Chapter 1

General Introduction

1.1 Preamble

Sound is present all around us in different forms. We experience some form of sound right from the moment we get up to the time we go off to sleep. We continue to experience sound even when asleep. What is remarkable of this form of energy is that we cannot protect ourselves from it naturally. If we do not wish to see a thing, we can close our eyes as we do so while sleeping, but we cannot do the same with sound as we continue to hear it even without being aware of it. Nature has given our visual capabilities limitations, wherein we cannot see something behind us but we can surely hear it. Sound happens because of quickly varying pressure wave within a medium due to a vibrating structure or an unsteady aerodynamic flow. The sound so produced may or may not be audible. Noise, on the other hand is the unwanted and undesirable product of this radiated sound.

A ship at sea or an aircraft in flight responds to the surface forces applied by the relative motion of the fluid they are traveling in. This response is in the form of vibration which can get converted to sound if the structure can impart kinetic energy to the surrounding fluid as pressure waves. These propagating pressure waves are audible as sound if their speed is greater than the speed of sound. Similarly, structures such as a floating airport,
a bridge, guideway, overhead crane, cableways, rails or roadways; are static while being subject to a moving load unlike a ship or an aircraft, where the structure is moving and the surface force is at rest. Advances in speed and weight of vehicles in all branches of transport in the last few decades have resulted into increased vibrations and hence increased noise levels. It has thus become essential to study effect of moving load on such transportation structures modeled as continuously supported beams and plates. May it be (a) a moving structure in static fluid; (b) a moving force over a static structure; or (c) both the force and the structure moving; the resultant effect is vibration, hence sound. Keeping in mind this thought process, the effect of moving load on the sound power is studied. Some areas where this nature of study has relevance are that of ships moving in water, floating ladder tracks, floating airports, tyres and transportation structures to name a few.

1.2 Prelude

The ocean is a noisy place. There are many sources of sound, and sound travels efficiently in water. Natural ocean sounds are produced by wind, waves, precipitation, cracking ice, seismic events, and marine organisms. The hearing ability of marine mammals has evolved to deal with these natural sounds of the ocean. Since the advent of the industrial age, sounds made by human beings have combined with natural ocean sounds, resulting in elevated noise levels, primarily in the frequency region below 1kHz and is assumed to affect the ability of marine mammals to communicate and to receive information about their environment. Such noise may interfere with or mask the sounds used and produced by these animals and thereby interfere with their natural behavior. Higher levels of human-made sounds can cause obvious disruptions; they may frighten, annoy, or distract the animals and lead to physiological and behavioral disturbances. They can cause reactions that might include disruption of marine mammals’ normal activities and, in some cases, short- or long-term displacement from areas important for feeding or reproduction. They may also disturb the species such as fishes, squids, and crustaceans upon which the marine mammals prey. At still higher levels, human-made
Fig. 1.1 Hearing curves for select teleost fishes (Popper (2008))

noise may cause temporary or permanent hearing impairment in marine mammals. Such impairment would have the potential to diminish the individual’s chances for survival. At greater range the underwater sound wave may not directly injure animals, but has the potential to cause behavioural disturbance or even physical or physiological disturbance. Hearing thresholds have been determined for perhaps 100 of the more than 29,000 living fish species (Figure 1.1). These studies show that, with few exceptions, fish are sensitive to low frequency sound below 1 kHz, and that marine mammals are very sensitive to sound of frequencies above 1 kHz.

1.3 Motivation

Warships, Figure 1.2, might look like all-powerful vessels but they are also highly vulnerable to being spotted by the enemy. The radiated noise signature of a warship is composed of a variety of noises; those generated by running machinery, sensor systems, crew activity, by the hull moving through the water (pressure signature) and by the propeller (such as cavitation and blade rate). The output of these noise sources defines the global level of the radiated signature and of any discrete components within it. The level of the radiated acoustic signature is therefore a critical parameter in the operational success of any warship. It is this signature that will affect the means by which, and the
range at which, the vessel may be detected, classified, tracked and engaged. Since the acoustic waves can propagate in the sea 4-5 times faster than in air and without substantial loss, any source of sound from a vessel gets detected in the sea more conveniently. One component of this noise signature is due to the ship’s movement; the relative motion of the ship and the sea water being the source. This relative motion between the ship and the sea water may be conceptualised as a *moving load on a static structure.*

**Very Large Floating Structures** (VLFS in short), Figure 1.3, with typical di-

mensions of 5 km long, 1 km wide, and only a few meters deep can be used to create
floating airports, bridges, breakwaters, piers and docks, storage facilities, wind and solar power plants, military stations, industrial space, emergency bases, entertainment facilities, recreation parks, mobile offshore structures and even for habitation and may be located near the coastline or in the open sea. Because of their relatively simple construction and ease of maintenance, pontoon-type VLFS (which just floats on the sea surface) are used as a floating airport or runway. They are constructed particularly in calm waters, often inside a cove or a lagoon and near the shoreline, to minimize ocean effects on them, where marine life is abundant in variety and quantity. This marine life is effected undesirably by noise pollution as confirmed by recent studies. Having a floating airport, increases the noise pollution in these sheltered areas due to activities such as movement of equipment, people, cargo (dry and liquid), variable ballast (dry and liquid), aircraft landing, crane handling, berthing and docking, connection and disconnection, running machinery onboard etc. Most of these sounds can be reduced or controlled, and have been studied independent of the VLFS. However, sound radiation due to an airplane taking off / landing has not been been reported in the literatures available.

A modern ladder rail track design, Figures 1.4 and 1.5, where steel rails are fixed onto successive ladder-like sections of two parallel longitudinal reinforced concrete sleepers up to 15 m long, when mounted upon discrete resilient supports on a concrete bed are called

Fig. 1.4 Train on a floating ladder track (Wikipedia)
as **floating ladders**. These have significant advantages which include robust structure, reduced settlement and much less vibration and noise. Floating ladder tracks are in use in Japan and with trains touching speeds of an upward of 500 km/h, the study of the effect of moving loads (in the form of fast moving trains) on floating ladders takes importance.

The subject of moving load problem becomes increasingly important owing to its broad applications in the **transportation industry** as seen in Figures 1.6 and 1.7. For instance, in recent years high-speed trains and automobiles are actively promoted for future surface transportation. At low speed, the dynamic effect of moving load to structures such as railways, highways, and airfields is insignificant. However, when vehicle speed increases, moving load effect can no longer be ignored and more sophisticated models of vehicle-structure interaction need to be considered. The study of the beam response to moving load also provides basis for vehicle detection, classification and weight-in-motion, structural health monitoring, and non-destructive evaluation.

Apart from these areas as discussed above, the relevance of moving load on structures has it’s importance in various other fields such as guideway, overhead crane or cableways. In
1.3 Motivation

Fig. 1.6 The fast moving transportation

Fig. 1.7 Modern transports on bridges (flickr.com)

some of these fields; the structure is static while being subject to a moving load; while in others such as a ship, the structure is moving and the surface force is at rest. May it be a moving structure in static fluid; a moving force over a static structure or both the force and the structure moving; the resultant effect is vibrations that result into sound. Such requirements have resulted in numerous investigations of moving load on continuously supported beams and plates but limited to vibrations. It is this lack of work in the study
of sound due to a moving load that the present study is motivated.

1.4 Literature Survey

Governments up until the 1970s viewed noise as a “nuisance” rather than an environmental problem. With an increasing awareness of “noise health effects” the science of sound (called acoustics) has developed increasingly in importance. Acoustics is an interdisciplinary science that deals with the study of all mechanical waves (in gases, liquids or solids) including vibration, sound, ultrasound and infrasound. The study of acoustics revolves around the cause, generation, propagation, reception and effect of mechanical waves and vibrations. These steps are the same for all problems may it be an earthquake, a rock band or a submarine using a sonar. The cause or the initiating force may be natural and/or man made while the generation process of acoustic energy, can be of various types. There is one fundamental equation that describes acoustic wave propagation which transfers energy through the propagating medium and is received by the receiver. The final effect may be purely physical or it may reach far into the biological or volitional domains.

Traditionally, noise problems have been identified and solved in a trial and error manner. The numerical analysis has invariably been relegated to a supporting role, and numerical results have often been greeted with skepticism. For large structures such as a ship or a floating airport, the trial and error methodology may not be a suitable means of study, owing to their size. Hence the numerical method for analysing sound from such large structures is considered as an acceptable norm. Since the sound produced by structures has been studied from the time of Rayleigh (1896), the available literature is vast and varied. We shall review in brief the work undertaken for beams and plates causing forced vibrations, the Very Large Floating structures, acoustics due to moving loads, mean flow, inplane loading and effect of noise on marine life.
1.4.1 Structural Geometry

Simple geometrical structures idealized as a membrane, a thin elastic plate or a thick elastic plate (with Timoshenko-Mindlin equation of motion) which are of infinite extent and are homogeneous have been studied extensively. This is done not only due to the frequent occurrences of these structures in several engineering disciplines, but also to simplify the mathematical structure of the governing equations compared with a full three-dimensional equation. The structure is assumed to have the same static inviscid fluid on both sides, or a vacuum on one side. The exciting fields such as a point source, a concentrated mechanical line force and moment or point-force have been studied. Owing to the available efficient analytical techniques, the results have been limited to very distant structural and fluid wave fields which are quantities of physical interest. Results for a membrane covering various distances from the excitation and frequencies have been given by Crighton (1983).

1.4.2 Forced Vibration

A “moving load”, which is the cause in the present study, makes structural analysis very difficult since the interacting force varies in time and space. Not withstanding the associated difficulties, the study of such moving load on structures has been of interest to researchers both theoretically and experimentally since the first railway bridge was built in the early 19th century. The investigations have hence resulted in a large number of publications. We shall break up our survey so as to limit it to beams and plates.

1.4.2.1 Beam

Theoretically, the problem of moving load was first tackled for a case in which the beam mass was considered small against the mass of a single, constant load. The original approximate solution is due to Willis et al. (1851), one of the early experimenters in the field. Since then these problems has become more dynamic in character mainly due to the increased vehicle speed and structural flexibility. Frýba (1999), while documenting a large number of related cases used the Fourier sine (finite) integral transformation and the
of the vehicle. Both Kargarnovin and Younesian (2004) and Lee (1998) considered the
Timoshenko beam on an elastic foundation subject to moving loads, by analytical ap-
proaches. Kim (2004) studied axially loaded beams on elastic foundations subject to
harmonic moving loads. The linear finite element analysis was applied to analyze a sim-
ply supported Euler-Bernoulli beam under the act of moving loads by Ju et al. (2006) and
Kidarsa et al. (2008). In Kargarnovin et al. (2005) and Younesian et al. (2006), nonlinear
dynamical behavior of Timoshenko beams with infinite length under the act of traveling
harmonic loads resting on viscoelastic foundations are studied. Dynamic response of an
inclined Euler-Bernoulli beam subjected to a moving mass has been investigated in Wu
(2005) using the linear finite element method considering transversal and longitudinal
displacements for the beam. In Yanmeni Wayou et al. (2004), the nonlinear vibration of
a horizontal pinned-pinned thin beam under the act of a moving mass considering the
influence of the load inertia and the nonlinearity caused by stretching effect of mid-plane
of the beam due to the immovable supports was studied. Recently, the nonlinear dynamic
analysis of an eccentrically prestressed beam under the act of a concentrated moving har-
monic load including damping effect has been studied in Simsek and Kocaturk (2009).
In that paper, the nonlinear deflections behavior of the beam is approximated by some
polynomial functions and material of the beam was assumed to follow the Kelvin-Voigt
model. Furthermore, the effects of large deflections, the internal damping of the beam,
the velocity of the moving harmonic load, the prestress load, and the excitation frequency
were discussed. Mehri et al. (2009) using a dynamic green function, presented the lin-
er dynamic response of uniform beams with different boundary conditions excited by a
moving load, based on the Euler-Bernoulli beam theory. An exact and direct modeling
technique is presented with various boundary conditions, subjected to a load moving at
a constant speed. Influence of variation in speed on the dynamic response was studied
and the results given in graphical and tabular form. Dehestani et al. (2009) Presented
an analytical-numerical method which can be used to determine the dynamic response
of beams carrying a moving mass, with various boundary conditions. Results illustrated
that the speed of a moving mass has direct influence on the entire structural dynamic
response, depending on its boundary conditions.
As can be seen, over the years many investigators have used different methods for solving the forced vibration problem. The most used method for determining these vibrations is the eigenfunctions method wherein the applied loads and the dynamic responses are expanded in terms of the undamped beams. This method gives the solution as an infinite series, which is truncated after a number of terms and an approximate solution obtained.

1.4.2.2 Plate

The dynamic response of rectangular plates to uniformly moving loads has been subjected to many investigations. Solutions have been presented by Holl (1950) and Piszczek (1959), Nowacki (1963) showing that a critical velocity existed for each vibrational mode. Steady state solutions of an infinite plate carrying a uniformly moving load were given by Livesley (1953), Reismann (1959) and Morley (1962). Gbadeyan and Oni (1994) and Fryba (1999) investigated the dynamic behaviour of beams and plates subjected to moving forces and moving masses. Manoach (1993) studied elastoplastic thick circular plates subjected to different types of pulses by using the Mindlin plate theory. Bert et al. (1994) used the differential quadrature method. Rossi et al. (2001) studied the forced vibration responses of a rectangular plate subjected to a stationary distributed harmonic loading. Zhu and Law (2000) and Marchesiello et al. (1999) studied the dynamic behaviour of bridges by considering the structure to be composed of rectangular plates. Shadnam et al. (2001a) considered a single force (or mass) moving along an arbitrary trajectory; however, they studied a straight path problem by means of analytical and numerical approach. The nonlinear plate version of the same problem was considered by Takabatake (1998), using a Galerkin approach, but with discontinuously changing plate thickness and later by Shadnam et al. (2001b). Wu et al. (2001) used the FEM and the Newmark direct integration method. Composite plates were considered by Lee and Yhim (2004) using finite elements and third-order plate theory and later by Au and Wang (2005) computed sound radiated from a composite plate due to moving loads. Kim et al. (2005) used a finite element model for a bridge while Li et al. (2005) extended the work by taking into account the wind loads on the vehicle. The bridge-vehicle interaction using a Lagrangian
approach was studied by Yagiz and Sakman (2006). The literature cited above are some of the main contributors who have undertaken the study of moving loads on plates and cannot be considered as complete.

1.4.3 Very Large Floating Structure

Many developed countries with long coastlines for want of land space have successfully reclaimed land from sea. Such activities have a negative ecological impact on the coastline surrounding the reclamation. In response to this requirement researchers and engineers proposed the construction of Very Large Floating Structures (VLFS in short) with typical dimensions of 5 km long, 1 km wide, and only a few meters deep. These structures can be constructed to create floating airports, bridges, breakwaters, piers and docks, storage facilities, wind and solar power plants, military stations, industrial space, emergency bases, entertainment facilities, recreation parks, mobile offshore structures and even for habitation and may be located near the coastline or in the open sea. Because of their relatively simple construction and ease of maintenance, pontoon-type VLFS (which just floats on the sea surface) are used for a floating airport or runway. They are constructed particularly in calm waters, often inside a cove or a lagoon and near the shoreline, to minimize ocean effects on them, wherein marine life is abundant in variety and quantity. Recent studies by marine biologists have confirmed undesirable effects of noise pollution on marine life. Having a floating airport, compounds the noise pollution in these sheltered areas due to activities such as movement of equipment, people, cargo (dry and liquid), variable ballast (dry and liquid), aircraft landing, crane handling, berthing and docking, connection and disconnection, running machinery onboard etc. Most of these sounds can be reduced or controlled, and have been studied independent of the VLFS over the years. However, the sound radiation by the VLFS when subjected to an airplane landing / taking off and efforts of reduction to protect the marine life from noise pollution, has not been studied.

The difficulties in solving this transient dynamic problem has resulted in only a few simplified studies to have been reported to date. Using a finite element (FE) program, Watanabe and Utsunomiya (1996) presented the numerical results for elastic responses
due to prescribed impulsive loading on a circular very large floating structure excited by impulsive loading. Kim and Webster (1998) and Yeung and Kim (1998) studied transient phenomena of an infinite elastic runway using a double Fourier transform approach. The former studied the added drag caused by the flexibility of the runway while the latter focused on the resonance phenomenon caused by the accumulation of energy near the moving load. Ohmatsu (1998) developed another kind of time-domain analysis method in which the structural response is obtained from the convolution integral of the frequency response function and impulse response function. Endo and Yago (1999) adopted a FE scheme and Wilson-θ method to investigate the transient behavior of an airplane taking off from and landing on a VLFS in rough sea conditions using a triangle time impulse load applied at the nodes of the structure to represent the loads introduced by the weight of the airplane. Endo (2000) calculated the behaviour of a VLFS and airplane during takeoff / landing run in wave condition allowing for the effects of hydroelasticity. Lee and Choi (2003) developed a FE-BE hybrid method to analyze transient hydroelastic response of VLFS. Kashiwagi (2004) presented the transient elastic deformation of a pontoon-type VLFS caused by the landing and takeoff of an airplane based on the mode superposition method using realistic numerical data from a Boeing 747-400 jumbo jet. Fleischer and Park (2004) used the modal analysis with Fourier series to solve the plane hydroelastics of a beam due to uniformly moving one-axle vehicle. Kyoung et al. (2006) developed a finite element method for the time-domain analysis of the hydroelastic deformation of a pontoon-type very large floating structure with fully nonlinear free-surface conditions. Jin and Xing (2007) proposed a mixed mode function-boundary element method to solve the dynamic responses of a floating beam excited by landing loads. Liu-chao and Liu (2007) proposed a time-domain finite element procedure to analyze the transient hydroelastic responses of VLFS subjected to dynamic loads. Liu-chao (2009) proposed a time-domain finite element model to analyze the fluid-structure interactive dynamical system. He validated the proposed approach by comparing the existing experimental and numerical results with the results obtained.

*The present study concentrates largely on the sound generated by such floating*
structures. So far no study of this nature has been reported in the literatures available.

1.4.4 Acoustics

The literature on analytical methods in acoustic radiation is vast and chronologically ranges from the late 19th century with the work of Rayleigh (1896), essentially defining the science of acoustics. Inherently the problem of acoustic radiation includes the problem of vibrations of the structures under consideration. The most important theories and analytical methods can be found in the classical acoustic textbooks Junger and Feit (1986), Fahy and Gardonio (2007), Pierce (1989), Cremer et al. (1988). Techniques for dealing with fully coupled motions of elastic plates and shells immersed in air or water were simply not available in Rayleigh’s time, but have become available in the past three decades or so.

1.4.4.1 Static load

Early investigations of sound radiation from a force excited, elastic, fluid-loaded plate by Gutin (1965), Maidanik and Kerwin (1966) and Feit (1967) were primarily concerned with the far field pressure and power radiated in to the acoustic medium. Ranganath Nayak (1970), using the Fourier integral representation of the solution was able to numerically evaluate the integral representation for the velocity response to determine the drive line admittance for a line-driven plate. Crighton in a series of papers Crighton (1972, 1977, 1979, 1983) had analysed both the near- and far field response of locally excited plates. Crighton’s results although probably the most complete to date in this field, are somewhat difficult to visualize due to the complexity of the problem and that they have not been displayed in a graphical form.

1.4.4.2 Moving Load

Sound radiation by beams due to moving loads is however a relatively newer area of interest. Keltie and Peng (1988) investigated the sound radiation from a fluid-loaded beam subject to a moving harmonic line force. Their study was the first of it’s kind
and showed some important characteristics of structures subject to a moving load. The results showed that for beams under light fluid loading, the coincidence sound radiation peak for a stationary force is split into two coincidence peaks due to the effects of the Doppler shift, while for beams under heavy fluid loading there are no pronounced sound radiation peaks. Following the study of Keltie and Peng (1988), Cheng (1999) formulated the vibration response of periodically simply supported beam on the whole structure in wavenumber domain through Fourier transform. This problem was an advance on traditional substructure methods. For an air-loaded beam subjected to a stationary line force, they showed that the radiated sound power exhibited peaks at certain wavenumber ratios. The wavenumber ratios at which radiation peaks occur nearly coincide with the lower bounding wavenumber ratios of the odd number of propagation zones. However, Cheng’s formulation did not include the presence of numerous wavenumber components induced from the elastic supports and is subject to the restriction that the external force is located on one of the elastic supports. Cheng et al. (2000, 2001) introduced a “wavenumber harmonic series” to discuss the vibro-acoustic response when a fluid-loaded beam on periodic elastic supports is subjected to a moving load. Results show that the response of a beam on an elastic foundation can be approximated using a periodically, elastically supported beam when the support spacing is small compared with the flexural wavelength. For such beams when the force is stationary a single radiation peak occurs which splits into two peaks due to Doppler shift when the force becomes traveling. Au and Wang (2005) studied the vibrations of a rectangular orthotropic thin plate with general boundary conditions traversed by moving loads. The effects of light and heavy moving loads were separately studied. Based on the Rayleigh integral and the analytical dynamic response of the plate, the acoustic pressure distributions around the plate were obtained in the time domain. It was concluded that the boundary conditions affect the acoustic pressure generated by moving loads. It was observed that the stiffer the plate, the higher the structural frequencies, and the larger the sound pressure caused by moving loads.

To solve the forced response of rectangular plates with elastic support, Yufeng and Qibai
(2009) adopted approximate solutions in conjunction with numerical methods. They investigated the effect of the moving force on the radiated sound from the rectangular plate with elastic support and concluded that the speed of the moving force can change the sound radiated from the plate with elastic support to become flat in special frequency range.

1.4.5 Mean Flow

The literature on fluid-structure interaction in the presence of fluid flow is relatively scarce. This is not because fluid flow is unimportant, but because problems often get too complicated when relative motion between an elastic structure and a surrounding fluid medium is involved. This is especially true when the dimension of a structure is finite, and the compressibility and viscosity of the fluid medium must be considered. One way of obtaining an understanding of the physics behind this type of fluid-structure interaction is to examine a simplified version of the problem in which the dimension of a structure extends to infinity, the fluid moves at a constant (mean) speed, and the effects of fluid compressibility and viscosity are neglected. Brazier-Smith and Scott (1984) present a numerical study of such a problem. Specifically, they study the dynamic response and instability of an infinite elastic plate in the presence of mean flow based on a causality analysis. Brazier-Smith and Scott’s numerical analysis is supported by Crighton and Oswell (1991) analytical work based on an asymptotic expansion analysis. In particular, Crighton and Oswell (1991) shed some light on the mechanism of the onset of convective instability. The mechanism of the onset of absolute instability was uncovered by Wu and Maestrello (1995). Further, Wu and Zhu (1995) show that the amplitude of the plate response is always bounded by the structural nonlinearities and instabilities depicted by Brazier-Smith and Scott (1984) never really occur.

1.4.6 Inplane loading

The first theoretical examination of plate under uniform compression was by Bryan (1891), who obtained a solution to the problem in 1891. Since then numerous researchers
have investigated local instability in plates under a wide variety of loading and boundary conditions using many different methods of analysis. There has been a number of excellent textbooks such as Timoshenko and Gere (2010) and Bulson (1970) that have described the main results of these investigations. An area analogous to floating airports is the study of ice packs wherein existence of compressive stresses has been long recognized in ship design for the Arctic. A well known example of a research vessel designed to withstand ice pack compression is the polar research ship Fram that was designed by Colin Archer for Fritjof Nansen in the 19th century. The effects of mean compression on wave propagation in the ice pack were recognized by Mollo-Christensen (1983b), Mollo-Christensen (1983a). He found that edge waves can have very low group velocity, and suggested it as an explanation of ice ride-up on shore. Schulkes et al. (1987) investigated the flexural gravity wave pattern excited on a floating ice plate by including the effect of compressive stress in the plane of the plate, uniform flow and stratification of the underlying water. Their work was followed by Liu and Mollo-Christensen (1988), who showed that wave energy can be concentrated because of pack compression through two very different mechanisms, namely a very small group velocity caused by high compressive stress and the increased instability of nonlinear modulations also caused by pack compression.

1.4.7 Effect of noise on Marine Life

Hearing in fishes was first demonstrated in cyprinids as early as 1903. Quantitative work on the range of frequencies over which fish hear and on representatives of several different families was later carried out by von Frisch and his colleagues in 1936. Since then most investigations on sound detection in fishes, have been performed under quiet laboratory conditions. Hearing thresholds have been determined for perhaps 100 of the more than 29,0009 living fish species. (see Fay (1988); Popper (2003); Ladich and Popper (2004); Nedwell et al. (2004) for data on hearing thresholds). These studies show that, with few exceptions, fish cannot hear sounds above about 3-4 kHz, and that the majority of species are only able to detect sounds to 1 kHz or even below. The following are the major effects of increased ambient noise on marine life:
1.4.7.1 Masking

Increased ambient noise causes masking of biologically significant sounds resulting into interference with clear reception of signals of interest. This disrupts activities such as breeding, navigation and communication.

1.4.7.2 Behavioural effect

Increased noise affects the behavior of marine mammals ranging from subtle to severe such that reactions to noise may range from a shift in orientation toward a sound source, to an escape or flight response.

1.4.7.3 Physiological effect

Noise can result in a range of physiological effects on marine mammals. Long-term noise exposure may cause stress responses in a manner similar to humans who live near busy highways or airports.

1.4.8 Parallel studies

A major motivation for the study of a moving load on a flexible beam or plate has been its application to transport systems (rail tracks, roads or runways), originally in temperate lands and subsequently in cold regions, where, in particular, floating ice sheets may be exploited. Acoustic pressure field radiated from a vibrating structures subjected to a moving load have been studied in various fields such as:

- **Road-bridge noise problems**: In recent years high-speed trains and automobiles are actively promoted for future surface transportation. At low speed, the dynamic effect of moving load to structures such as railways, highways, and airfields is insignificant. However, when vehicle speed increases, moving load effect can no longer be ignored and more sophisticated models of vehicle-structure interaction need to be considered. Studies devoted to vehicle-structure interaction include those by Frýba (1999); Sun (1996); Cebon (1999); Sun and Greenberg (2000); Deng and Sun (2000).
• **Vibration of railway bridges traversed by high speed trains**: A modern ladder rail track design, produces much less vibration and noise if it is mounted upon discrete resilient supports on a concrete bed, when it is called a floating ladder track. Several studies have been undertaken to reduce the sound generated by the fast moving trains on these tracks. (see Wilson (2004); Hosking and Milinazzo (2007); Yuan et al. (2009); Asanuma and Wakui (2012), Hosking and Milinazzo (2012)).

• **Floating Ice**: Ice sheets are used for ice roads, ice bridges, construction platforms, airstrips and recreational activities. It hence becomes very important to know when the ice is safe to use. Several major theoretical and experimental studies have been undertaken to understand the effect of aircraft and vehicle operation on floating ice (see Davys et al. (1985); Schulkes and Sneyd (1988); Milinazzo et al. (1995); Yeung and Kim (2000); Wang et al. (2004); Squire (2007)).

• **Sound Radiation from Tyres**: In modern times, ride improvement and quietness of vehicle are demanded due to growing concerns over environment. This has forced detailed study in tire performance. In this regards experimental work was initiated in 1966 followed by theoretical work (see Keltie (1982); Heckl (1985); Kim et al. (2006); Kropp (2011); Kropp et al. (2012)).

### 1.4.9 Conclusion

From the literature review undertaken above, one realizes that the structures can be idealized as Membrane, a thin elastic plate, a thick elastic plate or as a cylinder. Similarly the force can be applied as a point force, line force or a moment. The structure may have simple inhomogeneities such as ribs, supports or abrupt thickness change. The fluid present may be the same on both sides or may have vacuum on one side.

Keeping the range of possibilities and those already studied by others, we shall look at

• An idealized beam or plate as the floating structure.
• A line or distributed harmonic moving load.

• Consider no inhomogeneities.

• One side fluid (water) and other as vacuum.

1.5 Objectives

The main objectives of the present study are:

• For a ship (modeled as a floating beam) defined by a Timoshenko beam model, a Rayleigh beam model, a Shear beam model and an Euler-Bernoulli beam model, develop an analytical formulation for total sound power due to a moving harmonic load.

• Perform numerical analysis for the ship model for
  
  – Various beam models and compare the total sound power generated.
  
  – Timoshenko beam model to analyze effect of loss factor on total sound power.

• For a floating airport (modeled as a Timoshenko-Mindlin plate) subjected to the landing / takeoff of an airplane (modeled as a moving harmonic load)

  – Develop an analytical formulation for total sound power and structural response

  – Extend the analytical formulation to include effect of current.

  – Extend the analytical formulation to include effect of inplane loading.

• Perform numerical analysis for the floating airport model to

  – Study the total sound power generated.

  – Study effect of current on total sound power.

  – Study effect of inplane loading on total sound power.

  – Study the structural response due to harmonic and point moving load.
• Develop a **Graphical User Interface** as a tool for the user to obtain total sound power for a *ship* and a *floating airport* with different combinations of variables.

**Sound power level** is related to the amount of acoustical energy produced by a sound source and does not take into account its surroundings (unlike sound power level which takes into account the surroundings). If an object is rated at some sound power level, it means that that is the amount of power it is capable of radiating. When attempting to measure sound power level, an engineer will find that he cannot measure sound power emitted by the source. Instead, **sound power level** is calculated from several sound pressure measurements created by a source in a particular test environment using one of four common methods: **free-field**, **reverberation room**, **progressive wave (induct)**, and **sound intensity**. Once the sound pressure level is measured, the sound power level can then be determined mathematically.

Since the only accurate sound data an engineer / designer can provide is expressed as sound power, *the same would be calculated in this study* (by measuring sound intensity and then calculating sound power mathematically) to provide a common reference measurement that is independent of distance and acoustical conditions of the environment.

Since this type of study has been undertaken earlier in a limited form, the results obtained based on the application of the derived expression shall be analyzed based on the studies undertaken by various researchers and logic. The developed model has been validated with published results and discussed in section 5.2. The numerical results using the above formulation for different conditions shall be discussed. Application of the developed expression shall provide a first hand approximation of the sound power expected from a moving load on a structure floating on water.

### 1.6 Brief Outline of the Work Pursued in the Thesis

The content of the thesis is divided into *six* chapters for ease of explanation and the problems investigated.
Chapter 1, gives a **basic introduction and the motivation** behind the present study. The available literature relevant to the present study is reviewed thoroughly followed by a brief introduction of the research work pursued in this thesis. The basic physical equations, boundary conditions associated with acoustic problems associated with fluids and moving loads and the preliminary mathematical tools relevant to the thesis are discussed.

In Chapter 2, the generalized expression for the total sound power due to a **moving load on a ship** (modeled as a beam) as given by Keltie and Peng (1988) is formulated in detail for the various beam types, viz. Rayleigh beam, Shear beam and the Euler-Bernoulli beam.

Studies of sound generated from **floating airfields** due to the traveling load of starting, landing or taxiing planes is a natural extension of the ship (modeled as a floating beam) studied in the previous chapter. A dynamic analysis of a three-dimensional runway with time varying loading during landing / take-off however would be exceeding difficult. In Chapter 3, this analysis is made simpler by assuming that the runway behaves as a simple, infinitely long beam floating on water and supported by buoyancy. The model is assumed to be a one dimensional plate, described by the Timoshenko-Mindlin plate equation. The understanding of radiated sound power as established in chapter 2 has been extended to model a **floating airport**. The sound generated and platform response in frequency domain by the landing / taking-off of an airplane from such an airport, which would be akin to a moving load, has been analysed in section 3.2 of the chapter. Acoustic analysis in the presence of a **mean flow or current** complicates the analysis of a floating airport by modifying the effect of the moving load. The effect of mean flow on the response of a fluid-loaded structure has been studied in section 3.3. Even though a VLFS is structurally very long, the longitudinal strength does not play an important role in their design. The most severe type of loading for the bottom plate occurs when the structure is subjected to the combined action of uniformly distributed hydrostatic lateral loading and compression due to hogging. Similarly for the deck plate, maximum loading
occurs when the structure is subjected to compression and tension due to sagging and hogging respectively. The inplane loading plays an important role for such structures during berthing, plate connections at ends, initial deformation and corrosion to name a few and hence needs to be accounted for. This effect of inplane loading has been studied in section 3.4 by extending the formulation developed in section 3.2.

Having developed the general expressions for the ship (in chapter 2) and floating airport (in chapter 3), one needs to validate the model. This is done by using the published results for a Timoshenko beam by Keltie and Peng (1988). Once the model has been validated, a Graphical User Interface for undertaking numerical calculations using these mathematical formulations is developed and discussed in Chapter 4. The GUI has been generated to make the calculation procedure user friendly and speed up the user’s work especially for non-technical people.

The GUI developed in chapter 4 has been used to undertake numerical analysis to understand the total sound power radiated due to a moving load on a ship (modeled as a beam) and a floating airport (modeled as a plate) in Chapter 5. Using the beam model to represent the ship, we first analyse the total sound power produced by a Timoshenko beam, a Rayleigh beam, a Shear beam and an Euler-Bernoulli beam. We then compare them to understand which beam type produces the maximum sound power and the reasons associated with it in section 5.5. This is followed by the calculation and analysis of the total sound power from a Timoshenko beam due to varying loss factor in section 5.6. Having analysed the beam model, the plate model is used to undertake the numerical analysis of the total sound power from a floating airport due to a landing / taking off of an airplane. The analysis is carried out for aluminium and steel and the effect of the material on the total sound power is studied in section 5.7. The same plate model is then extended by modifying the governing equation Equation (3.3) to Equation (3.21) to incorporate the effect of mean flow and the numerical results of the total sound power are obtained and analysed in section 5.8. Another extension is obtained by incorporating the inplane loading to the plate model to get Equation (3.35). This is analysed nu-
merically next in section 5.9 to understand the effect of compressive and tensile inplane loads on the floating airport. The numerical analysis terminates with the **structural response** of the floating airport due to airplane landing / taking off modeled as a point and a harmonic moving load in section 5.10.

Finally, *Chapter 6*, summarizes the work done in the thesis followed by the future scope of research. Major contributions made in the thesis are also highlighted in this chapter.

Additional information used and derived is enumerated in the Appendices for clarity of the methods described in the chapters and as a starting point for future researchers working in this area.

- Appendix A: Time Averages of Products.
- Appendix B: Detailed derivation for the non-dimensionalized sound power.

### 1.7 Papers and Publications

The following papers have been written during the preparation of the present thesis and the details of their publication status are as given below.

**1.7.1 Published**


5. Agarwala, Nitin. and Nair, E. M. S. Fluid Loading in Structural-Acoustic Problems. *Communicated and accepted for publication by Commodore Garg Memorial Lecture, Published by INA, Delhi Chapter.

6. Agarwala, Nitin. and Nair, E. M. S. On Horizontal Beams and Sound Radiation due to a Moving Load. *The Online Journal of Science and Technology (TOJSAT), ISSN 2146-7390, 3(3), 120 - 133

7. Agarwala, Nitin. and Nair, E. M. S. Effect of inplane loading on sound radiation of a floating runway when an airplane is taking off. *Journal of Naval Architecture and Marine Engineering, ISSN 1813-8535 (Print), 2070-8998 (Online), 10(1), 41- 48

### 1.7.2 Under review


1.8 Understanding Sound

1.8.1 Basic Elements

In general, every noise problem involves a system of three basic elements: noise source, transmission path and a receiver.

1.8.1.1 Noise Source

A source is an excitation that results in noise, either at the point of excitation or elsewhere. Most sounds or noises we encountered in our daily life are from sources which can be characterized as point or line sources.

**Point Source**  In a free field condition, any source with its characteristic dimension being small compared to the wavelength of the sound generated, is considered as a point source. Alternatively a source is considered a point source if the receiver is at a large distance away from the source. If a sound source produces spherical spreading of sound in all directions, as seen in Figure 1.8, it is a point source. For a point source, the noise level decreases by 6 dB per doubling of distance from it.
Line Source  A line source is a source of air, noise, water contamination or electro-magnetic radiation that emanates from a linear (one-dimensional) geometry as seen in Figure 1.9. If the sound source produces cylindrical spreading, such as stream of motor vehicles on a busy road at a distance, it may be considered as a line source. For a line source, the noise level decreases by 3 dB per doubling of distance from it.

1.8.1.2 Transmission Path

The path is the route(s) that energy takes in traveling from a source to the radiator.

1.8.1.3 Receiver

The receiver is a person or an object where the sound is perceived and accessed.

1.8.2 Sound Fields

Sound fields are typically “near”, “far”, “free” and “reverberant” field.
1.8.2.1 Near Field

In the near field, close to the sound source, there can be a large amount of pressure variations, from one position to another position. Sound pressure level measurements in the near field are therefore forbidden in many applications. In the near field the sound field is very complex and complicated, and hence the sound pressure distribution and the sound intensity distribution may look completely different.

1.8.2.2 Far Field

In the far field, there is a more consistency in the sound field and sound pressure levels will not vary too much when the measurement position is moved slightly. The far field is also experienced if there is a free field situation.

1.8.2.3 Free Field or Direct Field

When a point source (or any source that radiates equally in all directions) radiates into free space, the intensity of the sound varies as $I/r^2$ where $r$ is the distance from the source, the intensity is given by $I = W/(4\pi r^2)$ and $W$ is the power of the source. This may be understood as a given amount of sound power being distributed over the surface of an expanding sphere with area $4\pi r^2$. An environment in which there are no reflections is called as a free field.

1.8.2.4 Reverberant Field or Diffuse Field

If the measurement position is close to some boundary like floors, walls and ceilings, there might be reflections of sound as well as direct sound. In such a sound field, a change in position further away or closer to the sound source may not give a significant change in the sound pressure level, because a lot of the sound pressure is caused by reflection. This is called a reverberant field or a diffuse sound field.
1.9 Sound Parameters

The three different quantities describing sound are sound pressure, sound intensity and sound power.

1.9.1 Sound Pressure

Sound pressure is a scalar, describing the pressure fluctuation at a given position and is measured in Pascal (Pa). Sound pressure is typically measured at the receiver’s position for evaluation of the harmfulness and the annoyance of a noise source.

1.9.2 Sound Intensity

Sound intensity is a vector quantity, that describes the amount and the direction of flow of acoustic energy at a given position. The unit for sound intensity is Watt per square meter (W/m²). It is used for noise source location, rating of noise sources and noise measurement in the medium at a listener’s location. For plane progressive waves with a harmonic source, the sound intensity over all time as given in Fahy and Gardenio (2007) is

\[ I(R, \phi, \theta) = \frac{|P(R, \phi, \theta)|^2}{2\rho c} \]

where \(|P(R, \phi, \theta)|\) is the amplitude of \(P(R, \phi, \theta, t)\) without the time dependent term \(e^{-i\omega t}\).

1.9.3 Sound Power

Sound power radiation can be defined as the rate of acoustic energy delivered by a source which can be obtained by integrating the sound intensity over a surface of convenience. It can only be calculated or determined either based upon sound intensity measurement or based upon sound pressure measurement. The main use of sound power is for noise rating so as to compare how noisy sources are. The unit for sound power is Watt (W). Sound power is an important index in acoustic radiation analysis because it is a single global quantity which can be used to characterize the strength of the sound generated by a source. Estimation of sound power is a common practice of identifying dominant sources.
prior to a noise control project. Many standard methods are available for sound power estimation. They may be based on pressure measurement or intensity measurement. Some of them require free-field environment while others require diffuse field (where there is preponderance of reflected or reverberant sound signal in relation to the direct sound signal) environment.

1.10 Sound Terminologies

1.10.1 Decibel

The scale often used for describing a measurement of sound is the decibel unit (dB). The decibel is a comparison of intensities or energy densities. It is not an absolute unit, but a ratio. Without a reference level, it means nothing. The decibel represents a measure of the sound power relative to a reference sound power, normally 1 pW (picoWatt). By definition, sound power level (PWL) is expressed as

$$PWL = 10 \log \frac{W}{W_{ref}}$$

where $W$ is the radiated sound power in Watts and $W_{ref}$ is the reference power $10^{-12}$ watts. Since the decibel scale is logarithmic, it is convenient for dealing with large differences in the measured quantities. Multiplication and division of decibel units becomes simple because they are reduced to an addition and a subtraction operation respectively.

1.10.2 The sound impedance ($Z$)

Sound impedance is a frequency dependent parameter and is very useful for describing the behaviour of sound producing object. The acoustic impedance at a particular frequency indicates how much sound pressure is generated by a given air vibration at that frequency. It hence varies strongly with the change of frequency. Mathematically, it is the ratio of the sound pressure $P$ and the product of the particle velocity $v$ and the surface area $S$, through which an acoustic wave propagates. Acoustic impedance can be expressed in
either its constituent units (pressure per velocity per area) or in rayls per square meter. Hence

\[ Z = \frac{P}{vS} \]

### 1.10.3 Subsonic Flows

The defining property of subsonic flow is that the flow speed is lesser than the local speed of sound. The condition for a flow to be subsonic in terms of the Mach number is \( M < 1 \) where \( M(= \frac{V}{C_0}) \) is the Mach number and \( C_0 \) is the local speed of sound. In subsonic flow at low Mach number the viscosity and heat conduction are normally important effects as the timescale is relatively long, \( M << 1 \) and density effectively remains constant, this is not so in supersonic flows.

### 1.11 Understanding Load

#### 1.11.1 Concentrated Load

Force applied at a single point is called as a concentrated load. Concentrated loads are useful mathematical idealizations, but do not exist in the real world. A concentrated load can be applied at more than one location on a structure, and multiple loading points may exist on the same structure.

#### 1.11.2 Distributed Load

Concentrated / point forces are models. These forces do not exist in the exact sense. Every external force applied to a body is distributed over a finite contact area. The following distinctions are made, depending on the dimension of the area of application:

##### 1.11.2.1 Line Load

Load distributed along a line is called as a line load. The intensity of this force is expressed as force per unit length of line (N/m).
1.11.2.2 Surface Load

Load applied over an area. The intensity of these forces (called pressure) is expressed as force per unit area. The units for the surface force are $N/m^2$.

1.11.2.3 Volume Load

Forces distributed over the volume of the body. Also called as body forces. The units for the intensity of body forces are $N/m^3$.

1.12 Understanding Moving Load

1.12.1 Moving Load Types

The moving load itself is modeled in various forms. Of the many the most commonly used are

1.12.1.1 Concentrated force

For a single point force the Dirac (impulse, also delta) function that expresses the concentrated load as $p(x, t) = \delta(x)P$, where $P$ is the vehicle weight as seen in Figure 1.10.
1.12.1.2 Moving Harmonic force

The time variable concentrated force is given as \( p(t) = Q \sin \omega t \) where \( Q \) is the amplitude and \( \omega \) the circular frequency of the harmonic force.

1.12.1.3 Moving Continuous force

This is frequently used in the calculation of dynamic stresses in pipelines carrying moving fluids, large-span railway bridges, resulting from the traverse of a train of cars hauled by a locomotive. The moving continuous force as seen in Figure 1.11 is represented as \( p(x, t) = q(\xi, t) - \mu q(\xi) \frac{d^2v(x,t)}{dt^2} \) where \( \xi = x - ct \) and by neglecting the second order terms we get \( p(x, t) = q[1 - H(x - ct)] \)

1.12.1.4 Moving force arbitrarily varying in time

Forces in this form are used in calculations of structures exposed to the effects of explosion, pressure waves, impacts of flat wheels on large-span railway bridges etc. as seen in Figures 1.12 and 1.13.

1.12.1.5 Platoon Load

This is a mix of both the constant load and the harmonic load. Hence the type of moving dynamic loads considered are moving constant loads and moving harmonic loads.
1.12 Understanding Moving Load

Fig. 1.12 Moving force linearly increasing in time (Fryba (1999))

Fig. 1.13 Moving force linearly decreasing in time (Fryba (1999))

1.12.2 Moving Load Models

Moving loads have a great effect on dynamic stresses in structures, and cause them to vibrate intensively, resulting into sound, especially at high velocities. Their peculiar feature is that they are variable in both time and space. If we think of a moving load as of a mass body moving in a generally curved path over the structure being examined, we see that according to d’Alembert’s principle its effects are twofold: the weight effects (or gravitational) of the moving load, and the inertial effects of the load mass on the deformed structure. If only the weight effect is considered, and the mass of the moving load neglected against the mass of the structure, the computation of strains in the solid is not easy. It becomes more complicated when the structure mass is assumed to be negligible against the load mass. The most difficult of all is the problem involving both the gravitational and the inertial action of moving loads having masses commensurable
with the mass of the structure. Mathematically these models are expressed as

### 1.12.2.1 Moving force model

In the moving force model, the force is assumed to be constant and equal to the vehicle weight, hence $f_c(x,t) = P\delta(x - vt)$; where $P$ is the vehicle weight. This is the simplest vehicle model, since we have only two parameters: vehicle weight and speed.

### 1.12.2.2 Moving mass model

The moving mass model accounts for the inertial effect of the moving load. We apply this method by modeling the moving mass as an oscillator with large spring stiffness. The force is given by $f_c(x,t) = [(P - m\frac{d^2w(vt,t)}{dt^2})\delta(x - vt)]$. The difference between solutions of the moving force and moving mass models grows with increasing speed.

### 1.12.2.3 Moving oscillator model

In the moving oscillator model, the mass $m_0$ is attached to the continuum through a spring and dashpot, $f_c(x,t) = [P + k(z(t) - w(vt,t)) + c(\dot{z}(t) - \dot{w}(vt,t))]\delta(x - vt)$ where $k$ and $c$ are spring and dashpot coefficients and $z(t)$ is the vertical displacement of the mass. In this case, we have an additional degree of freedom and, thus, need an additional equation of motion, $m\ddot{z} = -k[z(t) - w(vt,t)] - c(\dot{z}(t) - \dot{w}(vt,t)]$

### 1.12.2.4 Moving Multiple Degrees of Freedom model

The moving MDOF model accounts for the facts that (i) the vehicle is not a lumped mass but rather consists of several elastically interacting components and (ii) it interacts with the structure carrying it at several contact points.
1.13 Understanding Material

1.13.1 Beams

A beam is a rod-like structural member that can resist transverse loading applied between its supports. By “rod-like” it is meant that one of the dimensions is considerably larger than the other two. This dimension is called the longitudinal dimension and defines the longitudinal direction or axial direction. Directions normal to the longitudinal directions are called transverse. The longitudinal axis of a beam is directed along the longitudinal direction and passes through the centroid of the cross sections. A beam is straight if the longitudinal direction is a straight line. A beam is prismatic if the cross section is uniform.

1.13.2 Plates

Plates are defined as plane structural elements with a small thickness compared to the planar dimensions. The typical thickness to width ratio of a plate structure is less than 0.1. A plate theory takes advantage of this disparity in length scale to reduce the full three-dimensional solid mechanics problem to a two-dimensional problem. The aim of plate theory is to calculate the deformation and stresses in a plate subjected to loads. Of the numerous plate theories that have been developed since the late 19th century, two are widely accepted and used in engineering. These are

- The Kirchhoff-Love theory of plates (classical plate theory)

- The Mindlin-Reissner theory of plates (first-order shear plate theory)

1.13.2.1 Kirchhoff-Love theory

Kirchhoff-Love theory is an extension of Euler-Bernoulli beam theory to thin plates. The theory was developed in 1888 by Love, using assumptions proposed by Kirchhoff. It is assumed that a mid-surface plane can be used to represent the three-dimensional plate in two dimensional form. The following kinematic assumptions are made in this theory:
• straight lines normal to the mid-surface remain straight after deformation

• straight lines normal to the mid-surface remain normal to the mid-surface after deformation

• the thickness of the plate does not change during a deformation.

1.13.2.2 Mindlin-Reissner Plate Theory

Mindlin’s theory assumes that there is a linear variation of displacement across the plate thickness but that the plate thickness does not change during deformation. This implies that the normal stress through the thickness is ignored; an assumption which is also called the “plane stress” condition. On the other hand, Reissner’s theory assumes that the bending stress is linear while the shear stress is quadratic through the thickness of the plate. This leads to a situation where the displacement through-the-thickness is not necessarily linear and where the plate thickness may change during deformation. Therefore, Reissner’s theory does not invoke the plane stress condition.

The Mindlin-Reissner theory is often called the first-order shear deformation theory of plates. Since a first-order shear deformation theory implies a linear displacement variation through the thickness, it is incompatible with Reissner’s plate theory.

The vibration equation for the elastic plate, including rotational inertia and transverse shear effects, as given by the Mindlin plate equation is

$$\left(\nabla^2 - \frac{\rho_v}{\kappa^2 G} \frac{\partial^2}{\partial t^2}\right)\left(D_s \nabla^2 - \frac{\rho_v h^3}{12} \frac{\partial^2}{\partial t^2}\right) u + \rho_v h \frac{\partial^2 u}{\partial t^2} = \left(1 - \frac{D_s}{\kappa^2 Gh} \nabla^2 + \frac{\rho_v h^2}{12\kappa^2 G} \frac{\partial^2}{\partial t^2}\right) [f(x, t) - p(x, y = 0, t)]$$

(1.1)
1.13.3 Need for inclusion of Rotational inertia and Transverse shear effect

For short, sturdy beams the shear effect cannot be neglected as in conventional analysis using the Bernoulli-Euler's beam theory. The situation occurs when the cross section of the beam is relatively large in comparison with the beam span. Although the correction for the shear effect may yield results only a few percent more accurate in frequency prediction than those from classical beam theory for a moderately thick beam, the accuracy improvement may be quite profound when performing dynamic response analysis. It is with this reasoning that a Timoshenko-Mindlin plate is utilized for the present study.

1.13.4 Finite Vs Infinite beams

1.13.4.1 Finite beam

For finite beams the problem is one-dimensional wherein the beam boundaries generate standing waves. Because of this, sound can be radiated at frequencies both below and above the critical frequency. However the radiation ratio ($\gamma$) of finite beam is not zero below the critical frequency. These plates demonstrate boundary effect.

1.13.4.2 Infinite beam

A beam is considered infinite if the excitation occurs far enough from the ends such that the reflected energy is negligible. Derivation of the input impedance for such cases is simpler than that for semi-infinite / finite beams. At the point of application of the force, the beam moves straight up and down without rotation.

At low frequencies, the input impedance is four times as large as that for a semi-infinite beam. As frequency increases, the reactive term increases more rapidly than the resistive term. The magnitude of the impedance therefore continues to increase with frequency rather than approaching a constant value, as does that for a semi-infinite beam. In an infinite beam sound radiation only occurs above the critical frequency. At frequencies
above the critical frequency finite beams behave in a similar manner to infinite beams, but this is not the case at lower frequencies. Above the critical frequency, the radiation ratio ($\gamma$), is the same in both cases.

1.14 Basic Mathematical Tools Relevant to the Thesis

1.14.1 Analysis Methods

The problem at hand of a structure subjected to a load (moving or static) is to determine the deflections and stress resultants of the structure under the action of load. In order to solve such problems the analysis may be undertaken in either of the two domains, namely the time domain or the frequency domain.

1.14.1.1 Frequency Domain Analysis

When the motion of a structure and the surrounding water are time harmonic, they can be analyzed in the frequency domain. The relevant analysis methods are known as frequency domain methods. In this case it is the amplitudes of the motions that are to be determined. The commonly-used approaches for the analysis of VLFS in the frequency domain are the modal expansion method and the direct method.

Modal Expansion method  This method consists of separating the hydrodynamic analysis and the dynamic response analysis of the structure. The deflection of the structure with free edges is decomposed into vibration modes that can be arbitrarily chosen. These modes may be that of the dry type or the wet type. Most analysts use the dry-mode approach because of its simplicity and numerical efficiency. Next, the hydrodynamic radiation forces are evaluated for unit amplitude motions of each mode. The Galerkin’s method, by which the governing equation of the structure is approximately satisfied, is then used to calculate the modal amplitudes, and the modal responses are summed up to obtain the total response.
**Direct method**  This method is straightforward, but must solve a large scale matrix equation and this means that a large amount of memory space and CPU time is required. In this method, the deflection of the structure is determined by directly solving the motion of equation without any help of eigenmodes. In this solution procedure, the potentials of diffraction and radiation problems are established, and the deflection of structure is determined by solving the combined hydroelastic equation via the finite difference scheme.

In sum, the principal difference between the modal superposition method and the direct method lies in the treatment of the radiation motion for determining the radiation pressure.

### 1.14.1.2 Time Domain Analysis

When the motion of a structure and the surrounding fluid are time dependent or transient, they must be analyzed in the time domain. The corresponding methods are known as time domain methods. In this case it is the *time histories of the structural motions* that are determined. The commonly-used approaches for the time-domain analysis are the *direct time integration method* and the *Fourier transform method*. Time-domain analysis for hydroelasticity of VLFS is necessary for design purposes since some of the design cases, such as the airplane landing and take-off, is a transient phenomenon and can not be solved in the frequency-domain. Only a few studies of transient problems have been reported to date, and most of them are still too computationally intensive for practical use in VLFS design.

**Direct Time integration method**  In the direct time integration method, the equations of motion are discretized for both the structure and the fluid domain.

**Fourier transform method**  In the Fourier transform method, we first obtain the frequency domain solutions for the fluid domain and then Fourier transform the results for substitution into the differential equations for elastic motions. The equations are then solved directly in the time domain analysis by using the finite element method or other
suitable computational methods.

As seen above, the analysis may be carried out in the frequency domain or in the time domain. However most hydroelastic analysis are carried out in the frequency-domain, being the simpler of the two. For transient responses and for nonlinear equations of motion (due to effects of a mooring system or nonlinear wave as in a severe wave condition), it is necessary to perform the analysis in the time-domain. It is well-known that the time-domain and frequency domain analysis are reversible through Fourier transformation. Although the solutions in the time and frequency domains can be conveniently linked through the Fourier and inverse Fourier transforms, it is time consuming and difficult to perform these transforms in some cases. One of the difficulties in undertaking an inverse Fourier transform from the frequency domain to time domain is the evaluation of the integrals over an infinite frequency range. Thus it is necessary to truncate the integral at a finite frequency and to assess the accuracy of the truncated interval.

1.14.2 Solution Methodologies

Differential equation solution for beams and plates is a vast topic with many variations. These can be broken down into two solution groups the first being exact (i.e analytical) solutions and the second being approximate (i.e numerical) solutions. The advantages of analytical methods are that they can predict results without many computations, in a shorter time and usually without frequency limitations. They can shed light on physical mechanisms involved in certain phenomena and can be used to quickly compare different design alternatives without the need for considerable analysis effort, by roughly seeing how different parameters influence quantities such as radiated sound pressure or sound power. This can be very useful in the conceptual stage. The main disadvantage of analytical methods is that they can be used only for simple problems (e.g. simple structures). On the other hand numerical methods are very powerful in accurately modeling real life situations including complex structures. The most widely used numerical methods are the Finite Element Method (FEM) and the Boundary Element Method (BEM). Both of them can be used for structural and acoustic analysis but due to certain advantages
and disadvantages FEM is mainly used for structural problems whereas BEM is used for acoustical problems. In contrast with analytical methods, numerical methods require a great number of computations and usually take a long time to be performed. Moreover, they need considerable manual effort to design detailed models. They are also only applicable for low-frequency analysis. Numerical methods are used extensively in the detailed design stage. For the purpose of minimisation of sound radiation by structures, numerical methods have been used in conjunction with numerical optimisation algorithms to achieve optimum designs.

1.14.3 Approaches available

Three approaches are used to obtain the total acoustic power radiated from planar sources;

- Far-field approach

- Surface integration approach

- Fourier transform approach

1.14.3.1 Far-field Approach

This approach was developed by Rayleigh for a plate vibrating in an infinite baffle. It uses the Rayleigh surface integral to calculate the acoustic pressure, acoustic intensity and hence the acoustic power radiation. Each elemental area on the plate surface is assumed to be a point source of sound wave and their individual contributions are summed to yield the total acoustic power radiation. The radial component of the acoustic intensity is integrated over an imaginary far-field hemisphere enclosing the source (Junger and Feit (1986)), and this explains the use of spherical coordinates for writing the far-field pressure expressions.
1.14.3.2 Surface Integration Approach

This approach is suitable for baffled and unbaflled plates. The real acoustic intensity is integrated over the surface of the vibrating structure. The acoustic pressure is determined in terms of surface velocity, using the Rayleigh integral.

1.14.3.3 Fourier Transform Approach

In this approach, the velocity and its complex conjugate are employed for approximating the sound power radiated from the vibrating surface. Thus

\[ \Pi = \frac{1}{2} \Re \left( \int_s p_s v^* dS \right) \]

where \( p_s \) is the surface pressure, \( v^* \) is the complex conjugate of surface velocity and \( S \) is the area of the radiating surface.

1.14.4 Fourier Transformation

Time is fundamental in our everyday life in the 4-dimensional world. We see things move as a function of time. On the other hand, although sound waves are composed of moving atoms, their movement is too small and the frequency of the vibration is too fast for us to observe directly. It is thus easier to describe sounds in frequency space rather than time space. We can transform sound, or other things in physics for that matter, from time space to frequency space by the technique of Fourier transform. The Fourier transform (FT) is one type of mathematical transformation which changes one axis variable to another variable. The exponential Fourier transforms, are defined as

\[ \tilde{w}(\xi) = \int_{-\infty}^{\infty} w(x) e^{-i\xi x} dx \]

\[ w(x) = \frac{1}{2\pi} \int_{-\infty}^{\infty} \tilde{w}(\xi) e^{i\xi x} d\xi \]  \hspace{1cm} (1.2)

The spatial transform variable, \( \xi \), has physical significance as the wave number, and the wave number response or spectrum, \( w(\xi) \), is simply the structure’s response in the
wave number domain. When using Fourier transforms, it is assumed that both transforms exist over their entire domain of definition, where the \( \sim \) denotes the transformed expressions.