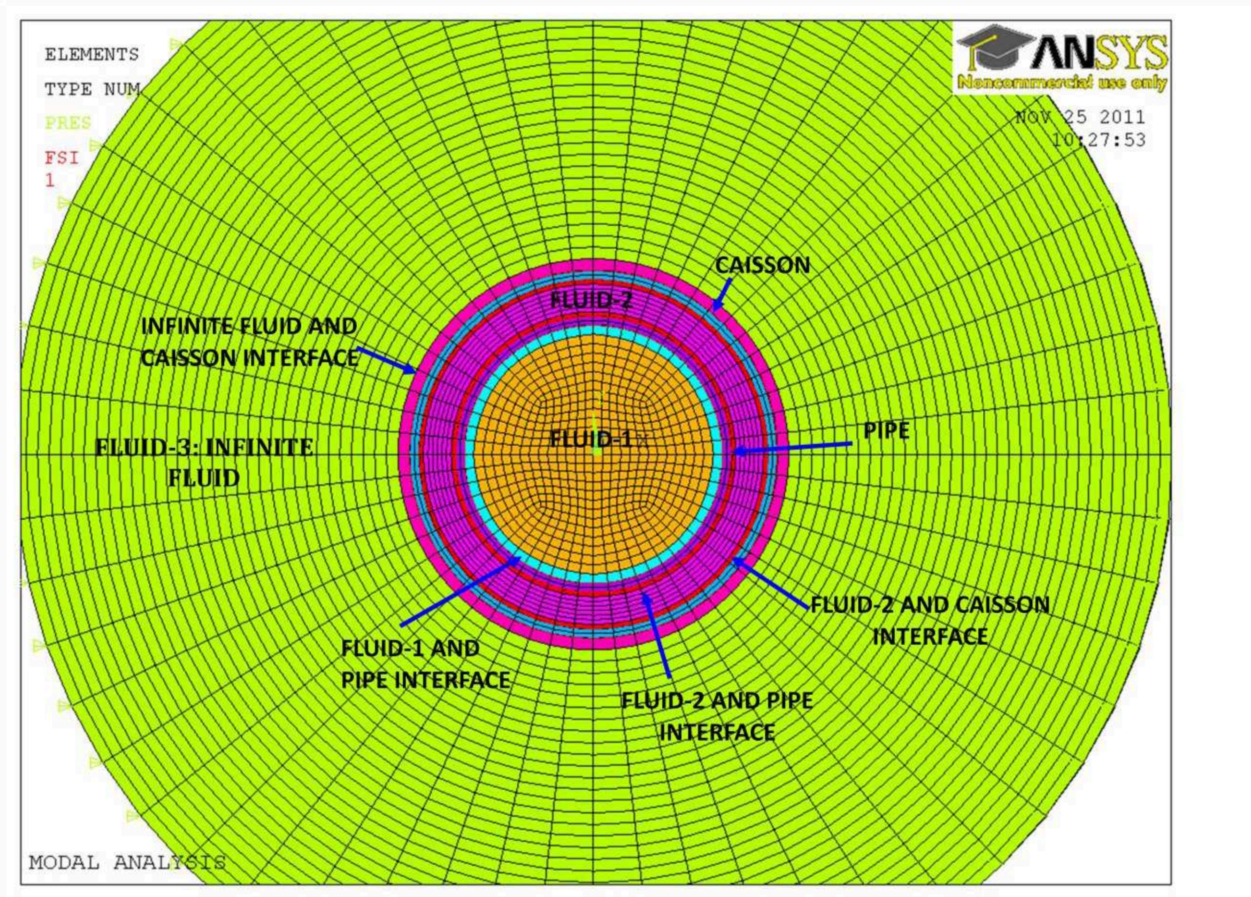


Figure 4.54 Finite element model of seawater OVBD system

4.3.3 Bending Vibration Characteristics of Dry Pipe and Caisson

It is essential to determined component vibration characteristic before going for assembled system vibration analysis. The first two and the first three bending natural frequency of dry pipe and caisson have been determined using finite element method (ANSYS) respectively. The pipe is considered fixed at two extreme points and the caisson is rigidly supported at four points as in assembled system.

The component natural frequencies are very important in determining the added mass because of the surrounding fluids.

Fig. 4.55 illustrate the first two natural frequencies and mode shapes of pipe supported rigidly at the two extremes. Likewise the first three natural frequencies and mode shapes of the caisson have been also determined as shown in Fig. 4.56. In all mode shape figures, the bottom of the system is at the origin of the coordinate system (z-coordinate-axis of the pipe).

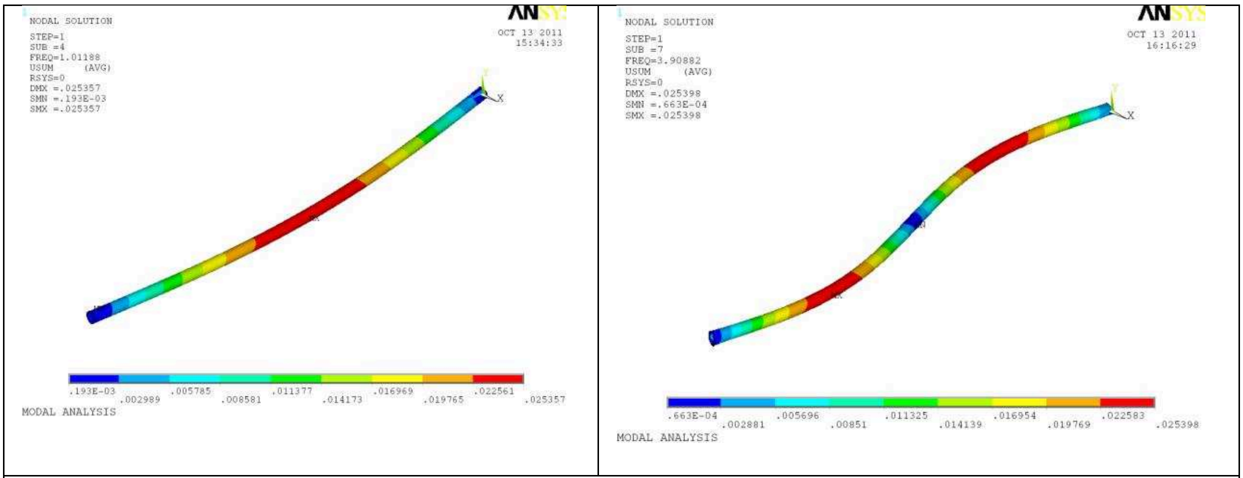
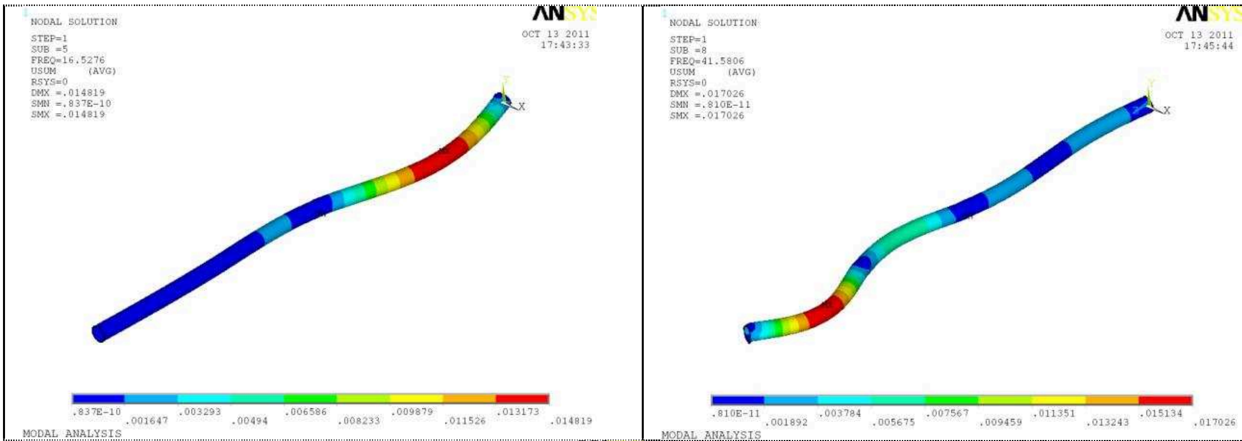


Figure 4.55 The first and second natural frequency and mode shape of pipe



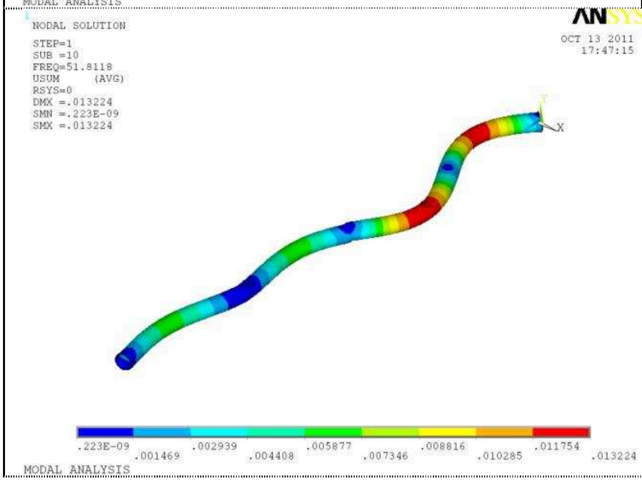


Figure 4.56 The first three bending natural frequency and mode shape of caisson

4.3.4 Bending Vibration Characteristic of Wetted Caisson

In fully assembled seawater OVBD system, the wetted surface of caisson will change depending on the draft and the ballast condition. Therefore, before proceeding to fully assembled vibration analysis of the system, the vibration characteristic of caisson with different wetted condition will be considered without integrating pipe to the system. This will clearly show the effect of ballast and inside fluid on natural frequency of caisson. In this project three cases have been considered, wetted in, out, and in and out as shown in Fig 4.57.

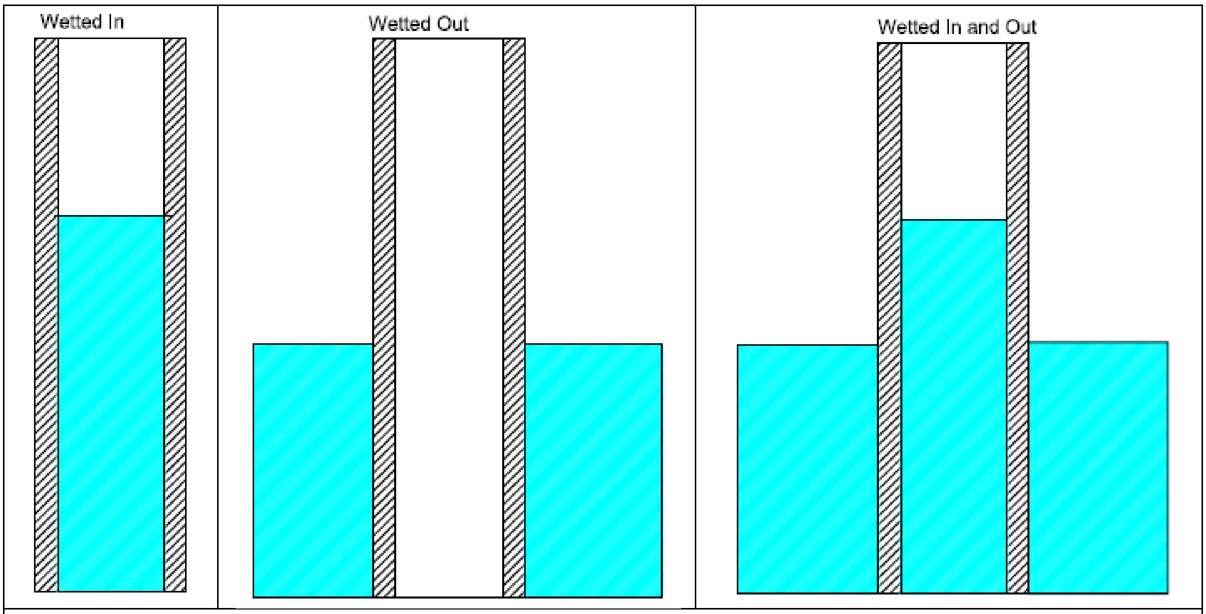


Figure 4.57 Different arrangement of caisson in a fluid (wetted in, out and in and out)

4.3.4.1 Wetted In

In this section, similar boundary condition of caisson as in fully assembled seawater OVBD system has been considered. But a simplified only caisson wetted in has been taken with different fluid level inside caisson as shown in Table 4.11. The first three mode bending natural frequency of caisson under different fluid level have been determined using finite element method ANSYS. As in PART-1, the fluid has been considered as acoustic fluid. As shown in Table 4.11 the bending natural frequency of caisson decreases with fluid inside, and it is also found that natural frequency depends on fluid level. Specially, mode-2 and mode-3 are much dependent on fluid level.

Table: 4.11 Bending natural frequency of caisson wetted in

Modes	Dry Caisson	Wetted In Caisson (1/4)	Wetted In Caisson (1/2)	Wetted In Caisson (3/4)	Wetted In Caisson (Full)
Mode-1	16.528	12.452	11.208	11.182	11.161
Mode-2	41.581	39.676	41.338	39.389	27.932
Mode-3	51.812	44.754	35.391	33.131	35.286

4.3.4.2 Wetted Out

This case attempted to determined the effect of fluid surrounding the caisson on natural frequency and to study the effect of fluid level. To investigate the effect of outside fluid level, quarter (1/4), half (1/2), upper quarter (3/4) and full water level have been considered and the corresponding first three mode bending natural frequencies were determined as shown in Table 4.12 using finite element model.

Table: 4.12 Bending natural frequency of caisson wetted in

Modes	Dry Caisson	Wetted Out Caisson (1/4)	Wetted Out Caisson (1/2)	Wetted Out Caisson (3/4)	Wetted Out Caisson (Full)
Mode-1	16.528	12.881	11.628	11.605	11.470
Mode-2	41.581	40.901	42.547	40.721	28.734
Mode-3	51.812	45.724	36.710	34.609	36.318

4.3.4.3 Wetted In and Out

The caisson filled with seawater with a draft of 24m (inside) and surrounded by ballast water which was considered changing according to the loading condition was considered for analysis. Acoustic fluid structure interaction finite element model with fine mesh has been used for analysis as shown in Fig. 4.58. In this analysis the level of inner fluid was kept constant and ballast water level was considered no, quarter, half, upper quarter, full as shown in Table 4.13 and the corresponding natural frequencies of caisson were calculated using modal analysis of acoustic fluid structure interaction finite element model using ANSYS.

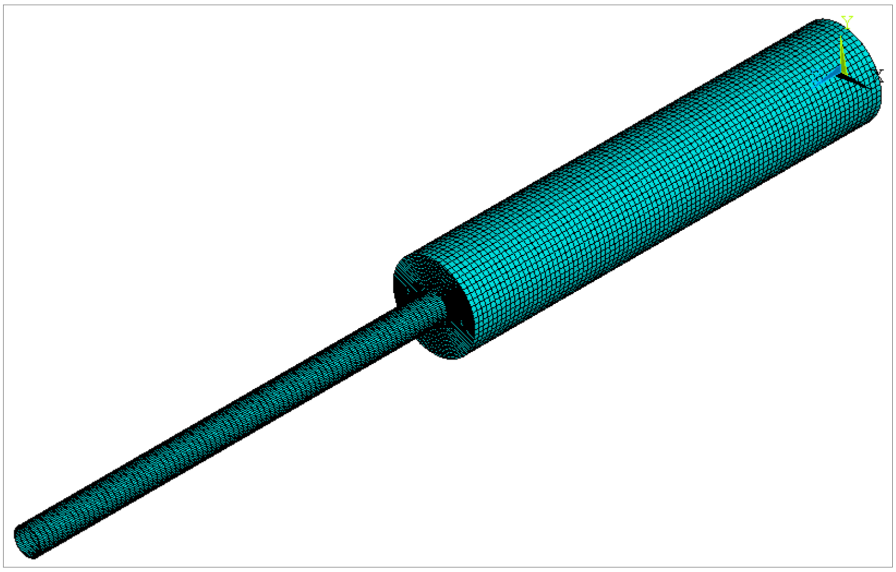


Figure 4.58 Half ballast fluid caisson filled with fluid and surrounded by ballast water

The result shows that in addition to change in natural frequencies caisson due to added mass, there is a significant effect from free surface level of the fluid as shown in Table 4.13. To present more precisely, the percentage decrement in natural frequency of caisson has been calculated with all considered ballast conditions. Fig. 4.62 shows that change in percentage decrement in natural frequency for lower level ballast condition is much as compared to change in percentage decrement in natural frequency because higher level ballast condition.

Table: 4.13 First three modes natural frequencies (Hz) of caisson under different ballast conditions

Modes	Dry	Wetted in only (0)	Quarter (0.25)	Half ballast (0.5)	Upper quarter (0.75)	Full ballast (1)
Mode-1	16.528	11.182	9.814	9.202	9.189	9.187
Mode-2	41.581	39.389	38.131	37.155	34.155	26.277
Mode-3	51.812	33.131	31.368	28.911	27.729	30.510

This reveal that free surface level has significant effect on natural frequency.

Similarly, figs. 4.63-4.66 which show mode-2 and mode-3 respectively reveal clearly the effect of ballast water condition, and also show that the natural frequency much depend on ballast water level. It also depend on the mode shape as well. It is also found that, the mode shape were also changing while the ballast water level changing. The first three mode shapes of the caisson with full ballast water have been shown as in figs. 4.59-4.61 to compare with mode shapes of dry caisson as shown in fig. 4.56.

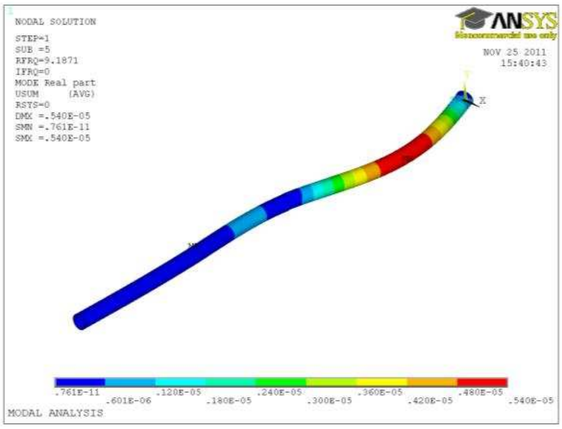


Figure 4.59 Caisson first mode shape with full ballast water

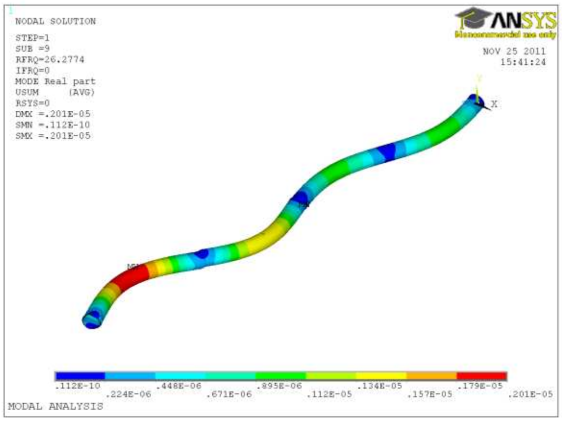


Figure 4.60 Caisson second mode shape with full ballast water

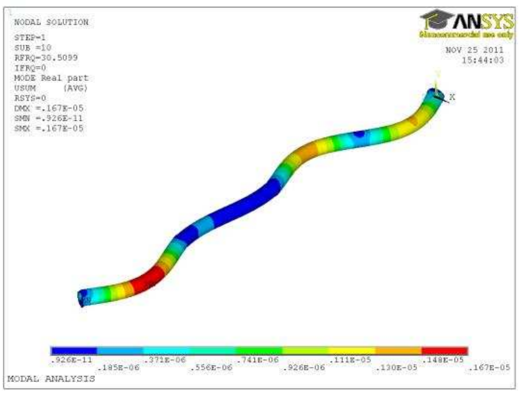


Figure 4.61 Caisson third mode shape with full ballast water

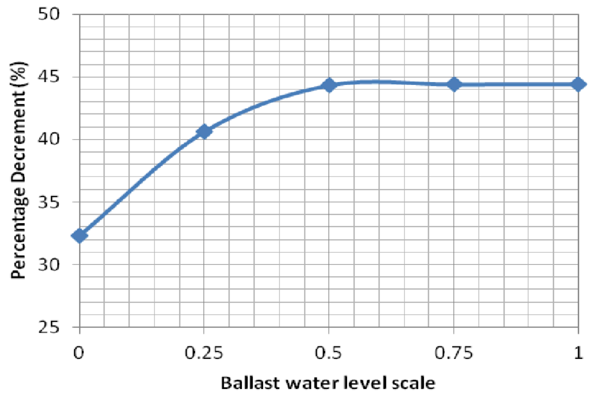


Figure 4.62 Caisson wetted in and out natural frequency percentage decrement Mode-1

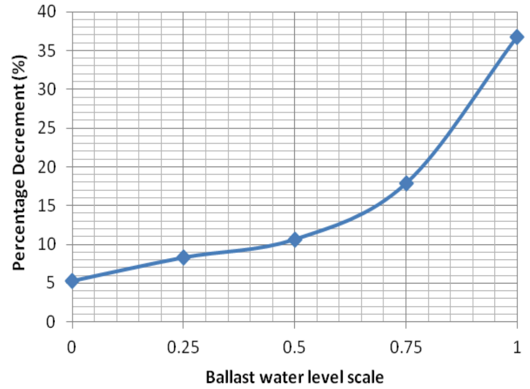


Figure 4.63 Caisson wetted in and out natural frequency percentage decrement Mode-2

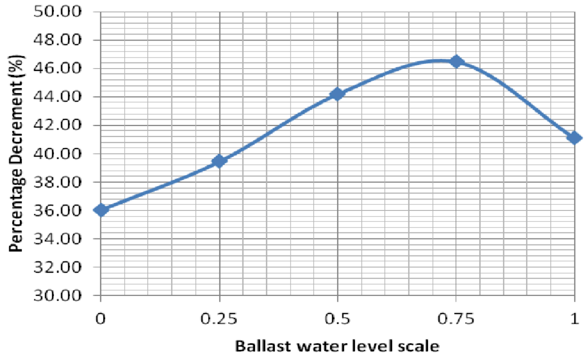


Figure 4.64 Caisson wetted in and out natural frequency percentage decrement Mode-3

4.3.5 Effect of Ballast Water on Seawater OVBD System

As discussed in problem description part, it is relevant to determine the vibration characteristics and added mass of the assembled system coupled with surrounding fluid. Therefore, the added mass and vibration characteristics of seawater OVBD system has been investigated using finite element method (ANSYS). The model has been developed using the draft of the ship as a fluid level for fluid between caisson and pipe which was kept constant throughout the study. To find out the influence of ballast water in affecting system natural frequency, it has been studied by changing the level of the ballast fluid. Figs. 4.65-4.67 shows that there is much decrement in natural frequency of the pipe due to the fluid inside and between the pipe and caisson. The first three mode of pipe vibration has been determined under different ballast conditions namely, no ballast (0), quarter ballast (0.25), half ballast (0.5), upper quarter (0.75), full ballast (1) level have been considered has shown in Figs. 4.65-4.66. The result reveal that, the effect of ballast fluid on pipe natural frequency is not significant. It is also shown that the percentage decrement in first three modes natural frequencies are almost similar.

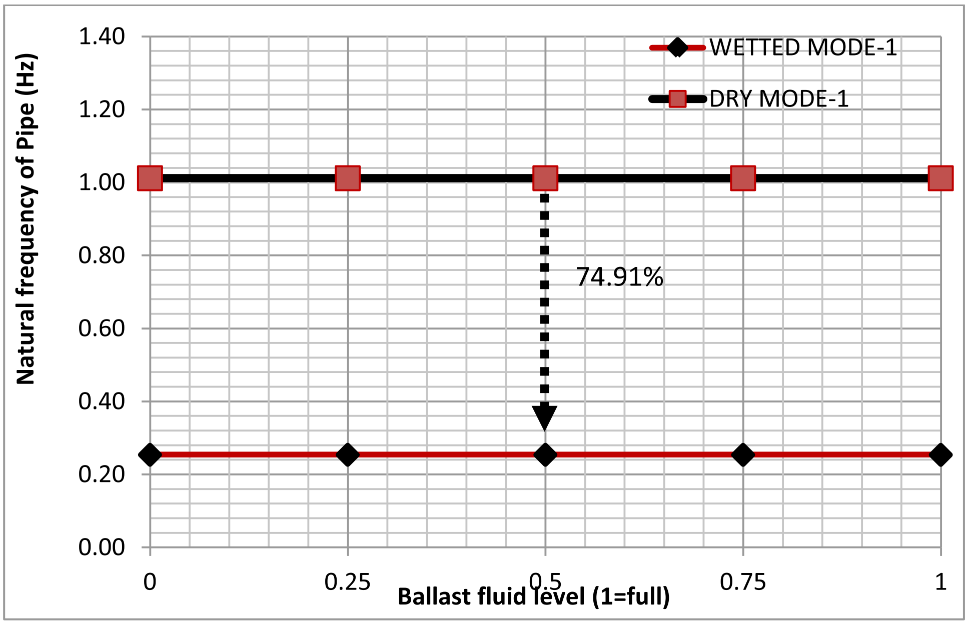


Figure 4.65 Decrement in natural frequency of pipe within OVBD system (Mode-1)

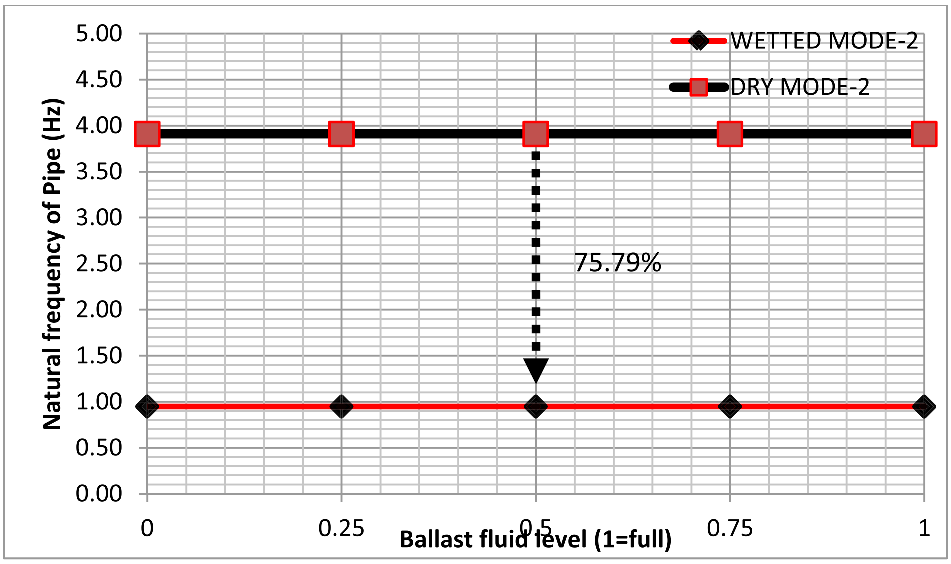


Figure 4.66 Decrement in natural frequency of pipe within OVBD system (Mode-2)

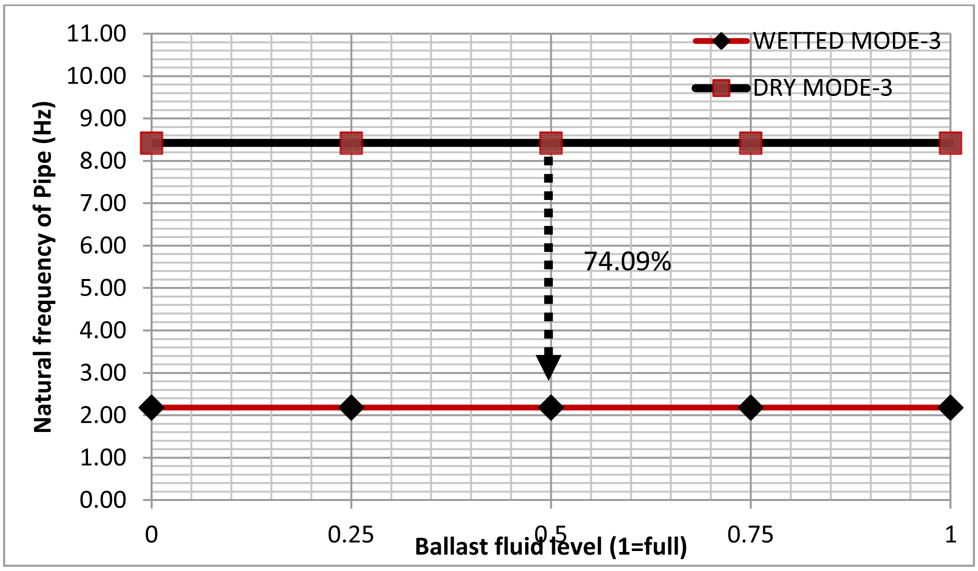


Figure 4.67 Decrement in natural frequency of pipe within OVBD system (mode-3)

Likewise the first natural frequency of the caisson has been determined with different ballast condition, and the result reveal that ballast fluid have much effect on the vibration characteristics of caisson. As shown in Fig. 4.67, five ballast conditions (no, quarter, half, upper quarter and full) have been considered for simulation, the result follows that the effect increases as ballast fluid increases. The percentage of decrement in first mode natural frequency of caisson has been noted on the corresponding ballast conditions as illustrated in Fig. 4.68.

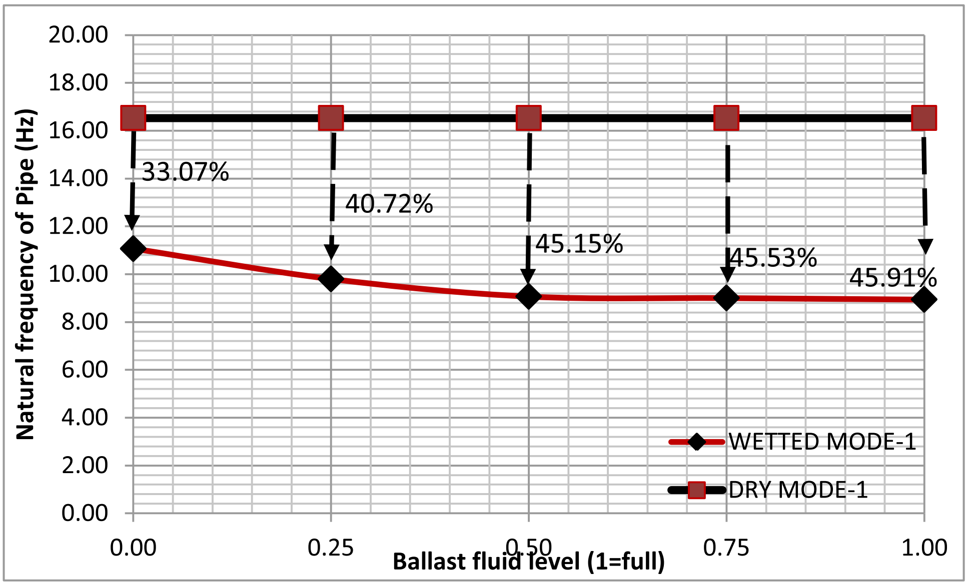


Figure 4.68 Decrement in natural frequency of caisson with OVBD system

4.3.6 Forced OVBD System Without Ballast Water

Harmonically forced no ballast, seawater OVBD system has been analyzed to investigate the transmission of vibration through fluid from pipe to caisson and vice versa and whether the

result from forced analysis coincide with modal analysis. Harmonically varying force (F=2000N) has been applied at the center of OVBD system on pipe over a frequency range (0, 14Hz) without ballast water. The system was kept similar with regards to the support and boundary conditions. The response has been taken at the center of the pipe and caisson.
Fig. 4.69 -4.70 illustrate the vibration displacement (VALU (m)) versus frequency (Hz) of pipe at the center. These figures clearly shows the peak frequencies of pipe (mode-1 (0.254Hz), mode-2 (0.9475Hz), mode-3 (2.1841Hz)) which coincide with the result from modal analysis.

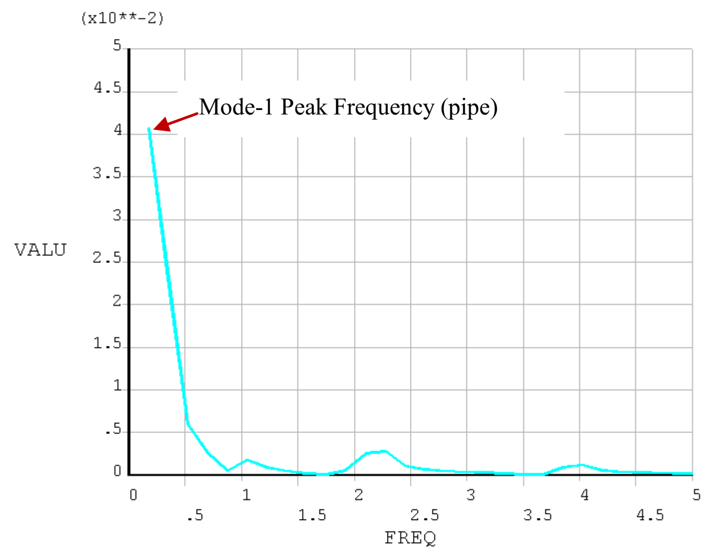


Figure 4.69 Displacement versus frequency response of harmonically forced pipe in OVBD system

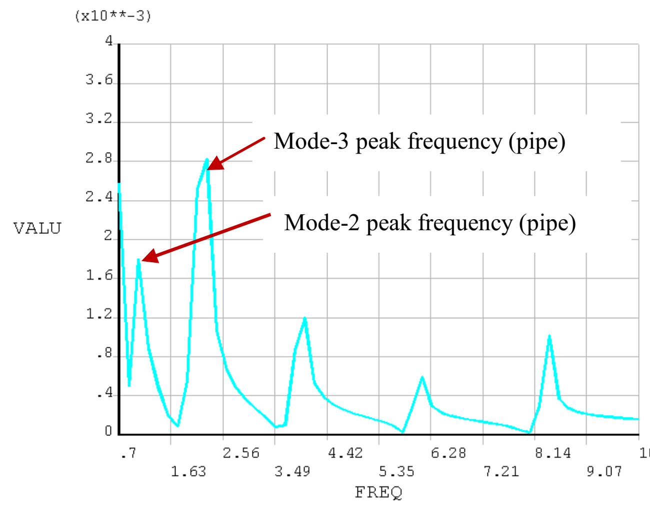


Figure 4.70 Displacement versus frequency response of harmonically forced pipe in OVBD system (Zoomed)

Likewise Fig. 4.71 illustrate the peak natural frequencies of caisson under the prescribed force and boundary conditions. The first mode-1 natural frequency of caisson has been determined in modal analyses as (11.063Hz) without ballast water. Under forced analysis as shown in Fig. 4.71 the biggest peak frequency occurred at the natural frequency of caisson itself. But it very interesting to note that there are peak frequencies below the natural frequencies caisson which resulted from transmission of vibration from pipe to caisson. As the reponse at caisson shows 100 to 1000 times smaller than response for pipe, the figures has been drwan separately. This difference in dispacement could be expected as the force has been applied on the pipe.

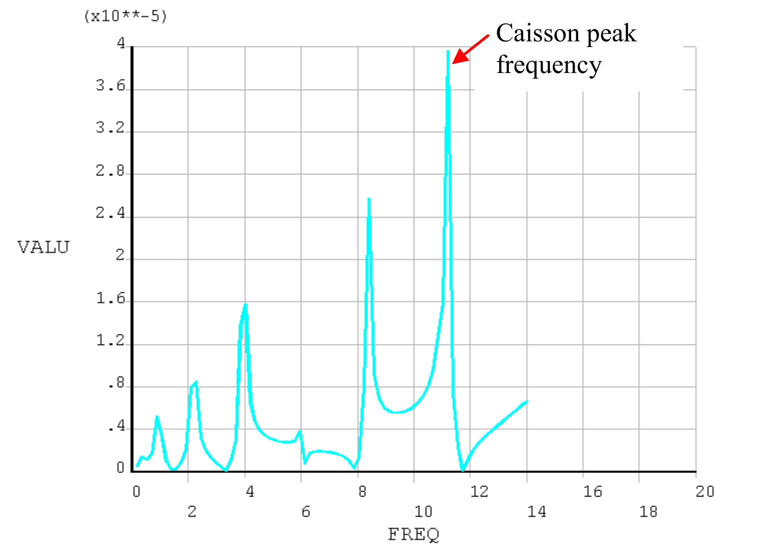


Figure 4.71 Displacement versus frequency response of harmonically forced pipe in OVBD system

CHAPTER 5

5. DISCUSSION AND CONCLUSION

This section of the project aims to summarize the discussions that has been made in the previous chapters of the project work and to conclude the relevance of the result and contribution of the project work for scientific developments.

5.1 Discussion

Fluid-structure interaction (FSI) problems occur in many applications in industry and science, of which the study of vibration characteristic of pipes and shafts arranged in fluid filled tubular spaces is a typical example. This multi-physics phenomenon needs to incorporate various aspects of complex mixed description of fluid and structural dynamics to investigate the complete dynamic behavior of the system as a whole. This field of study has received extensive research focus over the years and specially enormous effort has been paid to investigate the interaction between fluid and cylindrical structures as it is one of the most common construction members in a wide variety of engineering structures. This project work also stands from two important fluid cylindrical structure interaction engineering problems as well described in the previous chapters of the work. The main target was to investigate the bending vibration characteristic of shaft, tubes/pipes arranged in different way in tabular spaces, which is an important step in vibration analysis of fluid-cylindrical structures.

The analysis of the project work has been done using finite element method (ANSYS).The problems were considered as acoustic fluid structure interaction which is most typical for modal analysis of complex fluid structure interaction problem as it saves the memory space and computational time with relevant result.

For better presentation, the project was categorized into two parts (PART-1 and PART-2) as they are two different engineering problems. PART-1 deals with bending vibration analysis of shaft surrounded by fluid (oil) confined concentrically by outer cylindrical tube and immersed in infinite fluid (e.g., stern tube), and further categorized into four cases: from which CASE-1 to CASE-3 has been investigated using acoustic fluid structure interaction using finite element method and validated with available theoretical formulas. CASE-4 as the typical problem of the project work has been analyzed in a similar manner that has been used in CASE-2 and CASE-3 using acoustic fluid structure interaction finite element model (ANSYS). For the above engineering problems, systematic and parametric, free and forced vibration analyses have been investigated using finite element analysis, which include adaptation of mesh fineness, element type, boundary conditions, determination of dry and wetted natural frequencies and the corresponding mode shapes, effect of fluid density, parametric study and determination of added mass. Added mass is the most relevant parameter to study and design wetted structural elements under vibration.

This cases have been also compared with regards to the effect of fluid on vibration characteristic and hydrodynamics mass coefficients. It is found that CASE-4 is affected more than CASE-2 and CASE-3. It is also found that for CASE-4 percentage decrement in natural frequency of shaft increases as the radius of shaft increases while percentage decrement in tube natural frequency decreases. The percentage decrement of CASE-4 as tube radius decreases has been also computed and shows the same fashion with previous but with quite different magnitude which reveal that decrement/increment in natural frequency of such a system is not only dependent on radius ratio but also on absolute value of each.

The results from harmonically forced analyses of CASE-2 and CASE-4 have been also presented which further validate the result from modal analysis. More interestingly, the result for CASE-4 illustrate that, shaft and tube vibrate together and show peak/resonance frequency at each resonance as the vibration transmit through fluid. Alternative to harmonic analysis of fluid cylinderical structure interaction, more simpler interms of computational time and memory space, increased density harmonic analysis has been performed. This analysis has been employed considering the added mass obtained from modal analysis to increase the shaft density keeping the volume same. This could be used for first design stage.

PART-2 of the project deals with bending vibration of cylindrical tube filled with fluid and surrounded by confined partially filled fluid medium (sea water), confined concentrically by cylindrical outer tube surrounded by infinite fluid (e.g., Overboard discharge line). Similar procedures and trends of acoustic fluid structure interaction modeling using finite element method (ANSYS) have been used as in PART-1 to meet the aim of the project. Dry and wetted natural frequencies and corresponding mode shapes of the system have been determined. The influence of the surrounding fluid on natural frequency has been studied by changing the fluid level and revealed that ballast water condition has considerable effect on natural frequency of caisson but not on pipe. Therefore, the effect of ballast water on caisson has been further studied by taking out the pipe. So, three cases (wetted in, out, and in and out) of the caisson have been studied under different ballast conditions, and the result divulge that natural frequency is much dependent on wetted surface, mode shape and ballast fluid level. Obviously, the natural frequency of caisson with wetted in and out is affected much more than only wetted in and only wetted out. Also, wetted in has a significantly higher effect on natural frequency of the system as compared to wetted out case.

With regards to sea water OVBD system, pipe natural frequency decreased by around 75% for the first three mode shapes. The caisson natural frequency which depends on ballast condition also decreased by around 33% for no ballast to around 46% for full ballast condition.
Similar to PART-1, harmonically forced sea water OVBD system has been analyzed to validate the result from modal analysis and to find out the transmission of vibration and interaction of the systems as they are all coupled.

5.2 Conclusion

In order to improve design margins and ensure safety and satisfactory operating performance of any structure element, precise knowledge of their transient response is vital. Specially for complex fluid-structure interaction problems in which structural motion creates fluid deformation/flow which in turn influences the structural element motion should be well studied. For analysis of vibrating fluid-cylindrical structure interaction, an important step is the evaluation of their vibration modal characteristics, such as natural frequencies, mode shapes which play a key role in the design and vibration suppression of these structures when subjected to dynamic excitations. This project work laid important contribution to the design of fluid-cylindrical structure interaction problems by investigating the vibration characteristic of the complex interacting system of its kind. In addition this project work intended to determine simpler and quick empirical formulae to determine the added mass of fluid cylindrical structure interaction with hydrodynamic mass coefficient under considered parametric range.

Different fluid cylindrical structure interaction arrangements (PART-1 and PART-2) have been considered in this study and laid essential points with regards to their effect on each other. The effect of their interaction on cylindrical structure vibration characteristics has been carefully studied and the result obtained for different arrangement has been compared with regards to different parameters (decrement in natural frequency, added mass coefficient) for more awareness about the interaction of fluid cylindrical structure in different arrangements.

For both engineering problems the harmonically forced bending vibration analysis has been also performed to strengthen the result from modal analysis and to see the vibration transmission from one structural element to another through fluid element. And the result obtained reveal that natural frequency of one structural element much influences the vibration characteristic of other structural element as they are coupled by fluid element. In general, this project work laid important design parameters for further vibration investigation of considered structure elements.

5.3 Future Direction

In this project work, modal and harmonic bending vibration analyses of cylindrical structures arranged in fluid media have been made to determine the effect of fluid on natural frequency of the structure and to determine added mass coefficient to recommend easy and quick formulae for added mass determination for such arrangements. For this reason, parametric study has been made over some range of shaft and tube geometries, which are specific to the mentioned engineering problems. For general study, the consideration of a wider range of shaft and tube geometries will be essential.

Some of the studies made in this report have been validated with already existing formulations, and some others depend on previously validated models. For an even higher validity of the developed formulae and procedures, the results obtained in both parts of this work (PART-1 and PART-2) would be better supported with further possible experimental, analytical and numerical results.

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APPENDIX-1

Appendix-1 presents command listing that has been used in PART-1 of the project. As the report for PART-1 has been categorized into four different cases, the four command list will be presented as follow:

Appendix-1a

Command list for Modal analysis of dry Solid shaft(3D)

```
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
!Element type
 ET,1,SOLID95        ! structural element
!Material properties
MP,EX,1,2.068e11
MP,DENS,1, 8160
MP,NUXY,1,0
!Geometric of the model(Radius)
r1=0.2125
!Create a circles
CYL4,0,0,0,0,r1,90
!Extrude to required length
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,8.5
ESEL,ALL
NSEL,ALL
!Select, assign attribute to and mesh
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,16,1
LESIZE,3,,,16,1
LESIZE,2,,,16,1
LESIZE,7,,,120,1
LESIZE,8,,,120,1
LESIZE,9,,,120,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
!Reflect quarter circle into semicircle
about x-axis
nsym,x,1000000,all    ! offset node number
by 1000000
esym,,1000000,all
!Reflect semicircle into full circle about
y-axis
nsym,y,2000000,all    ! offset node number
by 2000000
esym,,2000000,all
NUMMRG,node        ! merge nodes
!Boundary conditions
csys,1
NSEL,S,LOC,z,0
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
csys,1
NSEL,S,LOC,z,8.5
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
nsel,all
esel,all
/Solution Module
ANTYPE,MODAL          ! MODAL ANALYSIS
MODOPT,LANB,20,0,200 ! BLOCK LANCZOS SOLVER
MXPAND,,,,YES         ! EXPAND MODE
SOLVE
FINISH
```

Appendix-1b

Solid shaft immersed in infinite fluid (CASE-2,3D)

```
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
! Element type
ET,1,SOLID95        ! structural element
ET,2,FLUID30        ! acoustic fluid
element with ux & uy
ET,3,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
!Material properties
MP,EX,1,2.068e11
MP,DENS,1, 8160.0000
MP,NUXY,1,0
MP,DENS,2,1030
MP,SONC,2,1460
!Geometry oft he model, Radius
r1=0.2125
r2=0.2625
! Create inner and outer quarter circles
CYL4,0,0,0,0,r1,90
CYL4,0,0,r1,0,r2,90
! Extrude to required length
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,8.5
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,2
VEXT,ALL, , ,0,0,8.5
ESEL,ALL
NSEL,ALL
! Select, assign attribute to and mesh area
1
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,14,1
LESIZE,3,,,14,1
LESIZE,2,,,14,1
LESIZE,11,,,80,1
LESIZE,12,,,80,1
LESIZE,13,,,80,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
! select, assign attribute to and mesh area
2
VSEL,S,VOLU,,2
VATT,2,2,2,0
LESIZE,5,,,6,1
LESIZE,7,,,6,1
LESIZE,6,,,14,1
LESIZE,4,,,14,1
LESIZE,18,,,80,1
LESIZE,19,,,80,1
LESIZE,20,,,80,1
LESIZE,21,,,80,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,2
! Reflect quarter circle into semicircle
about x-axis
nsym,x,10000000,all    ! offset node number
by 10000000
esym,,10000000,all
```

```
! Reflect semicircle into full circle about y-axis
nsym,y,20000000,all    ! offset node number by 20000000
esym,,20000000,all
NUMMRG,node            ! merge nodes only
! Modify outer 2 layers of el29 into type 3
esel,s,type,,1
nsle,s       ! Select those nodes attached to the selected elements
esln,s,0     ! Select those elements attached to the selected nodes
nsle,s       ! Select those nodes attached to the selected elements
cm,tmp_ele,elem ! Create component of elements
esel,s,type,,2
cmsel,u,tmp_ele  !select new set of component of elements
nsle,s
emodif,all,type,3
esel,all
nsel,all
! Flag interface as fluid-structure interface
csys,1
nsel,s,loc,x,r1
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
!Boundary conditions
csys,1
NSEL,S,LOC,z,0
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
csys,1
NSEL,S,LOC,z,8.5
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
! Define zero pressure at outer extreme
csys,1
NSEL,S,LOC,X,r2
D,ALL,PRES,0.0    ! SET PRESSURE AT OUTER RADIUS TO ZERO
nsel,all
esel,all
! Enter solution module
/SOLUTION
ANTYPE,MODAL             ! MODAL ANALYSIS
MODOPT,UNSYM,10,0,100  ! UNSYMMETRIC MATRIX
SOLVER
MXPAND,,,,YES          ! EXPAND MODE
SOLVE
FINISH
```

Appendix-1c

Solid shaft immersed in fluid filled tabular space (CASE-3, 3D,

```
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
!Element type
ET,1,SOLID45        ! structural element
ET,2,FLUID30        ! acoustic fluid element with ux & uy
ET,3,SOLID45         !Outer rigid cylinder
ET,4,FLUID30,,1,0  ! acoustic fluid element without ux & uy
! Material properties
MP,EX,1,2.068e11
MP,DENS,1, 8160.0000
MP,NUXY,1,0
MP,DENS,2,1030
MP,SONC,2,1460
MP,EX,3,2.068e11
MP,DENS,3, 8160.0000
MP,NUXY,3,0
!Model geometry
r1=0.2125          ! Shaft radius
r2=0.3555          ! Inner radius of tube
r3=0.3755          ! Outer radius of tube
L=8.5              ! length
! Create inner and outer quarter circles
CYL4,0,0,0,0,r1,90
CYL4,0,0,r1,0,r2,90
CYL4,0,0,r2,0,r3,90
! Extrude to required length
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,2
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,3
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
! Select, assign attribute to and mesh area 1
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,16,1
LESIZE,3,,,16,1
LESIZE,2,,,16,1
LESIZE,15,,,120,1
LESIZE,16,,,120,1
LESIZE,17,,,120,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
! Select, assign attribute to and mesh area 2
VSEL,S,VOLU,,2
VATT,2,2,2,0
LESIZE,5,,,20,1
LESIZE,7,,,20,1
LESIZE,6,,,16,1
LESIZE,4,,,16,1
LESIZE,22,,,120,1
LESIZE,23,,,120,1
LESIZE,24,,,120,1
LESIZE,25,,,120,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,2
! Select, assign attribute to and mesh area 3
VSEL,S,VOLU,,3
VATT,3,3,3,0
LESIZE,8,,,16,1
LESIZE,9,,,2,1
LESIZE,10,,,16,1
LESIZE,11,,,2,1
LESIZE,30,,,120,1
LESIZE,31,,,120,1
LESIZE,32,,,120,1
LESIZE,33,,,120,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,3
! Reflect quarter circle into semicircle about x-axis
nsym,x,100000,all    ! offset node number by 100000
esym,,100000,all
```

```
! Reflect semicircle into full circle about y-axis
nsym,y,200000,all ! offset node number by 200000
esym,,200000,all
NUMMRG,node      ! merge nodes only
! Modify outer 2 layers of el29 into type 4
esel,s,type,,1
nsle,s       ! Select those nodes attached to the selected elements
esln,s,0     ! Select those elements attached to the selected nodes
nsle,s       ! Select those nodes attached to the selected elements
cm,tmp_ele,elem ! Create component of elements
esel,s,type,,2
cmsel,u,tmp_ele  !select new set of component of elements
nsle,s
emodif,all,type,4
esel,all
nsel,all
!Modify inner fluid 4 layers of el29 into type 2
!Inner,
csys,1
NSEL,S,LOC,X,R2
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of elements
esel,s,type,,4
cmsel,R,tmp_ele
nsle,s
emodif,all,type,2
esel,all
nsel,all
! flag interface as fluid-structure interface
csys,1
nsel,s,loc,x,r1
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
csys,1
nsel,s,loc,x,r2
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
!Boundary conditions
NSEL,S,LOC,z,0
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,L
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,0
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,0,L
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,ALL
nsel,all
esel,all
! Enter solution module
/SOLUTION
ANTYPE,MODAL              ! MODAL ANALYSIS
MODOPT,UNSYM,10,0,100     ! UNSYMMETRIC
MATRIX SOLVER
MXPAND,,,,YES             ! EXPAND MODE
SOLVE
FINISH
```

Appendix- 1d-1

```
!Solid shaft immersed in fluid filled tabular space in infinite fluid (CASE-4-2D)
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
!Material type
ET,1,PLANE42           ! structural element
ET,2,FLUID29           ! acoustic fluid element with ux & uy
ET,3,PLANE42           ! Outer rigid cylinder
ET,4,FLUID29,,1,0      ! acoustic fluid element without ux & uy
ET,5,COMBIN14          !Structural element for spring and damper
ET,6,FLUID29           ! acoustic fluid element with ux & uy
ET,7,FLUID29,,1,0      ! acoustic fluid element without ux & uy
ET,8,COMBIN14          !Structural element for spring and damper
!Real Constants
!Define a thickness THK, for material-1 and 3, as 0.001m, pressure boundary for fluids, and spring constant for element number-4
R,1,0.001
R,2,0.1
R,3,0.001
R,4,0.1
R,5, 3108012.424  ! Stiffness of shaft
R,6,0.1
R,7,0.1
R,8, 5958323.956  ! Stiffness of tube
! Material properties
MP,EX,1,2.08e11
MP,DENS,1, 8.160E+06
MP,NUXY,1,0
MP,DENS,2,827
MP,SONC,2,1451
MP,EX,3,2.08e11
MP,DENS,3, 8.160E+06
MP,NUXY,3,0
MP,EX,5,2.068e11
MP,DENS,5, 8160
MP,NUXY,5,0
MP,DENS,6,1030
MP,SONC,6,1460
MP,EX,5,2.068e11
MP,DENS,5, 8160
!Geometry of the model
! Radius of the cylinder
r1=0.2125
r2=0.3555
r3=0.3755
r4=0.75
! Create inner and outer quarter circles
CYL4,0,0,0,0,r1,90
CYL4,0,0,r1,0,r2,90
CYL4,0,0,r2,0,r3,90
CYL4,0,0,r3,0,r4,90
! Select, assign attribute to and mesh area 1
ASEL,S,AREA,,1
AATT,1,1,1,0
LESIZE,1,,,16,1
LESIZE,3,,,16,1
LESIZE,2,,,16,1
MSHKEY,1
MSHAPE,0,2D ! mapped quad mesh
AMESH,1
```

```
! Select, assign attribute to and mesh area
2
ASEL,S,AREA,,2
AATT,2,2,2,0
LESIZE,5,,,10,1
LESIZE,7,,,10,1
LESIZE,6,,,16,1
LESIZE,4,,,16,1
MSHKEY,0
MSHAPE,0,2D ! mapped quad mesh
AMESH,2
! Select, assign attribute to and mesh area
3
ASEL,S,AREA,,3
AATT,3,3,3,0
LESIZE,8,,,16,1
LESIZE,9,,,2,1
LESIZE,10,,,16,1
LESIZE,11,,,2,1
MSHKEY,0
MSHAPE,0,2D ! mapped quad mesh
AMESH,3
! Select, assign attribute to and mesh area
4
ASEL,S,AREA,,4
AATT,6,6,6,0
LESIZE,12,,,16,1
LESIZE,13,,,16,1
LESIZE,14,,,16,1
LESIZE,15,,,16,1
MSHKEY,0
MSHAPE,0,2D ! mapped quad mesh
AMESH,4
! Reflect quarter circle into semicircle
about x-axis
nsym,x,10000,all    ! offset node number by
10000
esym,,10000,all
! Reflect semicircle into full circle about
y-axis
nsym,y,20000,all ! offset node number by
20000
esym,,20000,all
NUMMRG,node      ! merge nodes only
! Modify inner fluid 2 layers of el29 into
type 4
esel,s,type,,1
nsle,s       ! Select those nodes attached
to the selected elements
esln,s,0     ! Select those elements
attached to the selected nodes
nsle,s       ! Select those nodes attached
to the selected elements
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,2
cmsel,u,tmp_ele  !select new set of
component of elements
nsle,s
emodif,all,type,4
esel,all
nsel,all
!Modify inner fluid 4 layers of el29 into
type 2
!Inner,
csys,1
NSEL,S,LOC,X,R2
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,4
cmsel,R,tmp_ele
nsle,s
emodif,all,type,2
esel,all
nsel,all
! modify outer 6 layers of el29 into type 7
!outer
Csys,1
NSEL,S,LOC,X,R3
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,6
cmsel,u,tmp_ele
nsle,s
emodif,all,type,7
esel,all
nsel,all
! flag interface as fluid-structure
interface 1
csys,1
nsel,s,loc,x,R1
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
! flag interface as fluid-structure
interface 2
csys,1
nsel,s,loc,x,R2
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
! flag interface as fluid-structure
interface 3
csys,1
nsel,s,loc,x,R3
esel,s,type,,6
sf,all,fsi,1
nsel,all
esel,all
!Create spring element and connect with a
shaft element
N,88888,-0.9   !Definition of node number
and position
N,77777,0.9    !Definition of node number
and position
TYPE,5
REAL,5
E,88888,10039
E,77777,39
nsel,all
esel,all
!Create spring element and connect with a
Tube element
N,99999,-1   !Definition of node number and
position
N,66666,1    !Definition of node number and
position
TYPE,8
REAL,8
E,99999,10440
E,66666,440
nsel,all
esel,all
!Boundary conditions
KBC,1    ! Step boundary condition
D,88888,ALL        ! Constrain UY DOF
D,77777,ALL
D,99999,ALL        ! Constrain UY DOF
D,66666,ALL
!Fix element 1 in other directions
esel,s,type,,1
nsle,s
D,all,UY,UZ,ROTZ,1
nsel,all
esel,all
!Fix element 3 in other directions
Esel,s,type,,3
nsle,s
```

```
D,ALL,UY,UZ,ROTZ,1
nsel,all
esel,all
!Set pressure zero at the boundary
NSEL,S,LOC,X,R4
D,ALL,PRES,0.0    ! SET PRESSURE AT OUTER
RADIUS TO ZERO
nsel,all
esel,all
! enter solution module
/SOLUTION
ANTYPE,MODAL              ! MODAL ANALYSIS
MODOPT,UNSYM,20,0,200     ! UNSYMMETRIC
MATRIX SOLVER
MXPAND,,,,YES             ! EXPAND MODE
SOLVE
FINISH
```

Appendix- 1d-2

```
! Solid shaft immersed in fluid filled tabular space surrounded by infinite fluid (CASE-4, 3D)
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
!Material properties
ET,1,SOLID45        ! structural element
ET,2,FLUID30        ! acoustic fluid
element with ux & uy
ET,3,SOLID45         !Outer rigid cylinder
ET,4,FLUID30        ! acoustic fluid
element with ux & uy
ET,5,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
ET,6,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
! Material properties
MP,EX,1,2.08e11
MP,DENS,1, 8160
MP,NUXY,1,0
MP,DENS,2,1030
MP,SONC,2,1460
MP,EX,3,2.08e11
MP,DENS,3, 8160
MP,NUXY,3,0
MP,DENS,4,1030
MP,SONC,4,1460
! Geometry of the model (Radius and length
of the cylinder)
R1=0.2125
R2=0.3555
R3=0.3755
R4=0.75
L=8.5
! create inner and outer quarter circles of
the shaft and  the tube
CYL4,0,0,0,0,R1,90
CYL4,0,0,R1,0,R2,90
CYL4,0,0,R2,0,R3,90
CYL4,0,0,R3,0,R4,90
! Extrude to required length (Extrude)
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,2
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,3
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,4
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
! select, assign attribute to and mesh area
1
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,16,1
LESIZE,3,,,16,1
LESIZE,2,,,16,1
LESIZE,19,,,100,1
LESIZE,20,,,100,1
LESIZE,21,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
! Select, assign attribute to and mesh area
2
VSEL,S,VOLU,,2
VATT,2,2,2,0
LESIZE,5,,,16,1
LESIZE,7,,,16,1
LESIZE,6,,,16,1
LESIZE,4,,,16,1
LESIZE,26,,,100,1
LESIZE,27,,,100,1
LESIZE,28,,,100,1
LESIZE,29,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,2
! select, assign attribute to and mesh area
3
VSEL,S,VOLU,,3
VATT,3,3,3,0
LESIZE,8,,,16,1
LESIZE,9,,,2,1
LESIZE,10,,,16,1
LESIZE,11,,,2,1
LESIZE,34,,,100,1
LESIZE,35,,,100,1
LESIZE,36,,,100,1
LESIZE,37,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,3
! Select, assign attribute to and mesh area
4
VSEL,S,VOLU,,4
VATT,4,4,4,0
LESIZE,12,,,16,1
LESIZE,14,,,16,1
LESIZE,13,,,16,1
LESIZE,15,,,16,1
LESIZE,42,,,100,1
LESIZE,43,,,100,1
LESIZE,44,,,100,1
LESIZE,45,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,4
! Reflect quarter circle into semicircle
about x-axis
nsym,x,100000,all    ! offset node number
by 100000
esym,,100000,all
! Reflect semicircle into full circle about
y-axis
nsym,y,200000,all ! offset node number by
200000
esym,,200000,all
NUMMRG,node      ! merge nodes only
! Modify inner fluid 2 layers of el29 into
type 5
esel,s,type,,1
nsle,s       ! Select those nodes attached
to the selected elements
```

```
esln,s,0     ! Select those elements
attached to the selected nodes
nsle,s       ! Select those nodes attached
to the selected elements
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,2
cmsel,u,tmp_ele  !select new set of
component of elements
nsle,s
emodif,all,type,5
esel,all
nsel,all
!Modify inner fluid 5 layers of el29 into
type 2
!Inner,
csys,1
NSEL,S,LOC,X,R2
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,5
cmsel,R,tmp_ele
nsle,s
emodif,all,type,2
esel,all
nsel,all
! Modify outer 4 layers of el29 into type 6
!outer
Csys,1
NSEL,S,LOC,X,R3
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,4
cmsel,u,tmp_ele
nsle,s
emodif,all,type,6
esel,all
nsel,all
! Flag interface as fluid-structure
interface 1
csys,1
nsel,s,loc,x,R1
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 2
csys,1
nsel,s,loc,x,R2
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 3
csys,1
nsel,s,loc,x,R3
esel,s,type,,4
sf,all,fsi,1
nsel,all
esel,all
!Set pressure zero at the outer boundary
NSEL,S,LOC,X,R4
D,ALL,PRES,0.0  ! SET PRESSURE AT OUTER
RADIUS TO ZERO
nsel,all
esel,all
!Boundary conditions (Simply supported at
the two extreme)
NSEL,S,LOC,z,0
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,L
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,0
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,UX

NSEL,S,LOC,z,L
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,UX
nsel,all
esel,all
! Enter solution module
/SOLUTION
ANTYPE,MODAL          ! MODAL ANALYSIS
MODOPT,UNSYM,20,0,200 ! UNSYMMETRIC MATRIX
SOLVER
MXPAND,,,,YES         ! EXPAND MODE
SOLVE
FINISH
```

Appendix-1e

Solid shaft immersed in fluid filled tabular space surrounded by infinite fluid,CASE-4, 3D, harmonically forced analysis

```
/BATCH,LIST
/VERIFY,EV129-1S
/PREP7
/TITLE,HARMONIC ANALYSIS
! Material type
ET,1,SOLID45        ! structural element
ET,2,FLUID30        ! acoustic fluid
element with ux & uy
ET,3,SOLID45         !Outer rigid cylinder
ET,4,FLUID30        ! acoustic fluid
element with ux & uy
ET,5,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
ET,6,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
! Material properties
MP,EX,1,2.08e11
MP,DENS,1, 8160
MP,NUXY,1,0
MP,DENS,2,1030
MP,SONC,2,1460
MP,EX,3,2.08e11
MP,DENS,3, 8160
MP,NUXY,3,0
MP,DENS,4,1030
MP,SONC,4,1460
! Geometry of the model
! Radius of the cylinder
r1=0.2125
r2=0.3555
r3=0.3755
r4=0.85
L=8.5
! Create inner and outer quarter circles
CYL4,0,0,0,0,r1,90
CYL4,0,0,r1,0,r2,90
CYL4,0,0,r2,0,r3,90
CYL4,0,0,r3,0,r4,90
! Extrude to required length
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,2
VEXT,ALL, , ,0,0,L
ESEL,ALL
```

```
NSEL,ALL
ASEL,S,AREA, ,3
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,4
VEXT,ALL, , ,0,0,L
ESEL,ALL
NSEL,ALL
! Select, assign attribute to and mesh area
1
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,16,1
LESIZE,3,,,16,1
LESIZE,2,,,16,1
LESIZE,19,,,100,1
LESIZE,20,,,100,1
LESIZE,21,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
! Select, assign attribute to and mesh area
2
VSEL,S,VOLU,,2
VATT,2,2,2,0
LESIZE,5,,,16,1
LESIZE,7,,,16,1
LESIZE,6,,,16,1
LESIZE,4,,,16,1
LESIZE,26,,,100,1
LESIZE,27,,,100,1
LESIZE,28,,,100,1
LESIZE,29,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,2
! Select, assign attribute to and mesh area
3
VSEL,S,VOLU,,3
VATT,3,3,3,0
LESIZE,8,,,16,1
LESIZE,9,,,2,1
LESIZE,10,,,16,1
LESIZE,11,,,2,1
LESIZE,34,,,100,1
LESIZE,35,,,100,1
LESIZE,36,,,100,1
LESIZE,37,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,3
! select, assign attribute to and mesh area
5
VSEL,S,VOLU,,4
VATT,4,4,4,0
LESIZE,12,,,16,1
LESIZE,14,,,16,1
LESIZE,13,,,16,1
LESIZE,15,,,16,1
LESIZE,42,,,100,1
LESIZE,43,,,100,1
LESIZE,44,,,100,1
LESIZE,45,,,100,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,4

! Reflect quarter circle into semicircle
about x-axis
nsym,x,100000,all    ! offset node number
by 1000
esym,,100000,all
! Reflect semicircle into full circle about
y-axis
nsym,y,200000,all ! offset node number by
2000
esym,,200000,all
NUMMRG,node      ! merge nodes only *use
KP*
! Modify inner fluid 2 layers of el29 into
type 4
esel,s,type,,1
nsle,s       ! Select those nodes attached
to the selected elements
esln,s,0     ! Select those elements
attached to the selected nodes
nsle,s       ! Select those nodes attached
to the selected elements
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,2
cmsel,u,tmp_ele  !select new set of
component of elements
nsle,s
emodif,all,type,5
esel,all
nsel,all
!Modify inner fluid 4 layers of el29 into
type 2
!Inner,
csys,1
NSEL,S,LOC,X,R2
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,5
cmsel,R,tmp_ele
nsle,s
emodif,all,type,2
esel,all
nsel,all
! Modify outer 6 layers of el29 into type 7
!outer
Csys,1
NSEL,S,LOC,X,R3
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,4
cmsel,u,tmp_ele
nsle,s
emodif,all,type,6
esel,all
nsel,all
! Flag interface as fluid-structure
interface 1
csys,1
nsel,s,loc,x,R1
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all

! flag interface as fluid-structure
interface 2
csys,1
nsel,s,loc,x,R2
esel,s,type,,2
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 3
csys,1
nsel,s,loc,x,R3
esel,s,type,,4
sf,all,fsi,1
nsel,all
esel,all
!Set pressure zero at the boundary
NSEL,S,LOC,X,R4
```

```
D,ALL,PRES,0.0    ! SET PRESSURE AT OUTER
RADIUS TO ZERO
nsel,all
esel,all
!Boundary conditions
NSEL,S,LOC,z,0
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,8.5
NSEL,R,LOC,X,0,r1
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,0
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,UX
NSEL,S,LOC,z,8.5
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,UX
nsel,all
esel,all
! enter solution module
/SOLU
ANTYPE,HARMIC      ! Harmonic response
analysis
HROPT,FULL         ! Full harmonic response
HROUT,OFF ! Print results as amplitudes and
phase angles
OUTPR,BASIC,1
NSUBST,40 ! 80 Intervals within freq. range
HARFRQ,7,20        ! Frequency range from 0
to 40 HZ
KBC,1             ! Step boundary condition
F,100583,FX,2000
SOLVE
FINISH
! Postprocessor
/POST26
NSOL,2,3751,U,X,2UX! Store UX Displacements
NSOL,3,54723,U,X,3UX
/GRID,1   ! Turn grid on
/AXLAB,Y,DISP      ! Y-axis label disp
PLVAR,2,3 ! Display variables 2 and 3
FINISH
```

APPENDIX -2

Appendix-2 presents the command used to model seawater OVBD system in ANSYS.

PART - 2

```
! Command list for finite element model analysis of seawater OVBD system using ANSYS
/BATCH,LIST
/PREP7
/TITLE,MODAL ANALYSIS
! Material type
ET,1,FLUID30  ! acoustic fluid element with
ux & uy in Pipe
ET,2,SOLID95  ! structural element (Pipe)
ET,3,FLUID30  !acoustic fluid element with
ux &uy sea water
ET,4,SOLID95  !Outer rigid cylinder
(Caisson)
ET,5,FLUID30  !acoustic fluid element with
ux & uy Ballast
ET,6,SOLID95  ! structural element (outer
structure)
ET,7,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
ET,8,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
ET,9,FLUID30,,1,0  ! acoustic fluid element
without ux & uy
! Material properties
MP,DENS,1,1030
MP,SONC,1,1460
MP,EX,2,11.1e9
MP,DENS,2, 1790
MP,PRXY,2,0.37
MP,DENS,3,1030
MP,SONC,3,1460
MP,EX,4,2.08e11
MP,DENS,4, 8160
MP,PRXY,4,0.3
MP,DENS,5,1030
MP,SONC,5,1460
MP,EX,6,2.08e11
MP,DENS,6, 8160
MP,NUXY,6,0
! Geometry definition (Radius of the
cylinder)
R1=0.357
R2=0.381
R3=0.4825
R4=0.508
R5=1.6  ! For infinite case take 1.6m which
is 3X
L1=32   ! Length(level of fluid in pipe)
L2=32   ! Length of pipe
L3=24   ! Length(Fluid level between pipe
and caisson)
L4=32   ! Length of caisson
L5=16   ! Length (ballast water level or
height)
! Create inner and outer quarter circles
CYL4,0,0,0,0,R1,90
CYL4,0,0,R1,0,R2,90
CYL4,0,0,R2,0,R3,90
CYL4,0,0,R3,0,R4,90
CYL4,0,0,R4,0,R5,90
! Extrude to required length
ASEL,s,AREA, ,1
VEXT,ALL, , ,0,0,L1
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,2
VEXT,ALL, , ,0,0,L2
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,3
VEXT,ALL, , ,0,0,L3
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,4
VEXT,ALL, , ,0,0,L4
ESEL,ALL
NSEL,ALL
ASEL,S,AREA, ,5
VEXT,ALL, , ,0,0,L5
ESEL,ALL
NSEL,ALL
! Select, assign attribute to and mesh area
1
VSEL,S,VOLU,,1
VATT,1,1,1,0
LESIZE,1,,,14,1
```

```
LESIZE,3,,,14,1
LESIZE,2,,,14,1
LESIZE,23,,,180,1
LESIZE,24,,,180,1
LESIZE,25,,,180,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,1
! Select, assign attribute to and mesh area
2
VSEL,S,VOLU,,2
VATT,2,2,2,0
LESIZE,5,,,2,1
LESIZE,7,,,2,1
LESIZE,6,,,14,1
LESIZE,4,,,14,1
LESIZE,30,,,180,1
LESIZE,31,,,180,1
LESIZE,32,,,180,1
LESIZE,33,,,180,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,2
! Select, assign attribute to and mesh area
3
VSEL,S,VOLU,,3
VATT,3,3,3,0
LESIZE,8,,,14,1
LESIZE,9,,,8,1
LESIZE,10,,,14,1
LESIZE,11,,,8,1
LESIZE,38,,,135,1
LESIZE,39,,,135,1
LESIZE,40,,,135,1
LESIZE,41,,,135,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,3
! Select, assign attribute to and mesh area
5
VSEL,S,VOLU,,4
VATT,4,4,4,0
LESIZE,12,,,14,1
LESIZE,14,,,14,1
LESIZE,13,,,2,1
LESIZE,15,,,2,1
LESIZE,46,,,180,1
LESIZE,47,,,180,1
LESIZE,48,,,180,1
LESIZE,49,,,180,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,4
! Select, assign attribute to and mesh area
5
VSEL,S,VOLU,,5
VATT,5,5,5,0
LESIZE,16,,,14,1
LESIZE,17,,,24,1
LESIZE,18,,,14,1
LESIZE,19,,,24,1
LESIZE,54,,,45,1
LESIZE,55,,,45,1
LESIZE,56,,,45,1
LESIZE,57,,,45,1
MSHKEY,1
MSHAPE,0,3D ! mapped quad mesh
VMESH,5
! Reflect quarter circle into semicircle
about x-axis
nsym,x,1000000,all    ! offset node number
by 1000000
esym,,1000000,all
! Reflect semicircle into full circle about
y-axis
nsym,y,2000000,all ! offset node number by
2000000
esym,,2000000,all
NUMMRG,node      ! merge nodes only
! Modify inner fluid 2 layers of el29 into
type 4
esel,s,type,,2
nsle,s       ! Select those nodes attached
to the selected elements
esln,s,0     ! Select those elements
attached to the selected nodes
nsle,s       ! Select those nodes attached
to the selected elements
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,1
cmsel,u,tmp_ele  !select new set of
component of elements
nsle,s
emodif,all,type,7
esel,all
nsel,all
!Modify inner fluid 4 layers of el29 into
type 2
!outer,
Csys,1
NSEL,S,LOC,X,R2
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,3
cmsel,u,tmp_ele
nsle,s
emodif,all,type,8
esel,all
nsel,all
!Modify inner fluid 4 layers of el29 into
type 2
!Inner,
csys,1
NSEL,S,LOC,X,R3
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,8
cmsel,R,tmp_ele
nsle,s
emodif,all,type,3
esel,all
nsel,all
! Modify outer 6 layers of el29 into type 7
!outer
Csys,1
NSEL,S,LOC,X,R4
Esln,s,0
Nsle,s
cm,tmp_ele,elem ! Create component of
elements
esel,s,type,,5
cmsel,u,tmp_ele
nsle,s
emodif,all,type,9
esel,all
nsel,all
! Flag interface as fluid-structure
interface 1
csys,1
nsel,s,loc,x,R1
esel,s,type,,1
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 2
csys,1
nsel,s,loc,x,R2
esel,s,type,,3
```

```
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 3
csys,1
nsel,s,loc,x,R3
esel,s,type,,3
sf,all,fsi,1
nsel,all
esel,all
! Flag interface as fluid-structure
interface 4
csys,1
nsel,s,loc,x,R4
esel,s,type,,5
sf,all,fsi,1
nsel,all
esel,all
!Set pressure zero at the boundary
NSEL,S,LOC,X,R5
D,ALL,PRES,0.0    ! SET PRESSURE AT OUTER
RADIUS TO ZERO
nsel,all
esel,all
!Free surface of ballast water,
NSEL,S,LOC,z,L5
NSEL,R,LOC,X,r4,r5
NSEL,R,LOC,Y,0,360
D,ALL,PRES,0.0
nsel,all
esel,all
! Free surface of sea water
NSEL,S,LOC,z,L3
NSEL,R,LOC,X,r2,r3
NSEL,R,LOC,Y,0,360
D,ALL,PRES,0.0
nsel,all
esel,all
!Boundary conditions-1 for PIPE
NSEL,S,LOC,z,0
NSEL,R,LOC,X,r1,r2
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
NSEL,S,LOC,z,L2
NSEL,R,LOC,X,r1,r2
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
!Boundary conditions-2 for CAISSON
NSEL,S,LOC,z,0
NSEL,R,LOC,X,r3,r4
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
NSEL,S,LOC,z,L4
NSEL,R,LOC,X,r3,r4
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
!Support at different levels
NSEL,S,LOC,z,14.8
NSEL,R,LOC,X,r3,r4
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
NSEL,S,LOC,z,22.9
NSEL,R,LOC,X,r3,r4
NSEL,R,LOC,Y,0,360
D,ALL,UX,UY
nsel,all
esel,all
! enter solution module
/SOLUTION
ANTYPE,MODAL          ! MODAL ANALYSIS
MODOPT,UNSYM,40,0,200 ! UNSYMMETRIC MATRIX
SOLVER
MXPAND,,,,YES         ! EXPAND MODE
SOLVE
FINISH
```