Dynamic Loading Analysis of the Telescopic Jetty Gangway

by

Ashiran Bin Hamid

Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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BACHELOR OF ENGINEERING (Hons)
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TRONOH, PERAK

Jan 2010

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

ASHIRAN HAMID

ABSTRACT

This report basically discusses the research done and basic understanding of the chosen topic, which are **Dynamic Loading Analysis of the Telescopic Jetty Gangway**. This project is an inclusive research study regarding the harmonic response on a telescopic jetty gangway structure. The project is related to the study on the dynamic characteristics of the jetty gangway structure which exerted the harmonic force at the free end of the beam structure. From the studies, we could find the values of the harmonic response on the telescopic jetty gangway and the mode shape under the influence of harmonic force of sea wave.

The mathematical models are developed based on simplified I-beam geometry by using Newtonian method. The mathematical solutions of the jetty gangway were conducted in two different techniques. In the first technique, the upper and the lower beam were treated as separated parts. The separated calculation processes are conducted in both upper and load beam structure. For the second technique, the jetty gangway structure was treated as a single structure with different value of cross sectional areas at the beginning, middle and far end of the jetty gangway.

The mode shapes and the responses of the beam structure from both techniques were plotted and the data taken were compared with the mode shapes values and patterns of the single uniform beam taken from the previous references and studies.

ACKNOWLEDGEMENTS

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Special thanks to my parents and family members for supporting and encouraging me to pursue this degree. Also special thanks to my best friend, Miss. Asba Madzidah Abu Bakar for helping and supporting me in completing this report.

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Lastly, I offer my regards and blessings to all of those who supported me in any respect during the completion of the project.

CHAPTER 1

INTRODUCTION

1.1 Background

Telescopic jetty gangway is one of the new designs of gangway which give the flexibility to the shore personnel to board and off-board from/to the vessel and ship at the port. This design is created for countering the limitations of the conventional design of the jetty gangway. The limitations are including the rigidity of the gangway structure and the length of the gangway cannot be extended or adjusted to desired length. Many small jetties in the Asia pacific region are facing obstacles in operating their shore based (stationary) gangways due to these limitations. The figure 1 is representing the example of conventional jetty gangway.

Figure 1.1: Conventional Jetty Gangway [Wikipedia.com]

The study on the static load of the telescopic gangway structure had developed and only emphasized on the shear forces, bending moment, shear stress and bending stress of the gangway structure. The dynamic analysis on the telescopic jetty gangway is not conducted for the moment and had been highlighted from the previous study in order to be perform. This study is necessary for evaluating the stability of the telescopic gangway structure for public use.

1.2 Problem Statement

This project is a continuation project of the "Study on Telescopic Walkway Design of Jetty Gangway" conducted by Mr P. Dhinesh Kumar in his Final Year Project at University Teknologi PETRONAS. From the previous project, the design concept of the jetty gangway as well as the study on the loading analysis on the gangway structure had been performed. The study on the loading analysis provides the insightful design parameters for development of the detail design of a jetty gangway. However, the loading analysis only concentrated on the static load analysis and did not covers the study on the dynamic load analysis especially on the dynamic load due to the wave motion at the jetty. The dynamic loading due to the wave motion would create a harmonic motion of the jetty gangway which could result in the destruction of the jetty structure. Therefore, it is necessary to carry out the dynamic load analysis in order to make sure that the jetty gangway can withstand both static and dynamic load.

1.3 Objective

The objective of this project is to conduct the dynamic load analysis on the telescopic jetty gangway structure. The study only focuses on the jetty gangway at the full extension position. The purposes of the project are to collect sufficient data on the mode shape of every modes condition of the jetty gangway and values of the responses of the structure which depends on the frequencies of the wave.

1.4 Scope of Study

The project gives more attention on the dynamics analysis which discuss about the mathematical relations of the parameters of the jetty gangway structure. The study mainly concentrates on finding the relation of the wave (frequency and amplitude) with mode shape and responses (displacement) of the gangway structure. The mathematical relations determined from the study are focusing on the behavior of jetty gangway structure when the harmonic force form the wave is imposed at the free end of the jetty gangway. The study also investigates the stability of the structure in full extension modes with the wave factor that exist in the sea area. The parameter of the wave factor will be narrowed down to South East Asia region. The lists of parameters for the force harmonic are listed below.

- Wave amplitude = 1000 N
- Wave frequency range = 10Hz to 600 Hz

All of the data gained from the study will be collected and arranged to make it more presentable and can be as a reference in the future for researchers, professors and students of Universiti Teknologi PETRONAS.

`CHAPTER 2

LITERATURE REVIEW

2.1 Structure of the Telescopic Jetty Gangway

Gangway is a bridge – like structure used at the berth to access the ship`s deck from the jetty and vice versa. The gangway is comprised of two parallel sides, on which handrails are movably hinged on both sides on a railing. The handrails and the sides form a parallelogram with each inclination and the sides each are supported at the lower end on a roller and rotatable base secured at the upper ends. The trapezoidal steps are flexible suspended with spacing from the sites in which the sides are secured. Gangways may be constructed in steel, aluminum or combination of steel and aluminum.

Usual practice is to construct gangways entirely from aluminum. Aluminum extrusion for the gangway structure shall be aluminum alloy 6061-T6, 6063-T5 and 6063-T6. Powered gangway support structures are usually constructed in steel, with the walkway form the support structure to the ship constructed in aluminum to minimize the gravity loading applied to the support structure and the size of the hydraulic control system required to maneuver the walkway to and from the vessel [P Dhinesh kumar, 2009].

Figure 2.1: Gangway used in ExxonMobil Refinery Jetty in Wakayama, Japan

Figure 2.2: Gangway used in ExxonMobil Refinery Jetty in Sriracha, Thailand

2.2 Study of Vibration

Most human activities involve vibration in one form or other. Any motion that repeats itself after an interval of time is called vibration or oscillation. The theory of vibration deals with the study of oscillatory motions of bodies and forces associated with them.

2.2.1 Vibration Analysis

A system executes an oscillatory motion defined as an aggregation of components acting collectively as a whole. For mechanical systems the oscillatory motion is normally referred to as vibration. Vibration in generally consists of the study on fundamental concept of Newtonian mechanics, components modeling, system modeling, derivation of system differential equation of motion, general excitation, response characteristics and motion stability. The derivation of the equation of motion can be carried out by means of methods of Newtonian mechanics or by methods of analytical dynamics. The fundamental tool in deriving the equation of motion is the free-body diagram, namely, a diagram for each mass in the system showing all the forces acting upon the mass.

A model consists of a collection of either individual components, or group of component or both. The objective of system modeling amounts is to devising a simplified model capable of simulating the behavior of an actual physical system. For vibration systems, the behavior is governed by the equation of motion. In order to derive the system response, the equation of motion needs to be derived in prior. In addition, there is a large variety of excitations, and each type of excitation tends to require a different approach to the solution [Singiresu S. Rao, 2005].

2.2.2 Eigenvalues

$\delta\omega^2$ is known as the characteristic determinant or characteristic polynomial. The characteristic polynomial is of degree n in ω^2 and possesses in general distinct roots referred to as eigenvalues. The n roots are donated by ω_1^2, ω_2^2,.... ω_3^2 and the square roots of these quantities are the system natural frequencies ω_r ($r = 1, 2,\dots,n$). The natural frequencies can be arranged in increasing order of magnitude, namely, $\omega_1^2 \le \omega_2^2 \le \dots \le \omega_3^2$.The lowest frequency ω_1 is referred to as the fundamental frequency and for many practical problems it is the most important one. In general, all frequencies ω_r are distinct and the equality sign never holds, except in degenerate cases which are very random and cannot occur in one degree-dimensional structures but they can occur in two-dimensional symmetric structures. In addition, for the harmonic motion type, the natural frequencies referred as ω_r ($r = 1, 2,\dots,n$) [Leonard Meirovitch, 2001].

2.2.3 Dynamic Analysis

A vibration system is a dynamic system which the input and output are time dependent. The response of a vibrating system generally depends on the initial conditions and the external excitations. Mostly, the real practical vibration problem is very complicated and the variables which included in the mathematical analysis are not totally all been considered and calculated. The complex systems are often been analyzed by a simple model. Usually, the dynamic analysis consists of mathematical modeling, derivation of the governing equations, solution of the equation, and interpretation of the results.

2.2.4 Mode Shape

In the study of vibration in engineering, a mode shape describes the expected curvature (or displacement) of a surface vibrating at a particular mode. To determine the vibration of a system, the mode shape is multiplied by a function that varies with time, thus the mode shape always describes the curvature of vibration at all points in time, but the magnitude of the curvature will change. The mode Shape is dependent on the shape of the surface as well as the boundary conditions of a particular surface [Wikipedia.org].

Figure 3 below describing the mode shape of a string at n = 1, 2 & 3 .

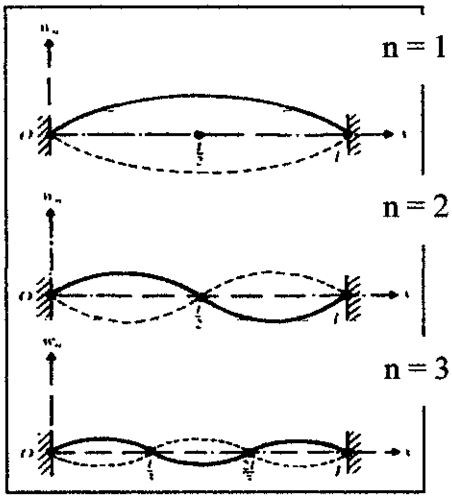


Figure 2.3: The Mode Shape of a String [Singiresu S. Rao, 2004]

2.2.5 Response

Response defined as the ratio of the output to the input, as for given frequency of a system operating under specified condition. The differential equation of motion was needed to be solved in order to derive the system response. Figure 2.4 described the transient and steady state response of a force vibration.

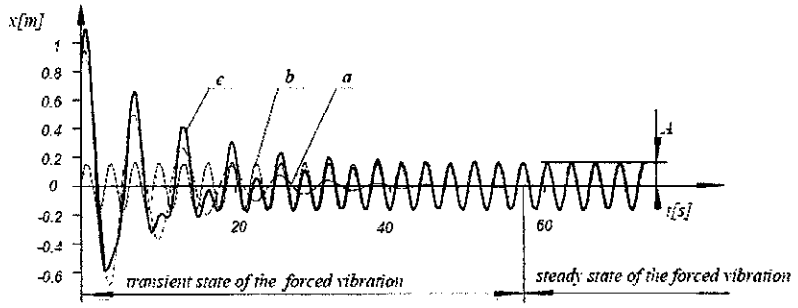


Figure 2.4: Transient and Steady State Response of Forced Vibration [Singiresu S. Rao, 2004]

The nature of the response depends on the excitations and on the system characteristics. The excitations represent external factors and consist of initial displacement and velocity and applied forces and/or moments.

2.2.6 Continuous System

Continuous system is a system of infinite degree of freedom in which it is not possible to identify discrete masses, damping and spring. It is necessary to consider continuous distribution of the mass, damping and elasticity and assume that each of the infinite number of point of the system can vibrate. When a system is modeled as a discrete system, the governing equations are ordinary differential equations which are easy to solve and if the system in modeled as a continuous system, the governing equation are partial differential equation which are more difficult [Singiresu S. Rao, 2004].

2.3 Method for Solving Continuous System

The continuous system is not similar compare to the discrete system. The motion of discrete system is governed by ordinary differential equations. In discrete system, the mass of beam, string and rod can be neglected and were treated as an equivalent springs. The mass of the elastic members which treated as continuous system is not neglected and the members can no longer be regarded as equivalent springs.

The methods to solve the continuous problem (displacement) are depended on two independent variables, x and t. The motion of continuous system is governed by partial differential equations and must satisfy the boundary conditions of the particular system. Due to the existing of boundary conditions, the continuous system also considered as a boundary-value problem (BVP).The methods to solve the continuous system can be divided into two approaches.

- Exact solution
- Approximate solution

In most part, problems in continuous system do not admit exact solution since the system parameters are appear in the form of coefficients in the partial differential equation and the boundary conditions of the system are depended explicitly on the spatial of variable x. As a result, the exact solution can only be used in a system with uniform mass and stiffness distributions [Leonard Meirovitch, 2001].

2.3.1Exact solution

The exact solution consists of two methods, which are:

- Newtonian Method
- Extended Hamilton Principle

Newtonian approach requires a free body diagram for a differential element of mass. It also applies the usage of sign conventions for forces and moments where *M(x,t)* is referring to the bending moment and *Q(x,t)* is referring to the shearing force. The figure below showed the example of free-body diagram with it respective sign.

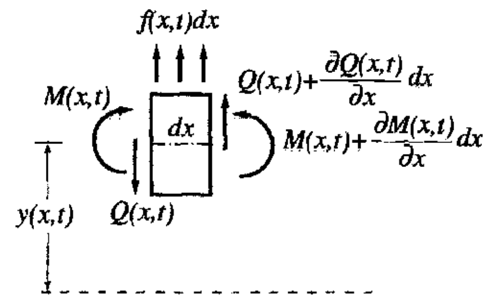


Figure 2.5: Free Body Diagram of Beam [Leonard Meirovitch, 2001]

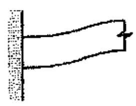
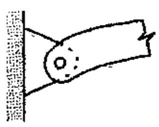
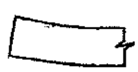
From the free body diagram, the force equation of motion and the moment equation of motion will be obtained and later will be manipulated in order to find the total solution equation. The total solution is referring to the equation of displacement for system with regard to x and t and must satisfied the boundary conditions of the particular solution.

By applying the partial differential method on force equation and moment equation, the equation of motion for structure can be derived which is shown below:

$$EL\frac{\partial^2 w}{\partial x^4}(x,t) + \rho A\frac{\partial^2 w}{\partial t^2}(x,t) = f(x,t) \ldots\ldots\ldots\ldots\ldots\ldots (2.3)$$

The boundary typical boundary conditions for the continuous system (in this case: the structure of beam system) are stated in the table below:

Table 2.1: Boundary Condition of a Beam Structure

No	Typical end of beam system	Boundary condition
1.	Fixed end 	• Displacement, $w(x) = 0$ • Slope, $\frac{dw(x)}{dx} = 0$
2.	Simply supported (pinned) end 	• Displacement, $w(x) = 0$ • Bending moment, $EI\frac{d^2 w(x)}{dx^2} = 0$
3.	Free end 	• Bending moment, $EI\frac{d^2 w(x)_C}{dx^2} = 0$ • Shear force, $-EI\frac{d^3 w(x)_C}{dx^3} = 0$

The extended Hamilton's principle is derived from the generalized d'Alembert's principle. The principle can be used to derive all the system equations of motion, regardless of whether the system is subjected to constraints or not. This principle also involves the kinetic energy, potential energy and the virtual work application in order to solve the system's problem. The Hamilton's approach also implemented the theory of energy conservation by mean of solving the system and structure problems. The extended Hamilton principle in the form of [Leonard Meirovitch, 2001]:

$$\int_{t1}^{t2} (\delta T - \delta V + \overline{\delta W}_{nc})dt = 0, \delta y(x,t) = 0, 0 \leq x \leq L, t = t_1, t_2$$

Where

$$T(t) = \frac{1}{2}\int_0^L \rho(x)\left[\frac{\partial y(x,t)}{\partial t}\right]^2 dx$$

is the kinetic energy and

$$\overline{\delta W_{nc}}(t) = \int_0^L f(x,t)\delta y(x,t)dx$$

is the virtual work of the non-conservative distributed force.

The variation relations in potential and kinetic energies were interpolated with regards to the geometry conditions of the structure in order to derive the equation which completes the derivation of the boundary-value problem. The equation is illustrated below [Leonard Meirovitch, 2001]:

$$T\frac{\partial y}{\partial x} + ky = 0, x = L$$

2.3.2 Approximate solution

The exact solution only compatible for systems characterised by uniformly distributed parameters and simple boundaries. In real life, however, most systems do not posses these properties. As a result, the approximate solutions need to be applied in order to solve the boundary-value problem. The approximate techniques normally model the continuous systems as discrete systems which amount to spatial discretization and truncation. The lists of common approximate solutions for solving the boundary-value problem are shown below [Singiresu S. Rao, 2004]:

- Rayleigh`s principle method
- Rayleigh-Ritz method
- Finite Element method

2.3.3 Rayleigh principle method

The Rayleigh`s quotient has a stationary value in the neighborhood of an eigenfunction and this stationary value is actually a minimum at the lowest eigenfunction. The Rayleigh`s principle can be presented by the equation below:

$$\lambda_1 = \omega_1^2 = \min R(Y) = R\ (Y_1)$$

If $\omega(x)$ is the mode shapes, then is equal to the square of the natural frequency of that mode. If ω is not a mode shape, the $R(\omega)$ is the scalar function for ω. For a discrete systems, $R(\omega)$ is a minimum when ω is a mode shape. Therefore, Rayleigh`s quotient can be used to approximate the lowest natural frequency for continuous system.

Rayleigh method can be applied to find the fundamental natural frequency of continuous systems. This method is much simpler than exact analysis for system with varying distribution of mass and stiffness. In order to apply Rayleigh`s method, we need to derive expression for the maximum kinetic, potential energies and Rayleigh quotient.

$$U_{max} = \frac{1}{2}\int \frac{M^2}{EI} dx = \frac{1}{2}\int EI\left(\frac{d^2y}{dx^2}\right)^2 dx$$

$$T_{max} = \frac{1}{2}\int \dot{y}^2 \, dm = \frac{1}{2}\omega^2 \int y^2 dm$$

$$\omega^2 = \frac{\int EI\left(\frac{d^2y}{dx^2}\right)^2 dx}{\int y^2 dm}$$

The usage of Rayleigh method is highly appreciated in continuous system. This is due to the significant of fundamental frequency in continuous system as the forced responses in many cases of continuous system are in large magnitudes. The Rayleigh method can be used to determine the fundamental frequency of a beam or shaft represented by a series of lumped masses [Singiresu S. Rao, 2004].

2.3.4 Rayleigh-Ritz method

The Rayleigh-Ritz method can be considered as an extension of Rayleigh method. It is base on the premise that the closer approximately to the exact natural mode can be obtained by superimposing a number of assumed functions than by using a single assumed function, as in Rayleigh method. The equation 2.8.1.1 below stated the deflection equation when n functions are chosen.

$$W(x) = c_1 w_1(x) + c_2 w_2(x) + \ldots + c_n w_n(x) \ldots\ldots (2.9)$$

If the assumed functions are suitably chosen, this method provided not only the approximate value of the fundamental frequency but also the approximate values of the higher natural frequencies and the mode shapes.

It is usual to approach the problem in vibration problem by using energy principles either with a Rayleigh-Ritz Method (continuous series) or with the finite element method. A number of studies have been made which employed different versions of the latter technique and these have dealt with a variety of boundary conditions. However, not all possible boundary conditions have been considered and there has also been the problem of obtaining convergence of the solution when the number of terms in the series solution is increased. The Rayleigh-Ritz approaches are suitable for the symmetric balanced and unbalanced cases. It is demonstrated that the solution by using Rayleigh-Ritz method possesses good numerical characteristics and convergence properties.

In Rayleigh Ritz method, an arbitrary number of functions can be used in order to get the accurate result as the number of frequencies can be obtained is equal to the number of functions used. However, the amount of computation required becomes prohibitive for the asymmetric case where the finite element technique should be used and implemented [Singiresu S. Rao, 2004].

2.3.5 Finite element method

The basic idea in the finite element method is to find the solution of a complicated problem by replacing it by a simpler one. Since the actual problem is replaced by a simpler one in finding the solution, we will be able to find only an approximate solution rather than the exact solution. The existing mathematical tools will not be sufficient to find the exact solution (and sometimes, even an approximate solution) of most of the practical problems. Thus, in the absence of any other convenient method to find even the approximate solution of a given problem, we have to prefer the finite element method. Moreover, in the finite element method, it will often be possible to improve or refine the approximate solution by spending more computational effort. The example of finite element structure of milling machines structures by finite element

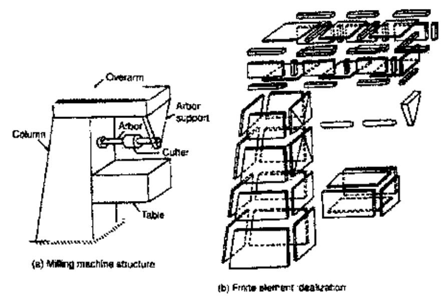


Figure 2.6 : The Diagram of the Finite Element Structure of a Milling Machine

The principal advantage of the finite element method is its generality; it can be used to calculate the natural frequencies and mode shapes of any linear elastic system. However, it is a numerical technique that requires a fairly large computer, and care has to be taken over the sensitivity of the computer output to small changes in input. For beam type systems the finite element method is similar to the lumped mass method, because the system is considered to be a number of rigid mass elements of finite size connected by massless springs. The infinite number of degrees of freedom associated with a continuous system can thereby be reduced to a finite number of degrees of freedom, which can be examined individually.

2.4 Method Used in the Project

In order to solve the project, I am choosing to use the Newtonian method as the primary calculation method. The main reason of using the Newtonian Method is because of the geometry condition of the beam structure which in uniformly structure. The Newtonian method may be considered as the best method in order to solve this boundary-value problem.

The result achieve from the Newtonian method will be accurately compare to the Rayleigh Principle and Finite element methods. Both methods can be relying on when more functions and iterations been included in the calculation. As a result, the calculation will be long and tedious and not suitable for manual analytical calculation.

2.5 Study Of Non-Linear Beam Analysis By Mesut Simsek

From the study on Mesut Simsek, researcher from Yildiz Technical University, Turkey, in his journal " Non-linear Vibration Analysis of a Functionally graded Timeshenko beam under action of a moving load" , he already specify the technique for study the non-linear beam structure. In this reference, the effect of large deflection, material distribution, velocity of the moving load and excitation frequency on the beam displacement, bending moments and stresses have been examined in detail.

In this journal, the result gain from the study then was compared with the existing result of a linear beam which was gain from the previous study. This is conducted in order to verify the result gain from his study where the result gain from linear beam will be treated as references values. Below are one of the results gain from the Mesut`s study which shown the transverse displacement of the beam with respected to the time for different value of the load velocity (20m/s) and the excitation of frequency (20 rad/s). Both line of the graph are represented the displacement of a linear and non-linear beam.

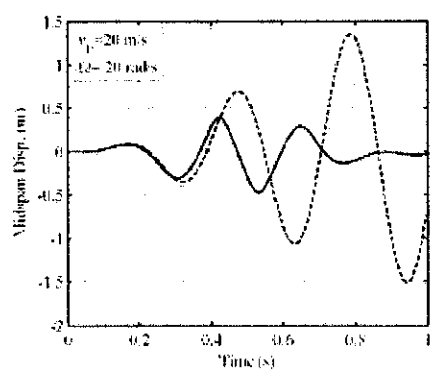


Figure 2.7: The Displacement of the Beam With Respect of Time for Linear (---) and Non-Linear (---) [Mesut Simsek,2010]

The comparison method of the linear and non-linear beam which been applied in this reference can be implemented in this project.

2.6 Dynamic Analysis of Uniform Cantilever Beam Subjected to Harmonic Force

Based on the study conducted by Demeter G. Fertis in his book, Mechanical and Structural Vibration, the solution of for the uniformly cantilever beam subjected to the vertical harmonic force at the free end of the beam are stated below:

The total solution of the beam given:

$$w(x,t) = W(x)\,.\,T(t)$$

where

$$W(x) = \frac{F}{2\beta^3 EI[1 + (\cosh\beta L)(\cos\beta L)]}[(\sinh\beta L + \sin\beta L)(\cosh\beta L - \cos\beta L) - (\cosh\beta L + \cos\beta L)(\sinh\beta L - \sin\beta L)]$$

$$T(t) = \sin\omega_f t$$

The notation given:

F = the amplitude of wave

ω_f = the frequency of the harmonic force.

The model of the beam from the book is illustrated below:

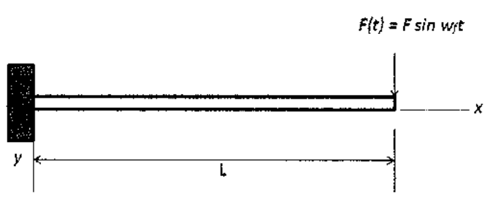


Figure 2.8: Beam Structure [Demeter G. Fertis,1995]

From the study conducted by Demeter G. Fertis , number of graph been plotted which illustrated the relationship of the displacement of the cantilever beam and the wave frequency of the harmonic force. The graph plotted base on the detail of the beam which are stated below:

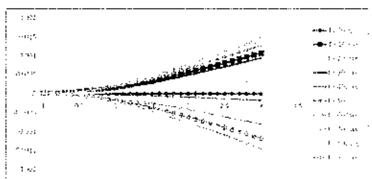
Length of beam, L = 5 meter

Modulus of elasticity, E = 69 Gpa (Aluminum 6061)

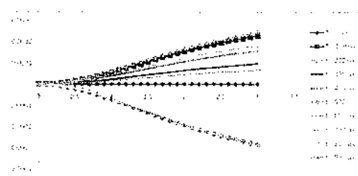
Cross sectional area, A = $0.0198m^2$

Moment of inertia, I = $0.000108\ m^4$

Graph of mode shape of beam with frequency varies from 0HZ to 700HZ are illustrated below:



(a)



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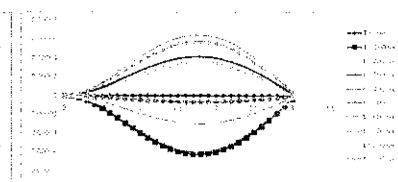

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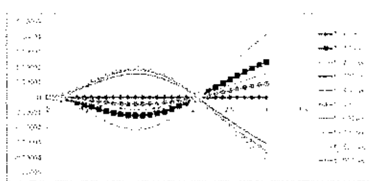

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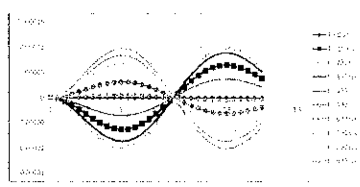

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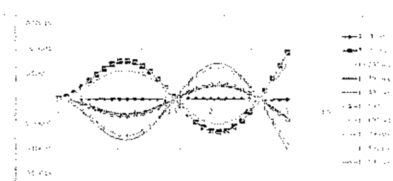

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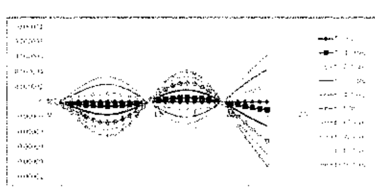

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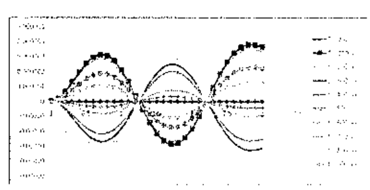

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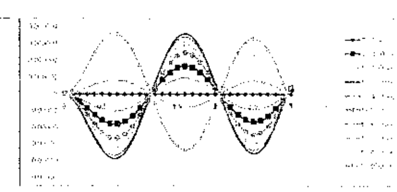
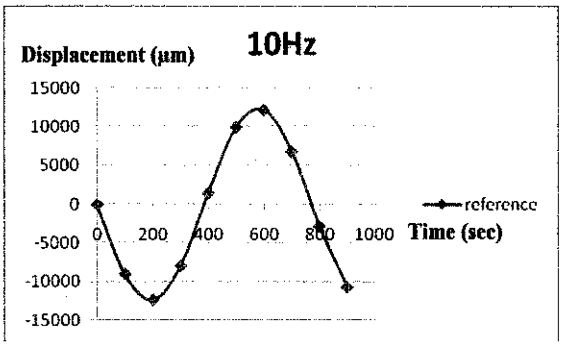


(i)

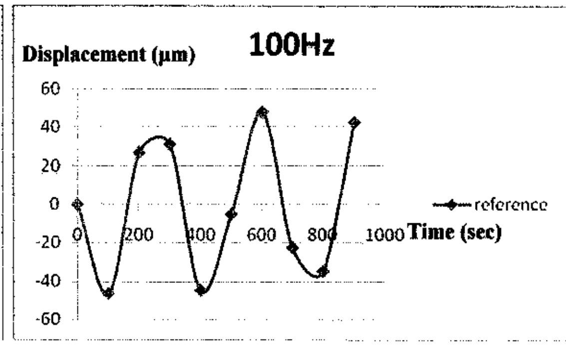
The graph (a) is the mode shape for frequency is 10Hz , (b) for 50Hz, (c) for 100Hz, (d) for 200Hz, (e) for 300Hz, (f) for 400Hz, (g) for 500Hz, (h) for 600Hz and (i) for 700Hz.

The relation of the displacement of the free end of the beam varies with the value of frequencies for 0Hz to 700Hz also had been plotted as shown below. The graph (a) is the displacement of the beam at frequency of 10Hz, varies from 0 sec until 900 seconds , (b) for 100Hz, (c) for 200Hz, (d) for 300Hz, (e) for 400Hz, (f) for 500Hz, (g) for 600Hz and (i) for 700Hz.



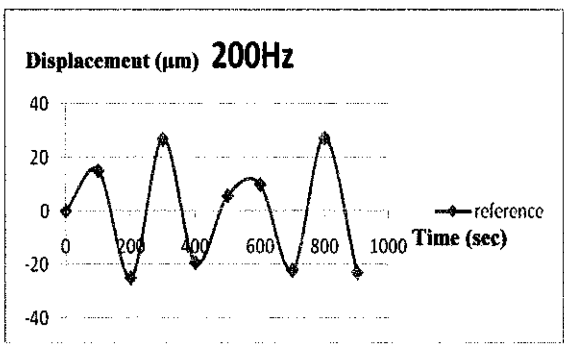
Displacement (μm)
10Hz
reference
Time (sec)

(a)



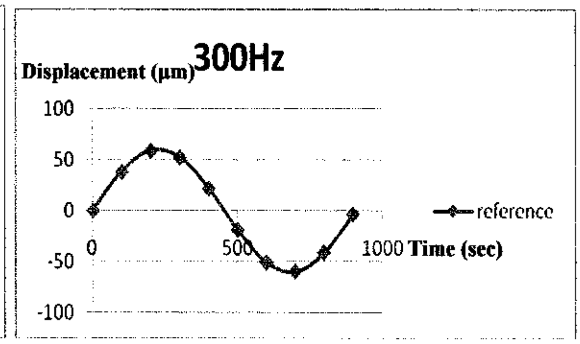
Displacement (μm)
100Hz
reference
Time (sec)

(b)



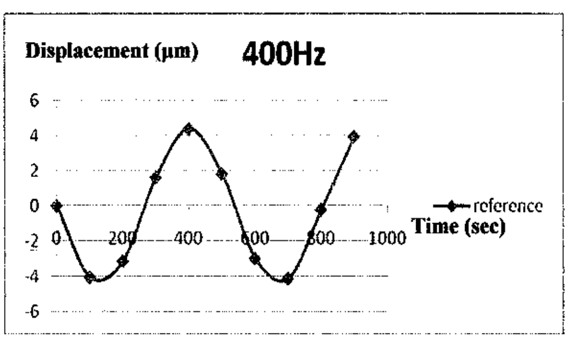
Displacement (μm)
200Hz
reference
Time (sec)

(c)



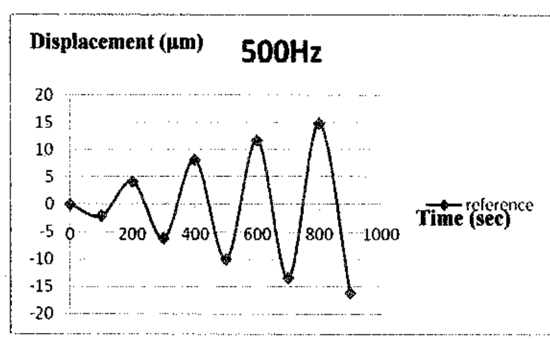
Displacement (μm)
300Hz
reference
Time (sec)

(d)



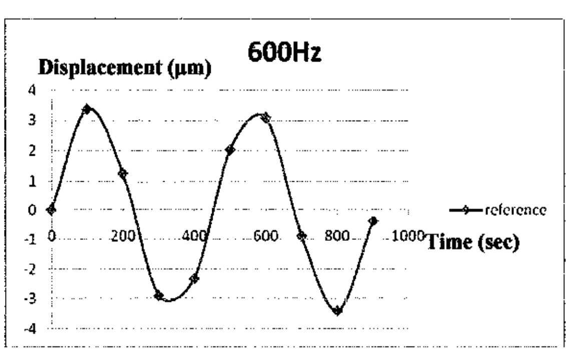
Displacement (μm)
400Hz
reference
Time (sec)

(e)



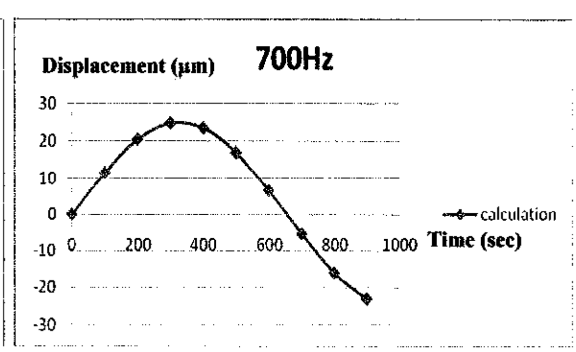
Displacement (μm)
500Hz
reference
Time (sec)

(f)



Displacement (μm)
600Hz
reference
Time (sec)

(g)



Displacement (μm)
700Hz
calculation
Time (sec)

(h)

CHAPTER 3

METHODOLOGY

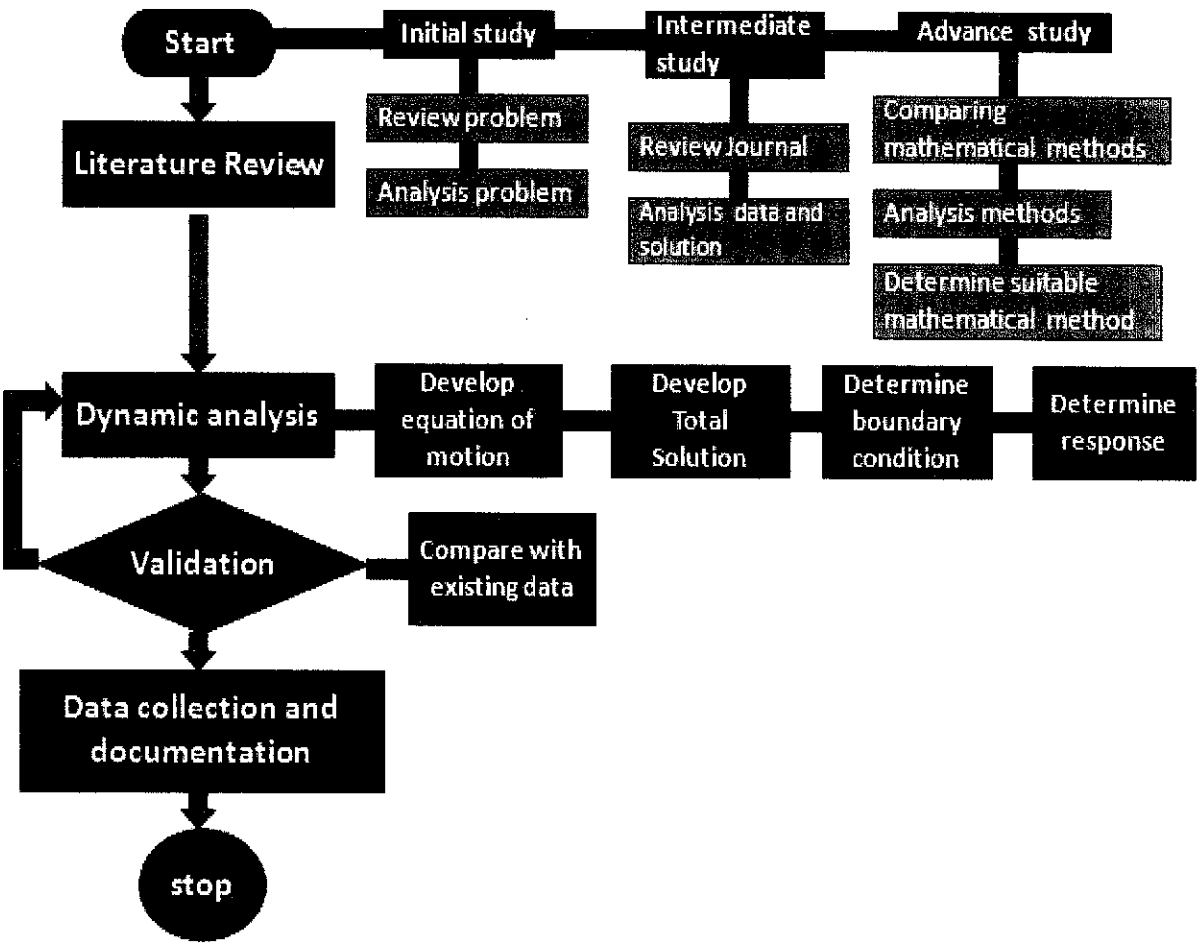


Figure 3.1: The Diagram of the Methodology for the Project

The projects will begin with literature review on several sessions. The session divided into 3 parts, which are:

- Initial Study
- Intermediate study
- Advance study

3.1 Initial study

The initial study was performed in order to know and develop a complete design structure of the telescopic jetty gangway with the full dimension referring to the standard of designing a gangway. By having a correct dimensional of a gangway, it will make the dynamics analysis more reliably and meaningful. The mechanical properties of the jetty gangway also were determined. The picture below represents the model of the jetty gangway which related to this project.

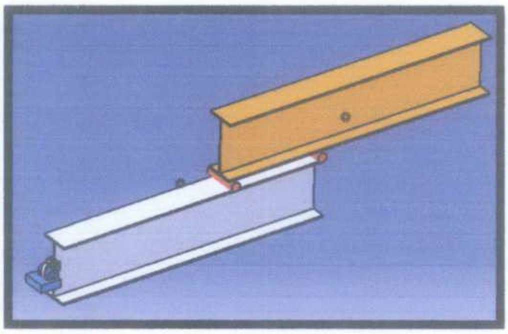
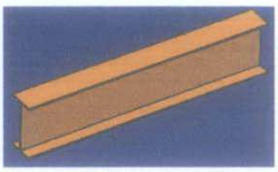


Figure 3.2: CAD design of the Jetty Gangway

Table 3.1: Detail and Properties of the Jetty Gangway

Figure	Detail Of Beam	
	• **Name : UPPER BEAM** • **Material type: Aluminum 6061** • **Density : 2700 kg/m^3** • **Modulus of Elasticity : 69 Gpa**	• **Mass : 159.259 kg** • **Length: 3 meter** • **Poisson`s ratio : 0.35** • **Cross sectional Area : 0.0198m^2**

	• **Name : LOWER BEAM** • **Material type: Aluminum 6061** • **Density : 2700 kg/m^3** • **Modulus of Elasticity : 69 Gpa** • **Mass : 159.259kg** • **Length: 3 meter** • **Poisson`s ratio : 0.35** • **Cross sectional Area : 0.0198m^2**

[Detail of information of the beam can be illustrated at **Appendix C**)

The position of the full extension of the jetty gangway is presented in figure below:

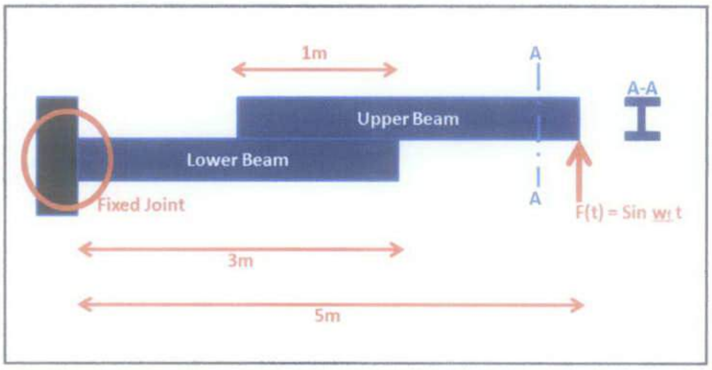


Figure 3.3: The Jetty Gangway Structure Diagram

3.2 Intermediate study

When the design of the gangway structure been finalized, a comprehensive studies related to the fundamental of a continuous system were performed. The studies provide the information on the behavior and responses (displacements) of the continuous system under the transverse and longitudinal vibrational forces. From the studies conducted, method of finding the natural frequency and mode shape of a continuous system were also determined. The studies related to the fundamental only were insufficient for solving this project. In order to make the research of the project more presentable and reliable, the literature review will be done from previous papers and journals which are related to the dynamics analysis of a structure. By conducting the literature review on the real research areas, a lot of ideas and methods which researchers used and emphasized on their studies were identified. The full information with regards to this topic can be reviewed in the previous chapter (literature review).

3.3 Advance study

When the studies on the dynamics analysis are completed, a set of methodology for solving the dynamics mathematical relations for this project been developed. The suitable approach for finding the total responses and mode shapes of a continuous system will be specified. The most suitable approach for this project is the Newtonian method. This approach been used in the solution and was referred at most time through out of the project period.

The mathematical solutions of the jetty gangway (by using the Newtonian approach) were conducted in two different techniques. In the first technique, the upper and the lower beam were treated as separated parts. The separated calculation process was conducted in both parts. The figure below shows the jetty gangway which the upper beam and the lower beam were treated as an individual beam structure.

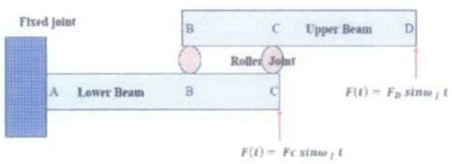


Figure 3.4: The Diagram Of Jetty Gangwat Sturcure For First Technique

For the second technique, the jetty gangway structure was treated as a single structure with different cross sectional areas at the beginning, middle and far end of the jetty gangway structure. The figure below illustrates the characteristic of the jetty gangway under the second technique.

Figure 3.5: The Diagram of Jetty Gangwat Sturcure For Second Technique

3.4 Elaboration of the First Technique

3.4.1 Develop the Equation of Motion

The Newtonian Method had been implemented In order to solve the problem. The picture below showed the physical model of continuous system of beam structure.

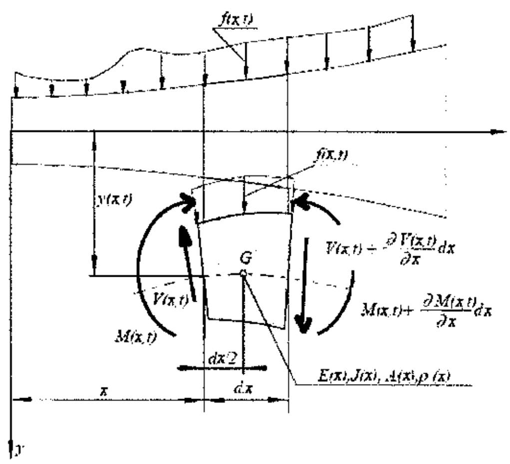


Figure 3.6: Free Body Diagram Of Beam [Leonard Meirovitch,2001]

There are some considerations/ assumptions applied in order to solve the problem:

1. Beam are assume to be as a elastic elements that are subjected to lateral loads which are the forces or moments that have their vector perpendicular to the centre line of a beam.
2. The beam performs vibrations due to the external distributed unit load , $f(x,t)$
3. Let consider $I(x)$ as beam moment of inertia, cross section $A(x)$, density $\rho(x)$ and the young`s modulus $E(x)$

From the free body diagram above, the force equation of motion in the y direction gives,

$$-(V+dV)+f(x,t)dx+V=\rho A(x)dx\frac{\delta^2 w}{\delta t^2}(x,t) \qquad (3.1)$$

The moment equation of motion about z axis passing through point G in figure x above leads to

$$(M+dM)-(V+dV)dx+f(x,t)dx\frac{dx}{2}-M=0 \qquad (3.2)$$

By writing $dV = \frac{\partial V}{\partial x}dx$ and $dM = \frac{\partial M}{\partial x}dx$, and disregarding terms involving second power in dx, both equation (3.1) and (3.2) can be written as

$$-\frac{\partial V}{\partial x}(x,t) + f(x,t) = \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) \tag{3.3}$$

$$\frac{\partial M}{\partial x}(x,t) - V(x,t) = 0 \tag{3.4}$$

Rearrange the equation (3.4) to be $V = \partial M/\partial x$ and then put into equation (3.3), then it becomes

$$-\frac{\partial^2 M}{\partial x}(x,t) + f(x,t) = \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) \tag{3.5}$$

Base on the Euler-Bernoulli theory, the mathematical relation of bending moment and deflection can be considered as

$$M(x..t) = EI(x)\frac{\delta^2 w}{\delta x^2}(x,t) \tag{3.6}$$

We then can modify equation (3.5) by using equation (3.6) and it becomes as

$$\frac{\partial^2}{\partial x^2}\left[EI(x)\frac{\delta^2 w}{\delta x^2}(x,t)\right] + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = f(x,t) \tag{3.7}$$

The equation (3.7) is the equation of motion for force lateral vibration. But the equation only true for non-uniform beam. Equation of motion for uniform beam can be expressed by reducing the equation (3.7) to be:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = f(x,t) \tag{3.8}$$

The equation (3.8) can be express as the general equation of motion for force vibration. The general equation of motion for free vibration of uniform beam then can be expressed as:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = 0 \tag{3.9}$$

The equation (3.8) then been modified by letting the force vibration to be $f(x,t) = qsin\omega_f t$ and then the equation become:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = qsin\omega_f t \tag{3.10}$$

The equation (3.10) express the equation of motion of a uniform beam with subjected to the harmonic force vibration where ω_f is the frequency, in radian per second.

3.4.2 Determine the Total Response Equation

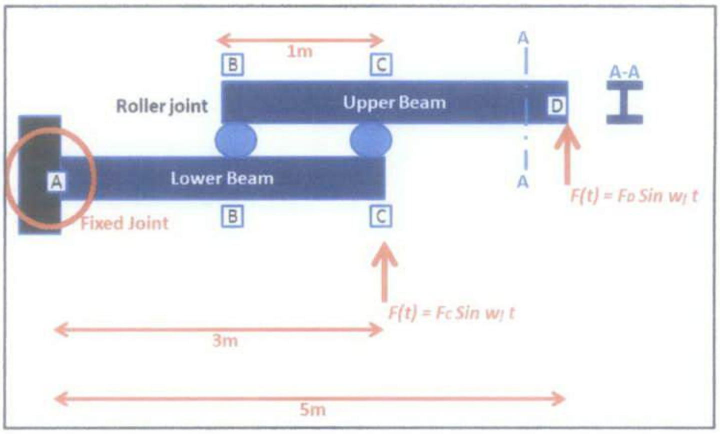


Figure 3.7: The Diagram of Jetty Gangway with the Harmonic Force at Point C and D

In this technique, the beam will be separated into two individual bodies which are: the lower beam and the upper beam. Each beam will be treated as an individual beam and the calculation for solving the boundary value problem will be conducted for both beam structures. The transfer function method is applied in order to relocate the harmonic force from point D to point C. The detail of the calculation step for transfer function method is stated in the **appendix A.**

3.4.3 Calculation for Lower Beam

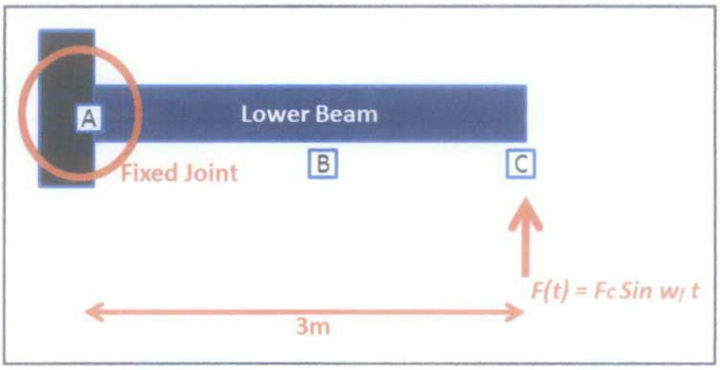


Figure 3.8: The Diagram of Lower Beam

The lower beam will be treated as a uniform cantilever beam and the calculation step for finding the total response for lower beam are stated below.

Consider now the lower beam is loaded with concentrated harmonic force F (t) at its free end by this expression:

$$F(t) = F_C \, sin\omega_f \tag{3.11}$$

Where, *FC* is referring to the amplitude of the wave force which is equivalent to 930.667N.

Let consider that, the governing differential equation of motion for the length L of the members between the fixed support and just to the left of the applied harmonic force $F(t) = F_C sin\omega_f t$ is the same as equation (3.10) when the load $qsin\omega_f t$ is made equal to zero which shown below:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = 0 \tag{3.12}$$

The steady-state motion cause by the force given by harmonic force will be determined. We assume the total deflection, w(x,t) of the jetty gangway to be of the form of:

$$w(x,t) = W(x).T(t) \tag{3.13}$$

Where, W(x) is a function of x only and T (t) is a function of t only.

The force vibration solution can be found by substituting equation (3.13) into equation (3.12) and perform separating variable, we will obtain:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4W(x)}{dx^4} = \frac{-1}{T(t)}\frac{d^2T(t)}{dt^2} \tag{3.14}$$

The right side and the left side of equation (3.14) will be equal to each other only if both sides are equal to a constant. So the constant is assumed to be equal to ω^2 , where it is a positive constant. The modified equation (3.14) shown below:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4W(x)}{dx^4}=\frac{-1}{T(t)}\frac{d^2T(t)}{dt^2}=\omega^2 \tag{3.15}$$

Since ω^2 is a positive constant and cannot be equal to zero, then equation (3.15) can be expressed in two separated equations:

$$\frac{d^4W(x)}{dx^4}-\frac{\rho A\omega^2}{EI}W(x)=0 \tag{3.16}$$

$$\frac{d^2T(t)}{dt^2}+\omega^2T(t)=0 \tag{3.17}$$

We can assume that, the solution for equation (3.17) can be expressed as

$$T(t) = sin\omega_f t \tag{3.18}$$

The expression of equation (3.18) is a reasonable assumption since the applied force is a sinusoidal. Equation (3.18) then was substitute into equation (3.17) and yield:

$$\omega^2 = \omega_f{}^2 \tag{3.19}$$

From expression of equation (3.19), Equation (3.16) can be modified by letting $\beta^4 = \frac{\rho A\omega_f{}^2}{EI}$ which lead to

$$\frac{d^4W(x)}{dx^4}-\beta^4W(x)=0 \tag{3.20}$$

The solution of equation (3.20) can be assumed to be:

$$W(x) = Ce^{sx} \tag{3.21}$$

Where C and S are constants and derive the auxiliary equation as

$$S^4 - \beta^4 = 0 \tag{3.22}$$

And the roots of the equation are

$S_{1,2} = \pm\beta, \quad S_{3,4} = \pm i\beta$

Hence the solution for equation (3.20) becomes

$$W(x) = C_1 e^{\beta x} + C_2 e^{-\beta x} + C_3 e^{i\beta x} + C_4 e^{-i\beta x} \tag{3.23}$$

Where, C_1, C_2, C_3,and C_4 are constants . Equation (3.23) above also can be expressed as

$$W(x) = C_1 cos\beta x + C_2 sin\beta x + C_3 cosh\beta x + C_4 sinh\beta x \tag{3.24}$$

The values ofC_1, C_2, C_3, and C_4 can be found from the boundary conditions and the value of β can be found from equation below:

$$\beta^4 = \frac{\rho A \omega_f^2}{EI}$$

3.4.4 Determine the boundary condition

The boundary conditions of the lower beam are stated below:

At the fixed joint (x=0),

$W(x = 0) = 0$,....... (3.25)

$\frac{dW(x=0)}{dx} = 0$,........ (3.26)

At the free end (x = L), where L = 3

$\frac{d^2 W(x=L)}{dx^2} = 0, \ldots\ldots\ldots (3.27)$

$-EI\frac{d^3 W(x=L)}{dx^3} = F_C, \ldots\ldots (3.28)$

From equation (3.25) and (3.24)

$C_1 + C_3 = 0 \ldots\ldots (3.29)$

From equation (3.26) and (3.24)

$C_2 + C_4 = 0 \ldots\ldots\ldots (3.30)$

From equation (3.27) and (3.24)

$C_1(-\cos\beta L) + C_2(-sin\beta L) + C_1(\cosh\beta L) + C_4(sinh\beta L) = 0 \ldots (3.31)$

From equation (3.28) and (3.24)

$C_1\beta^3(\sin\beta L) + C_2\beta^3(-\cos\beta L) + C_3\beta^3(\sinh\beta L) + C_4\beta^3(cosh\beta L) = \frac{-F_C}{EI}(3.32)$

Then, rearrange the equation (3.29), (3.30), (3.31) & (3.32) into the matrix form,

$$\begin{bmatrix} 1 & 0 & 1 & 0 \\ 0 & 1 & 0 & 1 \\ -cos\beta L & -sin\beta L & cosh\beta L & sinh\beta L \\ \beta^3(\sin\beta L) & \beta^3(-\cos\beta L) & \beta^3(\sinh\beta L) & \beta^3(cosh\beta L) \end{bmatrix} \begin{bmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ -\frac{F_C}{EI} \end{bmatrix}$$

From the matrix above, the values of constant C_1, C_2, C_3 & C_4 can be found by using MATLAB. The related steps of calculation are attached in **appendix B1** for references.

3.4.5 Calculation for upper beam

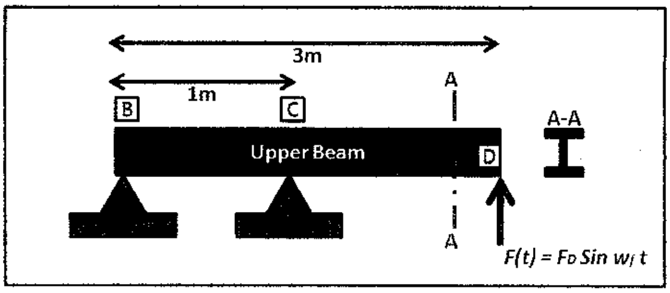


Figure 3.9: The Diagram of Upper Beam

The lower beam will be treated as a uniform cantilever beam and the roller joint at the point B and C are treated as a fixed joint since the roller will be lock up in order to be used by shore personnel. The calculation steps for finding the total response for lower beam are stated below.

Consider now the lower beam is loaded with concentrated harmonic force F (t) at its free end by this expression:

$$F(t) = F_D \, sin\omega_f \quad (3.33)$$

Where, *FD* is referring to the amplitude of the wave force which is equivalent to 1000N.

Let consider that the governing differential equation of motion for the length L of the members between the fixed support and just to the left of the applied harmonic force $F(t) = F_D sin\omega_f t$ is the same as equation (3.10) when the load $qsin\omega_f t$ is made equal to zero which shown below:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = 0 \quad (3.34)$$

The steady-state motion cause by the force given by harmonic force will be determined. We assume the total deflection, w(x,t) of the jetty gangway to be of the form of:

$$w(x,t) = W(x).T(t) \tag{3.35}$$

Where, W(x) is a function of x only and T (t) is a function of t only.

The force vibration solution can be found by substituting equation (3.35) into equation (3.34) and perform separating variable, we will obtain:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4W(x)}{dx^4} = \frac{-1}{T(t)}\frac{d^2T(t)}{dt^2} \tag{3.36}$$

The right side and the left side of equation (3.36) will be equal to each other only if both sides are equal to a constant. So the constant is assumed to be equal to ω^2, where it is a positive constant. The modified equation (3.36) shown below:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4W(x)}{dx^4} = \frac{-1}{T(t)}\frac{d^2T(t)}{dt^2} = \omega^2 \tag{3.37}$$

Since ω^2 is a positive constant and cannot be equal to zero, then equation (3.37) can be expressed in two separated equations:

$$\frac{d^4W(x)}{dx^4} - \frac{\rho A\omega^2}{EI}W(x) = 0 \tag{3.38}$$

$$\frac{d^2T(t)}{dt^2} + \omega^2 T(t) = 0 \tag{3.39}$$

We can assume that, the solution for equation (3.39) can be expressed as

$$T(t) = sin\omega_f t \tag{3.40}$$

The expression of equation (3.40) is a reasonable assumption since the applied force is a sinusoidal. Equation (3.40) then was substitute into equation (3.39) and yield:

$$\omega^2 = {\omega_f}^2 \tag{3.41}$$

Equation (3.38) can be modified by letting $\beta^4 = \frac{\rho A \omega_f^2}{EI}$ which lead to

$$\frac{d^4W(x)}{dx^4} - \beta^4 W(x) = 0 \tag{3.42}$$

The solution of equation (3.42) can be assumed to be:

$$W(x) = Ce^{sx} \tag{3.43}$$

Where C and S are constants and derive the auxiliary equation as

$$S^4 - \beta^4 = 0 \tag{3.45}$$

And the roots of the equation are

$S_{1,2} = \pm\beta, \quad S_{3,4} = \pm i\beta$

Hence the solution for equation (3.42) becomes

$$W(x) = C_1 e^{\beta x} + C_2 e^{-\beta x} + C_3 e^{i\beta x} + C_4 e^{-i\beta x} \tag{3.46}$$

Where, C_1, C_2, C_3,and C_4 are constants . Equation (3.46) above also can be expressed as

$$W(x) = C_1 cos\beta x + C_2 sin\beta x + C_3 cosh\beta x + C_4 sinh\beta x \tag{3.47}$$

The values ofC_1, C_2, C_3, and C_4 can be found from the boundary conditions and the value of β can be found from equation below:

$$\beta^4 = \frac{\rho A \omega_f^2}{EI}$$

3.4.6 Determine the boundary condition

Boundary conditions for the upper beam are stated below:

At the fixed joint (x=0),

$W(x = 0) = 0$,........ (3.48)

$\frac{dW(x=0)}{dx} = 0$,...... (3.49)

At the fixed joint (x=Lc=1m),

$W(x = L_C) = 0$,........ (3.50)

$\frac{dW(x=L_C)}{dx} = 0$,...... (3.51)

At the free end (x = LD=3m),

$\frac{d^2\, W(x=L_D)}{dx^2} = 0$,....... (3.52)

$-EI\frac{d^3 W(x=L_D)}{dx^3} = F_D$,...... (3.53)

The equation (3.47) and its respective derivative equations are stated below:

$$W(x) = C_1 \cos\beta x + C_2 \sin\beta x + C_3 \cosh\beta x + C_4 \sinh\beta x \quad (3.47)$$

$$\frac{dW(x)}{dx} = C_1\beta(-\sin\beta x) + C_2\beta(\cos\beta x) + C_3\beta(\sinh\beta x) + C_4\beta(\cosh\beta x) \quad (3.54)$$

$$\frac{d^2W(x)}{dx^2} = C_1\beta^2(-\cos\beta x) + C_2\beta^2(-\sin\beta x) + C_3\beta^2(\cosh\beta x) + C_4\beta^2(\sinh\beta x) \quad (3.55)$$

$$\frac{d^3W(x)}{dx^3} = C_1\beta^3(\sin\beta x) + C_2\beta^3(-\cos\beta x) + C_3\beta^3(\sinh\beta x) + C_4\beta^3(\cosh\beta x) \quad (3.56)$$

Since the numbers of boundary condition are exceeding the number of constants, the suitable 4 boundary conditions should be chosen from the existing 6 boundary condition. Then substitute the equation (4.38) and it derivatives into respective boundary condition equation

From equation (3.47) and (3.48)

$$C_1 + C_3 = 0 \quad \ldots\ldots\ldots\ldots (3.57)$$

From equation (3.47) and (3.50)

$$W(x) = C_1 cos\beta x + C_2 sin\beta x + C_3 cosh\beta x + C_4 sinh\beta x \quad \ldots.. (3.58)$$

From equation (3.52) and (3.55)

$$C_1\beta^2(-\cos\beta L_D) + C_2\beta^2(-\sin\beta L_D) + C_3\beta^2(\cosh\beta L_D) + C_4\beta^2(\sinh\beta L_D) = 0 \quad \ldots\ldots (3.59)$$

From equation (3.53) and (3.56)

$$C_1\beta^3(\sin\beta L_D) + C_2\beta^3(-\cos\beta L_D) + C_3\beta^3(\sinh\beta L_D) + C_4\beta^3(\cosh\beta L_D) = -\frac{F_D}{EI} \ldots (3.60)$$

Rearrange the equation (3.57), (3.58), (3.59) & (3.60) into the matrix form, as showed below:

$$\begin{bmatrix} 1 & 0 & 1 & 0 \\ cos\beta L_C & \sin\beta L_C & \cosh\beta L_C & \sinh\beta L_C \\ -\cos\beta L_D & -\sin\beta L_D & \cosh\beta L_D & \sinh\beta L_D \\ \beta^3(\sin\beta L_D) & \beta^3(-\cos\beta L_D) & \beta^3(\sinh\beta L_D) & \beta^3(cosh\beta L_D) \end{bmatrix} \begin{bmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ -\frac{F_D}{EI} \end{bmatrix}$$

From the matrix above, the values of constant C_1, C_2, C_3 & C_4 can be found by using MATLAB. The related steps of calculation are attached in **appendix B2** for references.

3.5 Elaboration of the second technique

3.5.1 Develop the equation of motion

The Newtonian Method had been implemented in order to solve the problem. The picture below showed the physical model of continuous system of beam structure.

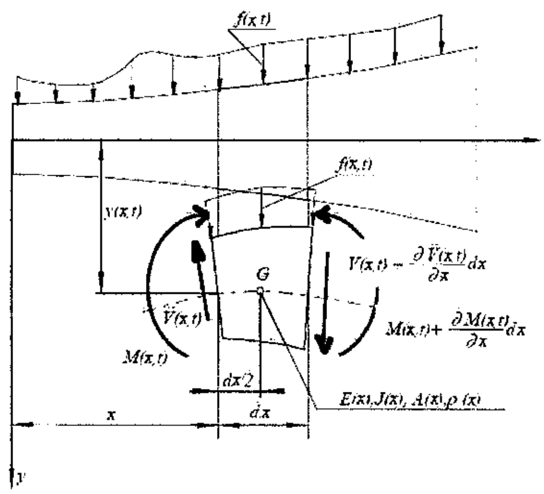


Figure 3.10: Free Body Diagram [Leonard Meirovitch,2001]

There are some considerations/ assumptions applied in order to solve the problem:

4. Beam are assume to be as a elastic elements that are subjected to lateral loads which are the forces or moments that have their vector perpendicular to the centre line of a beam.
5. The beam performs vibrations due to the external distributed unit load , $f(x,t)$
6. Let consider $I(x)$ as beam moment of inertia, cross section $A(x)$, density $\rho(x)$ and the young`s modulus $E(x)$

From the free body diagram above, the force equation of motion in the y direction gives,

$$-(V+dV)+f(x,t)dx+V=\rho A(x)dx\frac{\delta^2 w}{\delta t^2}(x,t) \tag{3.71}$$

The moment equation of motion about z axis passing through point G in figure x above leads to

$$(M + dM) - (V + dV)dx + f(x,t)dx\frac{dx}{2} - M = 0 \tag{3.72}$$

By writing $dV = \frac{\partial V}{\partial x}dx$ and $dM = \frac{\partial M}{\partial x}dx$, and disregarding terms involving second power in dx, both equation (3.71) and (3.72) can be written as

$$-\frac{\partial V}{\partial x}(x,t) + f(x,t) = \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) \tag{3.73}$$

$$\frac{\partial M}{\partial x}(x,t) - V(x,t) = 0 \tag{3.74}$$

Rearrange the equation (3.74) to be $V = \partial M/\partial x$ and then put into equation (3.73), then it becomes

$$-\frac{\partial^2 M}{\partial x}(x,t) + f(x,t) = \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) \tag{3.75}$$

Base on the Euler-Bernoulli theory, the mathematical relation of bending moment and deflection can be considered as

$$M(x.t) = EI(x)\frac{\delta^2 w}{\delta x^2}(x,t) \tag{3.76}$$

We then can modify equation (3.75) by using equation (3.76) and it becomes as

$$\frac{\partial^2}{\partial x^2}\left[EI(x)\frac{\delta^2 w}{\delta x^2}(x,t)\right] + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = f(x,t) \tag{3.77}$$

The equation (3.77) is the equation of motion for force lateral vibration. But the equation only true for non-uniform beam. Equation of motion for uniform beam can be expressed by reducing the equation (3.77) to be:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = f(x,t) \tag{3.78}$$

The equation (3.78) can be express as the general equation of motion for force vibration. The general equation of motion for free vibration of uniform beam then can be expressed as:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = 0 \tag{3.79}$$

The equation (3.78) then been modified by letting the force vibration to be $f(x,t) = qsin\omega_f t$ and then the equation become:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = qsin\omega_f t \tag{3.80}$$

The equation (3.80) express the equation of motion of a uniform beam with subjected to the harmonic force vibration where ω_f is the frequency, in radian per second.

3.5.2 Determine the total response equation

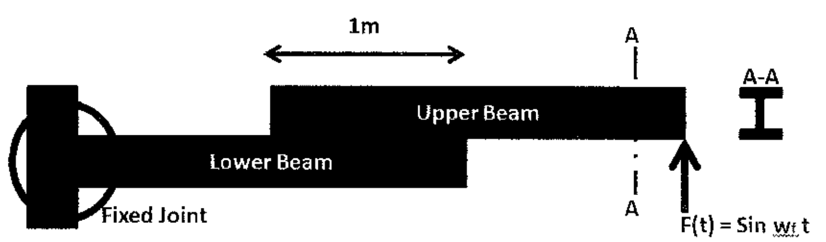


Figure 3.11: Structure of Jetty Gangway with the Harmonic Force at the Free End

Consider now the uniform cantilever beam which is in position as shown in the picture above, and is loaded with concentrated harmonic force F (t) at its free end by this expression:

$$F(t) = Fsin\omega_f t \tag{3.81}$$

There are some assumptions which applied in order to solve the problem. The assumptions are listed below:

1. The jetty gangway is at the full extension position as shown above and the joint between the lower beam and upper beam is a fixed joint
2. The joint connection between both beams is involving all surface related to be adhere to each other without any single void in between.
3. ω_f is the frequency of the harmonic force which is obtain for frequency range of (0Hz – 700 Hz)
4. The time duration for each dynamic analysis is been fixed up to 900 sec or 15 hours in particular day.
5. The amplitude of the harmonic force is a constant value of 1000N
6. Let consider L as the total length of the full extension for jetty gangway.

Let consider that the governing differential equation of motion for the length L of the members between the fixed support and just to the left of the applied harmonic force $F(t) = Fsin\omega_f\, t$ is the same as equation (3.80) when the load $qsin\,\omega_f t$ is made equal to zero which shown below:

$$EI(x)\frac{\delta^4 w}{\delta x^4}(x,t) + \rho A(x)\frac{\delta^2 w}{\delta t^2}(x,t) = 0 \qquad (3.82)$$

The steady-state motion cause by the force given by harmonic force will be determined. We assume the total deflection, w(x,t) of the jetty gangway to be of the form of:

$$w(x,t) = W(x).T(t) \qquad (3.83)$$

Where, W(x) is a function of x only and T (t) is a function of t only.

The force vibration solution can be found by substituting equation (3.83) into equation (3.82) and perform separating variable, we will obtain:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4 W(x)}{dx^4} = \frac{-1}{T(t)}\frac{d^2 T(t)}{dt^2} \qquad (3.84)$$

The right side and the left side of equation (3.84) will be equal to each other only if both sides are equal to a constant. So the constant is assumed to be equal to ω^2, where it is a positive constant. The modified equation (3.84) shown below:

$$\frac{EI}{\rho A(x)W(x)}\frac{d^4W(x)}{dx^4} = \frac{-1}{T(t)}\frac{d^2T(t)}{dt^2} = \omega^2 \tag{3.85}$$

Since ω^2 is a positive constant and cannot be equal to zero, then equation (3.85) can be expressed in two separated equations:

$$\frac{d^4W(x)}{dx^4} - \frac{\rho A\omega^2}{EI}W(x) = 0 \tag{3.86}$$

$$\frac{d^2T(t)}{dt^2} + \omega^2 T(t) = 0 \tag{3.87}$$

We can assume that, the solution for equation (3.87) can be expressed as

$$T(t) = sin\omega_f\, t \tag{3.88}$$

The expression of equation (3.88) is a reasonable assumption since the applied force is a sinusoidal. Equation (3.88) then was substitute into equation (3.87) and yield:

$$\omega^2 = \omega_f^{\ 2} \tag{3.89}$$

Equation (3.86) can be modified by letting $\beta^4 = \frac{\rho A\omega_f^{\ 2}}{EI}$ which lead to

$$\frac{d^4W(x)}{dx^4} - \beta^4 W(x) = 0 \tag{3.90}$$

The solution of equation (3.90) can be assumed to be:

$$W(x) = Ce^{sx} \tag{3.91}$$

Where C and S are constants and derive the auxiliary equation as

$$S^4 - \beta^4 = 0 \tag{3.92}$$

And the roots of the equation are

$S_{1,2} = \pm\beta, \qquad S_{3,4} = \pm i\beta$

Hence the solution for equation (3.90) becomes

$$W(x) = C_1 e^{\beta x} + C_2 e^{-\beta x} + C_3 e^{i\beta x} + C_4 e^{-i\beta x} \tag{3.93}$$

Where, C_1, C_2, C_3,and C_4 are constants . Equation (3.93) above also can be expressed as

$$W(x) = C_1 cos\beta x + C_2 sin\beta x + C_3 cosh\beta x + C_4 sinh\beta x \tag{3.94}$$

The values ofC_1, C_2, C_3, and C_4 can be found from the boundary conditions and the value of β can be found from equation below:

$$\beta^4 = \frac{\rho A \omega_f^{\,2}}{EI} \tag{3.95}$$

3.5.3 Determine the boundary condition

Before we can go to estimate what are the boundary conditions for the jetty gangway, we first need to understand the condition of the beam structure. The structure is not a normal uniform structure and the structure can be divided into 3 main parts which shown in the picture below:

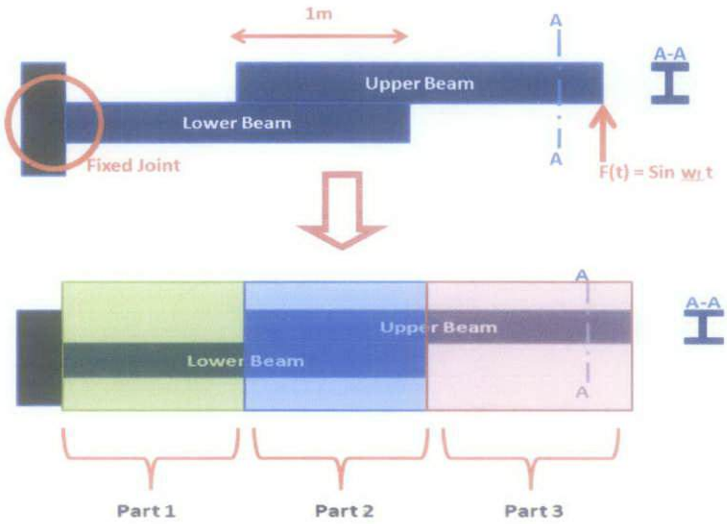


Figure 3.12: Diagram of the Jetty Gangway with 3 differents parts

The details of each part are stated below:

Table 3.2: The Detail of the Cross Sectional and Moment of Intertia for Each Parts

Part 1	Part 2	Part 3
Cross sectional area, A_A= 0.0198 m^2 Moment of inertia, I_A= 0.000108 m^4	Cross sectional area, A_B = 0.0396 m^2 Moment of inertia, I_B = 0.000216 m^4	Cross sectional area A_C = 0.0198 m^2 Moment of inertia, I I_C = 0.000108 m^4

The boundary conditions are already stated below:

At X = 0m (at the fixed joint)

- Displacement , $w(x)_A = 0$ (3.96)
- Slope, $\frac{dw(x)_A}{dx} = 0$ (3.97)

At X = 2m

- Displacement , $w(x)_A = w(x)_B$ (3.98)
- Bending moment, $EI\frac{d^2 w(x)_A}{dx^2} = 0$ (3.99)
- Bending moment, $EI\frac{d^2 w(x)_B}{dx^2} = 0$ (3.100)
- Shear force , $-EI\frac{d^3 w(x)_A}{dx^3} = -EI\frac{d^3 w(x)_B}{dx^3}$ (3.101)

At X = 3m

- Displacement , $w(x)_B = w(x)_C$ (3.102)
- Bending moment, $EI\frac{d^2 w(x)_B}{dx^2} = 0$ (3.103)
- Bending moment, $EI\frac{d^2 w(x)_C}{dx^2} = 0$ (3.104)
- Shear force , $-EI\frac{d^3 w(x)_B}{dx^3} = -EI\frac{d^3 w(x)_C}{dx^3}$ (3.105)

At X = 5m

- Bending moment, $EI\frac{d^2 w(x)_C}{dx^2} = 0$ (3.106)
- Shear force , $-EI\frac{d^3 w(x)_C}{dx^3} = F$ (3.107)

3.5.4 Determine the total response

Figure 3.13: The Profile of Jetty Gangway Structure with 3 Different Parts

Since we have differences profiles of beam in each part, the mode shapes result from the jetty gangway structure may be different and have difference mode shapes and responses with respect to each part. The picture below elaborates the scenario mentioned:

$$W(x)_A = C_{1A}\cos\beta x + C_{2A}\sin\beta x + C_{3A}\cosh\beta x + C_{4A}\sinh\beta x \quad (3.108)$$

$$W(x)_B = C_{1B}\cos\beta x + C_{2B}\sin\beta x + C_{3B}\cosh\beta x + C_{4B}\sinh\beta x \quad (3.109)$$

$$W(x)_C = C_{1C}\cos\beta x + C_{2C}\sin\beta x + C_{3C}\cosh\beta x + C_{4C}\sinh\beta x \quad (3.110)$$

But the $W(x)_A$ just valid for length between 0m to 2m , $W(x)_B$ just valid for length between 2m to 3m and $W(x)_C$ just valid for length between 3m to 5m .

So the total response for jetty gangway will be:

$$w(x,t) = \quad W(x)_A.T(t)_A + W(x)_B.T(t)_B + W(x)_C.T(t)_C \quad (3.111)$$

But$T(t)_A$, $T(t)_B$ and $T(t)_C$ is equal to $\sin\omega_f t$, so $T(t)_A = T(t)_B = T(t)_C = T(t)$ and equation (3.111) can be modify to:

$$w(x,t) = \quad [W(x)_A + W(x)_B + W(x)_C] . T(t) \quad (3.112)$$

So, all constants C_{1A}, C_{2A}, C_{3A}, C_{4A}, C_{1B}, C_{2B}, C_{2B}, C_{2B}, C_{1C}, C_{2C}, C_{3C} & C_{4C} can be found by boundary conditions and value of β.

The matrix 12 x 12 is derived from the equation of boundary conditions and equation (3.108), (3.109) and (3.110) in order to find the value if each constant C. The matrix form used in the calculation is shown below:

$$
\begin{bmatrix}
\cos(\beta_1\cdot 0) & \sin(\beta_1\cdot 0) & \cosh(\beta_1\cdot 0) & \sinh(\beta_1\cdot 0) & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-\sin(\beta_1\cdot 0) & \cos(\beta_1\cdot 0) & \sinh(\beta_1\cdot 0) & \cosh\beta_1\cdot 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-\cos(\beta_1\cdot 2) & -\sin(\beta_1\cdot 2) & \cosh(\beta_1\cdot 2) & \sinh\beta_1\cdot 2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\cos(\beta_1\cdot 2) & \sin(\beta_1\cdot 2) & \cosh(\beta_1\cdot 2) & \sinh(\beta_1\cdot 2) & -\cos(\beta_2\cdot 2) & -\sin(\beta_2\cdot 2) & -\cosh(\beta_2\cdot 2) & -\sinh(\beta_2\cdot 2) & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & -\cos\beta_2\cdot 2 & -\sin\beta_2\cdot 2 & \cosh\beta_2\cdot 2 & \sinh\beta_2\cdot 2 & 0 & 0 & 0 & 0 \\
0.5\beta_1^2(\sin\beta_1\cdot 2) & 0.5\beta_1^2(-\cos\beta_1\cdot 2) & 0.5\beta_1^2(\sinh\beta_1\cdot 2) & 0.5\beta_1^2\cosh\beta_1\cdot 2 & -\beta_2^2(\sin\beta\cdot 2 & \beta_2^2(\cos\beta\cdot 2) & -\beta_2^2(\sinh\beta\cdot 2) & -\beta_2^2(\cosh\beta\cdot 2) & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & \cos(\beta_2\cdot 3) & \sin(\beta_2\cdot 3) & \cosh(\beta_2\cdot 3) & \sinh(\beta_2\cdot 3) & -\cos(\beta_3\cdot 3) & -\sin(\beta_3\cdot 3) & -\cosh(\beta_3\cdot 3) & -\sinh(\beta_3\cdot 3) \\
0 & 0 & 0 & 0 & -\cos(\beta_2\cdot 3) & -\sin\beta_2\cdot 3 & \cosh\beta_2\cdot 3 & \sinh\beta_2\cdot 3 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\cos(\beta_3\cdot 3) & -\sin(\beta_3\cdot 3) & \cosh(\beta_3\cdot 3) & \sinh(\beta_3\cdot 3) \\
0 & 0 & 0 & 0 & 2\beta_2^2(\sin\beta_2\cdot 3) & 2\beta_2^2(-\cos\beta_2\cdot 3) & 2\beta_2^2(\sinh\beta_2\cdot 3) & 2\beta_2^2(\cosh\beta_2\cdot 3) & -\beta_3^2(\sin\beta_3\cdot 3) & \beta_3^2(\cos\beta_3\cdot 3) & -\beta_3^2(\sinh\beta_3\cdot 3) & -\beta_3^2(\cosh\beta_3\cdot 3) \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -\cos(\beta_3\cdot 5) & -\sin(\beta_3\cdot 5) & \cosh(\beta_3\cdot 5) & \sinh(\beta_3\cdot 5) \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \beta_3^2(\sin\beta_3\cdot 5) & \beta_3^2(-\cos\beta_3\cdot 5) & \beta_3^2(\sinh\beta_3\cdot 5) & \beta_3^2(\cosh\beta_3\cdot 5)
\end{bmatrix}
$$

$$
\times \begin{bmatrix} C_{1A} \\ C_{2A} \\ C_{3A} \\ C_{4A} \\ C_{1B} \\ C_{2B} \\ C_{3B} \\ C_{4B} \\ C_{1C} \\ C_{2C} \\ C_{3C} \\ C_{4C} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ -f/EIc \end{bmatrix}
$$

matrix method.

[A] x [B] = [c]

Imv [A] x [C] = [B]

The calculation of matrix for this case is been conducted by using MATLAB software due to the matrix involved 12x12 which are to difficult to be solve manually. Below are the value for each constant C with respect to the value of frequency of the harmonic force.

F(Hz)	10	20	30	40	50	60	100	200	300	400	500	600	700
Ba	0.410258	0.5802	0.710588	0.820516	0.917365	1.004923	1.29735	1.83473	2.247076	2.5947	2.900963	3.177845	3.432465
Bb	0.410443	0.580455	0.710909	0.820887	0.917779	1.005377	1.297936	1.835559	2.248091	2.595872	2.902273	3.179281	3.434016
Bc	0.410258	0.5802	0.710588	0.820516	0.917365	1.004923	1.29735	1.83473	2.247076	2.5947	2.900963	3.177845	3.432465
C_{1A}	-0.0001	-0.0001	-0.0001	0.0001	0	0	0.0035	0	0	0	0	0	0
C_{2A}	0.0002	0.0001	0.0001	-0.0001	0	0	-0.0029	0	0	0	0	0	0
C_{3A}	0.0001	0.0001	0.0001	-0.0001	0	0	-0.0035	0	0	0	0	0	0
C_{4A}	-0.0002	-0.0001	-0.0001	0.0001	0	0	0.0029	0	0	0	0	0	0
C_{1B}	-0.0018	-0.0003	0	-0.0002	-0.0001	0	-0.0043	0	0	0	0	0	0
C_{2B}	0.0017	0.0002	0.0001	-0.0001	0	0	-0.0158	0	0	0	0	0	0
C_{3B}	-0.0021	-0.0004	0.0003	-0.0009	-0.0003	-0.0002	-0.1195	-0.0001	0.0007	0.0002	0.0002	0.0003	0.0016
C_{4B}	0.0031	0.0005	-0.0003	0.0009	0.0004	0.0002	0.1202	0.0001	-0.0007	-0.0002	-0.0002	-0.0003	-0.0016
C_{1C}	0.0064	0.0011	0.0003	0.0001	0	0	-0.0372	0	0	0	0	0	0
C_{2C}	-0.0012	0.0004	0.0003	0.0003	0.0002	0.0001	0.0194	0	0	0	0	0	0
C_{3C}	0.0111	0.0035	0.0017	0.0017	0.0012	0.001	0.8061	0.002	-0.0083	-0.0035	-0.0043	-0.0068	0.0083
C_{4C}	-0.0126	-0.0036	-0.0017	-0.0018	-0.0012	-0.001	-0.8062	-0.002	0.0083	0.0035	0.0043	0.0068	-0.0083

1.0e-003
*

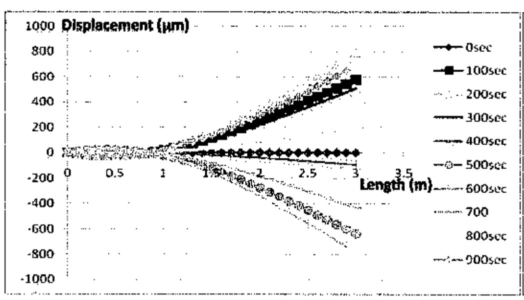
Then the analytical analysis was performed in order to practically find the values and equations of the displacement and mode shape of a gangway under the influences of sea wave. When the calculations were successfully done, all of the data collected was documented and the related graphs were plotted. The results gained were compared with the existing results from the previous related journals and reports. The patterns and trends of the graphs were evaluated with refer to the existing results gain in order to make sure that the results will be validated and reliable for the future used.

CHAPTER 4

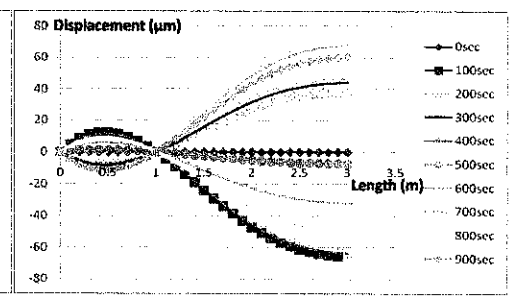
RESULT AND DISCUSSION

4.1 Result for First Technique

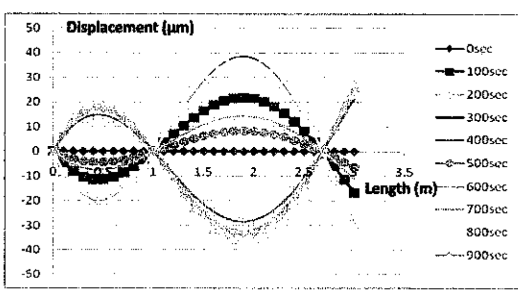
Result for upper beam



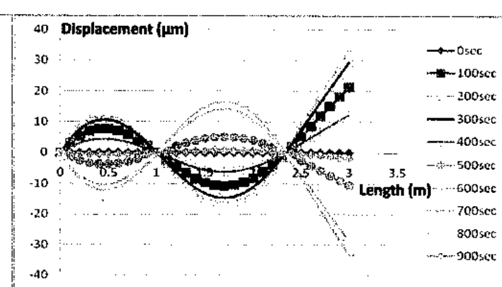
(a)



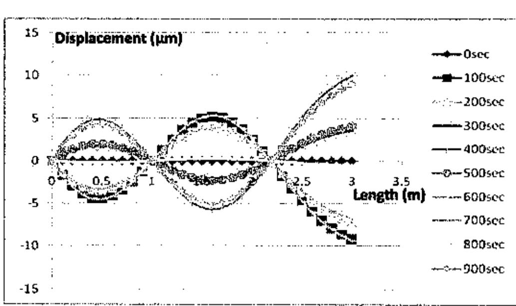
(b)



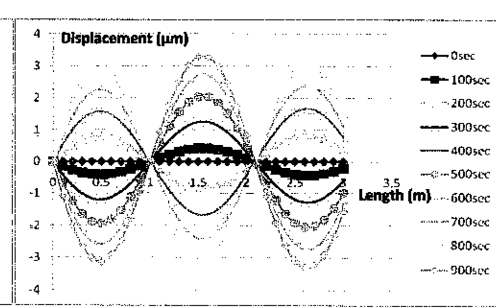
(c)



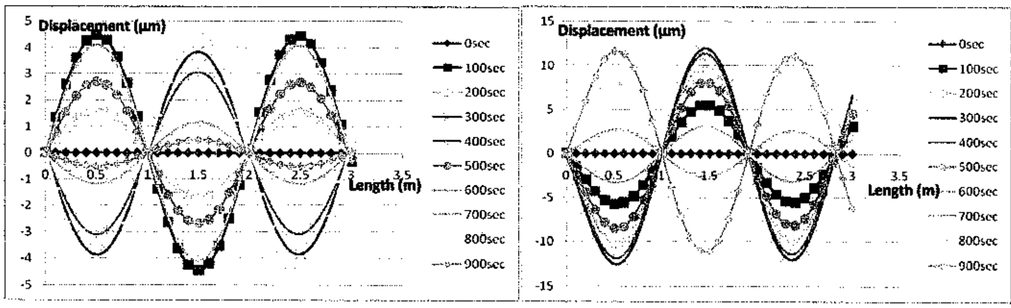
(d)



(e)



(f)



(g) (h)

Figure 4.1: The Responses of the Upper Beam Which Varies From 0 Sec to 900 Sec in Frequency Range of 10Hz to 600Hz.

Figure above is referring to the displacement of the upper beam structure which varies from frequency 10Hz to 600Hz. Figure 4.23(a) refer to 10Hz, Figure 4.23(b) refer to 100Hz, Figure 4.23(c) refer to 200Hz, Figure 4.32(d) refer to 300Hz, Figure 4.23(e) refer to 400Hz, Figure 4.23 (f) refer to 500Hz, Figure 4.23 (g) refer to 600Hz and Figure 4.23(h) refer to 700Hz. The mode shape of the upper beam structure, in general, is following the typical mode shape pattern without any abnormal graphs shown. From the figure above, the responses of the beam will show clear sinusoidal characteristic as the value of wave frequencies exerted at the free end of the beam increased. It can be visibly observed since the number of peak and dip of beam displacement increased as the value of frequencies increased.

The results gained from this calculation step were compared with the graphs taken from the study conducted by Demeter G. Fertis in his book, Mechanical and Structural Vibrations. The figure below shows the mode shape of the beam structure at 100^{th} seconds, with 3 deferent values of wave frequency (10Hz, 200Hz and 400Hz).

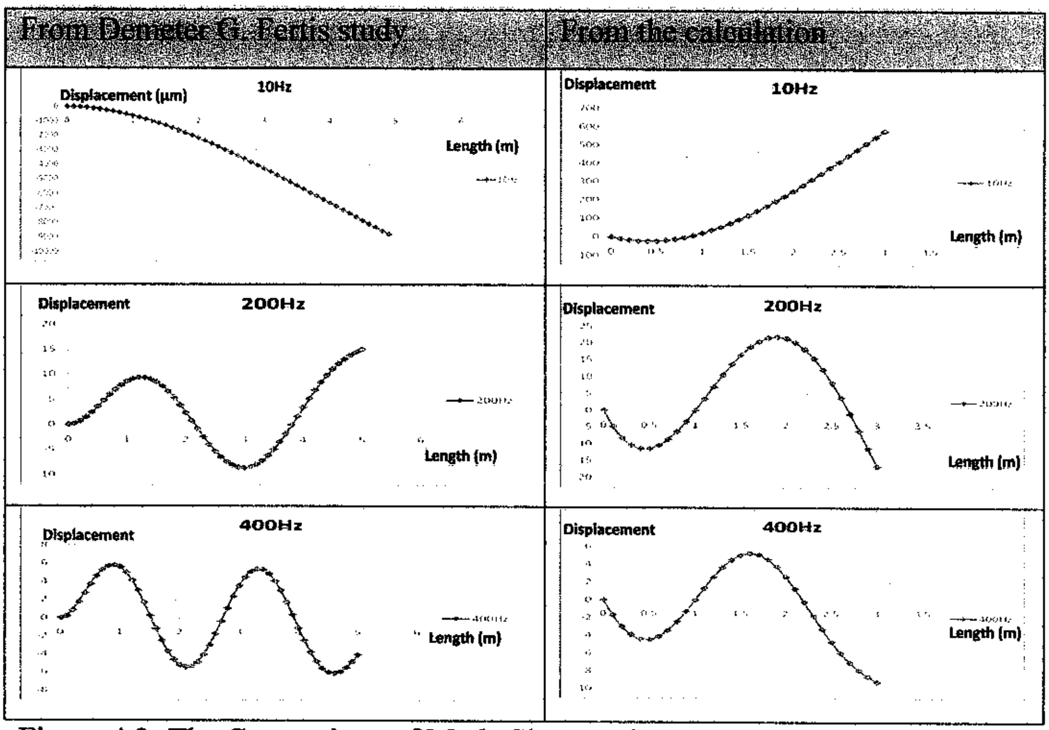
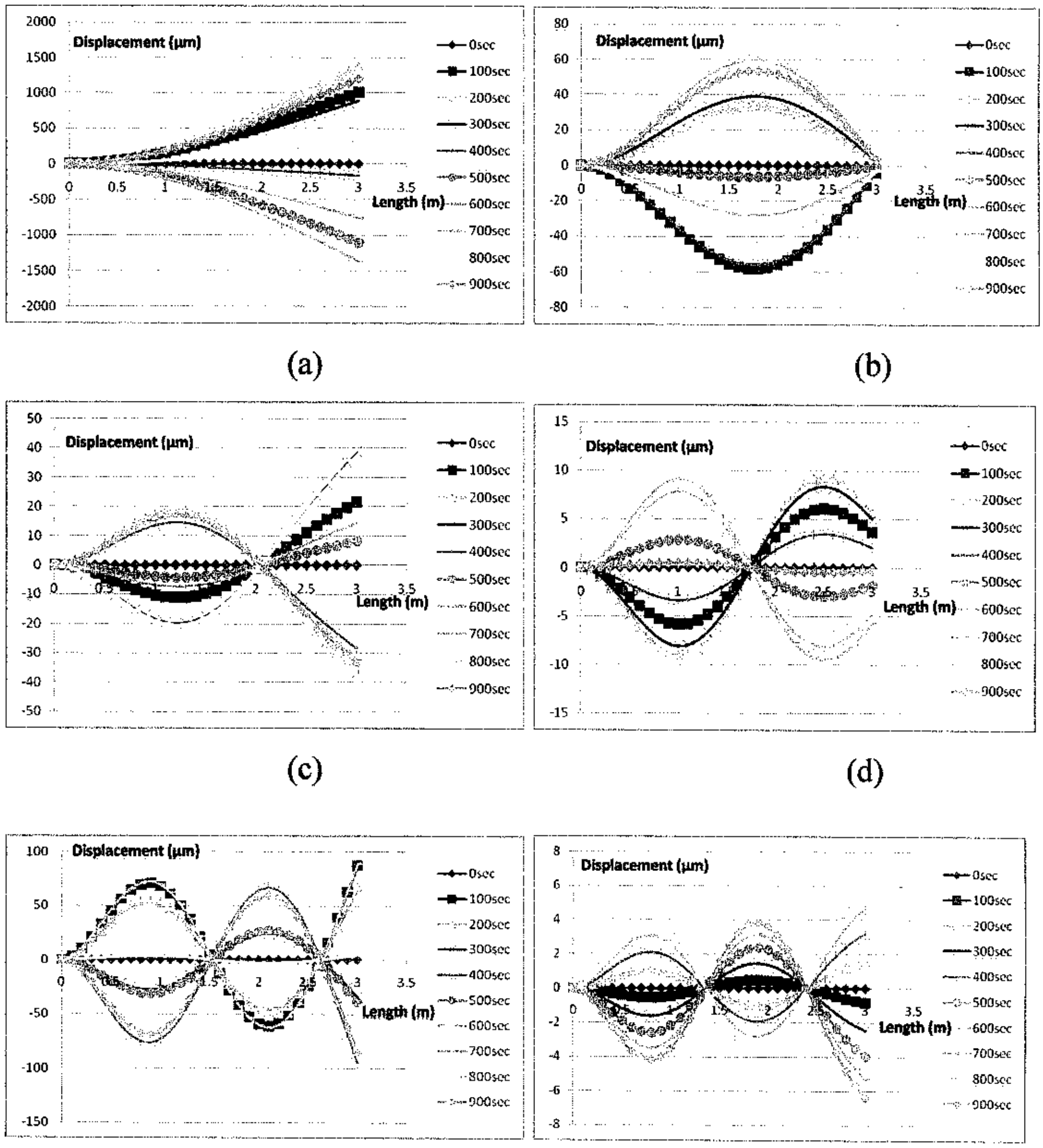


Figure 4.2: The Comparison of Mode Shape Taken From Calculation Step and Reference.

From the graph above, the mode shape of upper beam at 10Hz is considered as 1st mode shape. The mode shape then transformed to be the 2nd mode shape at frequency 200Hz and still remained as 2nd mode shape at frequency of 400Hz. Base on the data from the reference; the mode shape is at 1st mode when the frequency is at 10Hz. The mode shape then transformed to be as 2nd mode at frequency value of 200Hz and at the 3rd mode when the frequency is at 400Hz. It clear that, the mode shape of upper beam from the calculation, transformed to be at higher mode is slower compared to the data from the reference. This is due to the parameter factors as the upper beam has 2 fixed joint at the edge and the middle of it structure. The different profile will leads to the different set of boundary conditions. The boundary conditions then affect the value of constants C. As a result, the response and mode shapes of the upper beam will be influenced and the value of response/displacement will be differed from other type of beam profile.

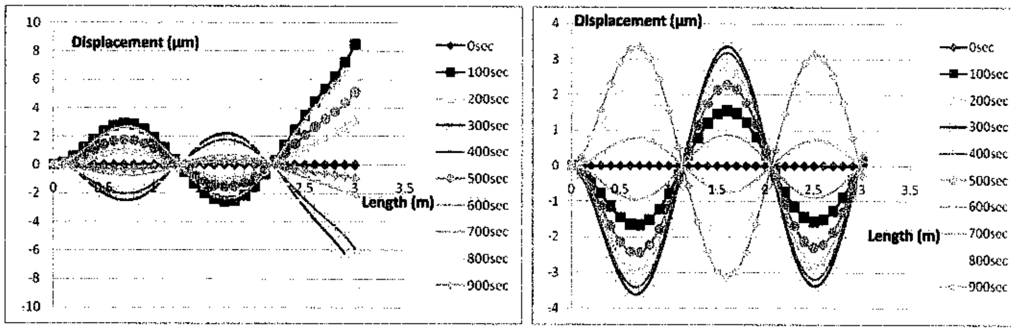
Result for lower beam



(a) (b)

(c) (d)

(e) (f)



(g) (h)

Figure 4.3: The Responses of the Lower Beam Which Varies From 0 Sec to 900 Sec in Frequency Range of 10Hz To 600Hz.

Figure above is referring to the displacement of the lower beam structure which varies from frequency 10Hz to 600Hz. Figure 4.25(a) refer to 10Hz, Figure 4.25 (b) refer to 100Hz, Figure 4.25 (c) refer to 200Hz, Figure 4.25(d) refer to 300Hz, Figure 4.25(e) refer to 400Hz, Figure 4.25 (f) refer to 500Hz, Figure 4.25 (g) refer to 600Hz and Figure 4.25(h) refer to 700Hz. The mode shape of the lower beam structure, in general, is following the typical mode shape pattern without any abnormal graphs shown. From the figure above, the responses of the beam will show clear sinusoidal characteristic as the value of wave frequencies exerted at the free end of the beam increased. It can be visibly observed since the number of peak and dip of beam displacement increased as the value of frequencies increased.

The results gained from this calculation step were compared with the graphs from the reference. The figure below shows the mode shape of the beam structure at 100th seconds, with 3 deferent values of wave frequency (10Hz, 200Hz and 400Hz).

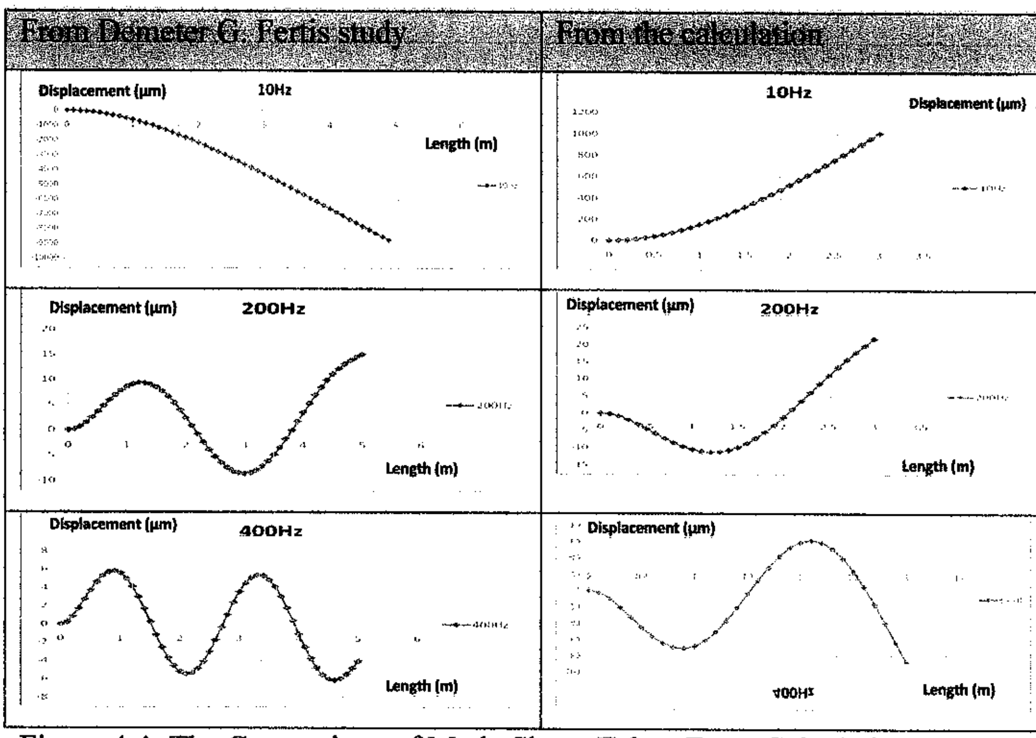
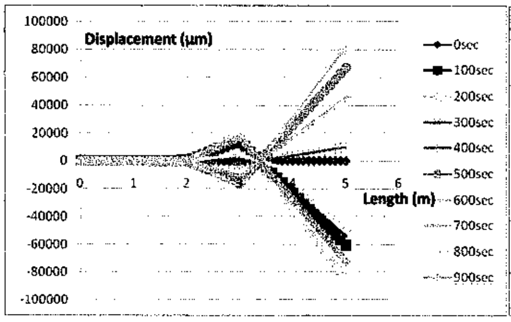


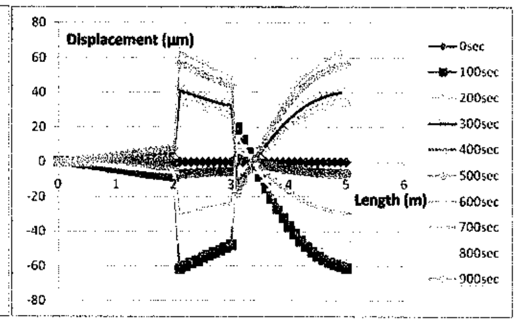
Figure 4.4: The Comparison of Mode Shape Taken From Calculation Step and Reference.

From the graph above, the mode shape of lower beam at 10Hz is considered as 1st mode shape. The mode shape then transformed to be the 2nd mode shape at frequency 200Hz and 3rd mode at frequency of 400Hz. The mode shapes of the lower beam are similar to the mode shape of the reference. This is due to the similar condition of beam, which both beam taken from the journal and the calculation method is treated as a cantilever beam. The only different which shown from the above figure is the length of the beam. The lower beam is 3 meter maximum and the parameter of beam from the reference is 5 meter maximum. The amplitude of the force which is imposed to the free end of the lower beam also lower if compared to the amplitude force of the beam from the references. This is due to the transfer function method which is applied in the calculation in order to relocate the source of harmonic force from the free end of upper beam to the free end of the lower beam. The calculation of transfer function method can be refer from **Appendix A**.

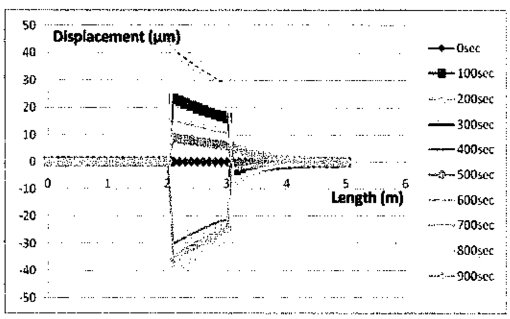
4.2 Result for Second Technique



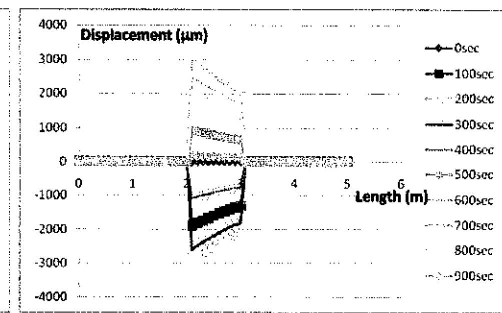
(a)



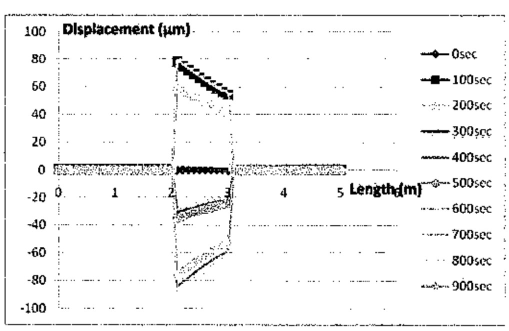
(b)



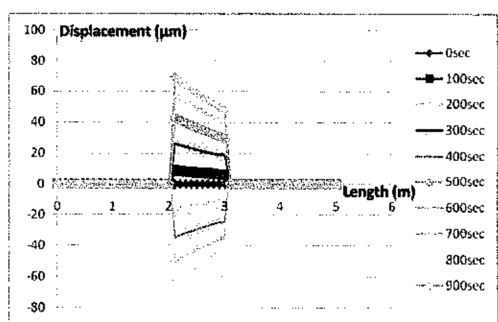
(c)



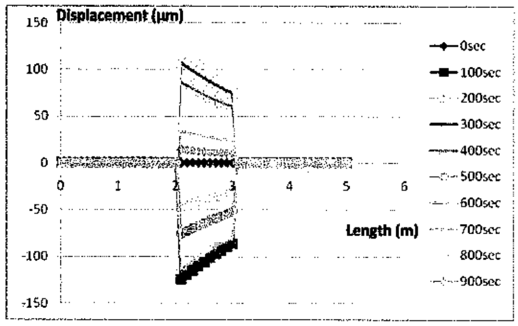
(d)



(e)



(f)

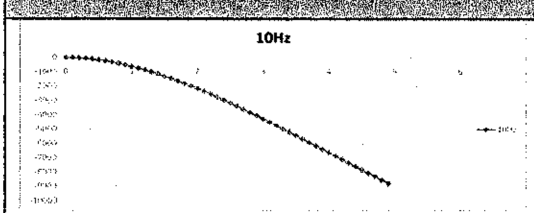
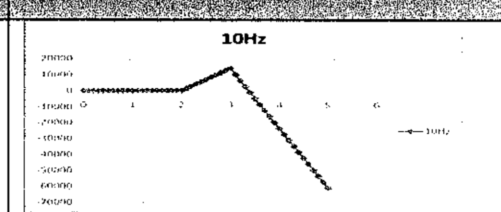


(g)

Figure 4.5: The Responses of the Beam Structure Which Varies From 0 Sec to 900 Sec in Frequency Range Of 10Hz To 600Hz.

Figure above is referring to the displacement of the beam structure which varies from frequency 10Hz to 600Hz. Figure 4.27(a) refer to 10Hz, Figure 4.27 (b) refer to 100Hz, Figure 4.27 (c) refer to 200Hz, Figure 4.27(d) refer to 300Hz, Figure 4.27(e) refer to 400Hz, Figure 4.27 (f) refer to 500Hz and Figure 4.27 (g) refer to 600Hz.

The mode shape of the beam structure, in general, is showing an abnormal pattern of the mode shape. The mode shape gained did not show any characteristic of sinusoidal function. The comparison between the mode shape of the calculation and mode shape taken from the previous study are illustrated below:

From Demeter G. Fertis study	From the calculation
	

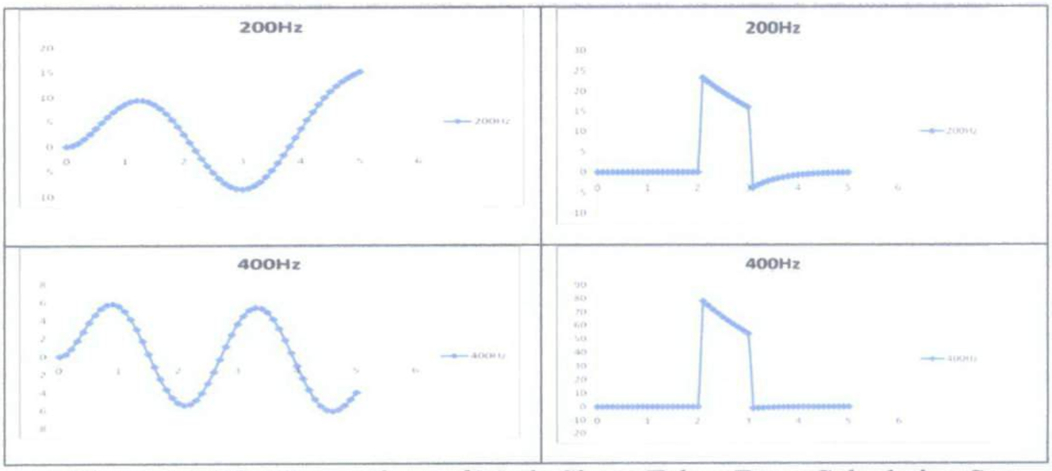


Figure 4.6: The Comparison of Mode Shape Taken From Calculation Step and Reference

The figure above shows the mode shape of the beam structure at 100^{th} seconds, with 3 deferent values of wave frequencies (10Hz, 200Hz and 400Hz). From the figure, the pattern of the mode shape at 10Hz is not similar to the mode shape of the beam at 200Hz and the mode shape of the beam at 200Hz is similar to the mode shape at 400Hz. In general, the patterns of the mode shape of the beam structure are changed from 10 Hz to 200Hz and the pattern is not changed from 200Hz to 400Hz.

From the above figure, we can observe clearly that, the pattern of the mode shape gained from the calculation steps is not similar to the pattern of the mode shape of the reference. This is due to several factors which influenced the character of the mode shape. The profile o the beam is not uniform as show from the figure below.

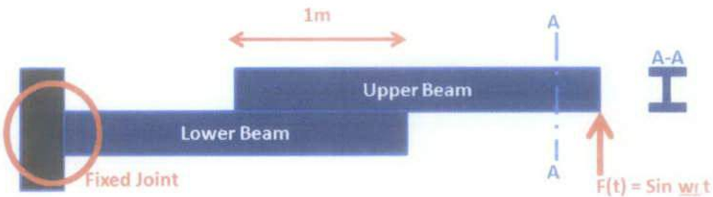


Figure 4.7: Diagram of Jetty Gangway Structure

The number of boundary conditions for this model is 12 boundary conditions, which 2 boundary condition at the fixed joint, 8 boundary conditions at the middle of the beam structure and another two boundary conditions at the free end of the beam structure. There are 12 unknown of constant C value. This leads to the derivation of 12 equations. The equations cannot be simplified due to the large quantity of equations, and the only way to solve the equations is by using a matrix form. The matrix 12 x 12 then been constructed an MATLAB software used for finds the value of each constant C. The values fond in this equation may be round of to certain decimal places and alter the real value of each constant. This may contributed in abnormality of the responses graph of the structure.

The boundary conditions chosen in the calculation may not be fit to the profile of the beam structure. The wrong boundary conditions may lead the abnormality to the beam response and mode shape pattern.

The graph of the displacement at the free end of the beam structure varies with the frequency value also been plotted and shown at the figure below:

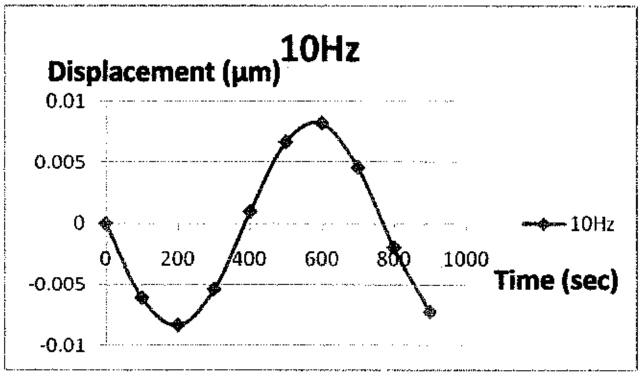


Figure 4.8: The Displacement of the Free End of the Beam with Frequency Value of 10Hz

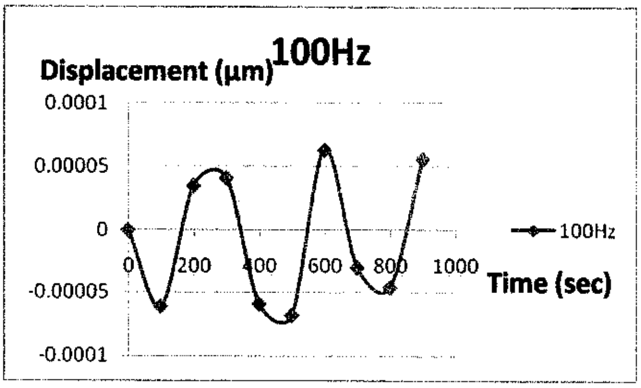


Figure 4.9 : the displacement of the free end of the beam with frequency value of 100Hz

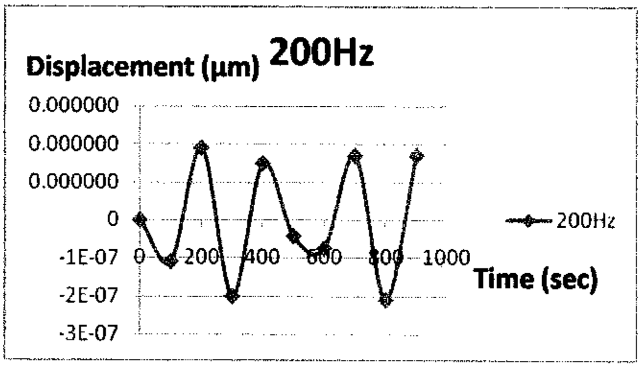


Figure 4.10: the displacement of the free end of the beam with frequency value of 200Hz

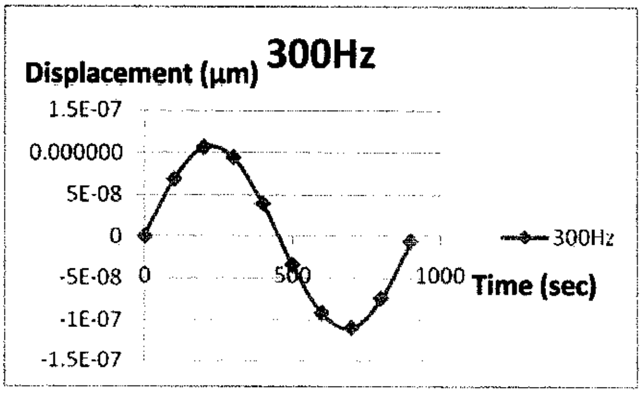


Figure 4.11: the displacement of the free end of the beam with frequency value of 300Hz

The displacement value of at the free end of the beam also compared to the value taken from the study conducted by Demeter G. Fertis. The comparison of both values illustrated in the figure below. Four sample of the displacement values taken from both reference and calculation in order to make a comparison. The displacement values are taken at the frequency value of 10Hz, 100Hz, 200Hz and 500Hz.

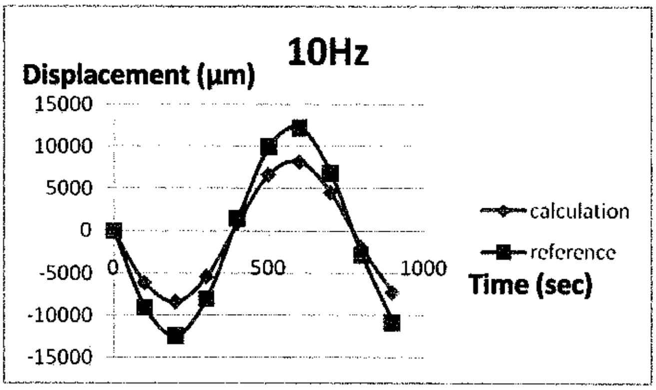


Figure 4.15: The Displacement Value at the Free End Taken From Calculation & Reference

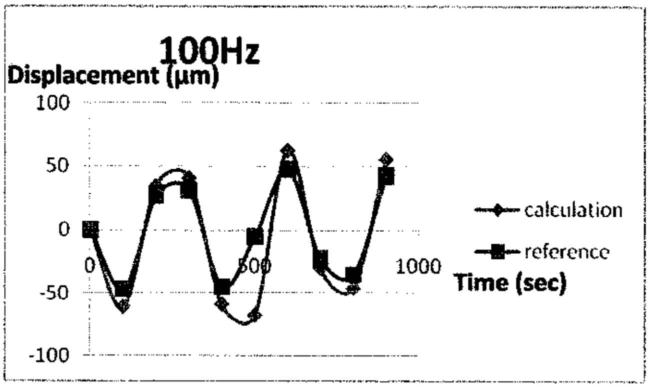


Figure 4.16: The Displacement Value at the Free End Taken From Calculation & Reference

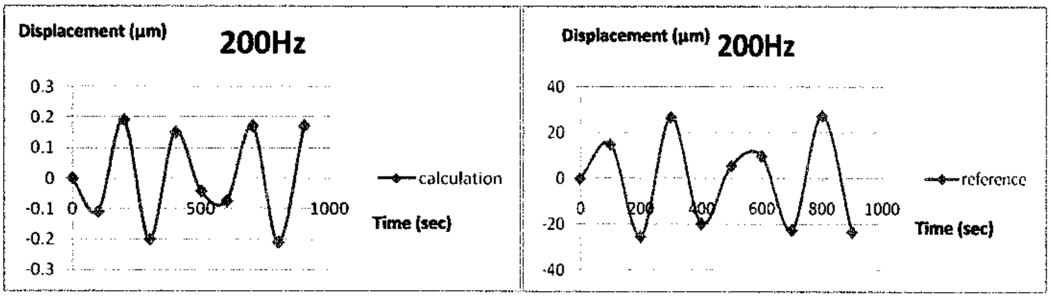


Figure 4.17: The Displacement Value at the Free End Taken From Calculation & Reference

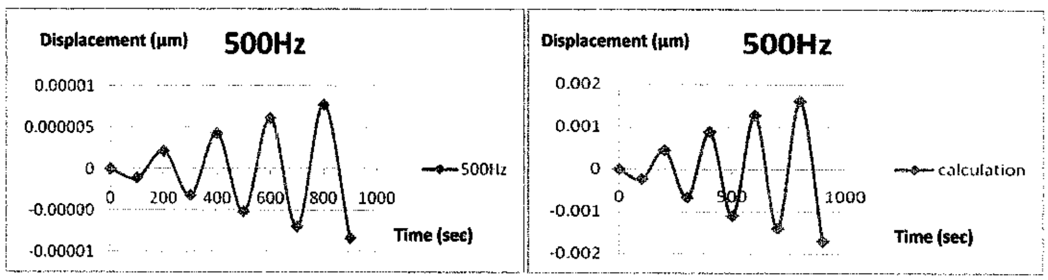


Figure 4.18: The Displacement Value at the Free End Taken From Calculation & Reference

Base on the data given from figure above, it is clear that the pattern of the displacements at the free end of the beam structure are given a same patter and the value of the displacement all most similar at the value of frequency 10Hz and 100Hz. The value then shrinks but at the same time, still maintained the pattern when the frequency values are increased. It mean than the displacement or the displacement motion of the beam at the free end for the jetty gangway structure is less if compared to the displacement values taken from the uniform beam structure of the reference.

In addition, the data of the motion of the beam at the free end of the jetty gangway when the jetty gangway is treated as a single structure is valid. This is due the boundary conditions were chose correctly and suitable to be used. The uncertainty of displacement at the middle of the beam structure can be considered derived due to the complexity of the beam profile and the selection of the boundary may be not suitable for the calculation.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

As a conclusion, this project was a comprehensive research study about harmonic response on a telescopic jetty gangway structure. The project was related to the study on the dynamic characteristics of the jetty gangway structure which exerted the harmonic force at the free end of the beam structure. From the studies, we found the values of the harmonic responses on the telescopic jetty gangway and the mode shapes under the influence of harmonic force of sea wave. The mathematical models were developed based on simplified I beam geometry by using Newtonian method. The mathematical solutions of the jetty gangway were conducted in two different techniques. In the first technique, the upper and the lower beam were treated as separated parts. The separated calculation processes were conducted in both upper and load beam structure. For the second technique, the jetty gangway structure was treated as a single structure with different value of cross sectional areas at the beginning, middle and far end of the jetty gangway.

The values of the mode shapes and responses of the jetty gangway were already calculated. The objective of the project was successfully achieved. The values and pattern of the mode shapes gained from both techniques were compared with the value from the reference for verification. The pattern of mode shapes from the first techniques shows the similarity when compared to the data of the reference. The pattern of the mode shape gained from the 2nd technique shows the abnormality and did not show the sinusoidal characteristic. However the values and patterns of the relation between the displacement at the free end and the value of wave frequencies are showing the similarity when compared with the values taken from the study conducted by Demeter G. Fertis.

The values of mode shapes and responses of the second technique can be defined as valid and reliable values. The abnormal patterns of mode shapes of the beam were caused of the complexity at the middle of the beam profile.

5.2 Recommendation

The study can be improved by comparing the result gained from other mathematical approaches, which are stated below:

- Extended Hamilton Principle
- Finite element method

The extended Hamilton principle has the advantages in dealing with the nonlinear and non uniform structure. As the beam shows the complex beam profile, this principle can be applied by treating the beam as non-linear structure.

The result also can be improved by using the finite element method. The principal advantage of the finite element method is its generality; it can be used to calculate the natural frequencies and mode shapes of any linear elastic system. The finite element method may require more computational effort in order to get the values better than other approximate solutions.

The studies also need to be extended by operating the simulation by using software. The related software like ANSYS and MATLAB can be applied in order to find the mode shapes and responses for telescopic jetty gangway. The simulation data are highly needed since the data and the simulation model can be implemented for other beam profile and other wave parameters. Variation of data can be collected and yet can be implemented in any location of the sea for any beam profiles.

CHAPTER 6: REFERENCES

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APPENDIX A: TRANSFORM FUNCTION METHOD

VALUE OF FORCE AND REACTION FORCE

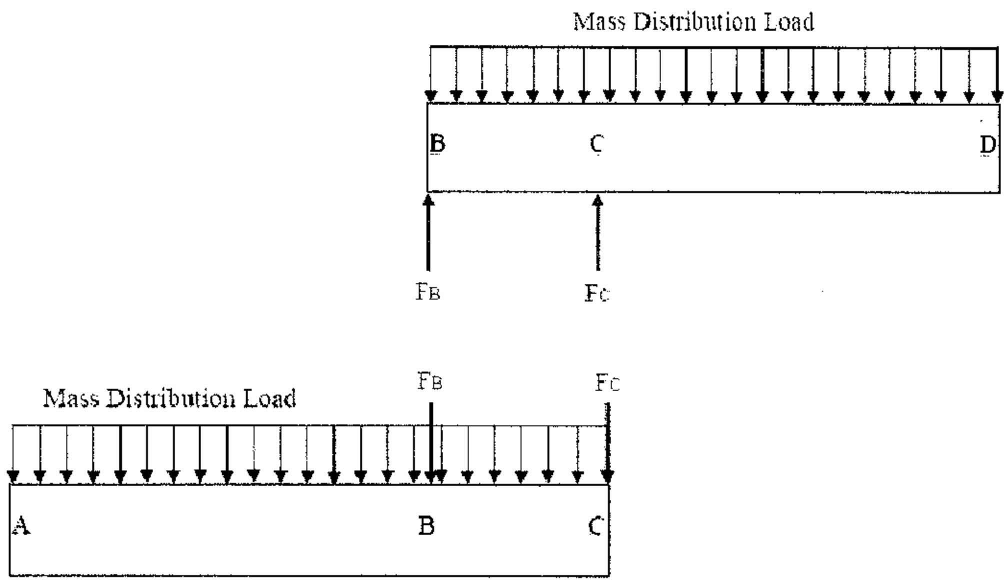


Figure A.1: The schematic diagram of upper beam

Detail of Beam

E, Modulus of elasticity: 69 *Gpa*

I, Moment of Inertia: 0.000108 m^4

m, Mass of beam: 159.259 *kg*

L, Total length: 3.0 *m*

a, Length between point B and point C: 1*m*

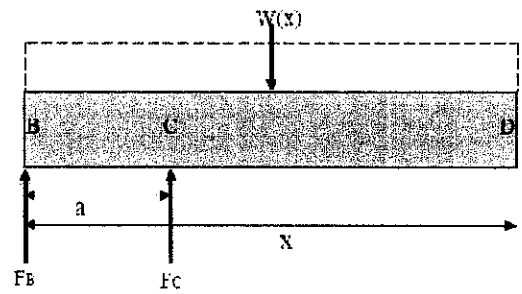
Distribution of mass loading

$W = \frac{m*g}{L}$, take g = 10 m/s^2

= 530.86 N/m

Upper beam

Free body Diagram



Let L = x,

Let $+\uparrow \sum F = w * x + F_{B+} F_c = 0$,

$\therefore F_B + F_c = w * x$ .... (4.1)

Let + $\sum M_D = 0$,

$\therefore wx\left(\frac{x}{2}\right) - F_C(x-a) - F_B x = 0$,

$\therefore F_C(x-a) + F_B x = wx\left(\frac{x}{2}\right)$ .... (4.2)

From equation (1),

Let $F_B = wx - F_c$, ... (4.3),

Then substitute (4.3) into (4.2)

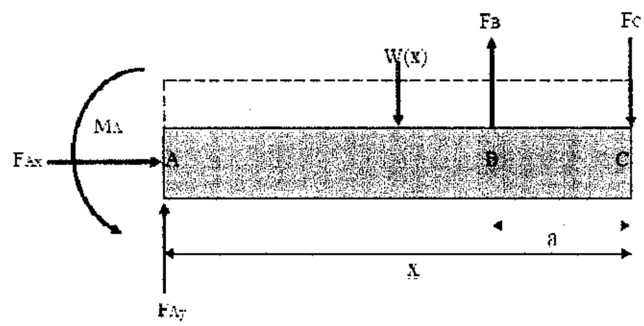
$\therefore F_C = \mathbf{2388.9N}$ .... (4.4),

Then substitute (4.4) into (4.3),

$\therefore\ F_B = \mathbf{-796.31}\boldsymbol{N}$

Lower beam

Free body diagram



Let $+\uparrow\sum F\ = 0,$

$$+\uparrow\sum F = F_{Ay} - wx + \ 796.31N(x-a) - 2388.9N\ (x)\ , \text{.................................}\ (4.5)$$

$\therefore\ F_{Ay} = \mathbf{-3185.17}\boldsymbol{N}$

Let $+\circlearrowleft\ \sum M_C = 0,$

$$\therefore +\circlearrowleft\ \sum M_C = -F_B(a) + wx\left(\frac{x}{2}\right) - F_{Ay}(x) + \ M_A, \text{.................................}\ (4.6)$$

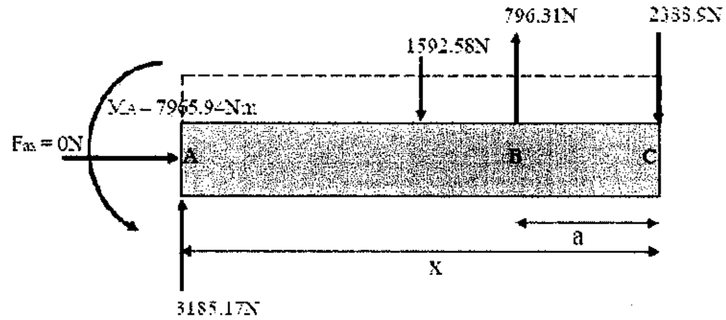
$\therefore +\circlearrowleft\ M_A = \mathbf{7965.94}\boldsymbol{Nm}$

Let $+\rightarrow\sum F\ = 0, \quad \therefore\ F_{Ax} = \mathbf{0}\boldsymbol{N}$

CALCULATION OF THE DEFLECTION OF BEAM

Lower beam

Free body diagram



By using the superposition method,

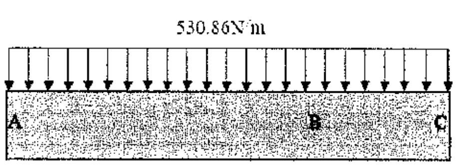
$\sum\delta_{total} = \delta_1 + \delta_2 + \delta_3$, ... (4.7)

Where

- δ1 due to distributed load
- δ2 due to load at point B
- δ3 due to load at point C

The forces and moment at point A will not be included in the calculation since Point A is the fixed joint (no deflection).

Value for δ1



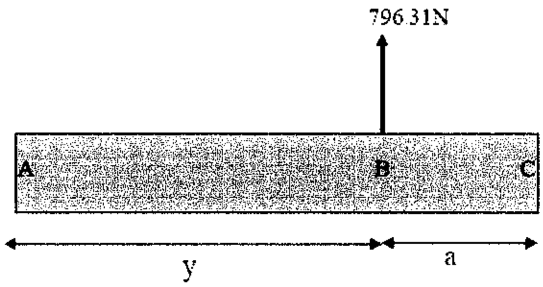
Let w = 530.86N/m,

$$\delta_{Max,Point\ C}(x = L) = \frac{-W(L^4)}{8EI} \text{.........................} (4.8)$$

$$\therefore \delta_{Max,Point\ C} = \frac{(-5374.9575)}{EI}\ m$$

$$\delta_{Point\ B}(x = 2m) = \frac{-W(x^2)}{24EI}(x^2 - 4Lx + 6L^2) \text{.........................} (4.9)$$

$$\therefore \delta_{Point\ B} = \frac{(-3008.20667)}{EI}\ m$$



Value for δ2

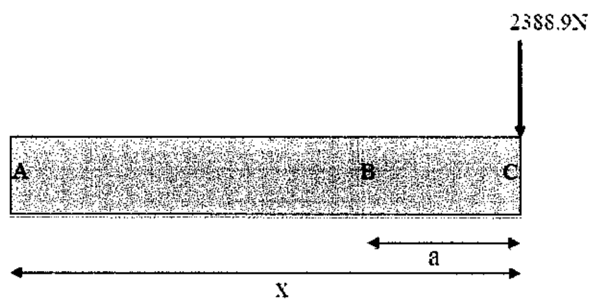
Let w = 796.31N,

$$\delta_{Point\ C} = \frac{W(y^2)}{6EI}(2y + 3b) \dots\dots\dots\ \dots\dots\ (4.10)$$

$$\therefore \delta_{Point\ C} = \frac{3716.06667}{EI}\ m$$

$$\delta_{Point\ B} = \frac{W(y^3)}{3EI} \dots\dots\dots\dots\dots\dots\dots\ (4.11)$$

$$\therefore \delta_{Point\ B} = \frac{2123.4667}{EI}\ m$$



Value for δ3

Let w = 2388.9 N,

$$\delta_{Point\ C}(x = L) = -\frac{W(L^3)}{3EI} \dots\dots\dots\ (4.12)$$

$$\therefore \delta_{Point\ C} = \frac{-21500.1}{EI}\ m$$

$$\delta_{Point\ B}(x = 2) = \frac{-W(x^2)}{6EI}(3L - x) \dots\dots\dots\ (4.13)$$

$$\therefore \delta_{Point\ B} = \frac{-11148.2}{EI}\ m$$

Total deflection value at lower beam

$$\sum\delta_{total\ at\ point\ B} = \delta_{1b} + \delta_{2b} + \delta_{3b}\ , \ldots\ldots\ldots\ldots (4.14)$$

$$= \frac{(-3008.20667)}{EI} m + \frac{2123.4667}{EI} m + \frac{-11148.2}{EI} m, \ldots\ldots\ldots\ldots (4.15)$$

$$= \frac{(-12032.959)}{EI} m$$

$$\sum\delta_{total\ at\ point\ C} = \delta_{1c} + \delta_{2c} + \delta_{3c} \ldots\ldots\ldots\ldots (4.16)$$

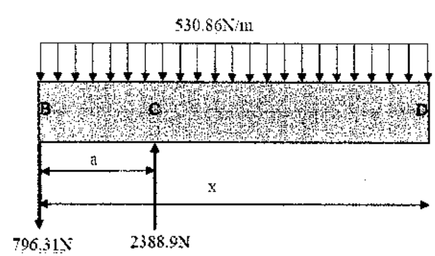
$$= \frac{(-5374.9575)}{EI} m + \frac{3716.066677}{EI} m + \frac{-21500.1}{EI} m$$

$$= \frac{(-23159.025)}{EI} m$$

Upper beam

The only load which influence the deflection is the distributed load. The forces at points B & C will not been considered since A & C are the simply supported joint.

Deflection only occurred at point D and by using the Super position method:

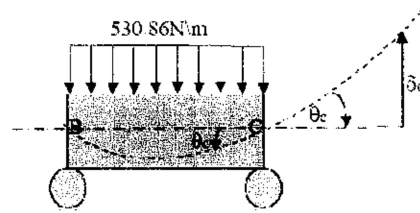


$$\sum\delta_{total\ at\ point\ D} = \delta_{1d} + \delta_{2d} + \delta_{3d}$$

Where

- δ1d due to distributed load at point B-C
- δ2d due to moment at just above point C
- δ3d due to distributed load at point C-D

Value for δ1



Slope θc = $\frac{-wL^4}{24EI}$....... (4.17), where L = length B to C= 1m

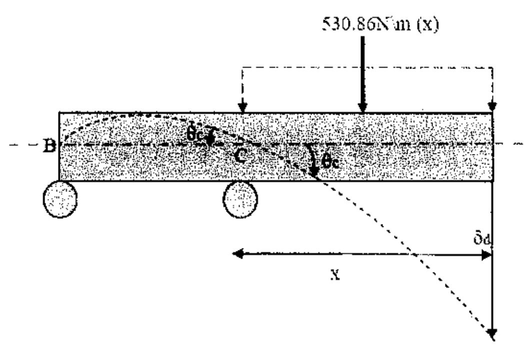
$$\therefore\ \theta_C = \frac{22.119}{EI}$$

$$\theta_C \approx \tan\theta_C \quad ....(4.18)$$

$$\therefore\ \delta_D = x\tan\theta_C$$, where x = length C to D = 2m

$$\therefore\ \delta_D = \frac{44.238}{EI}$$

Value for δ2



Let Moment at above point C = M>c,

$$M_{>C} = 530.86x\left(\frac{x}{2}\right)\text{ Nm}$$

Slope θc = $\frac{ML}{3EI}$, let L = length between B & C= 1m,

$$\therefore\ \theta_C = \frac{88.3334x^2}{EI}$$

$$\theta_C \approx \tan\theta_C \quad ....(4.19)$$

$$\therefore\ \delta_D = x\tan\theta_C$$, where x = length C to D = 2m

$$\therefore\ \delta_D = \frac{-707.8133}{EI}$$

Value for δ3

$$\delta_{Point\ D}(x = 2) = -\frac{W(X^4)}{8EI} \ ....(4.20) \quad \therefore\ \delta_D = \frac{-1061.72}{EI}$$

Total deflection value at upper beam

$$\sum\delta_{total\ at\ point\ D} = \delta_{1d} + \delta_{2d} + \delta_{3d}\ , \ldots\ldots\ldots\ldots (4.21)$$

$$= \frac{44.238}{EI}\ m + \frac{-707.8133}{EI}\ m\ + \frac{-1061.72}{EI}\ m, \ldots\ldots\ldots\ldots (4.22$$

$$= \frac{-1725.2953}{EI}\ m$$

After we already got all of the deflection values in both beam, then we will combine them for determine the overall deflection of the system as elaborated by figure below:

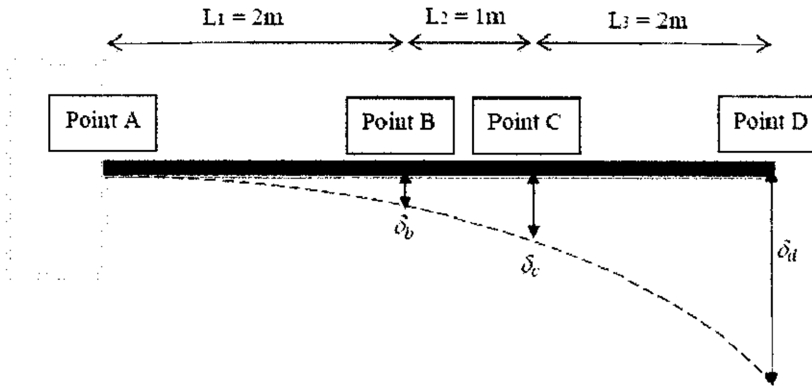


Figure A.2: The deflection value of total beam system

The total deflection for beam system

$$\delta_B = \frac{-12032.95889}{EI} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.23)$$

$$\delta_C = \frac{-23159.0245}{EI} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.24)$$

$$\delta_D = \frac{-1725.3058}{EI} + \delta_C = \frac{-24884.3302}{EI} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.25)$$

Apply the value of *E* and *I*, then

Table A.1: The deflection value of the beam

Deflection	Value in meter
δB	0.0016147m
δC	0.0031080m
δD	0.0033400m

APPENDIX B1: SOLUTION OF MATRIX FORM BY MATLAB

The step is representing the solution by MATLAB for lower beam at frequency value of 100Hz.

```
>> Fc = 930.667

Fc =

 930.6670

>> E = 6.9e10

E =

 6.9000e+010

>> L = 3

L =

   3

>> B = 1.29735

B =

  1.2974

>> A = [1 0 1 0; 0 1 0 1; -cos(B*L) -sin(B*L) cosh(B*L) sinh(B*L); B^3*(sin(B*L)) B^3*(-cos(B*L)) B^3*(sinh(B*L)) B^3*(cosh(B*L))]

A =

  1.0000        0          1.0000        0

     0        1.0000        0          1.0000

  0.7314      0.6820      24.5158      24.4954

 -1.4892      1.5970      53.4880      53.5326

>> b = [0; 0; 0; -Fc/(E*0.000108)]
```

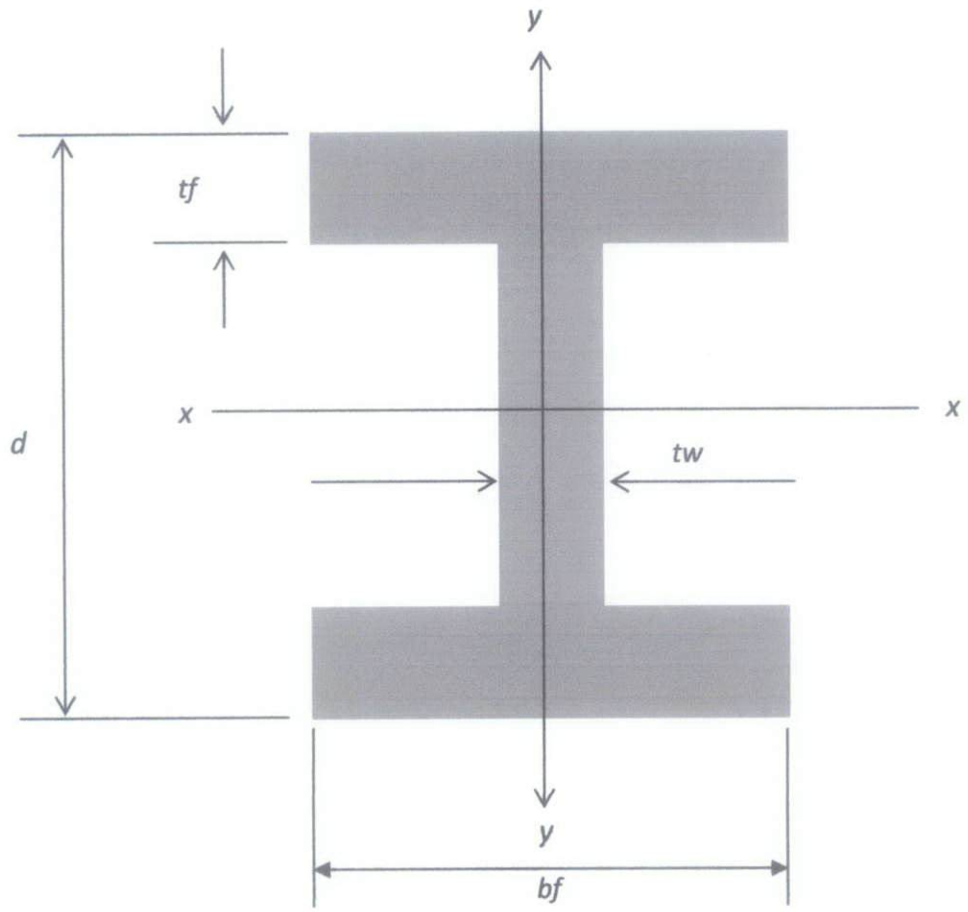
APPENDIX B2: SOLUTION OF MATRIX FORM BY MATLAB

The step is representing the solution by MATLAB for upper beam at frequency value of 100Hz.

```
======================
100Fz
=====================================
>>  Fd = 1000

Fd =

      1000

>> Lc = 1

Lc =

    1

>> Ld = 3

Ld =

    3

>> E = 69000000000

E =

  6.9000e+010

>> I = 0.000108

I =

  1.0800e-004

>> B = 1.29735

B =

   1.2974
```

```
>> A = [1 0 1 0; (cos(B*Lc)) (sin(B*Lc)) (cosh(B*Lc)) (sinh(B*Lc)); -(cos(B*Ld)) -
(sin(B*Ld)) (cosh(B*Ld)) (sinh(B*Ld)); B^3*(sin(B*Ld)) B^3*(-cos(B*Ld))
B^3*(sinh(B*Ld)) B^3*(cosh(B*Ld))]

A =

    1.0000         0    1.0000         0
    0.2701    0.9628    1.9664    1.6932
    0.7314    0.6820   24.5158   24.4954
   -1.4892    1.5970   53.4880   53.5326

>> b = [0; 0; 0; -Fd/(E*I)]

b =

  1.0e-003 *

         0
         0
         0
   -0.1342

>> c = inv(A)*b

c =

  1.0e-004 *

    0.4485
    0.0256
   -0.4485
    0.4347
```

APPENDIX C: DETAIL INFORMATION OF BEAM



Detail	Value
Designation (mm x kg/m)	W610 x 155
Area, A (mm^2)	19800
Depth , d, (mm)	611
Web thickness , tw (mm)	12.70
Flange width, bf (mm)	324
Flange thickness , tf (mm)	19.0
I , y-y, (10^6 mm^4)	108

[Information taken from, Mechanics of Material, R.C. Hibbeler2005, page 813]