VIBRATION ANALYSIS OF STRUCTURES

A THESIS SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF

Bachelor of Technology
in
Civil Engineering Department

By

C.KASHYAP

Department of Civil Engineering
National Institute of Technology
Rourkela
2007

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Under the Guidance of

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Department of Civil Engineering
National Institute of Technology
Rourkela
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National Institute of Technology

Rourkela

CERTIFICATE

This is to certify that the thesis entitled, "VIBRATION ANALYSIS OF STRUCTURES" submitted by Shri C.Kashyap in partial fulfillments for the requirements for the award of Bachelor of Technology Degree in Civil Engineering at National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University / Institute for the award of any Degree or Diploma.

Date: 03-05-2007

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C.Kashyap Date: 03rd May, 2007
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CONTENTS

ABSTRACT

The present focuses on dynamic nature of various structures present in an environment where they are bound to undergo vibrations. In such vibrating conditions when they are subjected to a resonance they experience high amplitudes, leading to the failure of the structure. Hence, the study of operating frequencies of

1) Machine Foundations
    i) Los Angels Abrasion Machine
    ii) Jaw Crushing Machine
2) Fiber Reinforced Glass Composites – varying the number of layers
    i) 16 layers
    ii) 12 layers
3) Steel Flats

In this study we have used a non computational technique for analysis of dynamic nature of structures. *Brüel&Kjær PULSE*™, Multi-analyzer System Type 3560 was used in the analysis.

The operating frequency ranges in case of Los Angels Abrasion Machine is found to be 48 Hz – first frequency and 73 Hz – second frequency. In case of Jaw Crushing Machine is 42.2 Hz – first frequency and 71.8 Hz – second frequency. Where as, in case of steel flat the operating frequency is found to be 41.50 Hz. The fiber reinforced glass composites were decreased in area in a regular pattern and the pattern of frequency variation was observed. In case of 16 layers the first frequency decreased from 284 Hz – 236 Hz and the second frequency also depicted similar pattern. In case of 12 layers the first frequency decreased from 190 Hz – 160 Hz and the second frequency varied from 588 Hz – 390 Hz. The observed trend is justified as the value of K decreases as we decrease the area of the sample.

We have also studied the determination of Buckling load from frequency study incase of a steel flat. When steel flat is subjected to increasing axial load the operating frequency is observed to decrease. When this operating frequency tends to zero the axial load nears

the buckling load of that structure. 30cm steel flat is tested in a UTM under increasing axial load. The initial frequency under no load condition is 260 Hz.   Under a load of 0.4 ton the first frequency decreases to 168 Hz. Extrapolating the decreasing trend we get the buckling load as 1.1739 ton. A similar trend was observed in case of second frequencies.

The vibration analysis of the foundations of various machines will help us in designing them such that their serviceability is increased. Similarly, fiber reinforced composites are being used in various structural members. These demands require a deeper understanding of fiber composite behavior. Composites offer great promise as light weight and strong structural materials. The study of dynamic behavior of a structure holds atmost importance in evaluating its engineering performance and serviceability.

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Chapter 1

GENERAL INTRODUCTION

1.1 Introduction

The vibration analysis of beams or beam-like structural elements and composites has been and continues to be the subject of numerous researches, since it embraces a wide class of problems with immense importance in engineering science. Every structure which is having some mass and elasticity is said to vibrate. When the amplitude of these vibrations exceeds the permissible limit, failure of the structure occurs. To avoid such a condition one must be aware of the operating frequencies of the machine foundation or the materials under various conditions like simply supported, fixed or when in cantilever conditions.

In practical application the vibration analysis assumes great importance. For example, vehicle-induced vibration of bridges and other structures that can be simulated as beams and the effect of various parameters, such as suspension design, vehicle weight and velocity, damping, matching between bridge and vehicle natural frequencies, deck roughness etc., on the dynamic behavior of such structures have been extensively investigated by a great number of researchers . The whole matter will undoubtedly remain a major topic for future scientific research, due to the fact that continuing developments in design technology and application of new materials with improved quality enable the construction of lighter and more slender structures, vulnerable to dynamic and especially moving loads.

Depending on the assumptions adopted, the type of analysis used, the kind of the loading or excitation and the overall beam characteristics, a variety of different approaches have been reported in the literature and a great number of both theoretical and experimental findings are related to beam dynamics.

The present study of vibration analysis of various structural elements uses the tool *Brüel&Kjær PULSE*™, Multi-analyzer System Type 3560 for generating the vibration spectrum and to get results for various structural elements. The entire setup details and the parts of this tool have been explained in detail at a later stage in the report.

1.2 Theory

1.2.1 Vibration

Structures experiences vibration to some degree and their design generally requires considerations of their oscillatory behavior.
The governing equations for free vibration is

$$[M]\{\ddot{q}\}+[C]\{\dot{q}\}+[K]\{q\}=\{F\}$$
$$[K]-\omega^2[M]=0$$

In practice, nearly all vibration problems are related to structural weaknesses, associated with resonance behavior (that is natural frequencies being excited by operational forces). It can be shown that the complete dynamic behavior of a structure (in a given frequency range) can be viewed as a set of individual modes of vibration, each having a characteristic natural frequency, damping, and mode shape. By using these so-called modal parameters to model the structure, problems at specific resonances can be examined and subsequently solved.

The first stage in modeling the dynamic behavior of a structure is to determine the modal parameters as introduced above:

- ✓ The resonance, or modal, frequency
- ✓ The damping for the resonance – the modal damping
- ✓ The mode shape

By using the modal parameters to model the structure, vibration problems caused by these resonances (modes) can be examined and understood. In addition, the model can subsequently be used to come up with possible solutions to individual problems.
The modal parameters can be extracted from a set of Frequency Response Function (FRF) measurements between one or more reference positions and a number of

measurement positions required in the model. A position is a point and a direction on the structure and is hereafter called a Degree of Freedom (DOF). The resonance frequencies and damping values can be found from any of the FRF measurements on the structure (except those for which the excitation or response DOF is in a nodal position, that is, where the mode shape is zero). These two modal parameters are therefore called ‘Global Parameters’. To accurately model the associated mode shape, frequency response measurements must be made over a sufficient number of DOFs to ensure enough detailed coverage of the structure under test. The extraction of the modal parameters from the FRFs can be done using a variety of mathematical curve-fitting algorithms. The FRFs are obtained using multi-channel FFT measurements. To arrive at these FRFs, the excitation force (from either an impact hammer or a shaker provided with a proper signal) and responding vibrations are measured. The FRFs can be represented in different ways depending on the measured response used:

- ✓ If the vibration response is measured in terms of acceleration, the FRF represents an accelerance function as it gives the complex ratio of acceleration over force in the frequency domain
- ✓ If the vibration response is measured in terms of velocity, the FRF represents a mobility function
- ✓ If the vibration response is measured in terms of displacement, the FRF represents a compliance function

When used for modal analysis, the three measurements contain the same information and are related to each other via integration or differentiation, which means division or multiplication by (jω) in the frequency domain, where (ω) is the angular frequency. In general, these types of measurements are referred to as mobility measurements as the FRFs determine how ‘mobile’ the structure is, i.e., how much vibration response per input force excitation.

If there are enough accelerometers available, they can be mounted on all the DOFs on the structure. This gives a larger corresponding mass loading. However, each DOF's mass loading will subsequently be the same in all the FRFs, providing better consistency in the data. In addition, if the number of measurement channels allows for measurements of all responses simultaneously, the measurement time could be minimized and data consistency maximized.

To avoid mass loading, response measurements can be performed using a Laser Doppler Vibrometer (single-point) or a Scanning Laser Doppler Vibrometer, which measures velocity in the direction of the laser beam.

1.2.2 System Response Model

The analysis of the input signals and the generation of the spectrum are done in the following way. The relationship between input (force excitation) and output (vibration response) of a linear system is given by:

$$\{Y\} = [H]\{X\} \tag{1}$$

Where $\{Y\}$ and $\{X\}$ are the vectors containing the response spectra and the excitation spectra, respectively, at the different DOFs in the model, and $[H]$ is the matrix containing the FRFs between these DOFs.

Equation (1) can also be written as:

$$Y_i = \sum_j H_{ij} X_j \tag{2}$$

Where *Yi* is the output spectrum at DOF i, *Xj* is the input spectrum at DOF j, and *Hij* is the FRF between DOF j and DOF i. The output is the sum of the individual outputs caused by each of the inputs.

The FRFs are estimated from the measured auto- and cross-spectra of and between inputs and outputs. Different calculation schemes (estimators) are available in order to optimize the estimate in the given measurement situation (presence of noise, frequency resolution, etc...).

There are two different types of Reference Modal Test that can be done using PULSE

1. Single-Reference Modal Test
2. Multiple-Reference Modal Test

Multi-reference modal testing is required in situations where a single-reference DOF featuring sufficient participation of all modes cannot be found, or where more modes exist at the same frequency. Multiple-Input Multiple-Output (MIMO) testing provides better distribution of the input force energy, which is especially important for large, complex and heavily damped structures. Additionally, it gives advantages in terms of improved consistency in the data and reduced test time.

The determination of frequency with the help of PULSE software requires the determination of the following factors

1.2.3 Determination of the Mode Shape

The simplest way of determining the mode shape for a structure is to use a technique called Quadrature Picking. Quadrature Picking is based on the assumption that the coupling between the modes is light. In practice, mechanical structures are often very lightly damped (<1%). This implies that the modes are lightly coupled. At any frequency, the magnitude of the frequency response function is the sum of the contributions (at the particular frequency) from all modes. When there is little modal coupling between the modes, the structural response at a modal frequency is completely controlled by that mode, and so Quadrature Picking can be used to unravel the mode shapes.

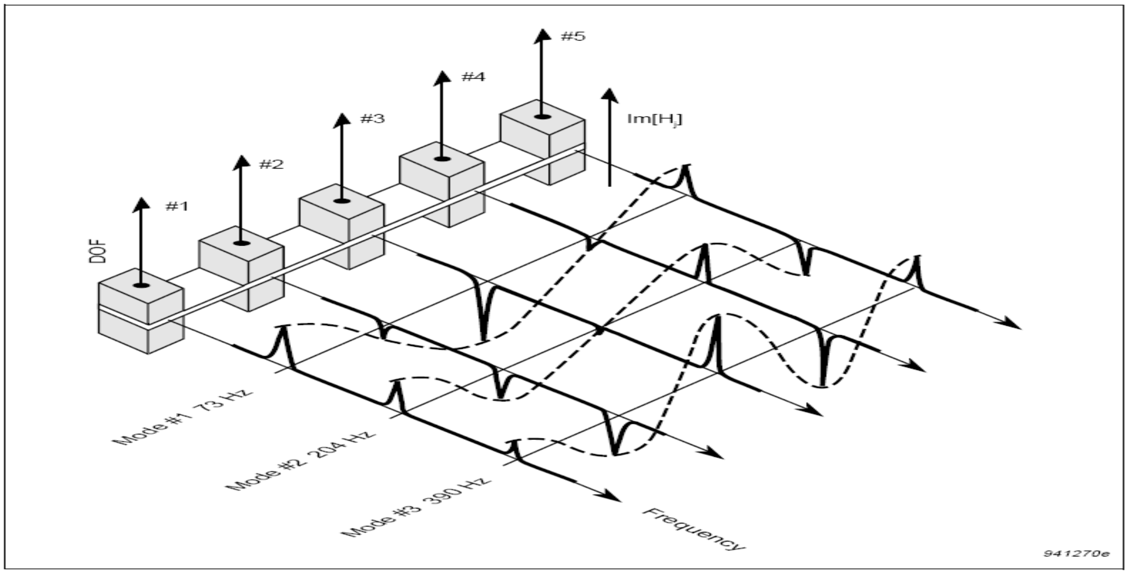
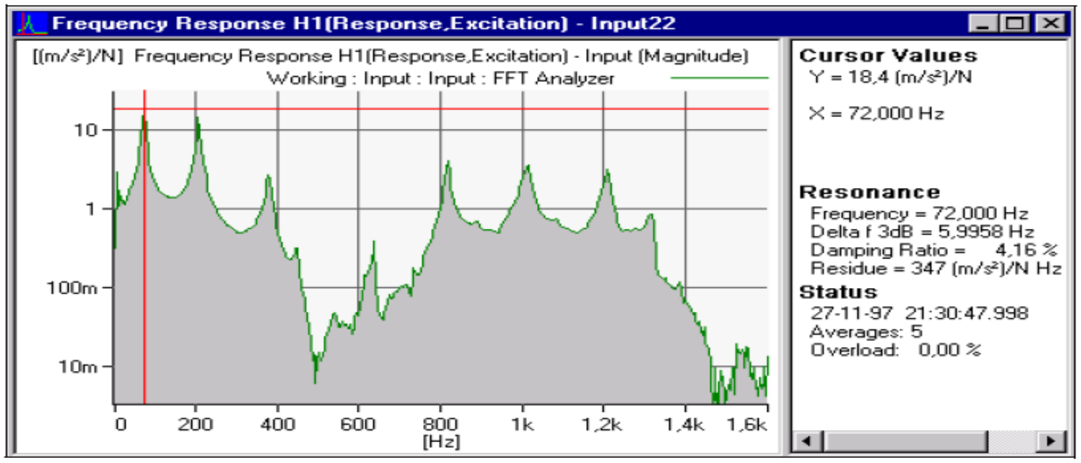


Fig. 1.1 The first three modes of vibration for the test structure. The modal displacements are found from the imaginary part of the frequency response function

1.2.4 Determination of the Modal Frequencies

The resonance frequency is the easiest modal parameter to determine. A resonance is identified as a peak in the magnitude of the frequency response function, and the frequency at which it occurs is found using the analyzer's cursor as shown in Fig. 6.


Frequency Response H1(Response,Excitation) - Input22

[(m/s²)/N] Frequency Response H1(Response,Excitation) - Input (Magnitude)
Working : Input : Input : FFT Analyzer

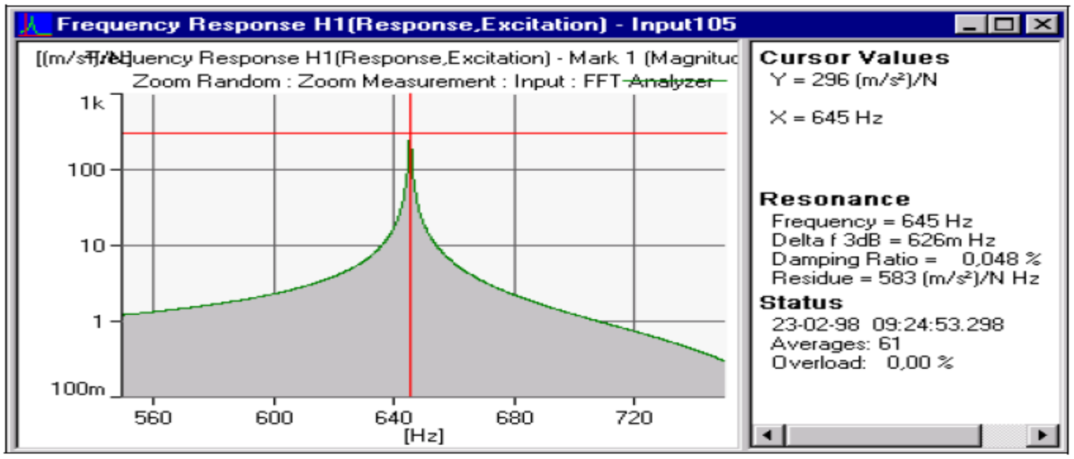
Cursor Values
Y = 18,4 (m/s²)/N
X = 72,000 Hz

Resonance
Frequency = 72,000 Hz
Delta f 3dB = 5,9958 Hz
Damping Ratio = 4,16 %
Residue = 347 (m/s²)/N Hz

Status
27-11-97 21:30:47.998
Averages: 5
Overload: 0,00 %

Fig. 1.2 Magnitude of the frequency response function (Accelerance) for the test structure

The frequency resolution of the analysis determines the accuracy of the frequency measurement. Greater accuracy can be attained by reducing the frequency range of the base-band measurement, for resonances at low frequencies, or making a zoom measurement around the frequency of interest, as shown in Fig. 7.


Frequency Response H1(Response,Excitation) - Input105

[(m/s²)/N] Frequency Response H1(Response,Excitation) - Mark 1 (Magnitud
Zoom Random : Zoom Measurement : Input : FFT Analyzer

Cursor Values
Y = 296 (m/s²)/N
X = 645 Hz

Resonance
Frequency = 645 Hz
Delta f 3dB = 626m Hz
Damping Ratio = 0,048 %
Residue = 583 (m/s²)/N Hz

Status
23-02-98 09:24:53.298
Averages: 61
Overload: 0,00 %

Fig. 1.3 Zoom measurement for achieving higher frequency resolution in the determination of the resonance frequency for the second mode. Determination of the damping for the second mode is identified by the half-power points and is read-out in the auxiliary cursor field

1.2.5 Composites

The other major components of study are the making and properties of composites. The composites that we used in our study are glass-fiber composites. They are cast with epoxy and a small percentage of hardener.

The quest of mankind for materials which can give him the best has initiated the advent of composite materials which have established itself as one of the most important engineering material today. The history of fiber reinforced composites is only three decades old. But during this short period there has been tremendous advancement in the science and technology of this new class material. The characteristics of composites that have made them one of the most important materials are

- ✓ Low density
- ✓ High strength
- ✓ High stiffness to weight ratio
- ✓ Excellent durability
- ✓ Design flexibility of reinforced polymers

are the primary reasons for their use in many structural components in aircrafts, automotives, marine and other industries.

The analysis and design of composite structures requires the input of reliable experimental data. As in the case of analysis, experimental characterization can be done on several scales, micro-mechanical, macro-mechanical and structural. Under the general objectives, specific types and applications of testing include the following.

- ✓ Characterization of constituent materials ie. Fiber, matrix and inter phase, for use in micromechanics analyses. Knowing these properties one can predict, in principle, the behavior of the lamina and hence of laminates and structures.
- ✓ Characterization of basic uni-directional lamina which forms the basic building blocks of all laminated structures.
- ✓ Determination of inter laminar properties.
- ✓ Material behavior under special conditions of loading, eg. Multi axial, fatigue, creep, impact and high rate loading.

✓ Experimental stress and failure analysis of composite laminates and structures, especially those involving geometric discontinuities such as free edges, cutouts, joints and ply drop offs.

A variety of experimental methods are used for the various applications above. Most of these deal with measurement for deformations or strains. Experimental methods for composite materials are much more complex than for isotropic materials and require significant modifications.

In particular the use of composites in safety critical applications lead to uneasiness since the mechanical response in crash applications is not well understood. Many complex processes occur in crash situations. The material is subjected to rapid accelerations and rapid straining with large pressure shocks, often accompanied by huge temperature increases of several hundreds degrees. This further stress the need for a full characterization of the behavior of fiber reinforced polymer composites under dynamic loading conditions and has prompted numerous investigations in recent years. However, when compared to metals, relatively few studies have been conducted to investigate polymer mechanical properties at high loading speeds.

1.2.6 Importance of the present study

Composite materials are extensively used for various components of civil, mechanical, metallurgical, naval and aerospace engineering structures.

Some applications of polymer matrix composites

Industrial Sector	Examples
Aerospace	Wings, fuselage, antennae, tail planes, helicopter, blades, landing gear, seats, floor, fuel tanks, rocket motor cases, nose cones, launch tubes.
Civil	Synclastic shell, anticlastic shell, folded plate structure, skeletal structure, movable

	prefabricated house, exterior and interior wall panels, doors and windows, partition walls, canopies, staircase and ladders, water tanks, pipes and drainages, pedestrian bridges.
Automobiles	Body panels, spoilers, lamp housings
Boats	Hulls, deck, masts, engine shrouds
Chemical	Pipes, tanks, hoppers, valves, pumps
Domestic	Chairs, tables, baths, ladders
Electrical	Panels, housing, switchers, insulators
Leisure	Motor homes, caravans, trailers, pools

Table 1.1 Listing the various uses of composites in different engineering spheres

This wide range of practical applications demands an understanding of the fundamental behavior of the composite materials. Fiber reinforced polymer (FRP) composite structures are subjected to a wide range of loading conditions in service. These include both static and dynamic loads and the life of a component will be determined by a failure criterion defining an acceptable level of stiffness & strength. The process such as fiber de-bonding, fiber pull out an matrix cracking have been shown an important role in the effect of fiber types & load speed of the laminates. Thus, the studies on the failure of different composites for varying rate of loading are of technical importance for understanding the behavior of composites.

1.2.7 Fiber reinforced Polymer (FRP) Composites

“Composite” material signifies that two or more materials are combined on a macroscopic scale to form a useful material. Fiber reinforced polymer (FRP) composite materials consist of fibers of high strength and modulus embedded in or bonded to a matrix with distinct interfaces between them. In this form, both fibers and matrix retain their physical and chemical identities, yet they produce a combination of properties that can not be achieved with either of constitutes acting alone. In general, fibers are the principal load carrying members, while the surrounding matrix keeps them in the desired location and orientation, act as a load transfer medium between them, and protects them from environmental damages due to elevated temperatures and humidity conditions.

The most common form in which fiber reinforced polymer composites are used in structural applications is called a laminate. It is obtained by stacking a number of thin layers of fibers & matrix and consolidating them into the desired thickness. Fiber orientation in each layer as well as the stacking sequence of various layers can be controlled to generate a wide range of physical and mechanical properties for the composite laminate.

1.2.8 Advantages of Composites

Composites are able to meet diverse design requirements with significant weight savings as well as high strength to weight ratio as compared to conventional materials. Some advantages of composites materials over conventional one are mentioned below.

- ✓ Tensile strength of composites is four to six times greater than of steel or aluminum.
- ✓ Improved torsional stiffness and impact properties
- ✓ Higher fatigue endurance limit (up to 60% of the ultimate tensile strength).
- ✓ 30 to 45% lighter than aluminum structures designed to the same functional requirements.
- ✓ Lower embedded energy compared to other structural materials like steel, aluminum etc.
- ✓ Composites are less noisy while in operation and provide lower vibration transmission than metals.
- ✓ Composites are more versatile than metals and can be tailored to meet performance needs and complex design requirements.
- ✓ Long life offers excellent corrosion resistance and fire retardancy.
- ✓ Improved appearance with smooth surfaces and readily incorporable integral decorative melamine are other characteristics of composites.
- ✓ Composites parts can eliminate joints/fasteners. Providing part simplification and integrated design compared to conventional metallic parts.

Processing of FRP Composites

The method of production and fiber reinforced polymer (FRP) selected by a manufacturer will depend on factors such as cost, shape component, number of components and required performance.

- ✓ Hand Methods
- ✓ Molding Methods
  - o Matched die molding
  - o Pultrusion
- ✓ Filament Winding

1.2.9 Fiber

Fiber constitutes the main bulk of reinforcements that are used in making structural composites. A fiber is defined as a material that has the minimum l/d ratio equal to 10:1, where 'l' is the length of the fiber and 'd' is its minimum lateral dimension. The lateral dimension 'd' (which is the diameter in the case of a circular fiber) is assumed to be less than 254μm. The diameter of fibers used in structural composites normally varies from 5μm to 200μm. A filament is a continuous fiber with the l/d ratio equal to infinity.

A whisker is a single crystal, but has the form of a fiber. A number of commercially available fibers as follow

- ✓ Glass fibers
- ✓ Carbon fibers
- ✓ Kelvar fibers
- ✓ Boron fibers
- ✓ Ceramic fibers

Fibers share the major portion of the load acting on a composite structure. Proper selection of the type, amount and orientation of fibers is very important. Since, it influences the following characteristics of a composite laminate.

- ✓ Specific gravity
- ✓ Tensile strength & modulus

- ✓ Compressive strength & modulus
- ✓ Fatigue strength as well as fatigue failure of mechanisms
- ✓ Electrical & thermal conductivities
- ✓ Cost

Glass Fibers

The glass fibers were manufactured more than 2000 years ago in Rome and Mesopotamia and were abundantly used in decoration of flower vases and glass wares in those days.

Glass was first made by man in 3000bc in Asia Minor. Continuous glass fibers were known to be used for decorative purposes in ancient times in Syria and Venice. The industrial manufacturing of glass fibers started in 1930's for use in filters and insulations. Glass fibers currently comprise more than 90% of fibers used in polymer composites. There are five major types of glass used to make fibers. These are

- ➢ A-glass (high alkali)
- ➢ C-Glass (chemical)
- ➢ D-Glass (low dielectric constant)
- ➢ E-Glass (electric)
- ➢ S-Glass (high strength)

Glass Fiber Type	SiO_2	Al_2O_3	CaO	MgO	B_2O_3	Na_2O
E Glass	54.5	14.5	17	4.5	8.5	0.5
S Glass	64	26	-	10	-	-

Table 1.2 Typical chemical composition of E&S Glass Fibers (in weight percentage)

Epoxy

Starting materials for epoxy matrix are low molecular weight organic liquid resins containing a number of epoxide groups, which are three membered rings of one oxygen atom and two carbon atoms. Epoxy matrix as a class, has the following advantages over thermo set matrices

- ✓ Wide variety of properties, since a large number of starting materials, curing agents, and modifiers are available.
- ✓ Absence of volatile matters during curing
- ✓ Low shrinkage during curing
- ✓ Excellent resistance to chemical & solvents
- ✓ Excellent adhesion to a wide variety of fillers, fibers and other substrates

The principal disadvantages are its relatively high cost and long cure time.

1.2.10 General Failure Behavior of Composite Laminate

The degradation of material property of the composite laminate during its service may be due to the following factors

- ✓ Plasticization of matrix
- ✓ Debonding of fiber matrix interface
- ✓ Micro cracking in the matrix
- ✓ Fiber fragmentation
- ✓ Cracks
- ✓ De-lamination
- ✓ Fiber pullout
- ✓ Fiber push

1.3 Literature Review

There has been immense interest in the scientific community about the vibration analysis of structures and there has been lot of research going on this subject. The focus of the study has however been on the numerical methods by which one can solve the vibration analysis of various structures.

M. Huanga, X.Q. Mab, T. Sakiyamaa, H. Matsudaa, C. Moritaa studied the Free vibration analysis of rectangular plates with variable thickness and point supports by a discrete method for analyzing the free vibration problem of rectangular plates with point supports. The fundamental differential equations involving Dirac's delta function are established for the bending problem of the plate with point supports. By transforming these differential equations into integral equations and using numerical integration, the solution of these equations is obtained and used as Green function to obtain the characteristic equation of the free vibration. The effects of the point support, the boundary condition, the variable thickness and aspect ratio on the frequencies are considered. This is one of the numerical methods for solving the vibration analysis problems.

The effect of structural defects on the vibration analysis is the step further. This has also been an area of immense interest. But this analysis is done by using numerical methods rather than adopting any experimental approach. E. Violaa,_, P. Riccia, M.H. Aliabadib studied the Free vibration analysis of axially loaded cracked Timoshenko beam structures using the dynamic stiffness method, the purpose is to investigate the changes in the magnitude of natural frequencies and modal response introduced by the presence of a crack on an axially loaded uniform Timoshenko beam using a particular member theory. A new and convenient procedure based on the coupling of dynamic stiffness matrix and line-spring element is introduced to model the cracked beam. The application of the theory is demonstrated by two illustrative examples of bending–torsion coupled beams with different end conditions, for which the influence of axial force, shear deformation and rotatory inertia on the natural frequencies is studied. Moreover, a parametric study to investigate the effect of the crack on the modal characteristics of the beam is conducted.

Similar to the above work, C. Meia,_, Y. Karpenkob, S. Moodyb, D. Allenb studied the Analytical approach to free and forced vibrations of axially loaded cracked Timoshenko beams. The dynamics of cracked structural members, especially beams, has been the subject of many research works mainly due to the growing interests in non-destructive damage evaluation of engineering structures using modal responses (natural frequencies and modeshapes) of a structure in the past two decades. The presence of a crack in a structural member introduces a local flexibility that affects its dynamic response. Numerous attempts to quantify local defects are reported to the literature. In general, there exist three basic crack models, namely the equivalent reduced section model, the local flexibility model from fracture mechanics and the continuous crack flexibility model. Various approaches have been applied in vibration analysis, mostly free vibration analysis, of cracked beams. Such approaches include finite element approach, Galerkin and local Ritz approach, approximate analytical approach, transfer matrix approach and dynamic stiffness matrix approach. In this paper, both free and forced vibrations are studied for an axially loaded cracked Timoshenko beam from wave standpoint, in which the vibrations are described in terms of wave propagation, transmission and reflection in waveguides. The reflection and transmission characteristics of flexural vibration waves have been studied by a number of researchers.

In this study, the transmission and reflection matrices for various discontinuities on an axially loaded cracked Timoshenko beam are derived. Such discontinuities include cracks, boundaries and change in sections. The matrix relations between the injected waves and externally applied forces and moments are also derived. These matrices can be combined to provide a concise and systematic approach to both free and forced vibration analyses of axially loaded cracked Timoshenko beams or complex structures consisting of such beam components. The effects of cracks (including crack size and crack location), axial loads and step sectional changes on the modes of vibrations are studied.

The focus in the present day has however shifted on to the experimental methods to get the vibration analysis and various vibration analysis software have been developed to get the frequency, amplitude and acceleration ranges of a particular vibrating structure. In line with this present trend, Ertuğrul Çam, Sadettin Orhan and Murat Lüy studied a problem of cracked beam similar to the above mentioned ones using impact echo method. Defects influence in a negative way the service life of structures. Thus, detection of them even at a very small size is a very important point of view to guarantee structural safety and to safe costs. The objective of this study is to obtain information about the location and depth of cracks in cracked beams. For this purpose, the vibrations as a result of impact shocks were analyzed. The signals obtained in defect-free and cracked beams were compared in the frequency domain. The results of the study suggest to determine the location and depth of cracks by analyzing the from vibration signals. Experimental results and simulations obtained by the software ANSYS are in good agreement

The use of accelerometers and laser vibrometers has been the latest trend in vibration analysis. Won-Suk Ohm, Lixue Wu, Peter Hanes, George S.K. Wong studied generation of low-frequency vibration using a cantilever beam for calibration of accelerometers. At low frequencies, performance of a conventional shaker is limited by small acceleration amplitudes and a high level of total harmonic distortion. The present article describes a low-frequency vibration generator that overcomes these limitations. The vibration generator consists of a cantilever beam excited by a conventional shaker. The cantilever beam is tuned to resonate at the desired excitation frequency, which leads to a relatively large vibratory motion at the beam tip with very small harmonic distortion. Analysis of the system is performed by means of model equations describing both the flexural and longitudinal components of vibration.

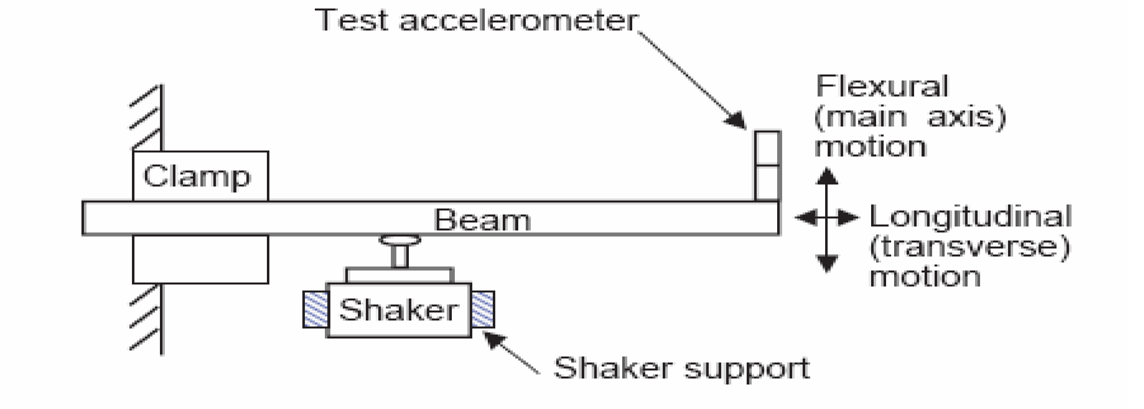


Fig 1.4 Test setup for using the accelerometer for vibration analysis

Similarly, C. Cristalli,_, N. Paone, R.M. Rodrı´guez studied the Mechanical fault detection of electric motors by laser vibrometer and accelerometer measurements the study presents a comparative study between accelerometer and laser vibrometer measurements aimed at on-line quality control carried out on the universal motors used in washing machines, which exhibit defects localized mainly in the bearings, including faults in the cage, in the rolling element and in the outer and inner ring. A set of no defective and defective motors were analyzed by means of the acceleration signal provided by the accelerometer, and the displacement and velocity signals given by a single-point laser vibrometer. Advantages and disadvantages of both absolute and relative sensors and of contact and non-contact instrumentation are discussed taking into account the applicability to real on-line quality control measurements and bringing to light the related measurement problems due to the specific environmental conditions of assembly lines and sensor installation constraints. The performance of different signal-processing algorithms is discussed. RMS computation at steady-state proves effective for pass or fail diagnosis, while the amplitude of selected frequencies in the averaged spectra allows also for classification of a variety of special faults in bearings. Joint time–frequency analysis output data can be successfully used for pass or fail diagnosis during transients, thus achieving a remarkable reduction in testing time, which is important for on-line diagnostics.

Vibration tests are critical and complex tests to be carried out in a production line due mainly to:

- ✓ Intrinsic complexity of the vibration phenomena in assembled systems operating in test conditions.
- ✓ Noisy environmental conditions in the production line.
- ✓ Short time available, usually the test must be performed in a few seconds.

An automatic system should fulfill the following requirements:

- ✓ Good repeatability,
- ✓ Reliability,
- ✓ Flexibility, adaptability to different types of motors and defects, and
- ✓ Capacity of being integrated in the production line.

The existing methods for mechanical diagnostics based on vibro-acoustic analysis are normally based on the use of accelerometers that have demonstrated a good performance in production processes, but installation may pose several problems. Microphones are often used to analyze noise emissions, but require acoustic isolation from other sound sources. Other measurement techniques, like the laser vibrometer, offer the advantage of being a non-contact method, avoiding the intrusive character of the accelerometer, and so to make sensor installation easier.

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Chapter 2

EXPERIMENTAL PROGRAM

2.1 Equipments

Brüel&Kjær PULSE™, Multi-analyzer System Type 3560

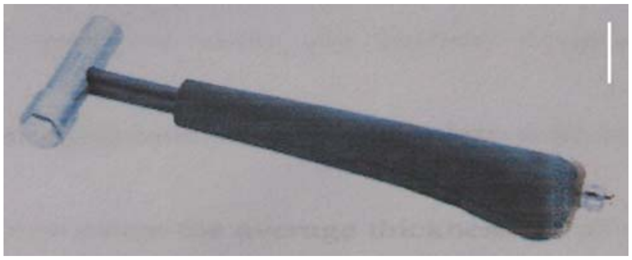
The PULSE software analysis was used to measure the frequency ranges to which the foundations of various machines are subjected to when the machine is running with no load and full load. This will help us in designing the foundations of various machines in such a way that they are able to resist the vibration caused in them. Below we present the analysis of frequency measurements for a few machine foundations measured in highway concrete lab in N.I.T. Rourkela.

Setup and Procedure (FFT analyzer)

- The connections of the FFT analyzer, laptop, transducers, model hammer along with the requisite power connections were made.
- The transducer was fixed to the foundation of the machine. The foundation made of concrete.
- To ensure perfect contact of the transducer the surface of the foundation was made clean and transducer was affixed properly.
- The machines were run with and without load to find out the frequency ranges in which the foundation is likely to vibrate.
- Each measurement was measured thrice to ensure no error creeps in measurement.

1. Accelerometer – 4507 type

2. Model Hammer – type 2302-05



3. Depicting the Various Components of the Experimental Setup

Fig. 2.1 Depicting the different parts and entire test setup of PULSE

2.2 Fabrication of Samples

2.2.1 Embedded Steel Flats

The steel flat was tested for its fundamental frequency. The steel flat was embedded in concrete base. A total of four samples were casted for testing purposes.

- ✓ Cantilever length of the steel flat was observed to be 29.5cm.
- ✓ Embedded length of steel flat is 10cm into the concrete base.
- ✓ The thickness of the flat is observed to be 0.45 cm
- ✓ The width of the flat is 1.9cm.

2.2.2 Glass Fiber Reinforced Polymer (GFRP) Composite

Two Glass-Epoxy composites were casted in order to test these samples for frequency measurements by varying the area of the samples. The two samples were one of

- ✓ 16 layers
- ✓ 12 layers

The areas of the two plates are
12 layers – 24 cm X 19.5 cm
16 layers - 28 cm X 19.7 cm
These rectangular plates were then reduced in size gradually. The sizes are as given in the table below

	16 –Layer Specimens				12 – Layers Specimens			
	1	2	3	4	1	2	3	4
Length (cm)	28	21	14	7	24	18	12	6
Breadth (cm)	19.7	19.7	19.7	19.7	19.5	19.5	19.5	19.5

Table 2.1 Depicting the sizes of specimens of composites

The length of each sample has been decreased in a regular pattern keeping the breadth same. Then the change in frequency pattern is being observed because of the change in the area of the sample.

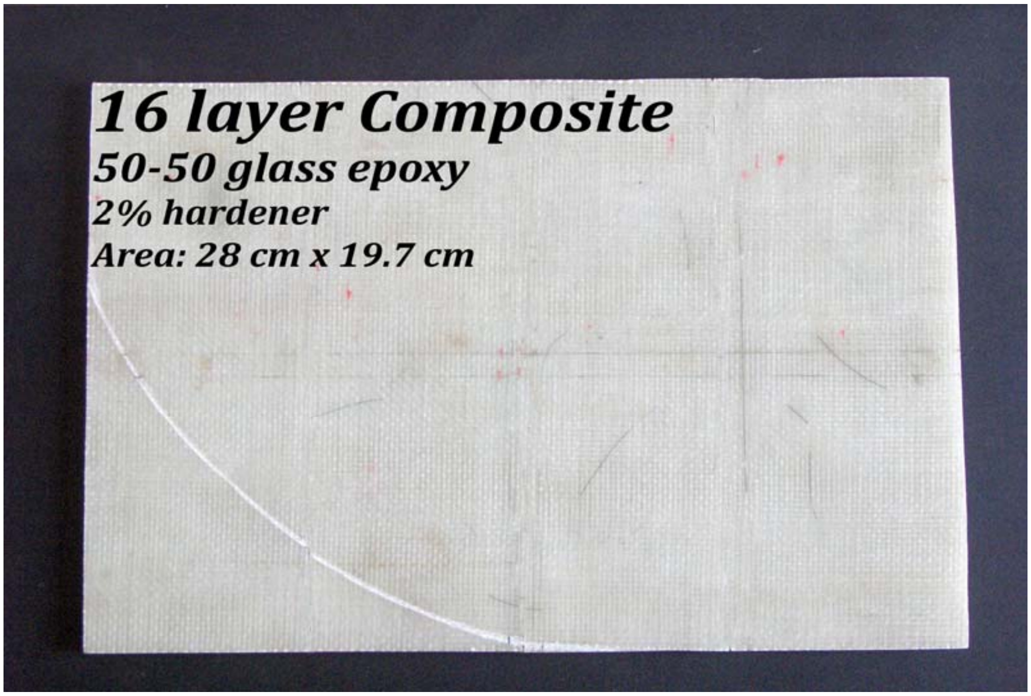


Fig 2.2 16 – layer sample of Glass Fiber Reinforced Polymer Composite

Fig 2.3 12 – layer sample of Glass Fiber Reinforced Polymer Composite

2.2.3 Vibration Analysis of Machine Foundations

The analysis was performed for the vibration of foundation of two machines

- ✓ Los-Angeles Abrasion Machine
- ✓ Jaw Crusher Machine

The machines with concrete foundations were selected and were tested in the lab. The setup has to done with extreme care as the sensitive equipment should be protected against dust and the distance has to be maintained from the machines for safety purposes.

2.2.4 Buckling Load of Rectangular Flat

A rectangular flat was taken with a length of 30cm. It was attached with two thin plates and then tested in a UTM machine under increasing axial load. The provision of thin plates on both sides of the flat ensures that the flat does not slip and the load gets transferred completely and effectively.

2.3 Experimental Program

The PULSE software was used to carry out the vibration analysis of the above mentioned cases. All the samples were tested for frequency ranges and the variations in their frequencies due to change in area of sample and by applying an axial load was imparted onto the sample. The results have been reported in the next chapter.

Chapter 3

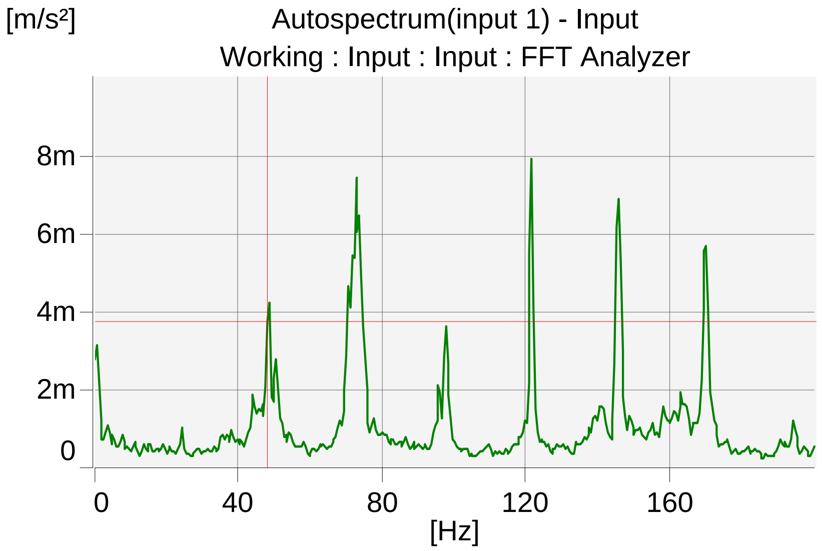
RESULTS AND DISCUSSIONS

3.1 The Pulse Report - Los Angels Abrasion Machine

The measurements were made when the machine was working with full load condition.

Report generated the 11/6/2006

First frequency measurement



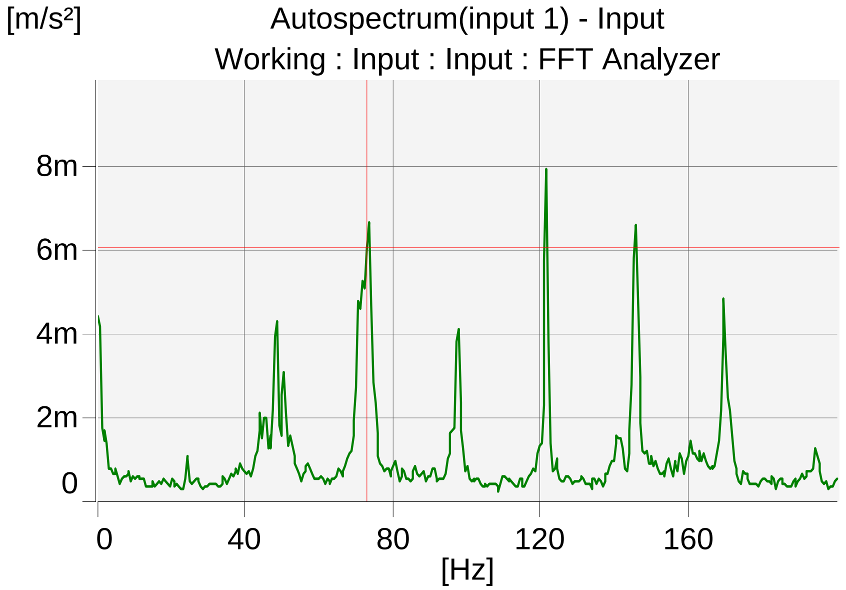
Cursor Values
Y = 3.70m m/s²

X = 48.00 Hz

Status
11/6/2006 13:53:38.908
Averages: 10
Overload: 0.00 %

Fig 3.1a Autospectrum (input 1) - Input

Second Frequency Measurement



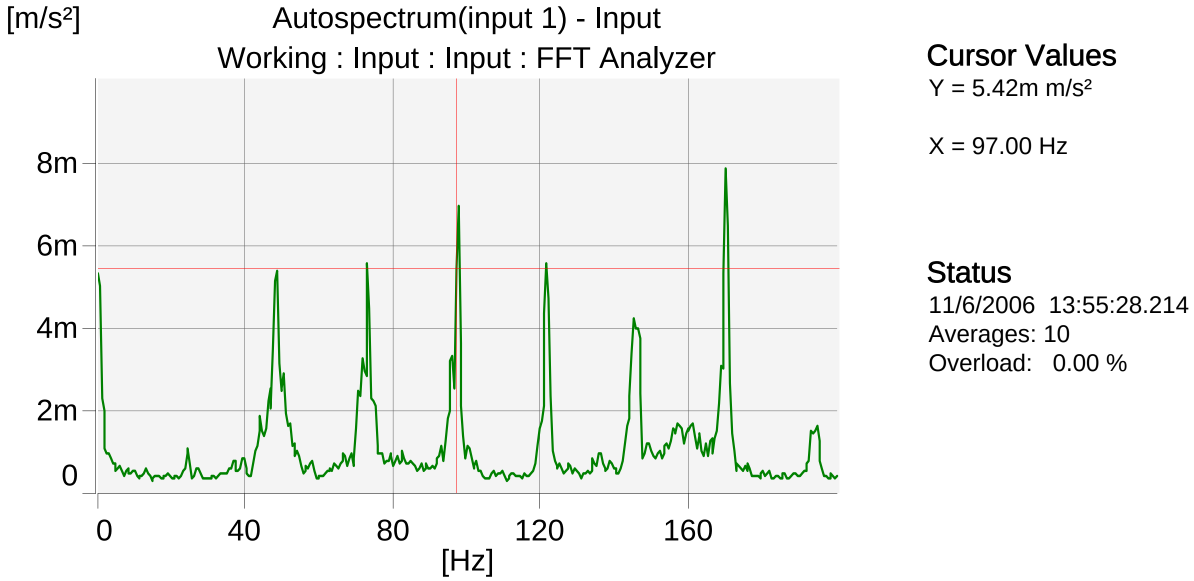
Cursor Values
Y = 6.02m m/s²

X = 73.00 Hz

Status
11/6/2006 13:54:03.568
Averages: 10
Overload: 0.00 %

Fig 3.1b Autospectrum (input 1) - Input

Third Frequency Measurement



Cursor Values

Y = 5.42m m/s²

X = 97.00 Hz

Status

11/6/2006 13:55:28.214

Averages: 10

Overload: 0.00 %

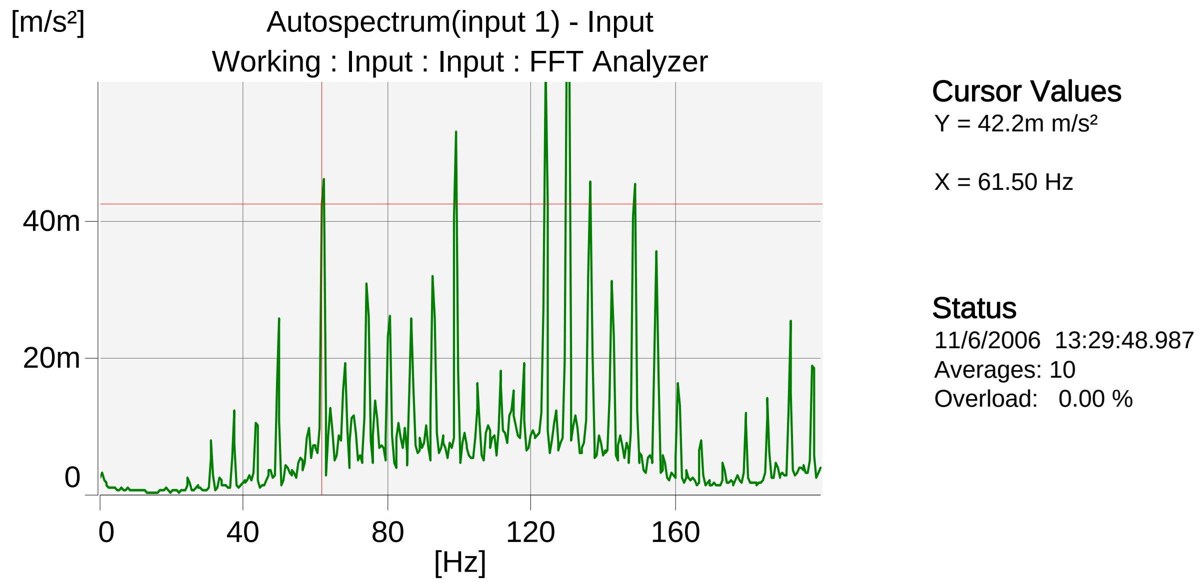
Fig 3.1c Autospectrum (input 1) – Input

3.2 The Pulse Report – Jaw Crusher Machine

The measurements were made when the machine was working with full load condition.

Report generated the 11/6/2006

First frequency measurement

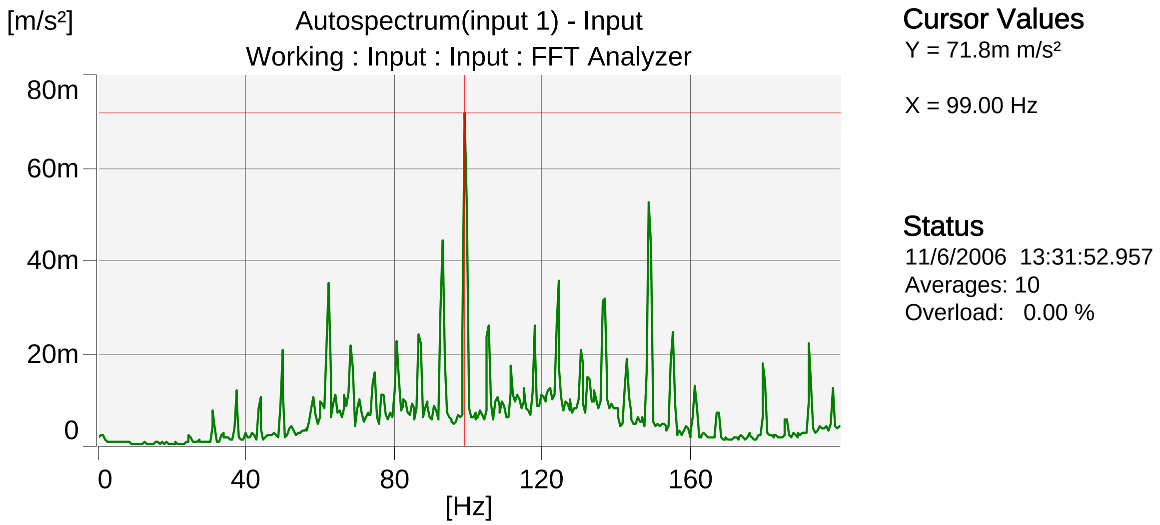


Cursor Values
Y = 42.2m m/s²
X = 61.50 Hz

Status
11/6/2006 13:29:48.987
Averages: 10
Overload: 0.00 %

Fig 3.2a Autospectrum (input 1) – Input

Second Frequency Measurement



Cursor Values
Y = 71.8m m/s²
X = 99.00 Hz

Status
11/6/2006 13:31:52.957
Averages: 10
Overload: 0.00 %

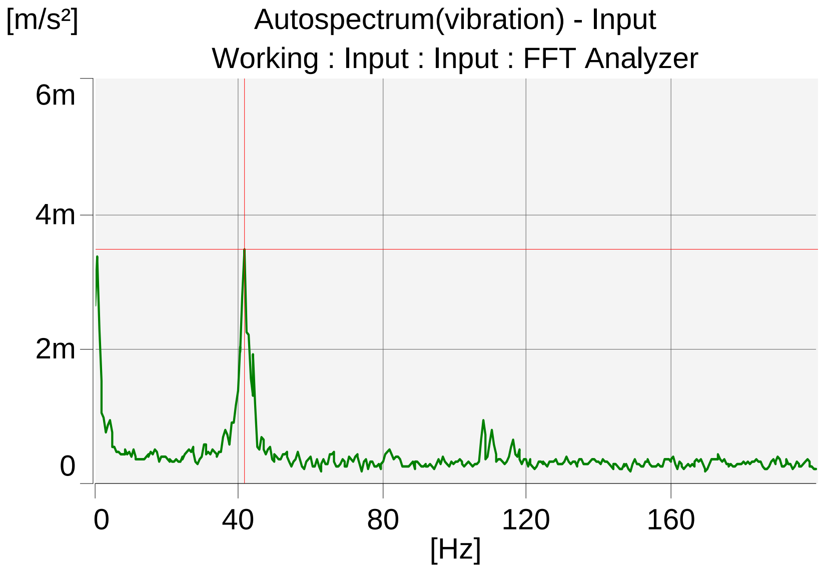
Fig 3.2b Autospectrum (input 1) – Input

3.3 The Pulse Report – Steel Flat

The steel flat was tested for its fundamental frequency. The steel flat was embedded in concrete base up to a length of 10cm and the cantilever length of the steel flat was observed to be 29.5cm for sample 1. A total of four samples were casted. The thickness of the flat is observed to be 0.45cm, the width of the flat is 1.9cm.
The sample 1 was tested by vibrating it with a model hammer and the accelerometer was affixed to the flat at various positions to verify the frequency measurement. Three measurements were made to minimize the error in measurement.

Report generated the 11/7/2006

First Frequency Measurement – First Observation



Cursor Values
Y = 3.44m m/s²

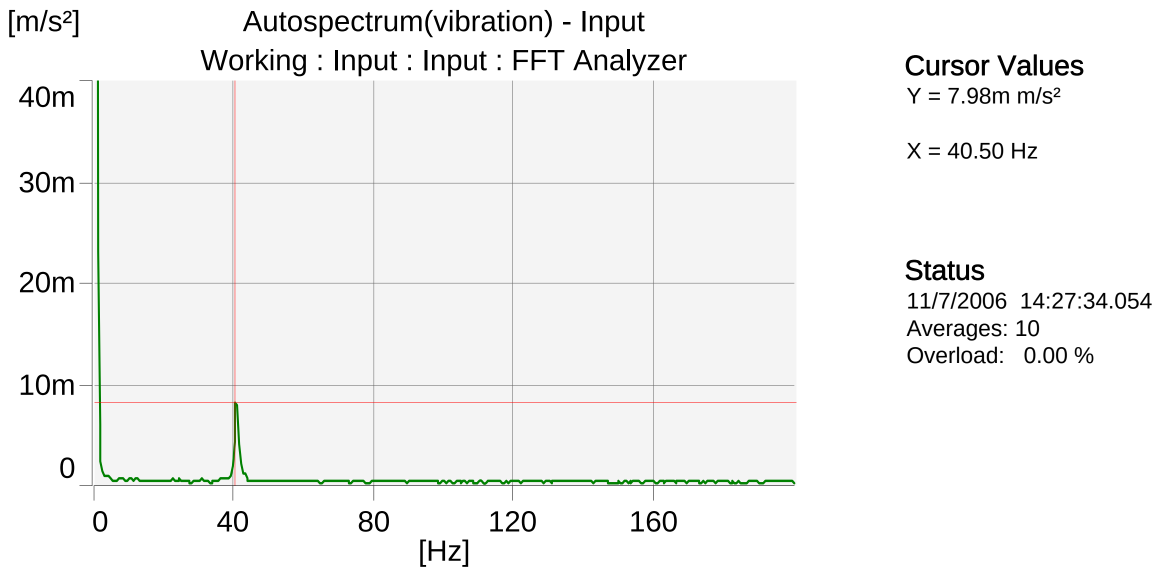
X = 41.50 Hz

Status
11/7/2006 14:26:18.739
Averages: 10
Overload: 0.00 %

Fig 3.3a Autospectrum (vibration) - Input

Here it is observed that there is no second frequency when the flat is vibrated with the model hammer and it is allowed to vibrate freely. The frequency measurement from observation 2 & 3 is listed below.

First Frequency Measurement – Second Observation



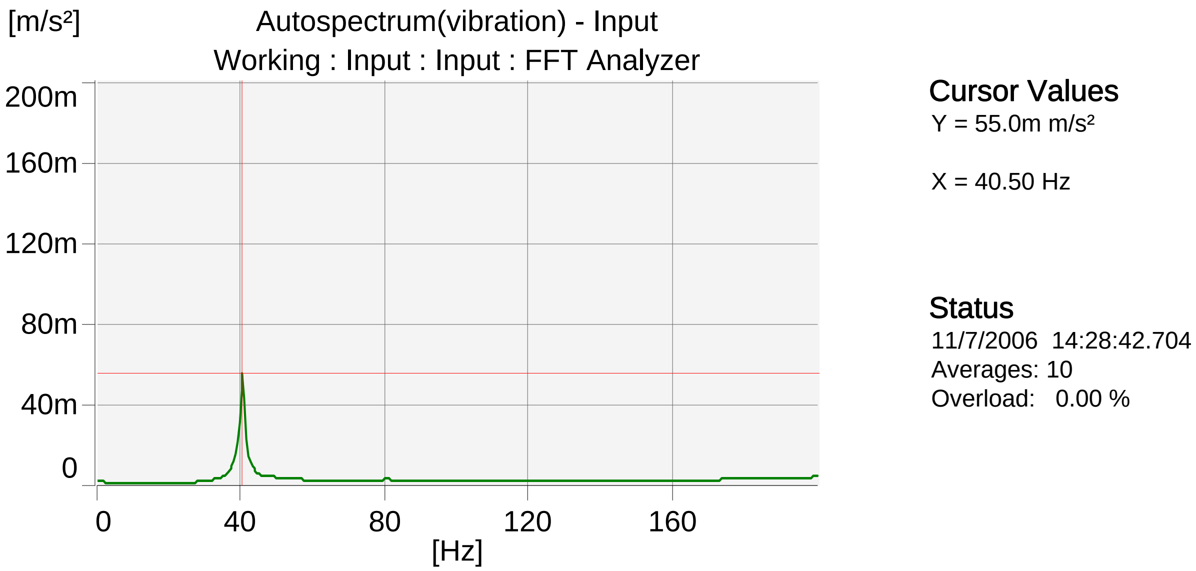
Cursor Values
Y = 7.98m m/s²

X = 40.50 Hz

Status
11/7/2006 14:27:34.054
Averages: 10
Overload: 0.00 %

Fig 3.3b Autospectrum (vibration) - Input

First Frequency Measurement – Third Observation



Cursor Values
Y = 55.0m m/s²

X = 40.50 Hz

Status
11/7/2006 14:28:42.704
Averages: 10
Overload: 0.00 %

Fig 3.3c Autospectrum (vibration) – Input

3.4 The Pulse Report – Composites

The two different composites were tested independently. The area of the composites was varied and the change in the frequency was observed.

3.4.1 The 16 – layers Glass Composite

3.4.1.1 Specimen 1 - 16 layers

Area of specimen 1 – 28 cm X 19.7 cm

First Frequency measurement - 284 Hz

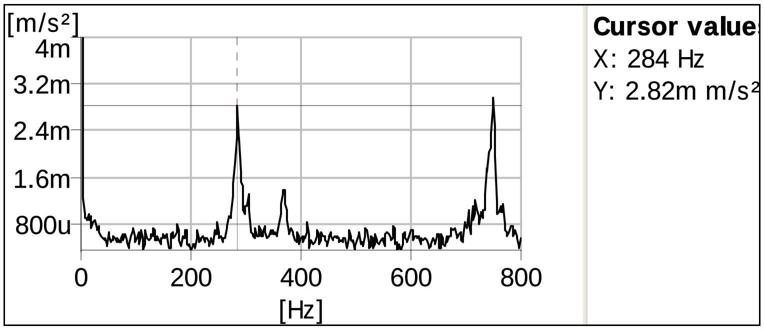


Fig 3.4.1.1a Pulse output showing first frequency output

Second Frequency measurement - 750 Hz

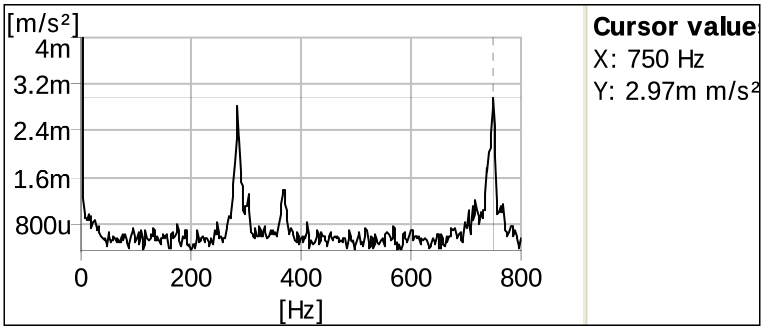


Fig 3.4.1.1b Pulse output showing second frequency output

3.4.1.2 Specimen 2 - 16 layers

Area of specimen 2 – 21 cm X 19.7 cm

First Frequency measurement - 284 Hz

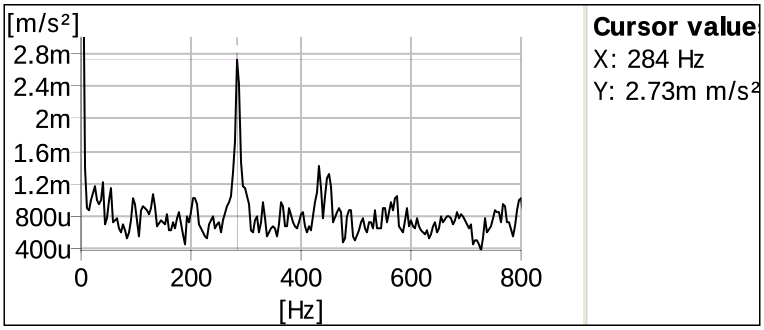


Fig 3.4.1.2a Pulse output showing first frequency output

Second Frequency measurement - 576 Hz

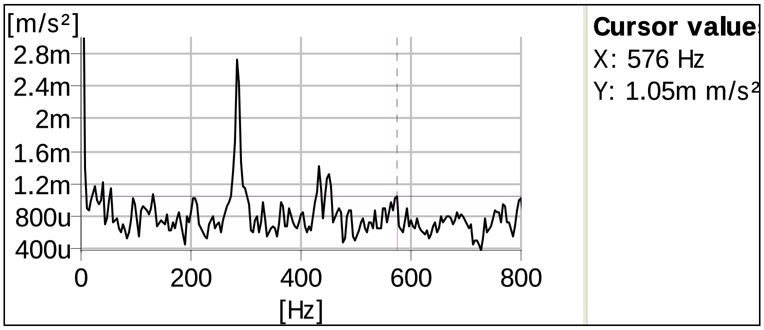


Fig 3.4.1.2b Pulse output showing second frequency output

3.4.1.3 Specimen 3 - 16 layers

Area of specimen 3 – 14 cm X 19.5 cm

First Frequency measurement - 270 Hz

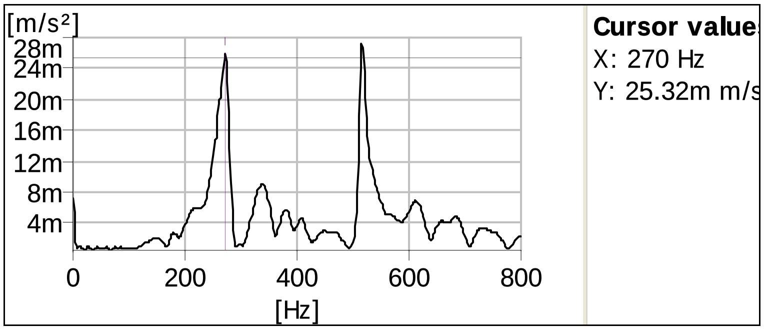


Fig 3.4.1.3a Pulse output showing first frequency output

Second Frequency measurement – 518 Hz

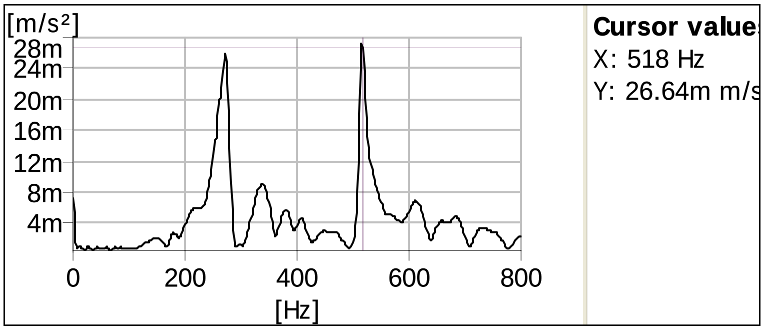


Fig 3.4.1.3b Pulse output showing second frequency output

3.4.1.4 Specimen 4 - 16 layers

Area of specimen 4 – 7 cm X 19.7 cm

First Frequency measurement - 236 Hz

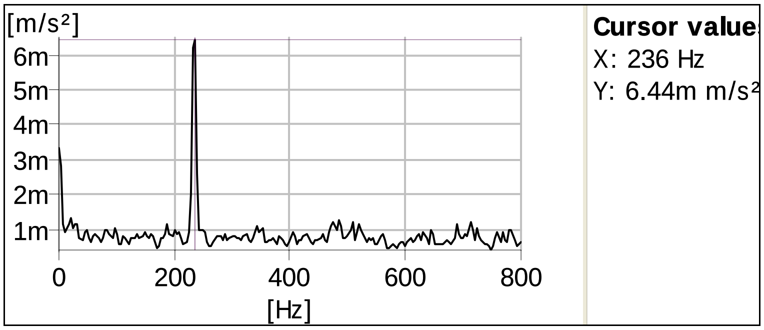


Fig 3.4.1.4 Pulse output showing first frequency output

3.4.2 Variation of Frequency – 16 layers

The frequency of vibration of the 16-layer composite is observed to decrease with the decrease in the area of the specimen. The same trend is being observed in both first and second frequencies. As the length of the composite decreases in a regular trend the frequencies are expected to decrease as the value of ‘k’ decreases.

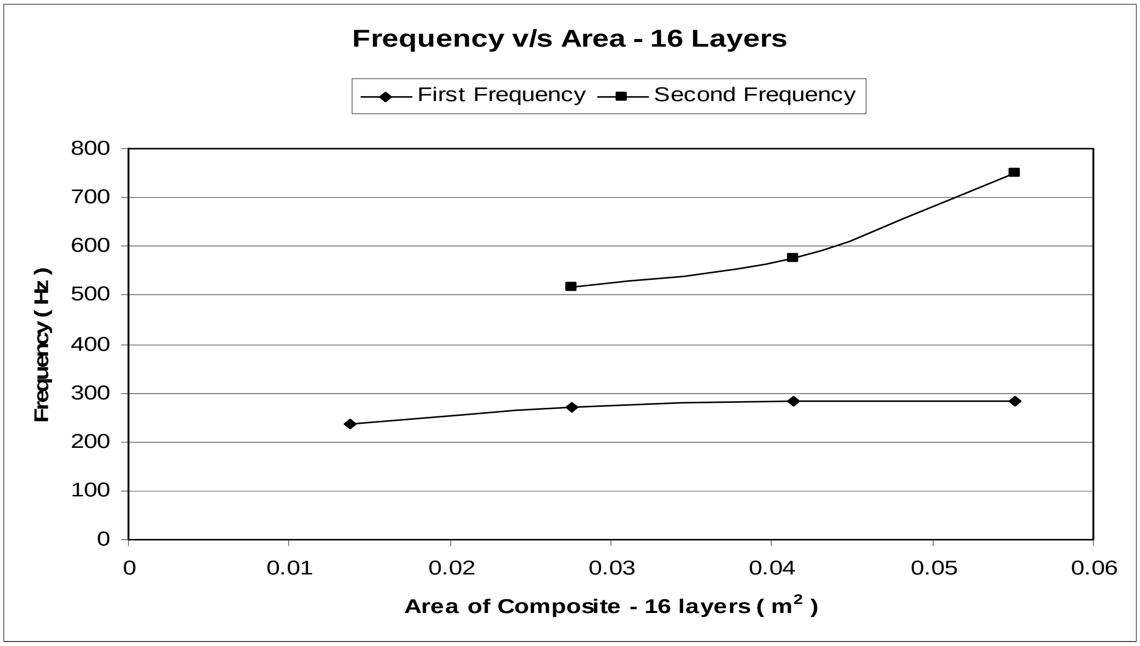


Fig 3.4.2 The variation of first frequency and second frequency with decreasing area of composite

3.4.3 The 12 – layers Glass Composite

3.4.3.1 Specimen 1 - 12 layers

Area of specimen 1 – 24 cm X 19.5 cm

First Frequency measurement - 190 Hz

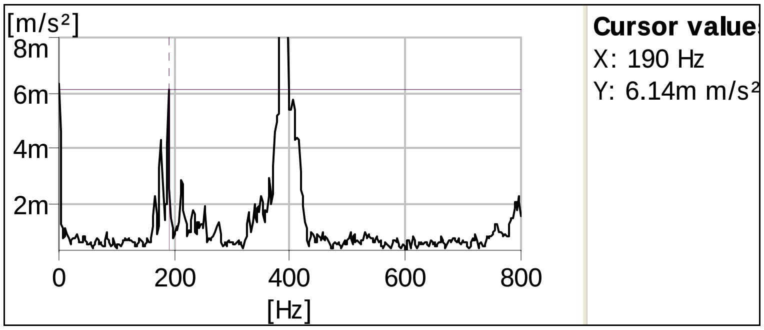


Fig 3.4.3.1a Pulse output showing first frequency output

Second Frequency measurement - 588 Hz

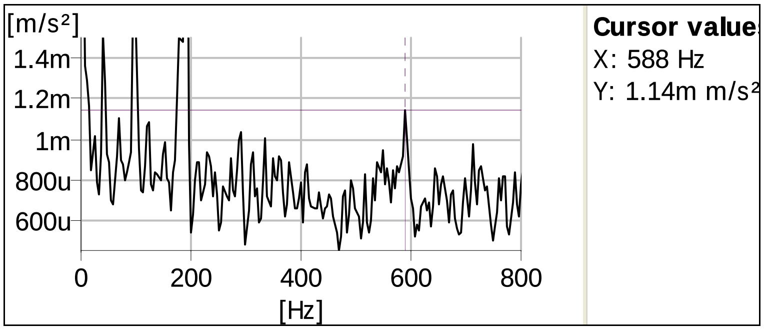


Fig 3.4.3.1b Pulse output showing second frequency output

3.4.3.2 Specimen 2 - 12 layers

Area of specimen 1 – 18 cm X 19.5 cm

First Frequency measurement - 188 Hz

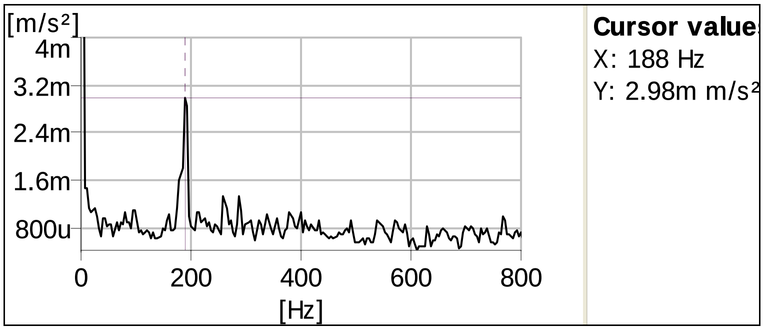


Fig 3.4.3.2a Pulse output showing first frequency output

Second Frequency measurement - 564 Hz

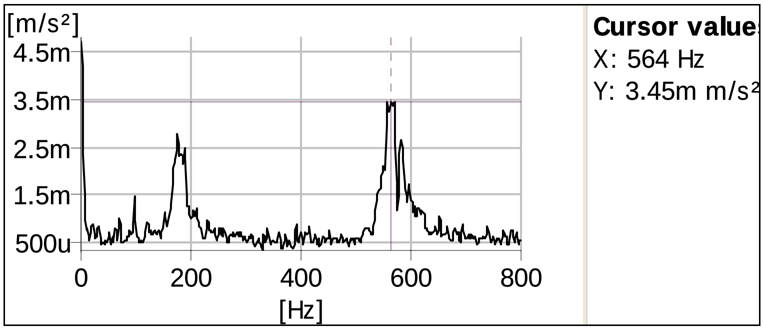


Fig 3.4.3.2b Pulse output showing second frequency output

3.4.3.3 Specimen 3 - 12 layers

Area of specimen 1 – 12 cm X 19.5 cm

First Frequency measurement - 176 Hz

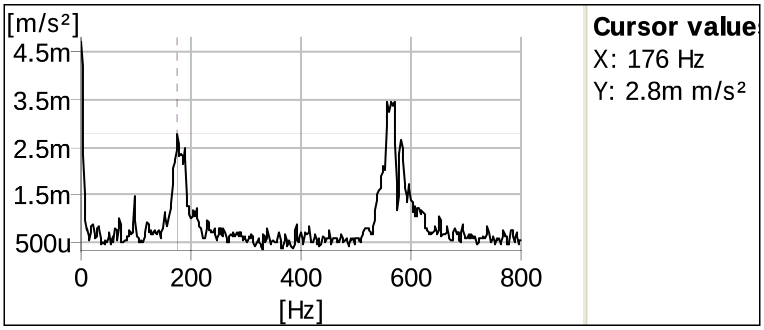


Fig 3.4.3.3a Pulse output showing first frequency output

Second Frequency measurement - 480 Hz

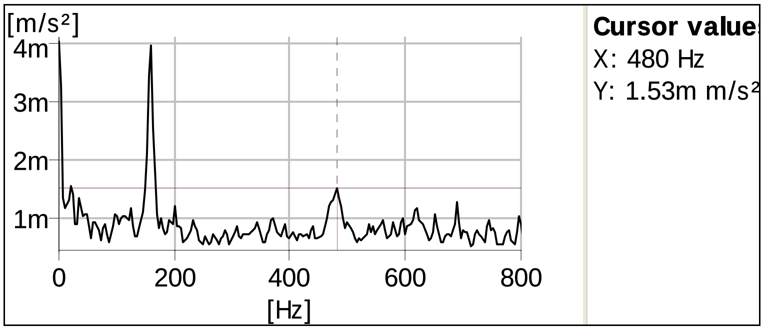


Fig 3.4.3.3b Pulse output showing second frequency output

3.4.3.4 Specimen 4 - 12 layers

Area of specimen 1 – 6 cm X 19.5 cm

First Frequency measurement – 160 Hz

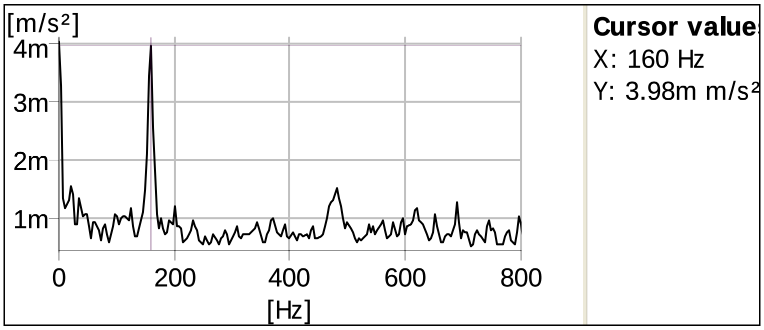


Fig 3.4.3.4a Pulse output showing first frequency output

Second Frequency measurement - 390 Hz

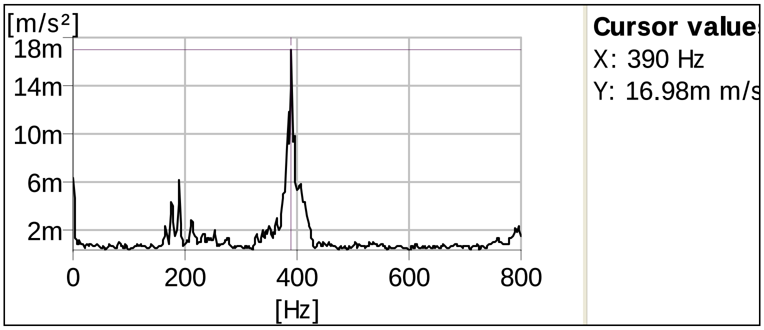


Fig 3.4.3.4b Pulse output showing second frequency output

3.4.4 Variation of Frequency – 12 layers

The frequency of vibration of the 12-layer composite is observed to decrease with the decrease in the area of the specimen. The same trend is being observed in both first and second frequencies. As the length of the composite decreases in a regular trend the frequencies are expected to decrease as the value of ‘k’ decreases.

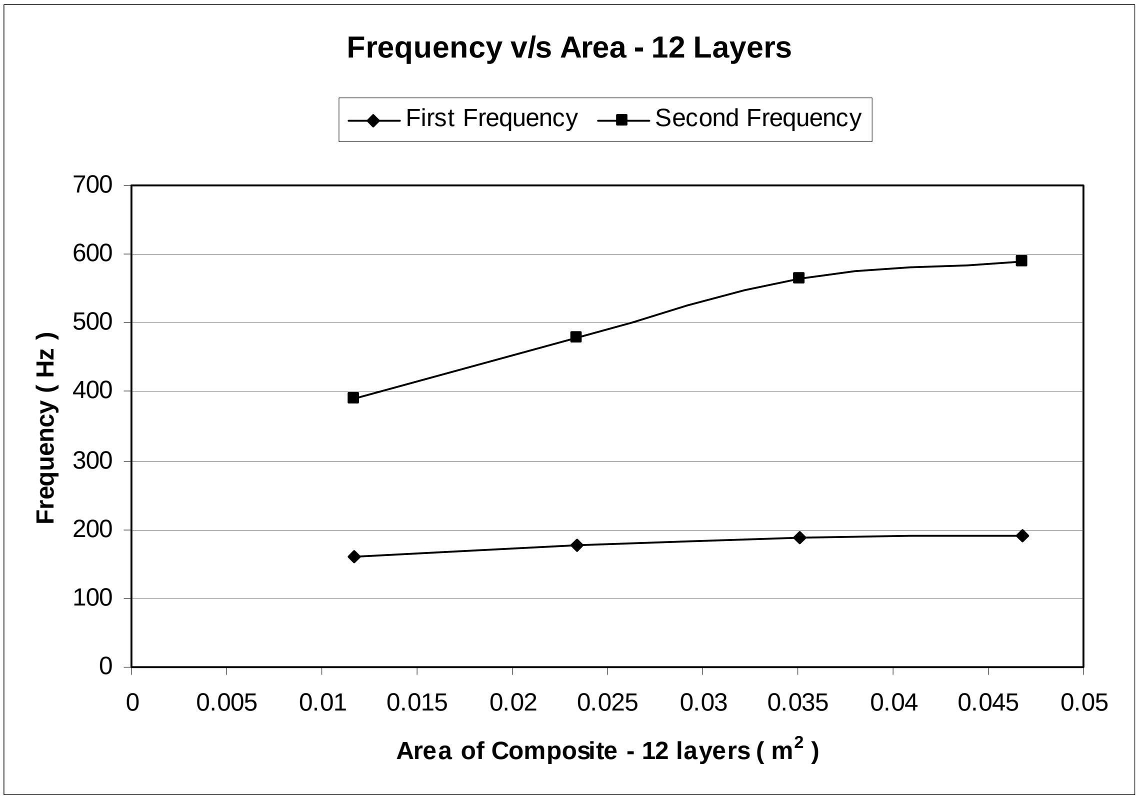


Fig 3.4.4 The variation of first frequency and second frequency with decreasing area of composite

3.5 The Pulse Report – Buckling Load of Flat Section

As mentioned above we have taken a steel flat of length 30cm. We have kept this in a setup of UTM and increased the load gradually by

- ✓ No load condition
- ✓ 0.2 ton load condition
- ✓ 0.4 ton load condition

3.5.1 *No Load Condition*

First Frequency measurement – 260 Hz

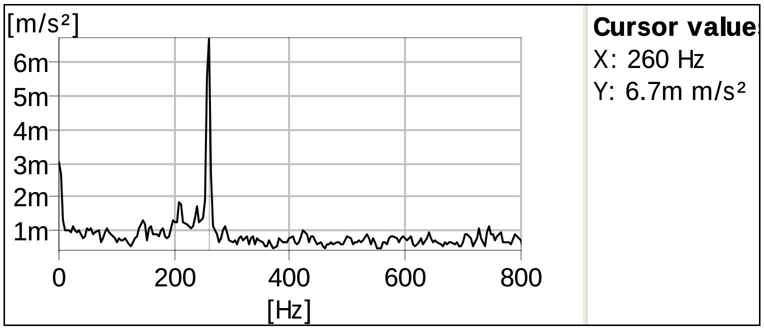


Fig 3.5.1a Pulse output showing first frequency output

3.5.2 *0.2Ton Load Condition*

First Frequency measurement – 244 Hz

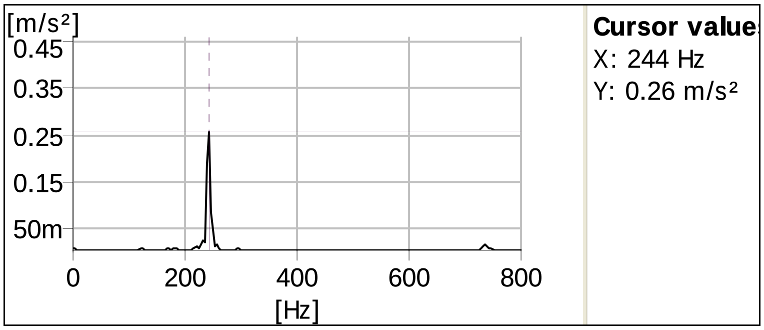


Fig 3.5.2a Pulse output showing first frequency output

Second Frequency measurement - 736 Hz

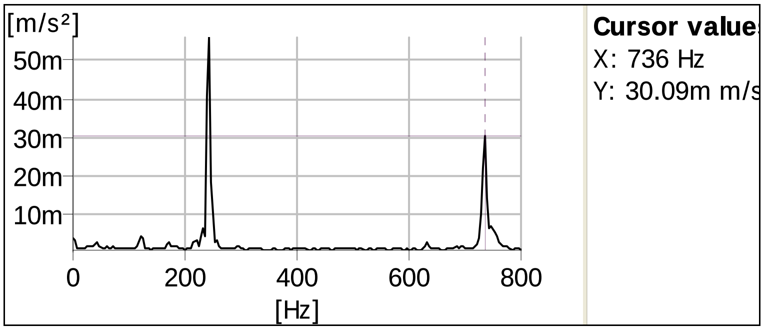


Fig 3.5.2b Pulse output showing second frequency output

3.5.3 *0.4Ton Load Condition*

First Frequency measurement – 168 Hz

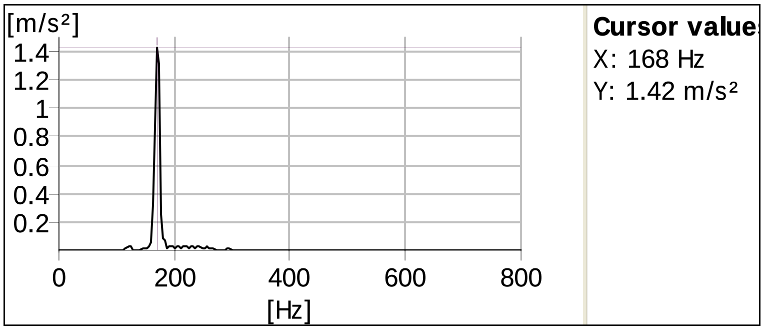


Fig 3.5.3a Pulse output showing first frequency output

Second Frequency measurement – 628 Hz

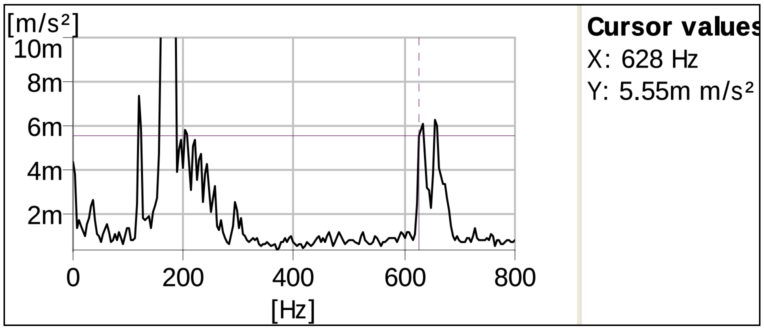


Fig 3.5.3b Pulse output showing second frequency output

3.5.4 Variation of Frequency with Increasing Axial Load

The Buckling load of given steel flat can be found out by increasing the axial load on it and checking the point where we can get zero frequency of vibration. As we observe in the graph given below the first and the second frequencies are decreasing with the increase in axial load. At a point where the graph meets the x-axis at that point the frequency becomes zero and the corresponding load is noted. From the equation of the curve of first frequency variation by extrapolating the graph we get the zero frequency load or buckling load to be 1.1739 ton.

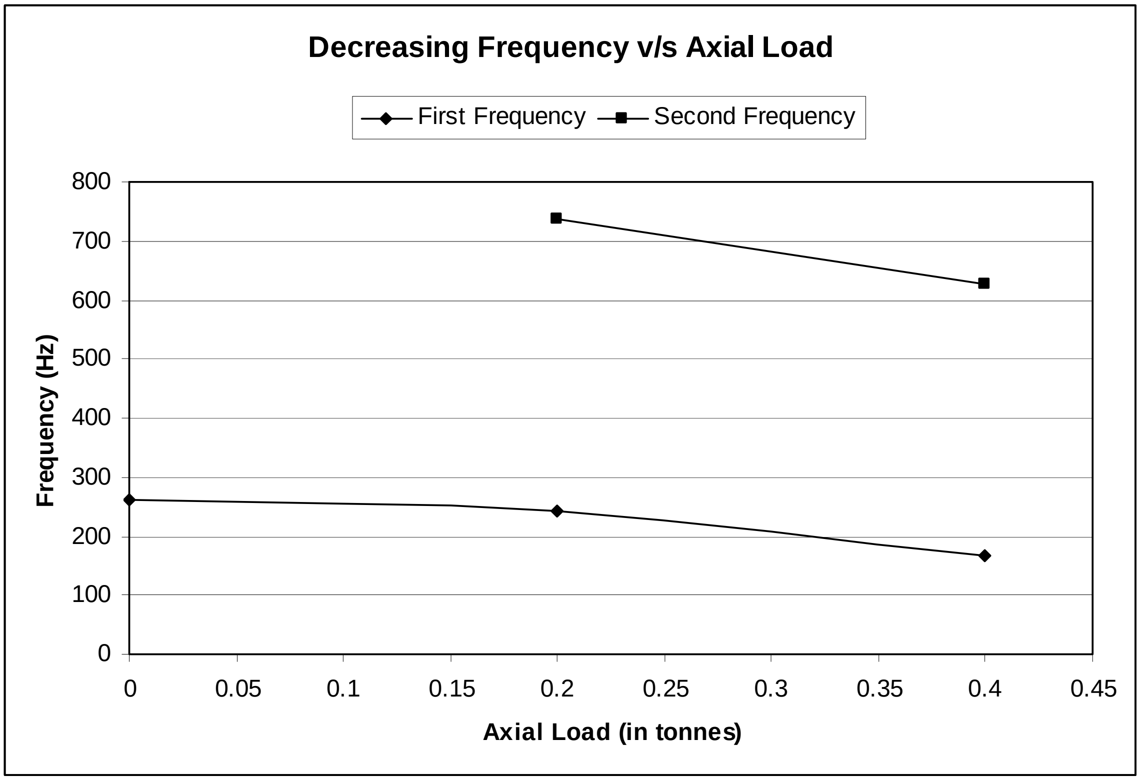


Fig 3.5.4 The Variation of frequency with increase in axial load

Chapter 4

CONCLUSION

4.1 Conclusion

The vibration analysis of a structure holds a lot of significance in its designing and performance over a period of time. The steel flats with slits will resemble any structural defects present in the structures consisting of the flats and their frequency analysis will help us understand the variations that are bound to occur in the frequency because of the defects present in them. The vibration analysis of the foundations of various machines will help us design the foundations such that their serviceability is increased. Henceforth, we suggest that the operating frequencies of the following machines should not be near the below mentioned values so as to avoid the condition of resonance.

- ✓ Los Angels Abrasion Machine – 48 Hz, 73 Hz.
- ✓ Jaw Crushing Machine - 42.2 Hz, 71.8 Hz.

The composites as mentioned earlier are used in a number of engineering applications and in many of them there is a possibility of composites subjected to a good amount of displacement and hence it is undergo vibration and the hence the study of the natural frequency is very important as we have to avoid resonance condition. The composites are used in different sizes in different applications and so we have varied the sizes and studied the frequency variations because of this factor. These frequency ranges have been discussed above. Henceforth, we have predicted the frequency ranges in which the composites are bound to fail.

The determination of buckling load by vibration analyses proves to be a powerful tool in finding out the maximum loading that steel flat can take without failure. The experiment conducted on 30 cm steel flat has given a buckling load of 1.1739 ton.

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APPENDIX

Fig 5.1 Depicting the entire PULSE setup for testing of Composites

Fig 5.2 Depicting the positioning of accelerometers on the composite sample for 16 – layers sample

Fig 5.3 Depicting the positioning of accelerometers on the steel flat sample

Fig 5.4 Depicting the yielding of the steel flat under the application of axial load