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Acoustic-Structure Interaction for a Floating Airport subject to a Moving Load

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Abstract:

Very Large Floating Structures used as floating airports are constructed inside sheltered waters to minimize ocean effects on them. These sheltered areas have abundant marine life which in general is affected by noise pollution. Sound generated by takeoff / landing of an airplane from a floating airport is the area of interest in this study. Traditionally, noise problems have been solved in a "trial and error" manner with numerical analysis being relegated to a supporting role and their results greeted with skepticism. For structures such as floating airports, "trial and error" can be costly and hence numerical methods are used. We propose a simple numerical method for calculating the sound radiation from floating structures due to such moving loads. The effect of moving load, current in the fluid below and axial loading has been studied. In developing the expression, Fourier transform methodology for a Timoshenko-Mindlin plate is utilized.

***Key words**: Moving Load; Floating Airport; Timoshenko-Mindlin plate; Heavy fluid-loading; Sound radiation; Mean flow; Inplane loading.*

1.Introduction

Structures vibrate due to surface forces applied on them. Larger the magnitude of the moving load, higher the vibration. High vibration levels generate noise or cause material failure, thus degrading the structure's performance. A pontoon type VLFS is a promising structure for floating air-ports or runways due to their simple construction and ease of maintenance. These structures are constructed near the shoreline in a cove or a lagoon to minimize ocean effects on them. Such sheltered areas have an abundance of marine life in variety and quantity which are subjected to noise pollution due to activities on VLFS such as movement of equipment, people, cargo (dry and liquid), variable ballast (dry and liquid), aircraft takeoff / landing, crane handling, berthing / docking, connection / disconnection, running machinery onboard etc. Recent studies by marine biologists have confirmed undesirable effects of noise pollution on marine life and hence the need to study sound radiation by these activities becomes important. Out of these, an airplane taking off / landing is possibly the largest contributor of sound radiation into the water. Other contributors such as presence of mean flow in the fluid below and the structure being subjected to an inplane loading due to a combined action of uniformly distributed hydrostatic lateral loading and compression due to hogging, berthing, plate connections at ends, initial deformation and corrosion to name a few, merit attention.

The phenomenon of acoustics of vibrating structures caught the attention of Lord Rayleigh [1] in 1896. 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. A standard reference on the analytical modeling is the book of Junger and Feit [2]. The sound radiation from a moving force excited, elastic structure is a relatively newer area of interest. Keltie [3] investigated the sound radiation from a fluid-loaded Timoshenko beam subject to a moving harmonic line force. Results show that for beams under light fluid loading, the coincidence sound radiation peak for a stationary force gets 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, Cheng and Chui [4] 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. [5, 6] introduced a "wavenumber harmonic series" to discuss the vibro-acoustic response of a fluid-loaded beam on periodic elastic supports 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.

To the best of the knowledge of the authors, the study of sound radiations from VLFS due to an airplane taking off has not been studied so far. In the present paper sound radiation by a VLFS due to an airplane takeoff / landing for two structural materials has been undertaken, while comparing them. The contribution of mean flow and inplane loading on the sound radiation are studied. An expression for the sound radiation has been developed for a wavenumber ratio of 0.1 to 2.2. In developing the expression, a Fourier Transformation in space for a Timoshenko-Mindlin plate is utilised as suggested in [3]. Results are presented at a range of frequencies both below and above coincidence for heavy fluid-loaded elastic plate. The present study thus provides a simple yet effective methodology in calculating the sound radiation into the marine environment from manmade structures.

2.Formulation

2.1.Structure Definition

To study acoustic effects of a VLFS, a dynamic analysis of a three-dimensional runway with time varying loading during take-off would be exceeding difficult. This analysis is made simpler by assuming that the runway behaves as a simple, infinitely long beam floating in water and supported by buoyancy. The model can be assumed to be a simple beam, described by a one dimensional Timoshenko-Mindlin plate equation.

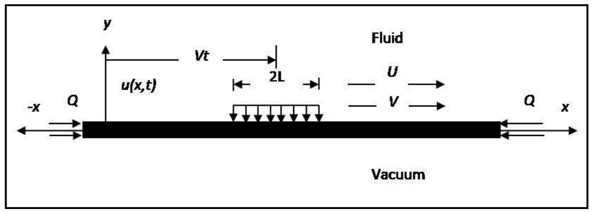

Figure 1: Schematic representation of the problem geometry

We assume that the floating airport behaves as a simple, infinitely long beam floating in water and supported by buoyancy. The structural damping is ignored since there is no apparent resonant mechanism in this problem. The water is assumed to be inviscid, and the flow resulting from the airplane take-off is irrotational. The *x*-axis is aligned with the length of the runway and the *y*-axis is directed vertically upwards, as seen in Figure 1. Because the floating runway is very narrow compared with its length, as a simplification, we will assume that the deformation and loading assumed not to vary across the runway. An excitation force of length 2L moving at a subsonic speed *V* is assumed to be acting on the floating airport. The space $y > 0$ is filled with an acoustic medium such as water. The other side of the plate is assumed to be vacuum. A subsonic mean flow of speed *U*, moving in the positive *x* direction is considered to be present in the water. A compressive inplane load of magnitude *Q* per unit width is considered to be present. If the load is tensile then it attains a magnitude -*Q*.

2.2.Governing Equation

A uniform distributed line force considered as the moving force is given by

$$f(x,t) = \frac{f_0}{2L}\left[H(x - Vt + L) - H(x - Vt - L)\right]e^{j\omega t}$$

The vibration equation for the one dimensional elastic plate, including rotational inertia, transverse shear effects and inplane loading, is given by the Timoshenko-Mindlin plate equation as

$$D\frac{\partial^4 u(x,t)}{\partial x^4} + Q\frac{\partial^2 u(x,t)}{\partial x^2} + \rho_v h\frac{\partial^2 u(x,t)}{\partial t^2}$$
$$-\rho_v I\left(1 + \frac{D\rho_v}{\kappa^2 G}\right)\frac{\partial^4 u(x,t)}{\partial x^2 \partial t^2} + \rho_v I\frac{\rho_v}{\kappa^2 G}\frac{\partial^4 u(x,t)}{\partial t^4}$$

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

(1)

2.3.Boundary Conditions

To account for the presence of current, the operator $(\frac{\partial}{\partial t})$ is replaced by the operator $(\frac{\partial}{\partial t}+U\frac{\partial}{\partial x})$ in the expressions of pressure distribution and the boundary condition at y = 0. The pressure distribution induced by the vibrating plate in the acoustic medium denoted by $p(x,y,t)$ thus satisfies the wave equation in two-dimensional space, given by

$$\left[\frac{\partial^2}{\partial y^2}+\frac{\partial^2}{\partial x^2}-\frac{1}{C_0^2}\left(\frac{\partial}{\partial t}+U\frac{\partial}{\partial x}\right)^2\right]p(x,y,t)=0 \quad (2)$$

For ρ_0 as the mass density of the acoustic medium, the boundary condition at y = 0 is modified as

$$\rho_0\left(\frac{\partial}{\partial t}+U\frac{\partial}{\partial x}\right)^2 u=-\frac{\partial p}{\partial y}\Big|_{y=0} \qquad (3)$$

2.4.Transformation

By applying the spatial Fourier transformation $FT()=\int_{-\infty}^{\infty}()e^{i\xi x}dx$, the force function for a harmonic line force in wave number domain may be written as

$$\tilde{F}(\xi,t)=f_0\frac{sin(\xi L)}{\xi L}e^{j(\omega+\xi[V-U])t} \qquad (4a)$$

the transformed displacement as

$$\tilde{U}_s(\xi,t)=U_s(\xi)e^{j(\omega+\xi[V-U])t} \qquad (4b)$$

and the transformed pressure as

$$\tilde{P}(\xi,y,t)=P(\xi,y)e^{j(\omega+\xi[V-U])t} \qquad (4c)$$

2.5.Combining Governing Equation

Upon substitution of equation (4a), (4b) and (4c) in the relevant plate equation and the acoustic equation, we get

$$U_s(\xi) = \frac{Z_F F(\xi)}{Z_m + Z_F Z_a} \qquad (5)$$

and

$$P(\xi, y=0) = \frac{j\rho_0(\omega + \xi[V-U])^2}{K_y} U_s(\xi) \qquad (6)$$

where the acoustic impedance operator (*Za*) is given by

$$Z_a = \frac{j\rho_0(\omega + \xi[V-U])^2}{K_y} \qquad (7)$$

the plate impedance operator (*Zm*) as

$$Z_m = D\xi^4 - Q\xi^2 - \rho_v h(\omega + \xi[V-U])^2 - \xi^2\left(\rho_v I + \frac{D\rho_v}{\kappa^2 G}\right)$$

$$\times(\omega + \xi[V-U])^2 + \rho_v I \frac{\rho_v}{\kappa^2 G}(\omega + \xi[V-U])^4 \qquad (8)$$

the *ZF* by

$$Z_F = 1 + \frac{D}{\kappa^2 G h}\xi^2 - \frac{\rho_v h^2}{12\kappa^2 G}(\omega + \xi[V-U])^2 \qquad (9)$$

and *Ky* is given by

$$K_y = \left[\begin{array}{ll} -j\sqrt{\xi^2 - (K_0 + \bar{M}\xi)^2} & for \quad \xi^2 > (K_0 + \bar{M}\xi)^2 \\ \sqrt{(K_0 + \bar{M}\xi)^2 - \xi^2} & for \quad \xi^2 < (K_0 + \bar{M}\xi)^2 \end{array}\right.$$

(10)

2.6.Total Acoustic Power

To find the total acoustic power, the surface acoustic intensity distribution may be integrated over the infinite length of the plate as

$$\Pi = \int_{-\infty}^{\infty} \frac{1}{2} Re[P(x, y=0, t)\dot{U}_s^*(x,t)]dx$$

Upon substituting the sound pressure (6) and the surface velocity (5) of the plate, the sound power radiated per unit width of the plate can be simplified as

$$\Pi = \frac{\rho_0}{4\pi} Re\Big[\int_{-\infty}^{\infty} \frac{(\omega + \xi[V-U])^3}{K_y} |U_s(\xi)|^2 \, d\xi\Big] \quad (11)$$

Limiting the study to subsonic motion of the moving load, the limits within which K_y is real is given by

$$\xi_1 = \frac{-K_0}{1+\bar{M}} \le \xi \le \xi_2 = \frac{K_0}{1-\bar{M}}$$

This allows us to rewrite the expression for the sound power as

$$\Pi = \frac{\rho_0}{4\pi} Re\Big[\int_{\xi_1}^{\xi_2} \frac{(\omega + \xi[V-U])^3}{K_y} |U_s(\xi)|^2 \, d\xi\Big] \quad (12)$$

2.7.Nondimensionalization

By using non-dimensional parameters, to present the numerical results, the dimensionless radiated sound power per unit width from a Timoshenko-Mindlin plate subjected to a moving load in the presence of a mean flow in the fluid and presence of inplane loading for a uniform distributed line force is obtained as:

$$W = \int_{\zeta_1}^{\zeta_2} \alpha^3 \beta \Big| Z_F \frac{sin(\zeta K_0 L)}{\zeta K_0 L} \Big|^2 |D_w|^{-2} \qquad (13)$$

where

$$\zeta_1 = \frac{-1}{1+\bar{M}} \le \zeta \le \zeta_2 = \frac{1}{1-\bar{M}}$$

$$\alpha = 1 + \bar{M}\zeta$$

$$\beta = \sqrt{\alpha^2 - \zeta^2}$$

$$D_w = \beta(D_1 - D_2 + D_3 - D_5) + jD_4 \; Z_F = 1 + \frac{2(1+\nu)\gamma^4}{\kappa^2}\Big(\frac{C_0}{C_L}\Big)^2\Big[\zeta^2 - \Big(\frac{C_0}{C_L}\Big)^2 \alpha^2 (1-\nu^2)\Big]$$

$$D_1 = \gamma^4 \zeta^4$$

$$D_2 = \alpha^2\Big[1 + [1 + \frac{2(1+\nu)}{\kappa^2(1-\nu^2)}]\gamma^4 \zeta^2 \Big(\frac{C_0}{C_L}\Big)^2 (1-\nu^2)\Big]$$

$$D_3 = \frac{2(1+\nu)}{\kappa^2} \alpha^4 \gamma^4 \Big(\frac{C_0}{C_L}\Big)^4 (1-\nu^2)$$

$$D_4 = Z_F \frac{\alpha_0 \alpha^2}{\gamma^2}$$

$$D_5 = \frac{Q}{\rho_v h}(\frac{\zeta^2}{C_0^2})$$

3.ANALYSIS

The investigation of the problem is covered in three parts:

- Evaluating the total radiated sound power for a Timoshenko-Mindlin plate of different structural materials.
- Effect of mean flow of the fluid on the total radiated sound power.
- Effect of inplane loading on the total radiated sound power.

Using (13) in MATLAB, the sound power is computed and then plotted against the wave number ratio (γ) or non-dimensional frequency for a Timoshenko-Mindlin plate immersed in water. The parameters used for the plate model are as given in Table 1.

Property	Value	Unit
E_{steel}	20 x 10^{10}	N / m^2
$\rho_{v\ steel}$	7800	Kg / m^3
D_{steel}	560	KNm
$E_{Aluminium}$	7.1 x 10^{10}	N / m^2
$\rho_{v\ Aluminium}$	2700	Kg / m^3
$D_{Aluminium}$	237	KNm
h	2.54 x 10^{-2}	m
v	0.3	
κ^2	0.85	
C_0	1481	m / s
ρ_0	1000	Kg / m^3
f_0	Unit magnitude	
U	Between -10 and 10	m / s
γ	Between 0.01 and 2.2	
T and Q	Between 5 and 200	MN

Table 1: Parameters used

4.Discussion

In the results shown below, four distinct frequency ranges exist: the very low frequency region ($\gamma < 0.1$); the low frequency region ($0.1 < \gamma < 1.0$); the frequency region near coincidence ($\gamma \sim 1.0$); and the frequency region above coincidence ($\gamma > 1.0$). In the very low frequency region and in the region above coincidence frequency, the sound powers radiated show no discernible difference. It is the low frequency region and the region near coincidence i.e ($0.1 < \gamma \leq 1.0$) which are of concern to us and need to be discussed.

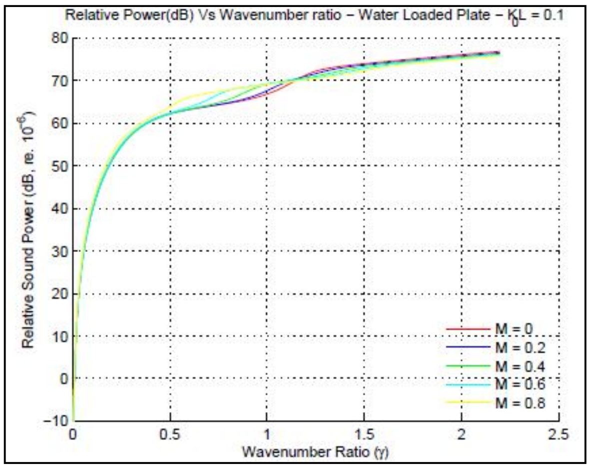

Figure 2: Relative sound power v/s wavenumber ratio for steel; K_0L = 0:1

4.1.Structural Material

Figure 2 and 3 shows the sound power from steel and an Aluminium structure respectively. As expected the sound produced from an Aluminium structure is lesser than that from a steel structure for all acoustic lengths. The difference of the acoustic power is however very large at low frequencies which reduces with increase in frequency to increase again beyond coincidence. This trend is noticed to be consistent for all values of K_0L. It is interesting to note that the differences tend to converge for varying convective speed of loading at higher frequencies as noted by Keltie and Peng [7].

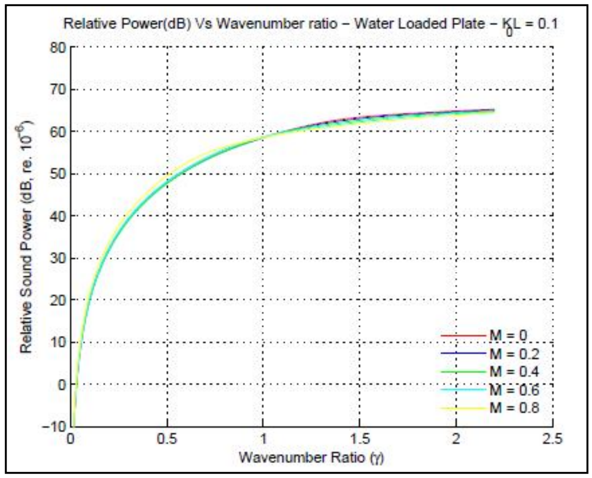

Figure 3: Relative sound power v/s wavenumber ratio for Aluminium; K_0L = 0:1

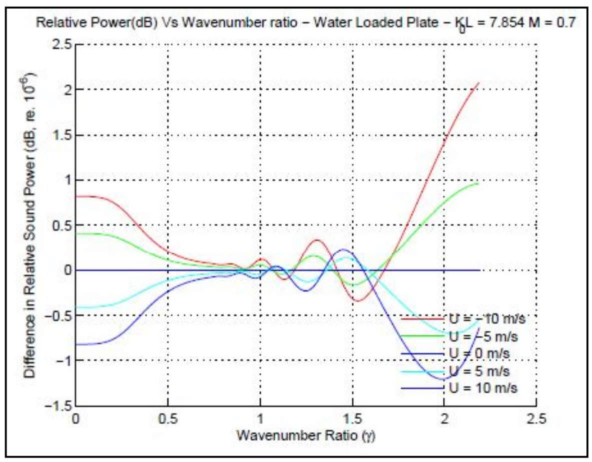

Figure 4: Difference in Relative sound power with current; $K_0L = 5\pi/2$ (non-integral multiple of π); M=0.7

4.2.Mean Flow

Presence of current on the sound radiation causes a proportional shift of the sound power curves. Presence of current in the direction of the subsonic moving force results into an increased Mach number and hence an increased sound power. The shift however is not very large. Since the effect of current is not large, we replot graphs as a difference curve with $U = 0$ as the reference to get Figure 4 and 5. Figure 4 is for fixed M and K_0L as non-integral multiples of π while Figure 5 is for K_0L as integral multiples of π. It is interesting to note that the trend of curves of integral multiples and non-integral multiples is different, but consistent. The variation due to convective speed of loading is increased magnitude for increased M, while the curve trends remain to be the same. It is noted that for non-integral multiples of π, for every step increase of $\pi/2$, there is an added node with the magnitude of the previous nodes being reduced. For integral multiples of π, for every step increase of π, there are two added nodes, with the previous nodes being reduced in magnitude. It is noted that the relative difference of sound power due to presence of mean flow is limited to 1 dB which may be considered as negligible.

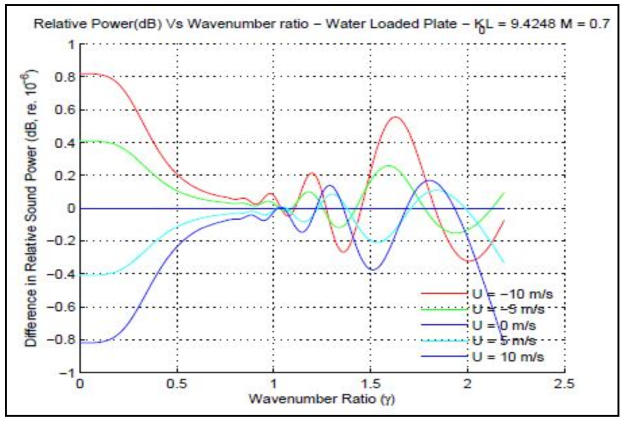

Figure 5: Difference in Relative sound power with current; $K_0L = 3\pi$ (integral multiple of π); M=0.7

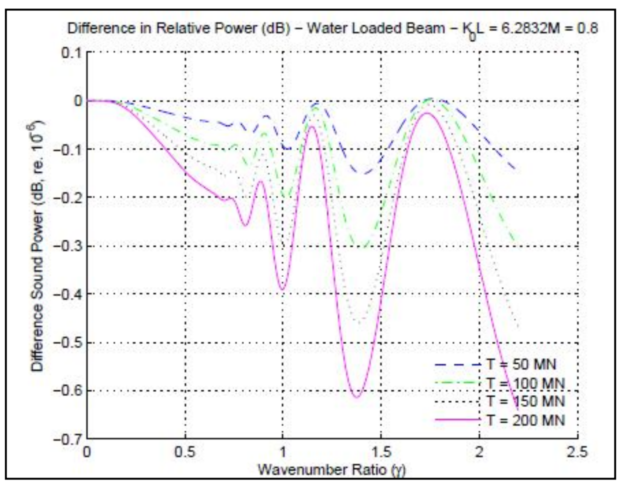

Figure 6: Relative sound power v/s wavenumber ratio under Tensile Load; $K_0L = 2\pi$; M = 0:8

4.3.Inplane Loading

The sound power generated by the moving load on a one dimensional Timoshenko-Mindlin plate subjected to inplane loading increases with increased loading. With increased speed, there is a marginal increase in the sound power generated, while an increased acoustic length *K0L* reduces the sound power level over the entire range of the frequency range. With increasing compressive inplane loading on such a plate, the sound power from the structure decreases. The Tensile loading on the other hand shows a corresponding increase in the sound power magnitude. The increase of the acoustic power is however not very large over the entire range of frequency as seen in Figure 6. What is interesting to note is the differences tend to converge for varying convective speed of loading at higher frequencies.

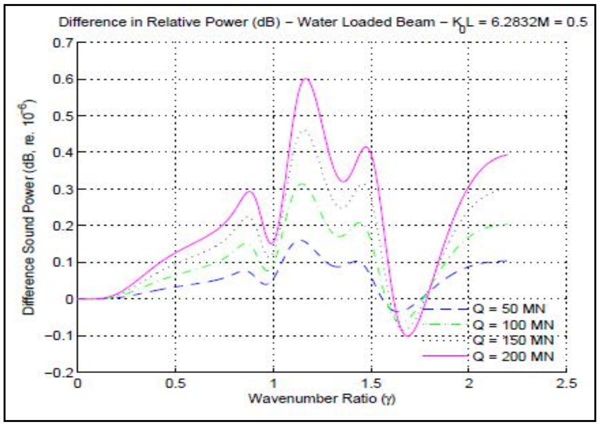

Figure 7: Relative sound power v/s wavenumber ratio under Compressive Load; K_0L = 2π; M = 0:5

5. Conclusion

Sound produced by an airplane taking off from a floating runway has been investigated for different structural materials, presence of mean flow and inplane loading independently. The entire analysis is carried out assuming a one dimensional plate in lieu of a three dimensional runway with time varying loading. The sound generated at various speeds of convective loading has been calculated and as expected an increase in sound is observed with increasing Mach number. The overall sound generated reduces with an increased acoustic length K_0L over the entire frequency range. No pronounced peaks are observed in the sound power curves due to the denser medium of water wherein the energy drain is faster disallowing peak formation. Changing of structural material from steel to Aluminium has an effect of higher sound power from steel as compared to Aluminium. When using Aluminium, increased Mach number has a similar effect as seen for steel, however the sound power from different speeds of convective loadings converge at higher frequencies. The presence of current does not alter the sound produced prominently and the change is seen to be in the range of 1dB. On analyzing the difference of sound power with current a unique trend of curves is observed for acoustic lengths of integral and non-integral multiples of π. The *inter se* trend however remains consistent. The methodology discussed herein provides the designer a simple tool for evaluating the total sound power radiated from a floating plate subject to a moving load such as a landing / take off of an airplane, mean flow below the floating structure and inplane load on such floating structures. Such a tool shall help in a better design of a VLFS for a safer marine environment.

6.Acknowledgements

The first author acknowledges the Indian Navy for the permission to undertake this research work.

- **Nomenclature**

ξ = Wave number variable

ζ = Nondimensional wave number variable

$\gamma (= \frac{K_0}{K_B})$ = Wave number ratio

ν = Poisson's ratio

ω = Angular frequency

ρ_v = Mass density of the structural material

ρ_0 = Mass density of the acoustic medium

$\kappa^2 (= \frac{\pi^2}{12}) \alpha_0$

$(= \frac{\rho_0 C_L}{\sqrt{12}\, \rho_v C_0})$ = Fluid loading parameter

$\delta (x - Vt)$ = Delta function

Π = Total acoustic power

h = Height of the beam

t = Time variable

x = Space variable in x direction

$p(x, y = 0, t)$ = Acoustic pressure acting on the surface of structure

$u(x, t)$ = Transverse displacement of the structure

f_0 = Strength of external force per unit width

$C_L (= \sqrt{\frac{E}{\rho_v}})$ = Longitudinal wave speed

C_0 = Speed of sound in acoustic medium

D = Flexural rigidity

$\bar{E} (= E(1 + \eta j))$ = Complex elastic modulus

E = Elastic modulus

$\bar{G} (= \frac{\bar{E}}{2(1+\nu)})$ = Complex shear modulus

$H(x)$ = Heavyside step function

$I (= \frac{h^3}{12})$ = Cross sectional moment of inertia per unit width

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