DESIGN OF CONTROLLER FOR SHAKER

A DISSERTATION

Submitted in partial fulfillment of the requirements for the award of the degree

of

MASTER OF TECHNOLOGY

in

ELECTRICAL ENGINEERING

(With Specialization in System Engineering and Operations Research)

By

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CANDIDATE'S DECLARATION

I hereby declare that the work that is being presented in this dissertation report entitled **"DESIGN OF CONTROLLER FOR SHAKER"** submitted in partial fulfillment of the requirements for the award of the degree of **Master of Technology in Electrical Engineering** with specialization in **System Engineering and Operations Research,** submitted in the **Department of Electrical Engineering, Indian Institute of Technology Roorkee,** Roorkee, is an authentic record of my own work carried out, during the period May 2007 to June 2008 under the guidance of **Prof. Hari Om Gupta** and **Prof. Pramod Agarwal,** Department of Electrical Engineering, Indian Institute of Technology Roorkee, Roorkee.

I have not submitted the matter embodied in this dissertation report for any other degree.

Date: 30/06/08
Place: Roorkee

(V. S. SRIRAM)

CERTIFICATE

This is to certify that the above statement made by the candidate is true to the best of our knowledge.

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ABSTRACT

This work presents the main results of a novel acceleration controller designed for acceleration control of an electrodynamic shaker. Vibration testing is performed to provide the test item vibration environments expected in its shipment and application environment. It is important and mandatory to do vibration testing for testing the mechanical reliability of articles. It is equally important that the test specifications are properly assessed; else it can lead to damage of the test item.

In this thesis work a linear power amplifier fed electrodynamic shaker is utilized to perform vibration testing. For this work the mathematical model of the shaker along with the power amplifier is obtained. A feedback loop is augmented along with the feed forward loop to control the acceleration levels of the shaker and also to smoothen out the shaker output waveform, making it more sinusoidal.

The tests are conducted in open-loop and in closed-loop for both no-load and loaded conditions. The results so obtained are compared and it is observed that better control is achieved in closed-loop system. Smoothening of waveform due to reduction of harmonic content and lesser percentage of steady state error in the magnitude is observed in the case of closed-loop control. The reasons for errors and limitations of the employed method are discussed and some suggestions are made to improve the overall performance of the system.

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I would like to dedicate this research work to my family.

Acknowledge Him in all Thy ways and He shall direct Thy paths.

(V. S. SRIRAM)

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CHAPTER 1

INTRODUCTION

1.1. GENERAL

Vibration testing is generally performed by applying a vibratory excitement to a test object and monitoring the structural integrity and performance of the intended function of the object. The technology associated with vibration testing started to develop rapidly after World War II and had been successfully applied to a wide range of products ranging from consumer electronics to structural systems.

Till recently most of the signal processing required for vibration testing was performed through analog methods. The obtained signal is converted into an electrical signal and passed through a series of electrical and electronic circuits to achieve the desired processing. Today's complex programs require fast and accurate processing of a large number of measurements. Digital processing for the analysis and processing of vibration signals began to replace their analog counterparts in the 1960's. The advantages of digital processing include the convenience and flexibility and the complexity of processing.

Vibration testing is usually achieved using shaker with a driving power amplifier. The item to be tested is fixed on the moving table to induce vibrations to it. The capability of a test item to withstand a predefined test environment depends on the dynamic response of the item and the functional operability variables [31]. Analysis of the response signals will aid in identifying the impending faults in the test item. The sensor output is particularly useful in analyzing the vibratory dynamics of the shaker, feedback control of shaker, frequency band equalization in real-time of the excitation signals, and synthesizing of future signals.

A complete knowledge in which the test item operates is never available to the test programmer. When performing vibration test either a deterministic or a random test environment could be chosen. In this dissertation work a deterministic signal, namely sinusoidal signal is chosen as the test environment. Sinusoidal signals do not exist in nature and sinusoidal vibration tests are conducted till date partly for historical reasons.

There are desirable and undesirable conditions in a mechanical vibration. The undesirable mechanical vibrations are those which cause failure of products or human discomfort. Clearly the idea is to suppress these vibrations which can be achieved either by isolation, design modification or control.

Isolation involves suppressing the vibrations before they undesirable effect on the system. It is always not feasible as the occurrences are random in nature. Design modification involves redesigning the whole system so that the vibration levels do not induce any undesirable effect on the system. This is a costly and tedious affair. In the control scheme the vibrations are sensed through sensor and what forces need to be acted on the system to counteract and suppress the vibrations are determined by a controller and the corresponding forces are applied to the system through actuators to the system.

1.2. LITERATURE SURVEY

Vibration testing as a formalized field began developing in the 1950's after the MIL (Military Standards) personnel's were given the task of simulating environments faced by the hardware in their operating conditions [30]. N. F. Hunter [8] has reviewed the state of art vibration testing in a general context, giving a deep insight into the process outline of the procedures to be followed.

A lot of authors have discussed about the effects of vibration and the importance of vibration testing on the performance and reliability of specific products like solder joints, electronics and machine conditions. Dennis H. Shreve [24] has given a useful insight into the vibration as an indicator of machine condition considering various factors of vibration.

Reliability of electronic devices is a crucial scope of interest to the electronic manufacturing industry. S. R. Wennberg et al. [1] has proposed vibration tests so as to minimize the cost and time involved in it along with relating the test failures to the actual field failures.

Pramod Malatkar et al. [19] has discussed in detail the necessity of proper modeling and test procedures of solder joint reliability when subjected to shock and vibration. Timothy P. Rothman et al. [4] had discussed about the physics-of-failure models to extract acceleration information under accelerated test.

Michael Kearney et al. [12] had discussed about various methods for fast and effective reliability enhancement testing of electronic equipment in the light of exposure to very harsh environments like step-stress, swept-sine among others.

Przemyslaw Matkowski et al. [15] has discussed about vibration tests as an accelerated test to evaluate the fatigue life of interconnections in electronics assembly. Based on the practical work a suitable vibration tests for such items is also proposed. Frank Fan Wang [9] has described how to relate the results of a sinusoidal to random vibration testing in electronics packaging. The method is based on the equivalent damage theory of the electronic packaging. John H. L. Pang et al. [14] has explained vibration fatigue analysis of solder joints using FEA methods.

Before implementation of any control scheme in the system, understanding the characteristics and response of the shaker is absolute necessary and important. Chulho Yang [16] has discussed about parametric and non-parametric system identification of non-linear mechanical systems. R Fair et al.[2] has discussed regarding the electrodynamic equivalent model of the shaker. A comprehensive representation of the mechanical and electrical equivalent of the shaker has been dealt with. The shaker has been analyzed for its various properties like heat dissipation capability, variation in the impedance levels with frequency. A design procedure has also been suggested. H.M. Macdonald et al. [3] regarding the modeling of the shaker over a wide range of frequencies and reasons were suggested for the double resonances present in the system. Then a control scheme was also proposed to get a desired response from the shaker. An excellent discussion on the Real-time Control Systems can be found in A. Gambier [25].

In vibration testing, a shaker is controlled to produce a specified vibration spectral density. But a shaker along with a fixture bolted to it will seriously distort the drive signal while producing a corresponding vibration signal. Many control schemes have been proposed, tested and results published in literature. The basic difference in the configuration of the system available in literature and on the system on which the results have been obtained in this work is that the systems in literature is based on inverter-fed-electrodynamic shaker, while here a linear power amplifier fed electrodynamic shaker is used. Linear power amplifiers suffer from the disadvantages of larger size and weight, lower conversion efficiency [6].

Yousuke Hirowatari et al. [23] has discussed about vibration control using a tracking filter. The system for consideration is not a shaker but an electro-hydraulic servo system. But the use of tracking filter provides a lot of advantages for vibration control like exact tracking of the magnitude and phase of the fundamental frequency waveform from a noisy signal.

Kuang-Chyi Lee [5] has proposed a controller for shaker based on fuzzy techniques to keep the vibration spectrum within the tolerable limits. Jingang Han et al. [16] have proposed a high performance switched mode power amplifier along with control schemes for driving a shaker. Leandro Della flora et al. [20,22] has proposed robust and adaptive acceleration controllers. M. Stefanello et al. [13] has discussed about both sinusoidal and random based on Robust Model Reference Control (MRC) law with a Repetitive Controller (RP). Yasuhiro Uchiyama et al. [18] proposed methods for acceleration and displacement control of the shaker using conventional controllers and designing them through μ-synthesis for extended frequency band. C. M. Liaw et.

al.[6,7,11] have discussed a lot in literature about control of shaker. They have proposed switching mode power amplifiers for shakers and also control schemes for sinusoidal and random tests using robust and current control.

1.3. OBJECTIVE OF THE WORK

This objective of this work is to study the frequency response of the shaker. A sinusoidal acceleration controller for proper tracking of the reference waveform is proposed and the proposed controller is implemented on a practical system and the results in open-loop and closed-loop for both no-load and loaded conditions is to be verified for better performance in closed-loop.

1.4. ORGANIZATION OF THE REPORT

Chapter 2 gives a comprehensive insight the art of Vibration Testing practice and its advancement over the years.

Chapter 3 gives the theory of operation and mathematical modeling of the electrodynamic shaker.

Chapter 4 uses a practical shaker system to obtain its mathematical model. The experimentation methods are explained and the results obtained are compared with the theoretical model

Chapter 5 proposes a novel acceleration controller for acceleration control of the shaker. Results were obtained both using simulation and practical setup. The results from the practical setup are obtained for both no-load and loaded conditions. Comparisons were drawn up and suggestions are given for further improvement.

CHAPTER 2

VIBRATION TESTING

2.1. INTRODUCTION TO VIBRATION TESTING

Vibration Testing is performed to provide the test item vibration environments expected in its shipment and application environment. There are many pitfalls to avoid, especially for the entry level engineer, when planning and conducting tests. Today's test equipment enables engineers and technicians, with limited testing experience, to conduct tests relatively quickly and efficiently. However, while computers and menu driven software have enabled testing laboratories to become increasingly more efficient, as far as testing productivity is concerned, a void has been created. This void is the lack of in-depth knowledge of testing that exists in many laboratories.

It is one thing to go through the motions and to setup and program a test. However, it is another thing to understand the rationale for conducting the testing and the details of conducting the test. It is still another thing to be able to interpret the data and to extract meaningful conclusions from the data. Consideration must be given to numerous variables, in order to conduct an effective test and to obtain the desired results. A good test begins at the planning stage. Understanding the test objective, and the selection of appropriate test procedures and requirements, are considerations that should be given as much emphasis as the actual testing itself. Test tailoring, when appropriate, and when permitted, should be employed. Certainly, test tailoring should not be thrust upon a neophyte test engineer. Test tailoring is something that should be left to knowledgeable, experienced testing experts.

Once a test has been properly planned, it is important to prepare for the testing. Even before samples for test are received, a great deal of work must be accomplished. It is important that everyone involved understand the test objectives. It is also necessary to determine what data is to be gathered. Data traceability is one aspect of testing that should not be overlooked during the planning and preparation stages of testing. All samples should be uniquely numbered and all data should be traceable to those samples. Data should also be traceable to a particular step or condition in the testing program, during which the data is to be taken. In addition to traceability the baselining of samples is something that should not be overlooked. Baselining of samples involves, at a minimum, a detailed visual exam, to identify the starting condition of the test samples, prior to the point at which they are placed into test. Baselining of samples can also involve parametric, functional checks of the test sample's operation and performance. By knowing how a sample performs "Out of the box," prior to the application of adverse environments, the test engineer can have data that can be used for comparison purposes later on in a test program.

Issues such as accelerometer location, fixturing, test levels, monitoring requirements and others must be considered, when developing and conducting a vibration test. Failure to specify and understand these issues could result in an improper test being conducted. Vibration testing is especially sensitive to variability from one test to the next, because of issues like accelerometer placement and fixturing. By varying these parameters, a completely different test could be conducted, even if the same test levels are to be applied. It is absolutely essential that the test engineer understand the effect that these parameters can have of the outcome of a vibration test.

Conducting a good laboratory test takes more than button pushing to spit out a number. It takes engineering talent and experience. It also takes effective planning, diligent setup and data gathering, and comprehensive reporting. It also requires that a system of procedures and policies be in place and be followed, so that a consistent and uniform approach to testing occurs. In addition to engineering talent, experience, procedures, and policies; the laboratory needs one other key element for successful testing, and that is common sense. Knowing what your test data is telling you, documenting it, and reacting to it appropriately are the key common sense elements to good testing. Engineering talent, policies and procedures are needed to facilitate the use of common sense [19].

2.2. HISTORY OF VIBRATION TESTING

Vibration testing is something which has been practiced since ancient times in an intuitive and practically oriented manner. Shipbuilders, for example, considered the repetitive impacts of wind and waves when designing the wooden vessels. Primitive archers intuitively designed their arrows to withstand the impact of encountering bone.

Formalized vibration testing began during and following World War II. The main reason for vibration testing to be developed was the nature in which various physical systems failed. The failure mode was related to the repetitive loads that appeared in the system in which case the response of the system was different to that under a static load [8].

2.3. QUANTIFYING VIBRATION

2.3.1. Definition of Vibration

Vibration is one possible motion in a mechanical system, an oscillating motion about some reference or equilibrium position. The motion can consists of a single component occurring at a single frequency, as with a tuning fork, or of several components occurring at different frequencies simultaneously, as for example, with the piston motion of an internal combustion engine.

Vibration signals in practice usually consist of many frequencies occurring simultaneously so it is not possible to make out its frequency components just by looking at its amplitude-time pattern. To measure and control vibratory motion an understanding of its basic nature is necessary [30].

2.3.2. Sources of Vibration

Vibration can be generated by almost any motion or mechanical force, including, but not limited to the combustion and rotation of the automobile engine and drive train, irregularities in the road surface, reaction forces of rockets, turbulent gas or liquid flow in pipes and ducts, also by acoustical energy from rockets and the other engine exhausts, aerodynamic flow over the automobile, over aircraft, missiles, buildings, bridges, electrical power lines, through wind musical instruments or by gunfire. Vibratory motion can be helpful or detrimental.

It also occurs because of the dynamic effects of manufacturing tolerances, clearances, rolling and rubbing contact between machine parts and out-of-balance forces in rotating and reciprocating members. Often, small insignificant vibrations can excite the resonant frequencies of some other structural parts and be amplified into major vibration and noise sources.

Sometimes though, mechanical vibration performs a useful job. For example, vibration is generated intentionally in component feeders, concrete compactors, ultrasonic cleaning baths, rock drills and pile drivers [30].

2.3.3. Effects of Vibration

Vibrations results in dynamic deflections of and within the material. The dynamic deflections may cause or contribute to structural fatigue, stress and mechanical wear of structures, assemblies and parts. Some of the vibration induced failures are:

- Loose fasteners/components
- Intermittent electrical contacts
- Electrical short circuit
- Deformed seals

- Optical or mechanical misalignment
- Cracked and/or broken structures
- Excessive electrical noise
- Loose wiring etc [32]

2.3.4. Classification of Vibration

Vibration can be classified as

- continuous and cyclic (such as sinusoidal and complex vibration)
- continuous and non-cyclic (such as random vibration)

The most uncertain part of a vibration test is the simulation of the test input. For example, the operating environment of a product such as a automobile is not deterministic and will depend on many random factors. Consequently, it is not possible to generate a single test signal that can completely represent various operating conditions. When performing the test either a random or deterministic signal could be used to meet the test requirement. The test signal applied is known as the Test Environment.

Based on the vibration testing specifications the test environment should be developed to have the required characteristics as explained below:

1. Intensity

The vibration amplitude is the characteristic which describes the severity of the vibration, and can be quantified in several ways.

The peak-to-peak value is indicates the maximum excursion of the wave, a useful quantity where, for example, the vibratory displacement of a machine part is critical for maximum stress or mechanical clearance considerations. The peak value is particularly valuable for indicating the level of short duration shocks. Peak values only indicate what maximum level has occurred, no account is taken of the time history of the wave.

The rectified average value, on the other hand, does take the time history of the wave into account, but is considered of limited practical interest because it has no direct relationship with any useful physical quantity.

The RMS value is the most relevant measure of amplitude because it both takes the time history of the wave into account and gives an amplitude value which is directly related to the energy content, and therefore the destructive abilities of the vibration.

2. Frequency content

The frequency content in the vibration affects the severity of the vibration in the sense that it can influence to resonate the object and, thus leading to its failure.

In a system when the forcing frequency nears the natural frequency the amplitude of the vibration can get extremely high. This phenomenon is called resonance and subsequently the natural frequency of a system is often referred to as the resonant frequency.

If resonance occurs in a mechanical system it can be very harmful-- leading to eventual failure of the system. Consequently one of the major reasons for vibration analysis is to predict when resonance may occur depending on the vibration content and to determine what steps to take to prevent it from occurring.

3. Decay rate

The decay rate of the vibration is an important factor as it indicates how fast the excursions to the peak value occurs as peak value is an indicative of the intensity of the vibration. It is also important as the physical system can be subjected to wide range of magnitudes in a very short time. It is of significant importance for describing transient behavior.

4. Phasing

If the test environment consist of many frequencies then their relative phase differences is of importance. Every wave has a third dimension, the first two being its magnitude and frequency, and the third being its phase. It is the phase difference which finally decides the wave shape of the test environment. The phasing phenomenon refers to the dynamic interactions in the device. The response of the device will be different to different components of the test environment. Generally phase analysis comes into picture in case of excessive vibration magnitudes.

In vibration testing, the excitation input (test environment) can be represented in several ways. The common representations are

- Time signal
- Response spectrum
- Fourier spectrum
- Power Spectral Density (PSD) function [31, 32].

2.4. REQUIREMENTS OF VIBRATION TESTING

Vibration testing is a mandatory requirement because it involves safety and reliability-not only of products but also life. Market reputations are gained or lost because of performance.

For some manufacturers vibration testing is a customer's demand that must be fulfilled. Others undertake it to support their warranty agreements. Some manufacturers offer extended warranties on their products, especially to remain competitive. When there is no pressure to prove long-term reliability, manufacturers are tempted to regard vibration testing as an unnecessary cost. But what they don't realize is that the cost of vibration testing is negligible compared to the cost of failures and a permanent loss of credibility in the market.

The capability of the test object to withstand a predefined environment is evaluated by monitoring the dynamic responses (acceleration, velocity, displacement, strain etc.) and functional operability variables (temperature, pressure, voltage, current).The analysis of the response will aid in detecting existing faults or impending faults in various components of the test equipment.

Incorporating vibration testing during the development of the product will ensure:

- Fast launch of new products
- Optimization of product design
- Customer confidence and satisfaction
- Reduced manufacturing costs
- Reduced variances in manufacturing
- Maintain consistency in quality standards
- Reduce warranty costs
- Reduce after sales cost service cost
- Product survival during transportation [24]

2.5. PROCESS OUTLINE OF VIBRATION TESTING

Vibration Testing is usually performed by applying a vibratory excitation to a test object and monitoring its structural integrity and performance of the intended function of the object. It means that the Device Under Test (DUT) is subjected to a predefined set of vibrations which the DUT can possibly undergo in its operational life in a laboratory environment. The vibrations that the DUT is exposed may be single type of vibration or a superimposed one like e.g. sine, random, sine-on-random (SoR) wave, and random-on-random (RoR) wave.

As far as possible most of the vibration testing should be done during the development of the new product, because that helps to identify any vibration weakness while the design changes are easy to incorporate.

The goals of vibration testing have changed and expanded with time to include:

- Ensuring unit survival in the expected field environment.
- Qualifying an analytical model of the test unit in such a manner that the analytical model can both interpolate response to environments reasonably related to the test environment, and extrapolates response to environment outside of the test envelope.

The process is outlined using a flowchart in fig.2.1. Each step of the process is also discussed:

1. Define the Test Environment --- Formulate First Principles Engineering Model

One major goal of vibration testing is the simulation of field vibrations in a controlled laboratory environment. Sophisticated engineering judgments are required to select the features of the field environment which most affect the test item. Some features, if are known prior to the test won't affect the DUT are omitted. The laboratory environment does reproduce field force or acceleration levels, frequencies, and to some extent test item boundary conditions. Vibration environments are traditionally separated into random and sinusoidal. With sophisticated digital control systems embodied, swept SoR and swept RoR environments are produced in the laboratory as improved simulations of certain field environments.

Random test levels are defined by the input power spectrum and sinusoidal tests by frequency bounds, sweep rate and peak g level. Test levels are based on "enveloping" the field measurements. In the case of the Power Spectrum enveloping, smoothens the measured field spectrum while simultaneously rounding toward higher values. This procedure is inherently conservative, since it rounds doubtful or noisy values upward. Problems occur if the rounding process is applied to deterministic values, for example, when low response levels are rounded upward in the vicinity of notches in the response transfer function. Since test values are rounded upward, testing a too low a level is not typically a problem. Testing at too high a level can, and often does, occur, and is referred to as overtesting.

An analytical model should be constructed prior to test definition to help define the test parameters. Prior to testing, the analytical model exercises the test structure in the field environment modifies the boundary conditions, and exercises the structure in the laboratory, assessing the effects of the changed boundary conditions and qualifying the fixture design.

Simultaneous excitation of several axes of a test item is a challenging problem. Controlled simultaneous excitation of multiple axes of vibration requires sophisticated mechanical coupling between several independently driven vibration machines. Further, the excitation signal for each vibration machine must, in general, take into account acceleration and/or force coupling between the vibration machines.

2. Design and Fabricate Fixtures---Analyze Fixtures in light of the Environment

The ideal vibration fixture transmits only the desired forces to the test item, while simultaneously simulating the field boundary conditions. Practical vibration fixtures are quite limited when compared to the ideal. Typical vibration fixtures are rigid and massive. This produces excellent transmission of forces to the test item, but generally does a poor job of simulating the more flexible boundaries present in the field. Fixtures are usually monolithic metal structures. Such structures are by nature lightly damped. The combined characteristics of light damping and massive rigid design mean that it is usually possible for powerful vibration machines to transmit sufficient force to the test item to achieve desired acceleration levels.

3. Define Test Instrumentation-Coordinate Instrumentation Locations with Analytical Model

The test instrumentation consists of four types of transducers: accelerometers, strain gauges, force transducers, and displacement gauges. Piezoelectric accelerometers are the most widely used. With increasing frequency, velocity field measurements are made using laser Doppler velocometers. These full field measurements are very useful for panel-like or 2 dimensional surfaces. Scanning vibrometers provide excellent spatial resolution for modal tests when sinusoidal excitation is used. Point velocity measurements are also useful during full level vibration tests. These laser devices are very useful, but have not supplanted accelerometers for several reasons, including the necessity of visual access to the surface and the relatively high cost of a scanning laser system.

In many cases intuition serves as the basis for selecting both the number of transducers and the response locations. A better method combines intuition with information about the system. For example, transducer locations may be selected for model validation, or for observations at locations of maximum response.

4. Select/Fabricate the Test Unit

The test unit is a sample item from a random population just as the test environment is a sample from a random population of possible environments. As a first approximation, test units may be assumed identical, but realistically test units vary. In many cases the variance is unknown, and in most cases the variance occurs in a high dimensional space. Sometimes multiple units are available, but usually a single unit is tested.

Maximizing information implies selection of a test item with a particular set of characteristics. Items of information include unit survival, boundaries to failure, and, components most likely to fail. The test should clarify critical parameters. Unfortunately, just as the test item population exists in a space of many dimensions, so does the information gained from the test. Test planning attempts

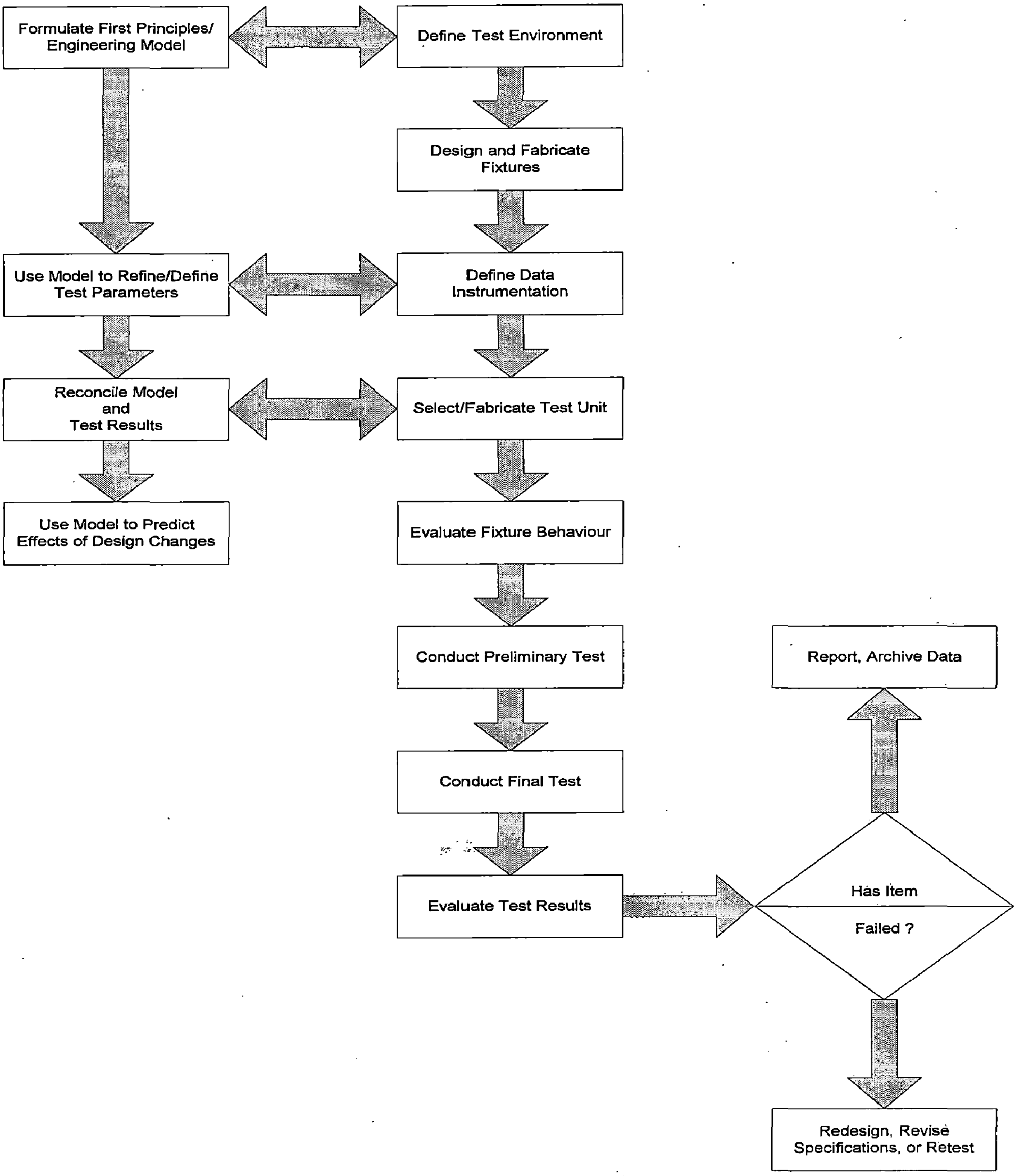


Fig.2.1 Process Outline of Vibration Testing (Source: [8])

to maximize the information obtained from a test, though often in the absence of a good analytical model.

5. Evaluate the fixture---Reconcile the Fixture and the Analytical Fixture Model

Most vibration tests take place in three stages. The first stage of is fixture evaluation. The second stage uses a mockup test unit. The third stage tests the actual unit. The first of these steps, fixture evaluation, exercises the bare or unloaded fixture under test conditions. The fixture transmits forces from the vibration machine to the test unit The fixture also serves as a boundary condition at the fixture-test unit interface. Both the test and field interface forces are usually non-uniform at frequencies above the first interface resonance. The pattern of the field interface forces typically does not match the force pattern in the laboratory environment. Vibration control systems are used to duplicate the average forces or accelerations observed in the field. Non phase sensitive spectral averages, which combine spectral magnitudes, are superior to phase sensitive time averages. At some frequencies phase sensitive averages allow cancellation of equal amplitudes of opposite phase. An average based on spectral magnitudes avoids this problem, which otherwise leads to very high input forces when a test unit rocks about an axis through the plane of the input accelerometers.

Reconciliation of the fixture model and the data from the fixture evaluation qualifies the analytical fixture model. This qualification includes a realistic, test based, fixture damping value and adjustment of the model frequencies and mode shapes to match the test values.

6. Conduct Preliminary Test---Reconcile Model and Test Results

A preliminary vibration test is conducted at levels less than the desired test level. Traditionally low level tests are conducted from –18 Db (1/8) level to –3 Db (0.707) level.

Low level testing exercises the fixture-test item interface. Vibration controllers operate by exercising the test system with low level vibration (usually random), determining the system transfer function, modifying the drive spectrum by the inverse of the transfer function, and repeating the process. Once control at a given level is achieved, the test level is increased. This process exposes the test item to a series of environments, whose total duration may exceed that of the field environments. The slow increase in test level allows the control system to adjust to a transfer function that change with test level as a result of system nonlinearities. Usually more severe nonlinear behavior occurs at the higher test levels.

Some nonlinearity is handled by the vibration controller, in the sense that the control input does not exceed the desired power spectrum. The drive signal, based on the inverse of the best transfer function estimate, adjusts for changes in resonant frequencies with increasing test level. Harmonic generation is another problem. Harmonics mean that the drive at a frequency f_o, produces a response at some harmonic nf_o. As the test level increases, the harmonic response at nf_o increases. Eventually the nf_o response exceeds the desired response level. The harmonic response is not truly random since it is related to the response at f_o.

7. Conduct Final Test---Reconcile Model and Test Result

The test is conducted to levels based on the field environment. Test duration may be realistic or based on some compromise. For some simulations, like missile flight, the field environment duration is equal to that of the test. For much longer duration tests, like truck transport, test duration is less than the duration of the field environment. A compromise is reached based on increasing test amplitude and decreasing test duration. Any such test depends on engineering judgment, perhaps rationalized by some fatigue related criteria.

8. Evaluate Test Results---Validate Analytical Model

Component failure may be a realistic reflection of a field failure or an artifact of the testing process. In practice, failure of a test item in a vibration environment leads to serious investigations of the possibility that some spectral component was overemphasized.

The behavior of the test unit during the full level test is the final validation of the test fixture and the analytical model. Reconciliation of the test item response and the corresponding analytical model response is the final qualification of the analytical model.

A major question regarding test-model correlation deals with the probabilistic nature of the test item properties. The probabilistic nature of the test item selection and the model selection should be considered.

In most cases a direct comparison of the response time series from the test and analytical model is inappropriate. Time series values are generally very sensitive to small perturbations. Some less sensitive measure is required for the initial comparison. This measure emphasizes important data features and condenses the time series values into a readily understandable format. Condensation reduces the effect of nondeterministic (from the measurement standpoint) events by averaging them out of the estimation process as noise.

Some suggestions for better test-model correlation include:

- Better definition of the test item assembly and its specific place in the random population of potential test items.
- Improved accuracy of test measurements in concert with better measurement locations.

- Improved understanding of the fundamental physics underlying the vibration behavior of structures, especially mechanical joints and damping.

9. Evaluate Test Result

Despite the increased precision, faith in models, and multitude of measurements, the fundamental fact of unit survival is important. The results of vibration tests fall into three categories:

1. The test unit survives with no obvious damage.
2. The test unit is damaged. The test specification is revised (usually based on either better field data) and the unit passes the revised test.
3. The unit fails, as do attempt to rationalize changes in the specification, and the unit is redesigned.

These cases, curiously enough, are in order of probability. It seems that most items survive vibration, but a few fail. Of those few that fail, revision of the specification eliminates most failures. Finally a few items still fail, and these are redesigned.

10. Final Model Qualification

Final qualification of the analytical model depends on successful prediction of an environment outside the bounds of the data used to reconcile the model to the test data. In the most straightforward qualification, some of the test data is used to qualify the model and the model is used to predict the remainder. This is, for example, a common procedure in statistical model validation. A more difficult challenge is the prediction of unit responses outside of the envelope of the test. This is much more challenging and likely requires full awareness of the probabilistic nature of the test item and of the environment [8, 32].

2.6. TEST ENVIRONMENT

Vibration Testing Environment is broadly classified into Sinusoidal, Random and Complex vibration.

i) Sinusoidal Vibration

Pure sinusoidal vibration is composed of a single frequency. Sinusoidal vibration is analogous to a laser beam, where the light wave is composed of a single frequency.

ii) Complex Vibration

Complex vibration is composed of specific, distinct and identifiable frequencies. It has frequencies which are discrete in nature. Complex vibrations are also periodic in nature.

iii) Random Vibration

Random vibration is composed of a multitude of frequencies. In fact, random vibration is composed of a continuous spectrum of frequencies. Random vibration is somewhat analogous to white light. White light can be passed through a prism to reveal a continuous spectrum of colors. Likewise, random vibration can be passed through a spectrum analyzer to reveal a continuous spectrum of frequencies.

A sub-classification of the schemes is Sine-on-Random (SoR) and Random-on-Random (RoR) testing.

In SoR testing a sine wave in superimposed on a random wave and this forms the test environment, while in RoR testing a random wave is superimposed on another random wave to generate the required test environment.

The traditional vibration testing has played an important role in the development of today's reliable and sophisticated electronic and electro-mechanical products. The core ideas of these methods were to define a set of specifications and create a test environment. These methods were generally in a single dimension and used only sinusoidal vibrations. If the device is still functional after the test, it is considered to have passed, but it had a drawback in the sense it could not identify the weakest link present in the device and it did not use any other test environments. Thus newer methods of testing known as HALT (Highly-Accelerated Life Testing) and HASS (Highly-Accelerated Stress Screening) have been developed. These methods employ multi-dimensional vibration and waveforms other than sinusoidal.

In HALT the goal of the test is to break the product so as to identify the weakest link in the device. After the device has failed, the weak component is upgraded. The revised product is again subjected to another round of HALT, so the product fails again. This identifies the next weakest link. By going through several iterations like this, the product can be made quite robust. With this approach, only the weak spots are identified for improvement. HALT should be performed during the product design phase to ensure basic design reliability.

In HASS testing the goal of the test unlike HALT is not to break up the product. It is an on-going screening test, performed on regular production units. The limits in HASS testing are based on the proper interpretation of HALT results. The importance of HASS testing lies in the fact that final assembly of a product is done by an industry which depends on other industries or its raw materials. Thus the final product is reliable only if all the components are reliable. Thus the best way to ensure reliability of the final product is through HASS testing.

2.7. CONCLUSION

Vibration Testing has definitely helped improve product quality over the years, but today's very high standards for product quality, requires reducing or eliminating field failures completely. Today's test methods are changing as the technical limitations that historically restricted the control and application of vibration for reliability testing disappear. If test is unavoidable, it should at least pay for itself by reducing field return costs through improved product reliability. For that reason, the correlation between a product's actual operating environment and the tests performed upon it has come under scrutiny. This has increased the use of multi-axis excitation, time replication, and HALT/HASS methods.

CHAPTER 3

THEORY AND MODELING OF ELECTRODYNAMIC SHAKER

3.1. INTRODUCTION

The process of estimating input-output dynamic system models and their parameters using measured data is referred to as system identification. The development of such mathematical models is critical for analyzing the mechanical behavior of dynamic systems especially when those systems exhibit nonlinear behaviors over a range of excitation amplitudes because many types of nonlinearity are not readily modeled using first principles. To develop an accurate parametric input-output model, it is essential to accurately determine nonlinear forms e.g. cubic stiffness, quadratic damping that are needed in the model and to subsequently identify parameters that complete those forms within the model.

The first step in developing a model of a nonlinear system is to characterize nonlinearities within the system. Characterization involves the detection, classification and location of all damping, stiffness or kinematic nonlinearities throughout the system. The characterization process utilizes input and output data in addition to any prior physical knowledge about the system. Characterization is the first step in identification because it provides prior information about a system's nonlinear structure that is needed to select an appropriate input-output model form. Once the system has been characterized, various system identification and parameter estimation techniques can be used to identify model parameters with varying degrees of accuracy.

After characterizing a system, an appropriate model form is selected and input-output experimental data is utilized to estimate unknown parameters or functions in the model. There are many different system identification techniques; however, two broad categories can be used for classifying identification schemes in general: parametric and nonparametric. Parametric models or white box models require prior knowledge of the model structure, whereas nonparametric models or black box models do not. In parametric methods, the identification procedure aims to estimate model parameters using specific algorithms for searching the parameter space to reduce the overall model error in some way. Smooth i.e. continuously differentiable mechanical and structural system nonlinearities that occur in stiffness or damping forces are generally modeled with polynomial forms.

Identification of nonpolynomial forms, such as drag force type quadratic damping, hysteretic damping, coulomb friction damping and bilinear stiffness have also been studied by many researchers. In some systems, sufficient information on the mathematical structure or class of the system is not available a priori to constrain the model in any way, so non-parametric methods can be used in these cases. For nonparametric identification, the input-output mapping is achieved using a series of functional or series of orthogonal functions. The parameter estimation problem in both parametric and nonparametric methods aims to optimally estimate the parameters and functions in an inverse process through which the deviations between experimental and theoretical measurements are minimized according to some cost function [16].

3.2. ELECTRODYNAMIC (ED) SHAKERS

Moving coil electrodynamic actuators (also called 'shakers') have become widely used in vibration testing systems as a means of applying mechanical motion to a test object. The main components of a typical system are illustrated in fig. 3.1 [3].

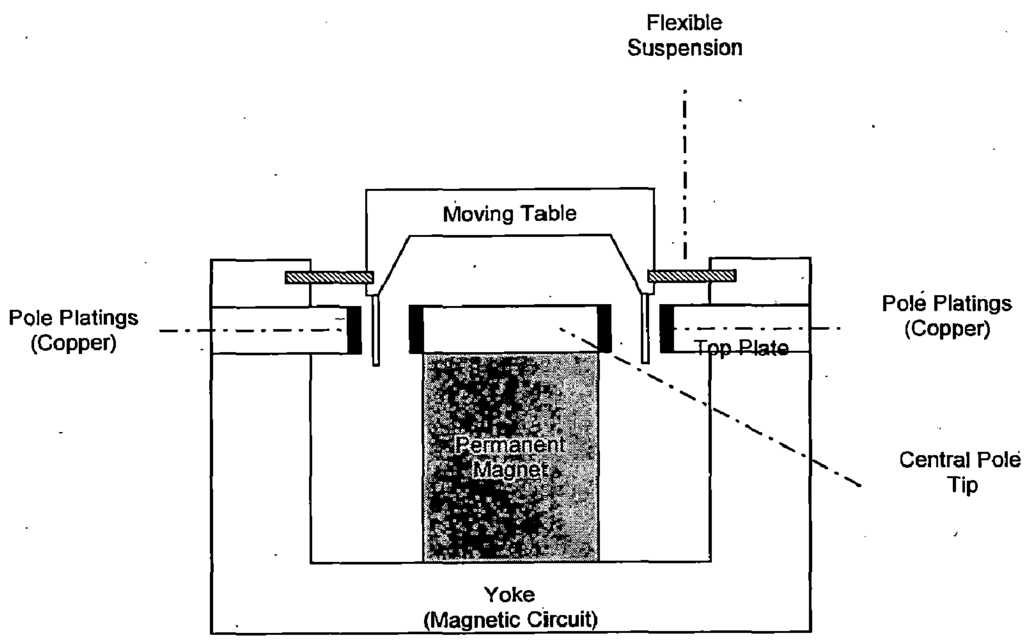


Fig. 3.1 Cross-sectional view of a shaker

ED shakers are generally involved in the reliability testing of aerospace and other electronics. In general, their frequency range is in the range of 5 or 10 to 2000Hz. ED shakers are electromechanical devices that operate on the same principle like a loudspeaker. It has two windings namely the armature and field winding.

The purpose of the field coil is to produce a steady magnetic field. Only large shakers have this field coil, where as in smaller shakers the steady magnetic field is produced by a permanent magnet.

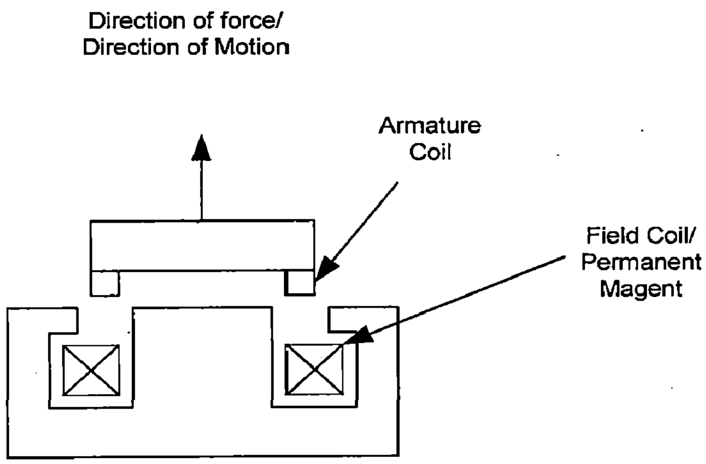


Fig. 3.2 ED Shaker cross-section

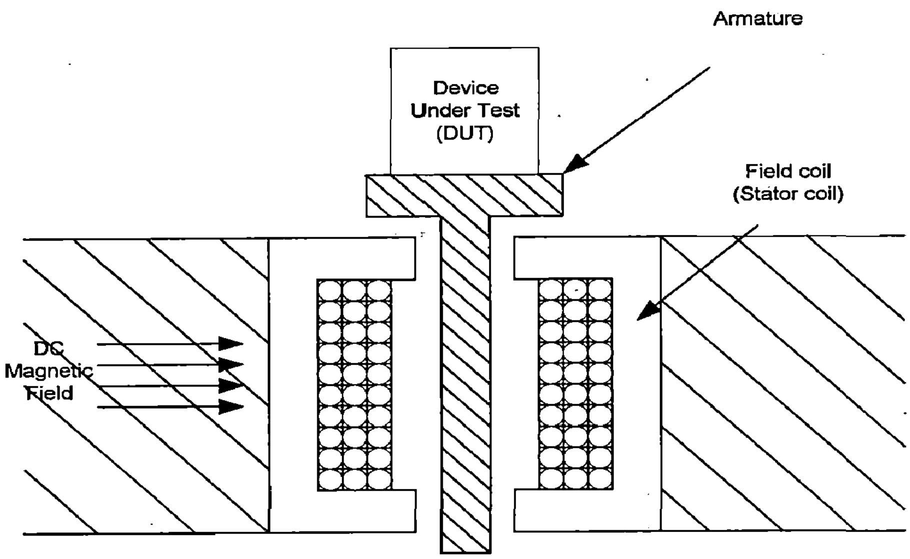


Fig. 3.3 ED Shaker cross-section showing the winding arrangement

The AC supply is given to the armature which is the shaking assembly. The DUT is fixed to the top of the armature assembly. The ideal armature has no mass and is infinitely stiff. Thus it never resonates during a vibration test. This ideal is heavily compromised in a practical design.

The force that the coil (armature) generates is proportional to the current flowing through it and this force is oscillatory in nature. The force generated is given according to Fleming's Left Hand Rule as

$$F = \oint IdL \times B \tag{3.1}$$

where I is the current flowing through the armature coil

dL is the differential length of the conductor in the magnetic field

B is the steady magnetic filed produced by the field coil

Thus this force simulates the test. The current through the coil and the motion of the armature cause a lot of heat to be generated especially at higher frequency of operation, so cooling of the shaker is necessary. The cooling arrangement may be air-cooled or water-cooled and depends on the rating of the shaker and the frequency of operation.

Water-cooled shakers are still widely used. They have an advantage over air-cooled shakers where high forces are needed or for very large payloads. In these cases, a great deal of heat is generated in the moving parts of the shaker, which is more effectively removed by a water circuit. Water-cooled shakers also are used in clean-room environments where air and dirt particles blowing around the test equipment are not desired, such as in aerospace/satellite testing applications.

It is clear from the description above that ED shakers are electromechanical in nature; therefore it should be modeled as an electromechanical device.

3.3. PARAMETRIC MODELING

Parametric modeling involves using the computer to design objects by modeling their components with real-world behaviors and attributes. The model of the shaker is to be modeled as a Single Degree of Freedom (DoF) electromechanical device. The analysis of the electrical and mechanical part are taken up separately and finally both these are combined together to obtain a final model. For obtaining a model which gives consistent results it is required to model it as a Single DoF device. A theoretical frequency response of the system is shown in fig 3.4.

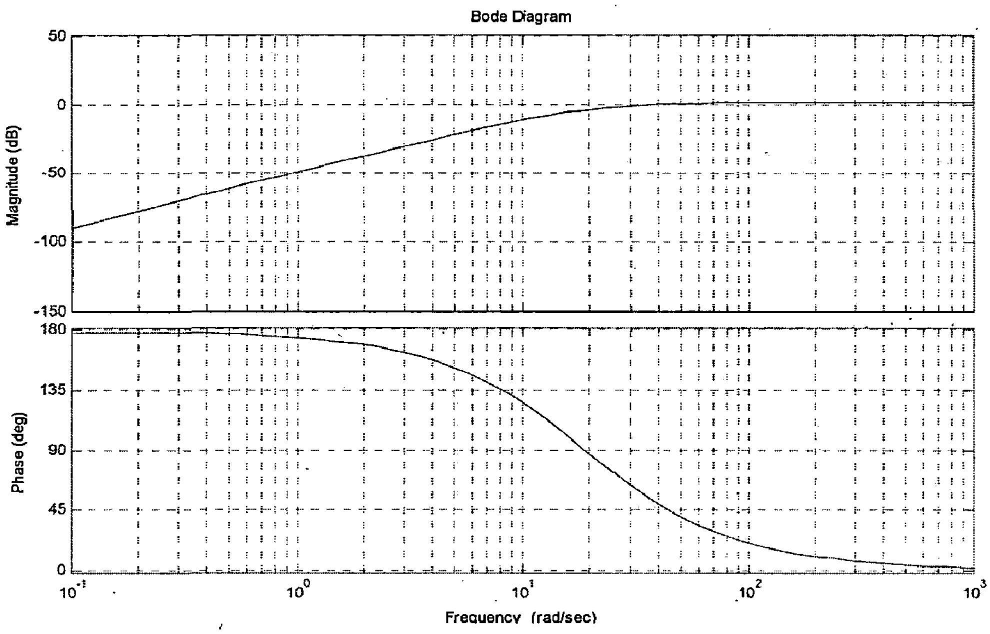


Fig. 3.4: Theoretical Bode Plot of ED Shaker

1. Mechanical equivalent

The mechanical parameters in the system involve the moving parts in the shaker, namely the armature table and the moving coil. The mechanical part is modeled as Single DoF mass, spring, dashpot combination. The parameters of the mechanical equivalent are given in fig. 3.5.

k is the spring constant of the flexible suspension.

c is the damping of the flexible suspension.

m_o is the mass of the moving table (without the test specimen).

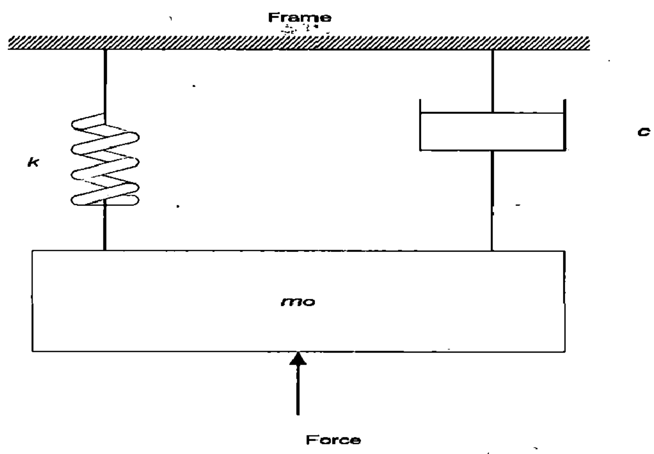


Fig. 3.5 Mechanical equivalent of the shaker

2. Electrical equivalent

The electrical parameters in the system involve representing the resistance, inductance and the back-emf induced in the coil due to the motion and effects of copper pole plating on the pole tips.

The equivalent circuit represented as impedances in fig.3.6 are now:

i) $R1$: resistance of moving coil at operating temperature

ii) $L1$: leakage inductance of moving coil due to flux set up by moving coil

(a) passing around the thin air region between the coil and the mid-plane in the coil to pole-face gap and (b) internal to the coil conductors

iii) L_m: moving coil main inductance due to flux passing across gap through plating and into the pole face iron. The radial flux density of the main flux varies axially as occurs with slot leakage flux in a conventional machine.

iv) Plating leakage inductance $L2$. This is the result of plating-current-driven flux passing through the pole face iron and the radially thin region between the plating surface and the mid-plane of the clearance between the plating and the moving coil. A very small internal plating leakage flux also occurs.

v) Plating resistance $R2$. This is simply taken as the resistance of the plating.

vi) Z_m is the equivalent impedance representation of the motional back emf generated in the winding, which is the ratio of E_b and I.

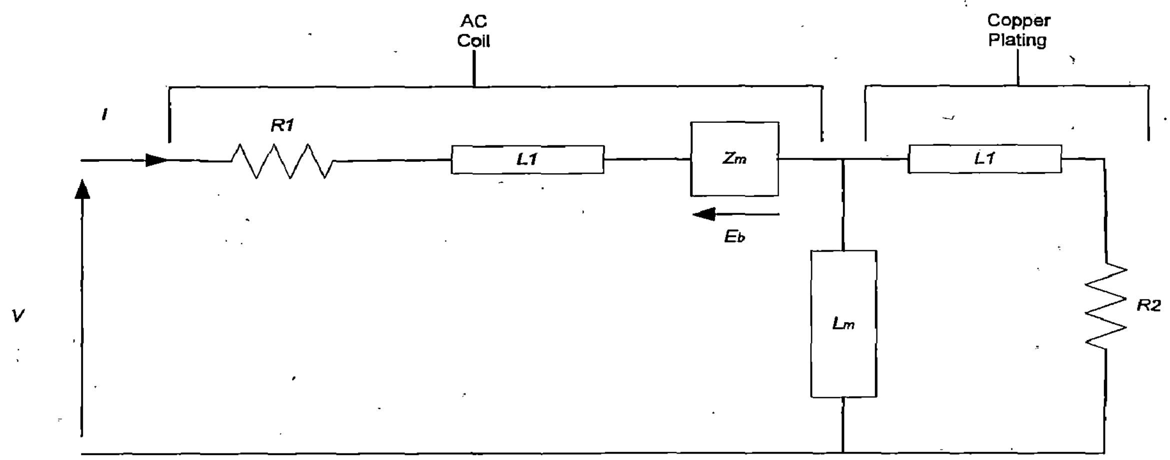


Fig. 3.6 Electrical equivalent including effect of the pole plating

$$Zm = \frac{E_b}{I} = \frac{j\omega K_f^2}{K_s - m\omega^2} \qquad (3.2)$$

This formula is obtained by modeling the moving element as a rigid body of mass m supported on an ideal suspension of constant stiffness K_s and zero mechanical damping. Solution of the resulting second order system for force $F = K_f I$ and $E_b = K_f \dot{x}$, where $\dot{x}$ is the velocity of the mass m.

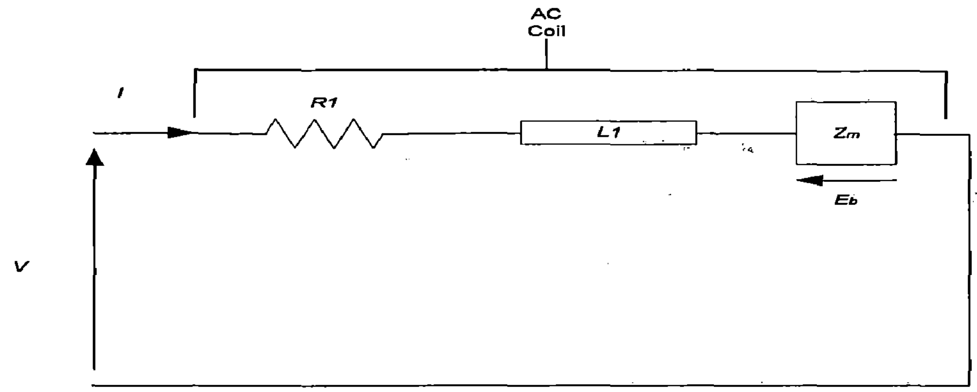


Fig. 3.7 Electrical equivalent excluding effect of the pole plating

There are practical difficulties in determining the values of L_m, $L1$ and $R2$, so a reduced model is used where all these 3 terms are neglected in the model as shown in fig.3.7. Though the electrical equivalent is represented as consisting of only the armature coil and the motional back emf induced in it, any change produced due to effect of the copper plating will be reflected as change in the current. It is only for the ease of understanding and modeling that such a equivalence is drawn up [2].

3. Integrated equivalent

The final model including the reduced electrical and mechanical equivalent is shown in fig. 3.8.

A transfer function of the model with acceleration *a(t)* of the moving table as the desired output and *I(t)* flowing through the armature coil as the input is given hence

$$\frac{a(s)}{I(s)} = \frac{Bls^2}{m_o s^2 + cs + k} \tag{3.3}$$

The above transfer function is a useful representation as generally acceleration is used as a measure of the output and since the acceleration is produced by the current, flowing through the armature windings. The relation between the voltage applied to the coils, the impedance and the current flowing through it is given by the relation below:

$$V(s) = (R + sL)I(s) + E_b(s) \tag{3.4}$$

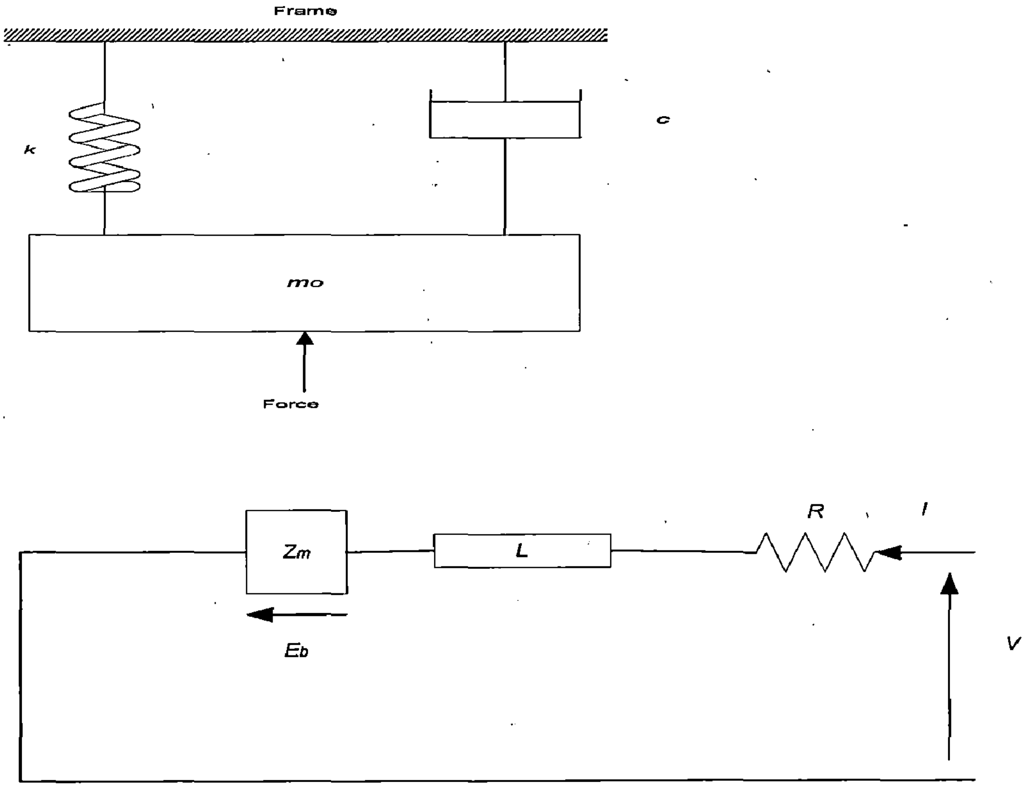


Fig. 3.8 Integrated equivalent shaker model

Both the resistance, R and the inductance, L have a non-linear frequency dependency due primarily to the action of the pole plating, which has the effect of increasing the resistance and decreasing the inductance with frequency. In addition, these electrical parameters are subject to secondary effects such as thermal variations and eddy current losses within the coil. For simulation purposes, the electrical impedance variation over the normal operating bandwidth is determined from measurements at the moving coil terminals with the armature clamped in its rest position by not applying the field supply [3].

A resonance is always indicated by the existence of complex conjugate poles. The frequency breakpoint at is a result of the resonance of the armature mass on its suspension system. The advantages, disadvantages and uses of ED shakers are discussed henceforth:

Advantages:

1. Can deliver purest sinusoidal vibration.
2. Wide range of operating frequencies.

3. Frequency and amplitude can easily be controlled by adjusting the power supply frequency and voltage.
4. Can reproduce random/complex waveforms with great fidelity.

Disadvantage:

1. The cost per kg force is high.

Table 3.1 Uses of ED Shakers

KINDS OF TESTS	ITEMS TESTED	PURPOSE OF TESTS
Functional tests	Car body mounted components, Car entertainment equipment, isolated equipments	Verify function or performance, Check of sound skip or other problem, Verify isolation is effective
Human sensibility tests	Cabin hardware, Drivers and passengers	Verify no buzz, squeak or rattle, comfort
Durability tests	Car body mounted components, Air bag inflation sensors, Engine mounts	Wear, durability, Check trigger delay time, Natural frequencies, Damping
Transportation tests	Appliances, computers, etc.	Evaluate packaging
Operational environment tests	Aerospace hardware	Proper functioning
Failure simulation tests	Varied products	Personnel safety
Calibrating sensors	Accelerometers, force sensors	Determine sensitivity

CHAPTER 4

SYSTEM DESCRIPTION AND EXPERIMENTAL MODELING

4.1. MEV-0050-VIBRATION SYSTEM [33]

MEV system is developed with indigenous design and manufacture for dynamic testing to determine the fundamental physical properties of materials and products. The series of MEV vibration system are the result of long experience in the design and manufacture of test system. It assists the user in determining the safety, reliability, durability and resonance frequencies of the product.

A vibration test system consists of a signal generator, a power amplifier and vibration exciter but in addition to this other instrumentation like accelerometer, microprocessor based vibration controller, resonance dwell unit, slip table and vertical load supports are required for sine wave testing to make the system more versatile.

Control Principle

The principle that makes all MEV test systems flexible and versatile is the use of closed loop control. It provides the exciter with a control signal which is changing in frequency between certain limits at a predetermined sweep rate. The signal is automatically adjusted to maintain a constant vibration level of all frequencies. This is achieved by using a control accelerometer, mounted on the exciter table in servo loop to provide a feedback signal by controller. The difference between the command and input signal is used to provide continuous correction signal to the vibration.

Applications

- Sweep Sine Testing
- Resonance Search Testing
- Endurance Test
- Calibration of vibration pick up

Electrodynamic Shaker system consists of the following items

- Electrodynamic Shaker MEV Series
- Power Amplifier MPA Series
- Cooling System MCS Series

4.1.1. Electrodynamic Shaker

The MEV Series of shakers are having drive armature connected rigidly to the moving platform and positioned in the magnetic field. When AC current flows in this drive coil it gives rise to a force by converting the electrical current into mechanical force which moves the platform. The shaker can operate in the frequency range from 5 Hz to 3000 Hz from either sine or random input waveform.

A modern armature suspension system is having four heavy duty rolling struts and a central linear bearing assembly which provide an armature with excellent cross-axial restraint in large shakers. The use of rolling strut design makes it easily convertible to long stroke applications required for half-sine bump tests with maintaining all dynamic qualities of the vibration exciter i.e. low distortion, high cross-axial stiffness and the ability to work with off centre loading.

A force air cooling unit is coupled with the exciter for cooling of armature, field and flexure to ensure continuous performance. The reinforcing struts are provided on the vibration platform for the specimen setting. The shakers of maximum force output of 2000 Kgf are on air-cooling type, reducing maintenance work on cooling systems. The shakers above 100 Kgf have the facility of low frequency isolation. The specifications of the shaker is given in table 4.1.

The function of a vibration generation system is to produce a selected waveform with required vibration level (i.e. acceleration/velocity/displacement) and frequency to test the specimen mounted

Table 4.1 Details of the shaker

Model	MEV 0050
Peak sine force	50 Kgf
Displacement	12mm (p-p) max,10mm (p-p) controllable
Armature mass	0.85 Kg
Moving armature suspension	Link-Arm type
Moving armature diameter	75mm
Shaker rotation	$\pm$ 90 degree from vertical
Cooling	Forced air
Protection	Field interlock, Over travel
Interface	Drive Coil Interface, Field Coil Interface, Vibrator interface
Duty	Continuous

on the shaker. The Electrodynamic Shaker is very much reliable as there are no moving parts to wear out and axial resonance frequency is kept quite high to avoid self resonance. The system force rating and moving element mass are the primary characteristics which determine the vibration level.

Salient Features

- Long term reliability
- Heavy duty rolling strut armature suspension system (for large shakers)
- Excellent cross-axial restraint
- Dual suspension system
- Forced air cooling facility

4.1.2. Power Amplifier

The MPA Series power amplifiers are all of solid state type permitting for excellent durability, and are equipped with self protective facility such as cooling facility check, temperature check, current supply check as well as interlock facility for disoperation to ensuring safety operation. The amplifiers are designed for continuous full power and transient overload condition. The amplifiers of upto rated output 15 KVA are of air-cooling type.

The function of power amplifier is to amplify the output signal of vibration controller/generator, sufficiently to drive the exciter to the vibration level. D.C. supply to energize the magnetic field of the electrodynamic shaker is built in the power amplifier. Power amplifier is having direct coupled circuit design for D.C. centering of moving platform to use full stroke length of exciter at low frequency for heavy payloads.

Salient Features

- Low distortion
- High power gain
- Internal filed support
- Wide range frequency bandwidth

The amplifier generates control signal which is to be fed to power amplifier for further amplification. It has a built in sinusoidal waveform generator which provides variable frequency sine wave output adjustable from 1 Hz to 10 KHz in four over lapping ranges. At auto position of the frequency selector, the control is transferred to vibration controller.

It has various protection facilities with the use of advanced analog-digital techniques. It has features of D.C. table centering, selectable current limit of power amplifier. The details of power amplifier and the built-in function generator are explained in table 4.2 and table 4.3.

Table 4.2 Details of power amplifier

Model	MPA 0500
Power output	500 VA
Coupling with shaker	Direct coupled
Input Impedance	10 KOhms
Protection	Output current limit, Field failure
Additional output	D.C. output sufficient to energize the field of $\pm$ 50 Kgf rating shaker
Input signal	Built-in/External/Vibration Controller
Interfaces	Armature Interface, Field Coil Interface, Vibrator Interface
Duty	Continuous

Table 4.3 Details of built-in sinusoidal function generator

Frequency range	1 Hz to 10 KHz in four over lapping ranges
Frequency adjustment	Coarse and fine through knob control
Operational modes	Manual control, control through vibration controller
Protection indicators	Output current limit, Field failure, Over travel
Sine output level	Matched with power amplifier
Status indicators	RUN and TRIP
External input	From external vibration controller through BNC connector

Salient Features

- Versatile design
- RESET and START indication
- Built-in sinusoidal waveform generator
- Integral metering of armature voltage and current

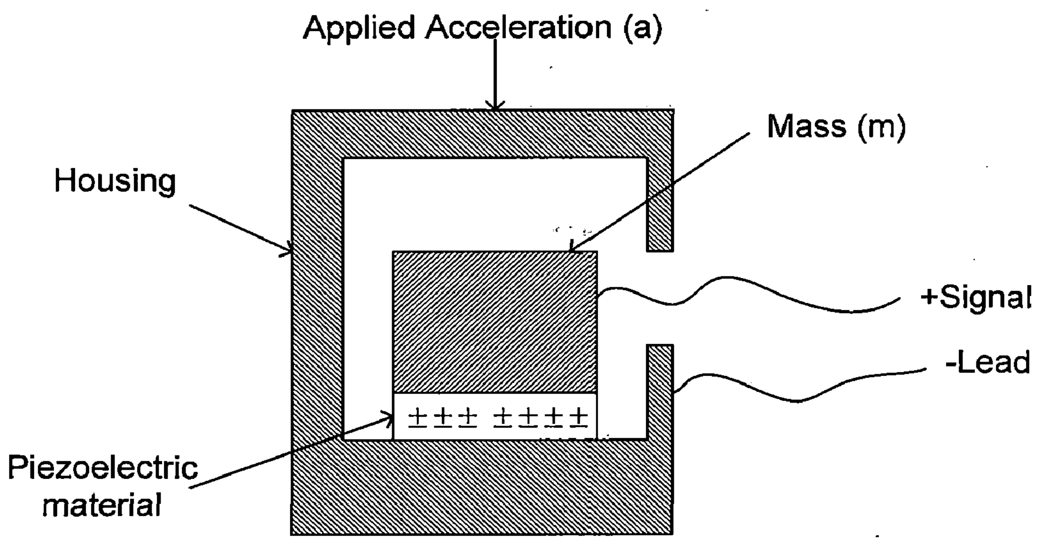
4.1.3. Cooling System

The cooling system which extracts the heat from the shaker field, armature coils also extract heat from transistor section of power amplifier. It is a standard equipment complete with two to three nos. of 3 m length suitable diameter hose pipes. The unit is totally enclosed and comprises of a centrifugal blower which is driven by a three phase motor. It is intended to be located remotely to minimize heat and noise in the vicinity of the vibration system. The details of the cooling system is provided in table 4.4.

Table 4.4 Details of cooling system

Model	MCS 015
Revolution	2800 rpm
Capacity	20 cubic meter/minute
Motor make	Crompton/ Kirloskar
Power source	220V $\pm$ 10% single phase A.C. supply

4.2. ACCELEROMETERS



Accelerometers

Fig. 4.1 Schematic of piezoelectric accelerometer

An accelerometer is an apparatus, either mechanical or electromechanical, for measuring acceleration or deceleration - that is, the rate of increase or decrease in the velocity of a moving object. They are used for detecting vibrations.

The measurement of acceleration or one of its derivative properties such as vibration, shock, or tilt has become very commonplace in a wide range of products.

The types of sensor used to measure acceleration, shock, or tilt include piezo film, electromechanical servo, piezoelectric, liquid tilt, bulk micro-machined piezo-resistive, capacitive, and surface micro-machined capacitive. Each has distinct characteristics in output signal, development cost, and type of operating environment in which it best functions.

The Piezoelectric Accelerometer

Among the desirable features of the piezoelectric (PE) accelerometer are accuracy, durability, large dynamic range, ease of installation, and long life span. Although these devices cost more than other types, in many situations their benefits outweigh the higher price. To provide useful data, PE accelerometers require proper signal conditioning circuitry. A schematic of the PE accelerometer is given in fig 4.1.

The PE accelerometer uses an internal PE element coupled with a loading mass to form a single-degree-of-freedom "mass-spring" system. The accelerometer is a charge-sensitive device; an instantaneous change in stress on the internal PE element produces a charge at the accelerometer's output terminals that is proportional to the applied acceleration. For interfacing purposes, the PE accelerometer can be modeled as a voltage generator, Eg, in series with an internal capacitance, Ci. The internal capacitance is an important characteristic because it can have a significant effect on overall system sensitivity. A typical PE accelerometer's sensitivity is specified in Pico coulombs per g (pC/g). Typical sensitivities are 0.5–1000 pC/g. PE accelerometers can be applied to measure vibration levels ranging from 10–4 g to >104 g. The useful measurement range of a given unit is often limited only by its signal conditioning and measurement systems.

The accelerometers can be used to measure very low frequencies. In practice, the low-frequency response is usually limited by the signal conditioning electronics in order to eliminate noise from sources such as thermal effects, strain on the accelerometer base, and tribo-electric noise generated in the connecting cable. The low-frequency cut-off is typically set around 2 Hz, but may be set higher if the lowest frequencies are not of interest to the user.

The accelerometer's useful upper frequency limit is dependent on its resonance frequency. The device will exhibit a sharp peak in its electrical output at the resonance frequency that must be compensated for. The upper resonance frequency is a function of the unit's mechanical characteristics and the way it is attached to the test object. As a general rule, the output sensitivity and upper resonance frequency of a PE accelerometer are dependent on the size (mass) of the accelerometer. For example, a larger accelerometer will have increased output sensitivity but a lower resonance frequency.

The specifications of the accelerometer are given below:

Type	AR-50F	Capacity	50g
Temperature	23°C	Humidity	65%

- Rated output — 553 $\mu V/V$
  (Strain, K=2) — 1106×10^{-6}
- Calibration coefficient — $0.0452g/1\text{x}10^{-6}$
- Frequency response range (± 5% at 23°C) — 0 ~ 240 Hz
- Non-linearity — 1% RO
- Input resistance — 122.8 Ω
- Output resistance — 122.8 Ω
- Connection cable — 0.05 mm^2 5 m
- Safe overload rating — 300% RO
- Safe temperature range — 0 ~ 50° C
- Safe excitation — 5 V
- When measured by strain meter (K=2)
  Acceleration = Calibration coefficient x Readout strain

4.3. DYNAMIC STRAIN AMPLIFIER [34]

Dynamic Strain Amplifier is designed for dynamic measurement of strain using strain gauge. The sensor is connected to the input of the amplifier through a 5 pin coaxial connector. The output is amplified and is passed through a 4 pole selectable low pass filter with Butterworth characteristics. Analog filtered output is available through BNC connector point.

Specifications:

- Bridge excitation — 4 to 10 V DC, screw driver adjustable
- Coarse balance — 11 steps
- Fine balance — Screw driver control (lock nut)
- GainSwitch selectable — 10, 50, 100, 500, 1000 and 5000

- Low pass filter — 3 selectable range 100 Hz, 300 Hz and 1000 Hz
- Analog output — ± 10 V
- Power source — Mains 230 V, 50 Hz, AC
- Operating temperature — 0 to 55°C

The measurement of strain is based on the method of Wheatstone bridge shown in fig. 4.2. The ratio of resistances are such that when no current flows through the galvanometer

$$R_1 / R_4 = R_2 / R_3 \tag{4.1}$$

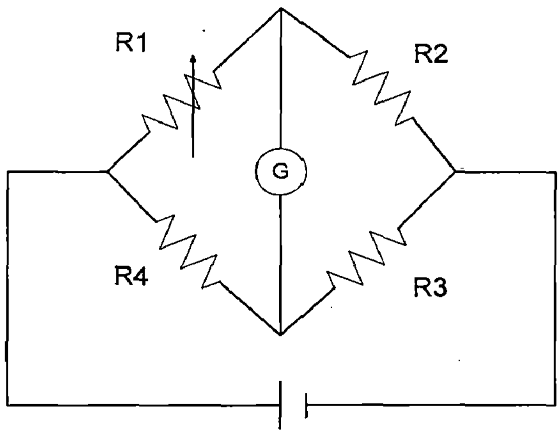


Fig. 4.2 Wheatstone Bridge

The resistance strain gage operated on the principle that the electrical resistance changes over with strain. If δR is the change in the gage resistance corresponding to a particular system condition and R is its initial value, then its gage factor is defined as

$$G = \frac{\delta R / R}{\delta L / L} \tag{4.2}$$

Where δL and L are the change in the length and the initial length of the element. The strain in this case is defined as

$$\varepsilon = \delta L / L$$

$$= \frac{\delta R / R}{G} \tag{4.3}$$

Relation between strain and bridge output

Let R= Gauge resistance and δR be the change in gauge resistance

This gauge is placed as one arm of the Wheatstone bridge, where R_2, R_3 and R_4 are the fixed resistances

Initially under zero current flow condition through the galvanometer

$R_1 / R_4 = R_2 / R_3$

Let $R_2 = R_3$ and under load conditions the value of R changes by δR, then the output voltage is given by

$$V_o = E_o\left(\frac{R}{R+\delta R+R}\right) - \frac{E_o}{2} \qquad (4.4)$$

4.4. FILTER CIRCUITS [37]

Second order Low-Pass Butterworth Filter

A stop-band response having a 40 dB/decade roll-off is obtained using the second order low-pass filter. LM 741 op-amp is used for designing this filter.

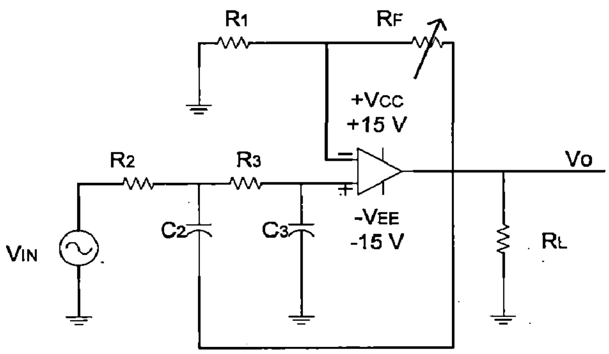


Fig. 4.3 Second order low pass filter

The gain of the second-order filter is set by R_1 and R_F while the high cut-off frequency f_H is determined by R_2, R_3, C_2 and C_3

$$f_H = \frac{1}{2\Pi\sqrt{R_2 R_3 C_2 C_3}} \qquad (4.5)$$

The voltage gain magnitude equation is given by

$$\left|\frac{v_o}{v_{in}}\right| = \frac{A_F}{\sqrt{1+(f/f_H)^4}} \qquad (4.6)$$

The factor A_F is known as pass band gain and is given by:

$$A_F = 1 + \frac{R_F}{R_1} \qquad (4.7)$$

Second order High-Pass Butterworth Filter

All the equations for the High-Pass filter design procedure is similar to that of Low-Pass filter, except that the positions of the resistances and capacitances are interchanged. Similarly as in the previous case LM 741 was used for implementing this filter.

These two filters were cascaded in series to obtain a band-pass filter so as to pass the output signal of the DAC to feed it the power amplifier. Before interfacing the filter circuit with the power amplifier, a voltage follower (buffer) circuit is implemented in series for the driving capabilities of two different circuits operating at different voltage levels.

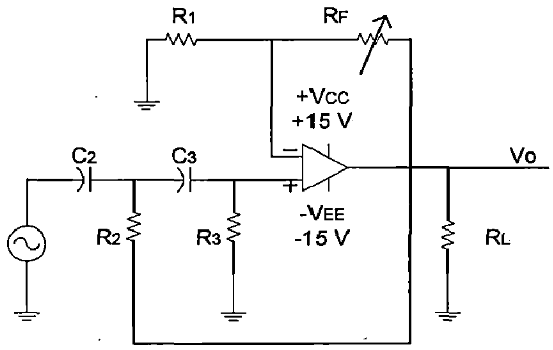


Fig. 4.4 Second order high pass filter

4.5. PCI-9112 ENHANCED MULTIFUNCTION DATA ACQUISITION CARD [35]

The PCI-9112 is an advanced performance, data acquisition card based on the 32-bit PCI Bus architecture. High performance designs and the state-of-the-art technology make this card ideal for data logging and signal analysis applications in medical, process control, and etc.

Features

The PCI-9112 PCI Bus Advanced Data Acquisition Card provides the following advanced features:

· 32-bit PCI-Bus

· 12-bit analog input resolution

· On-board A/D FIFO memory

· Auto-scanning channel selection

· Up to 110KHz A/D sampling rates

· 16 single-ended or 8 differential analog input channels

· Bipolar or Unipolar input signals

· Programmable gain of x0.5, x1, x2, x4, x8

· On-chip sample & hold

· Two 12-bit monolithic multiplying analog output channels

· 16 digital output channels

· 16 digital input channels

· 3 independent programmable 16-bit down counters

· Three A/D trigger modes: software trigger, programmable pacer trigger, and external pulse trigger.

· Integral DC-to-DC converter for stable analog power source

· 37-pin D-type connector for PCI-9112

· 100-pin SCSI-type connector for cPCI-9112

· Compact size: half-size PCB

Specifications

¨ **Analog Input (A/D)**

· Converter: B.B. ADS774, successive approximation type

· Input Channels: 16 single-ended or 8 differential

· Resolution: 12-bit

· Input Range: (Software controlled)

Bipolar: ±10V, ± 5V, ±2.5V, ±1.25V, ±0.625V

Unipolar: 0~10V, 0~5V, 0~2.5V, 0~1.25V

· Conversion Time: 8 m sec

· Overvoltage protection: Continuous ± 30V maximum

· Accuracy:

GAIN = 0.5, 1 0.01% of FSR ±1 LSB

GAIN = 2, 4 0.02% of FSR ±1 LSB

GAIN = 8 0.04% of FSR ±1 LSB

· Input Impedance: 10 MW

· Trigger Mode: Software, Timer Pacer, and External trigger

· Data Transfer: Program control, Interrupt, DMA (Bus mastering)

· Data Throughput: 110 KHz (maximum)

· FIFO memory: 2K Words (for cPCI-9112 only)

¨ **Analog Output (D/A)**

· Output Channel: 2 double-buffered analog outputs

· Resolution: 12-bit

· Output Range:

Internal reference: (Unipolar) 0~5V or 0~10V

External reference: (Unipolar) max. +10V or -10V

· Converter: AD 7541 or equivalent, monolithic multiplying

· Settling Time: 30 m sec

· Linearity: ±1/2 bit LSB

· Output Driving: ±5mA max

¨ **Digital I/O (DIO)**

· Channel: 16 TTL compatible inputs and outputs

· Input Voltage:

Low: Min. 0V; Max. 0.8V

High: Min. +2.0V

· Input Load:

Low: +0.5V @ -0.2mA max.

High: +2.7V @+20mA max.

· Output Voltage:

Low: Min. 0V; Max. 0.4V

High: Min. +2.4V

· Driving Capacity:

Low: Max. +0.5V at 8.0mA (Sink)

High: Min. 2.7V at 0.4mA (Source)

¨ **Programmable Counter**

· Device: 8254

· A/D pacer: 32-bit timer (two 16-bit counters cascaded together) with a 2MHz time base

· Counter: One 16-bit counter with a 2MHz time base

· Pacer Output: 0.00046 Hz ~ 0.5 MHz

¨ **General Specifications**

· I/O Base Address: 9 consecutive DWORD (double word) address location

· Connector: 37-pin D-type connector

· Operating Temperature: 0° C ~ 60° C

· Storage Temperature: -20° C ~ 80° C

· Humidity: 5 ~ 95%, non-condensing

· Power Consumption:

+5 V @ 475 mA max

+12V @ 150 mA max

· Dimension: Compact size only 98mm (H) X 173mm (L)

4.6. RESULTS AND DISCUSSIONS

Mathematical modeling of any device is an important step in the simulation studies involving that device. In this work a mathematical model of an ED shaker has been obtained for simulation purposes. The idea behind conducting practical experiments involved obtaining the frequency response of the ED shaker. It was formulated to keep current flowing in the armature coil as the input parameter, rather than voltage since it is current which produces the force. The output acceleration is chosen as the output parameter. One more reason for this is that we need not include R and L in the transfer function and hence avoid dealing with their frequency dependence, though their dependence affects the current directly.

Three different bode plots of the system is obtained. The plot $G_i(s)$ and $G_v(s)$ are estimated from the asymptotic approximation. The estimated $G_i(s)$ and $G_v(s)$ bode plots of the system are shown hence in fig.4.5 and fig.4.6.

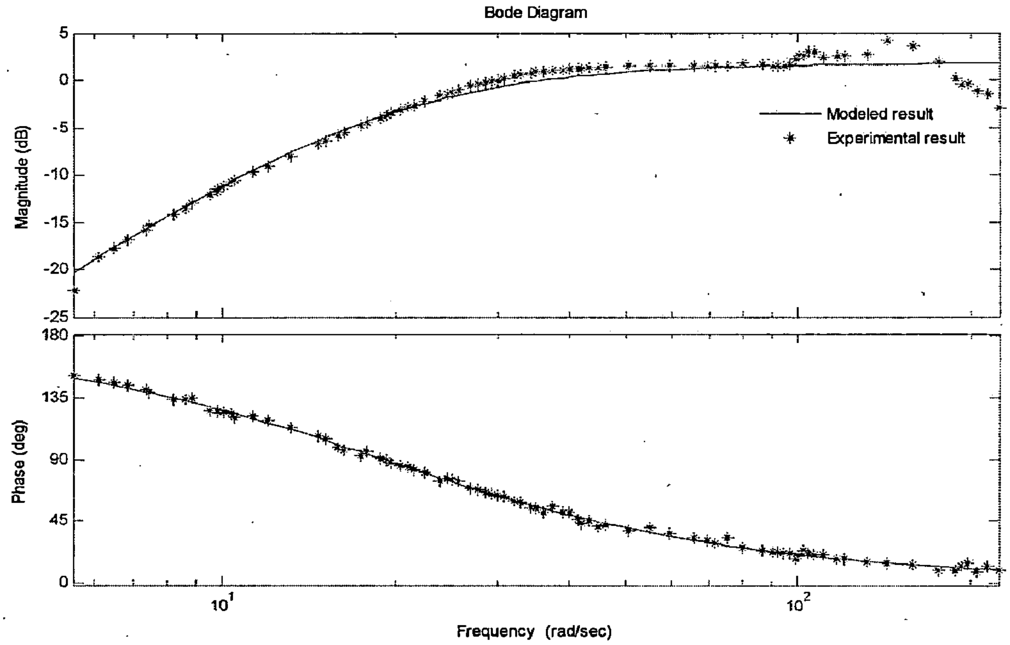


Fig. 4.5 Bode plot of $G_i(s)$

$$G_i(s) = \frac{1.23s^2}{s^2 + 36.1s + 361} \tag{4.8}$$

Here $G_i(s)$ represents the model of the shaker with respect to the output acceleration calibrated in terms of voltage to the input current flowing in the armature windings. $G_v(s)$ represents the model of the shaker with respect to the output acceleration calibrated in terms of voltage to the input voltage of the linear power amplifier. In this the gain of the power amplifier is also included. It is noted that order of the system doesn't change between $G_i(s)$ and $G_v(s)$, though theoretically it should have changed due to the (R+sL) factor as discussed. Based on this we can assume that this pole (R+sL) does not affect the model of the system. A correct explanation is that L is very small in comparison to R and can be effectively neglected.

$$G_v(s) = \frac{11.08s^2}{s^2 + 36.1s + 361} \tag{4.9}$$

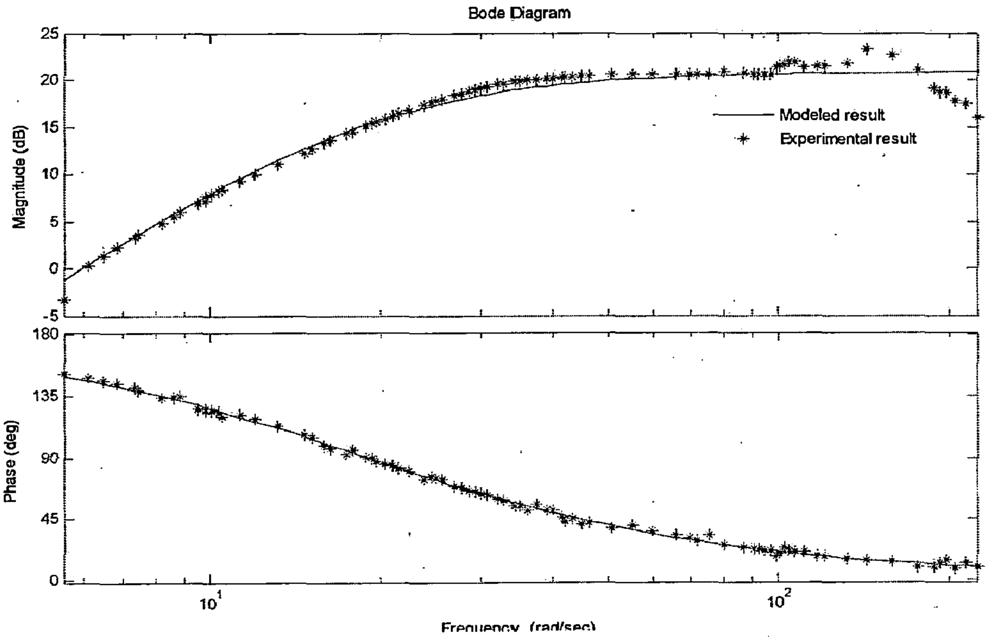


Fig. 4.6 Bode plot of $G_v(s)$

As a result of these assumptions we derive at the following parameters of the system:

m_o=0.85

k=306.85

c=30.685

Bl=1.0455

The gain of the power amplifier is approximately 9.01.

The transfer function of the whole system when signal from the PC is passed through the buffer, filter, power amplifier and shaker is obtained and the bode plot is shown in fig.4.7.

$$G(s) = \frac{650s^2}{s^3 + 118.1s^2 + 3321s + 29602}$$

$$=G_v(s)\left(\frac{58.717}{s+82}\right) \quad (4.10)$$

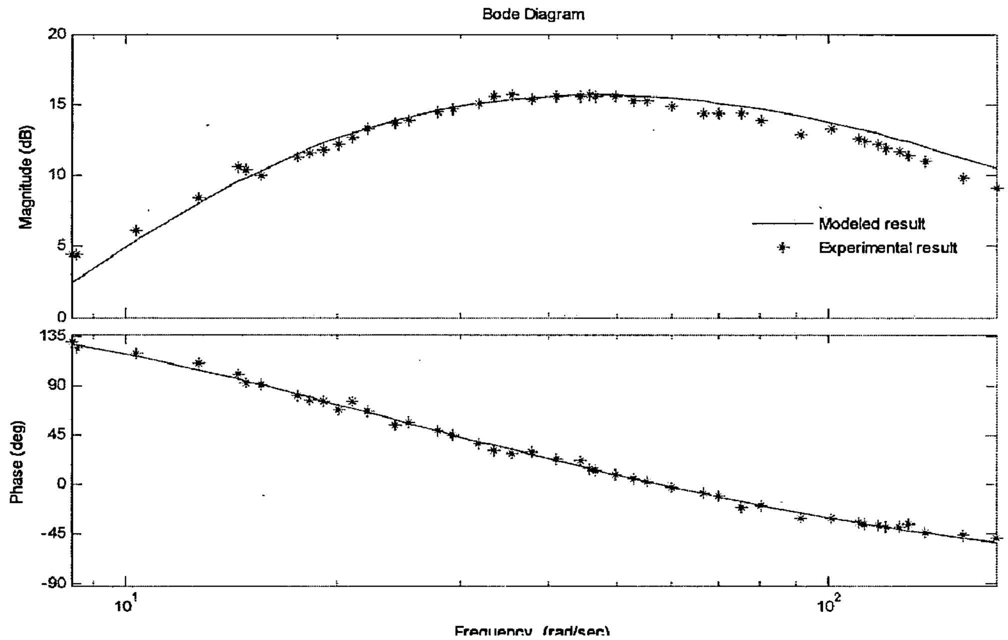


Fig. 4.7 Bode plot of *G(s)*

The variation of resistance (Ω) and inductance (H) of the armature winding with frequency is obtained and is shown in fig. 4.8. It is important to model the variation of resistance and inductance with frequency as their variation causes variation in the current values flowing in the armature coil. Since the design and control is based on the current feedback, all the factors affecting that should be taken into consideration. In the plot shown hence the field supply was cut out the values of R and L were obtained by noting the input voltage, output current and the phase difference between these two quantities. Applying basic circuit theory the values of R and L were obtained. The logic behind cutting out the field supply was that once movement of the armature is zero and hence the impedance model of the motional back-emf is avoided as it is difficult to measure it. The values of R and L during running conditions will be different due to effect of the back-emf. Also curve-fitting is done to the data so obtained.

As expected both the resistance, R and the inductance, L have a non-linear frequency dependency due primarily to the action of the pole plating, which has the effect of increasing the resistance and decreasing the inductance with frequency as shown above. In addition, these electrical parameters are subject to secondary effects such as thermal variations and eddy current losses within the coil.

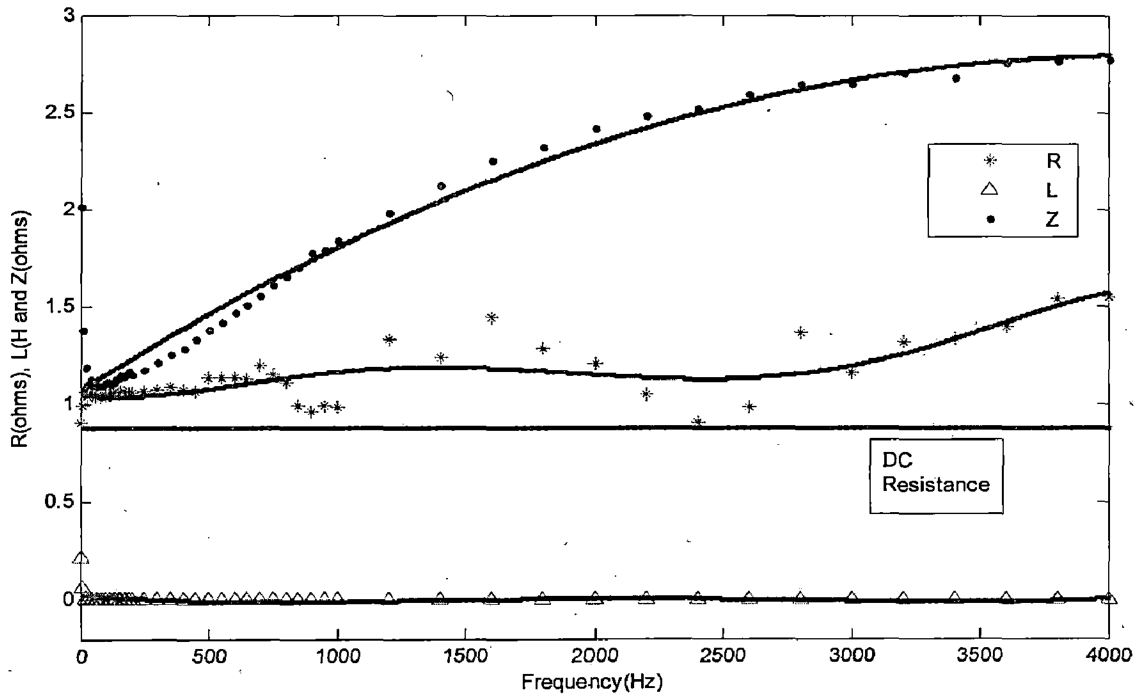


Fig. 4.8 Variation of R, L and Z with frequency

The harmonic analysis of the voltage and current is also done. It is observed that both voltage and current has harmonics in them. The percentage of harmonics in the voltage form is less, but in case of current the percentage is comparatively larger. The major harmonic components were the second, third and fifth, though there magnitude was different at different frequencies. If harmonics are present in the current waveform, a proper sinusoidal acceleration waveform is not obtained and the out waveform is distorted.

4.7. CONCLUSION

The mathematical model of the shaker and also of the open loop system involving other circuitry has been obtained. The results obtained were in compliance with the expected results. These results will used in the design of the acceleration controller.

CHAPTER 5

SINUSOIDAL ACCELERATION CONTROL

5.1. INTRODUCTION

Shaking systems which are capable of replicating an actual situation are being used in every industry for vibration-testing [18]. Vibration testing is a critical step in the development of a reliable product hence is very much popular in the fields of military, aerospace, automotive, electronics and so on. The controller in the shaker system is used not only for providing stable control but also to have a good tracking of the reference waveform. The history of sinusoidal vibration testing and its control started from the 1950's when the military specifications started to include the terms environmental in the specifications [30]. Even today sinusoidal vibration tests are partly conducted because of the historical reasons.

5.2. PROPOSED CONTROL SCHEME

Conventional controllers so far have been employing the open-loop iterative compensation through repetitive excitations [18]. This method has the drawback that each iteration produces damage to the test specimen, but the control action is taken in the next iteration. The requirement of an acceleration controller arises from the fact that if acceleration above certain levels is given it can damage the test article particularly. So it is always advisable that before the start of a vibration test to conduct a gentle sweep test to identify the resonant frequencies in the test object and the shaker. This can particularly be fatal if the test object is something like a satellite which will also be the actual launch piece. A set of specifications is defined for each test frequency and test object. Acceleration definition for each test article is a study of itself. Depending on the test article different profiles are defined. In this initially a profile is defined for different frequencies and control action is taken to keep these defined values within the tolerable limit. It is understood that for control of the output waveform of the shaker, three parameters of the waveform needs to be taken into consideration.

1. Magnitude
2. Frequency
3. Phase

It has been experimentally found out that frequency of input and output waveform is the same. Therefore the control action is to be taken to control the magnitude and phase difference between the actual and desired waveforms. Phase difference is potentially produced in any system between the input and output waveform, due to the transportation lag.

For the purpose of this an acceleration control scheme as outlined in fig.5.1 has been implemented and tested on the available shaker.

The main components in the control feedback loop are explained thus:

1. Tracking filter
2. RMS values and error
3. PI controller with rate limiter

All the data input sensor and data output to power amplifier were done through ADLINK PCI-9112 data acquisition card whose specification has already been explained in Chapter 4.

5.2.1. Tracking Filter

Tracking sinusoid signals in noise is a common procedure in communications, radar, and spectral analysis applications. Traditional techniques to track sinusoids include methods such as FIR filters combined with LMS algorithm (least mean square), phase-locked loops (PLL), and adaptive constrained IIR structures. Adaptive filters can, under the right conditions, achieve close to optimum solution but have the disadvantage of being computationally intensive and are slow to converge. Conventional PLL-based tracking filters are faster and easier to implement but they require a reference whose center frequency is relatively close to the input signal, this severely limits their range of operation. Finally, constrained IIR structures have the characteristic of adjusting only the center-frequency dependent coefficients of the filter in order to track the signal. Because only the center-frequency parameters are adjusted, constrained IIR structures are more stable, and are more computationally efficient than adaptive FIR filters, although their response is not as fast as PLL structures. Automatic sine tests work exceptionally well when the signal from the signal is purely sinusoidal in nature [30]. But shaker signals are seldom pure sinusoids. The various reasons maybe attributed to power amplifier distortion, shaker resonances, fixtures and so on. Even if the shaker motion is highly distorted for any of the reasons explained, the tracking filter can be used to eliminate the harmonics present in the waveform and extract out only the fundamental component.

Tracking Filter is a device which multiplies sensor signal by exiting signal Ref Sine Wave and Ref Cos Wave two times and integrates separately. Output of two loops in added and fed back to the input. Time constant and bandwidth of tracking filter are determined by integral constant K. The purpose of the tracking filter is to extract the fundamental component from the harmonic signal. The reason for this is that the acceleration of the shaker depends on the magnitude and frequency of the pure sinusoidal (fundamental) given as input. The sensor picks up noise signals from various sources

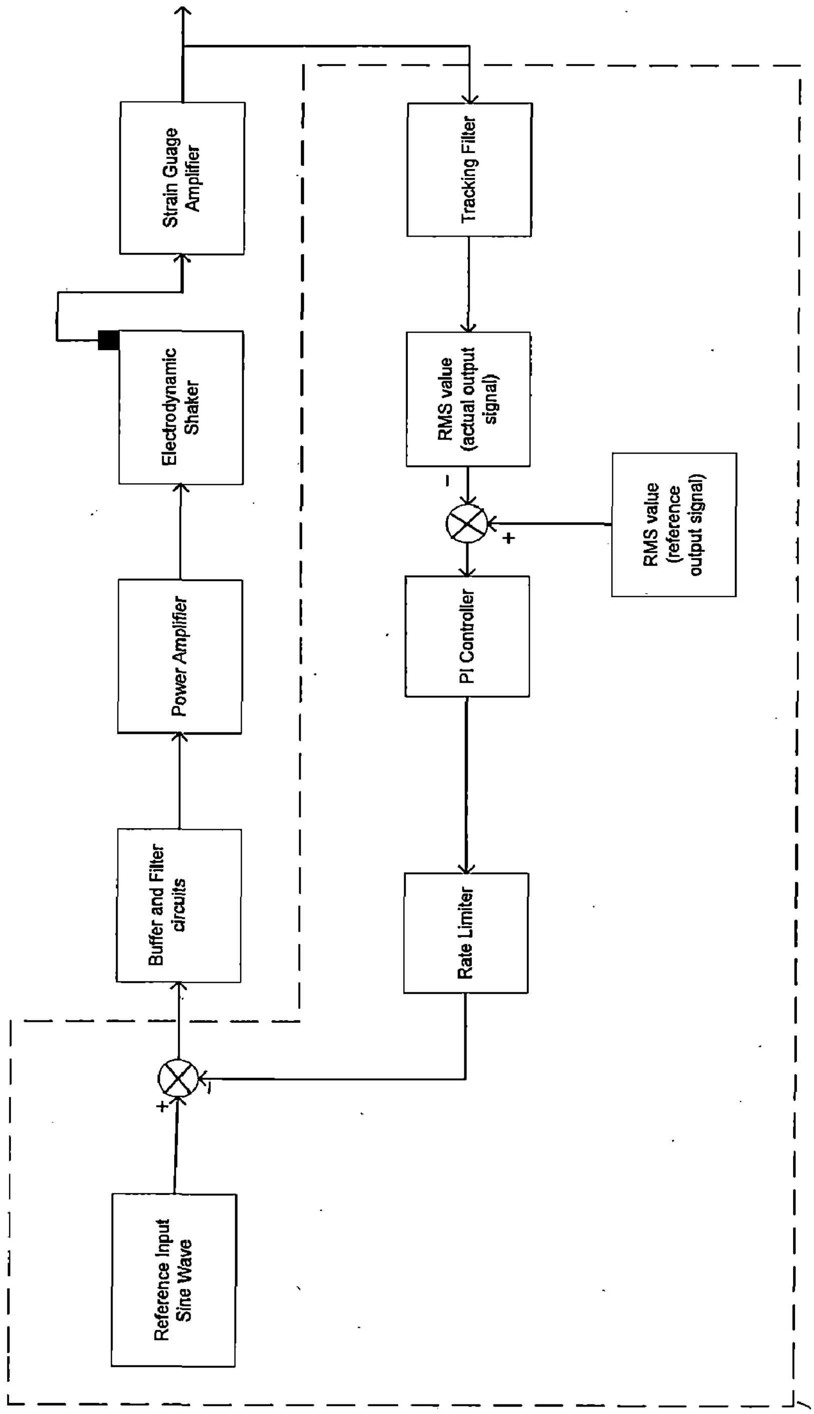


Fig.5.1 Proposed control scheme

These signals need to be eliminated and hence tracking filter is used. The scheme of tracking filter implemented is shown below. The SIMULINK model is shown here.

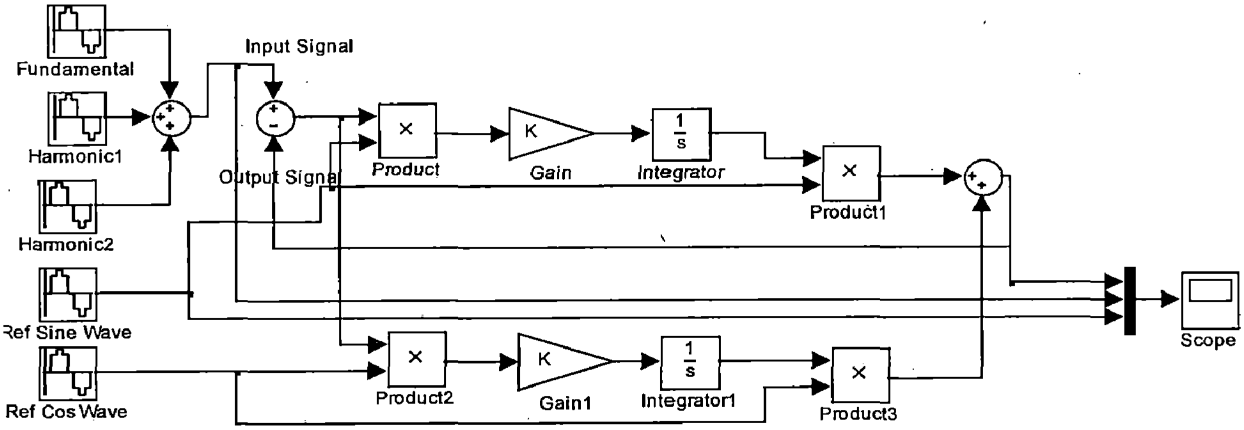


Fig. 5.2 Scheme of tracking filter

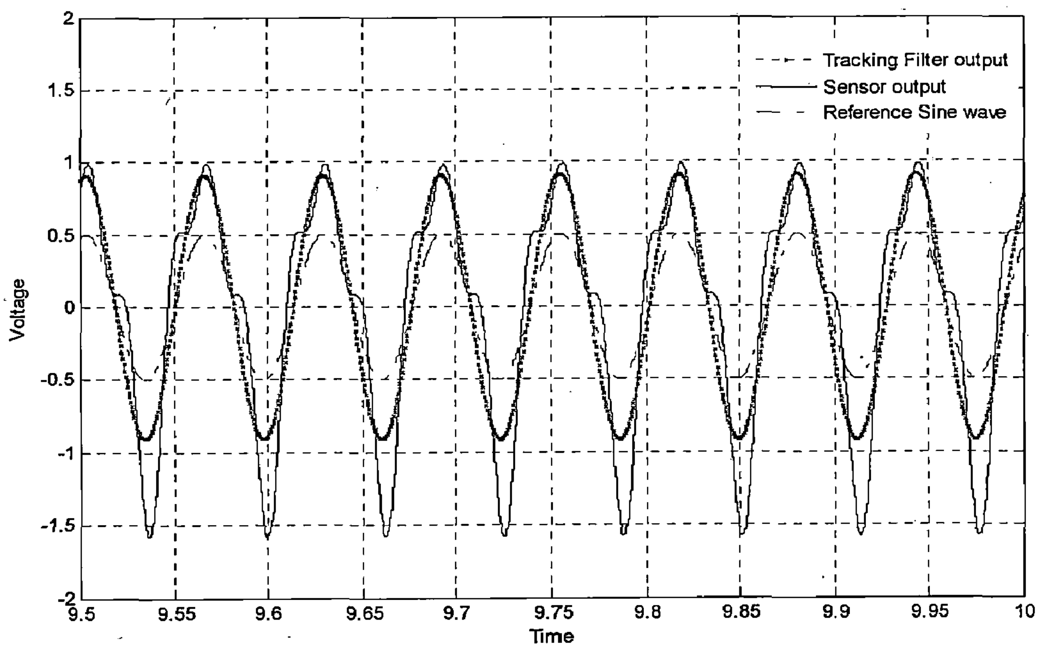


Fig. 5.3 SIMULINK diagram of tracking filter with its output.

It was found out by simulation that even by eliminating the Ref Cos Wave loop, tracking is precise. Hence this loop is avoided to speed up the control operation. This SIMULINK model is shown below

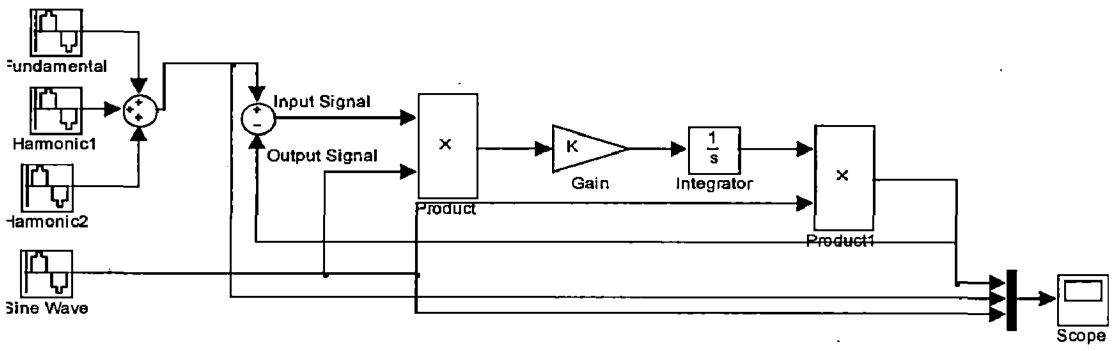


Fig. 5.4 Scheme of modified tracking filter

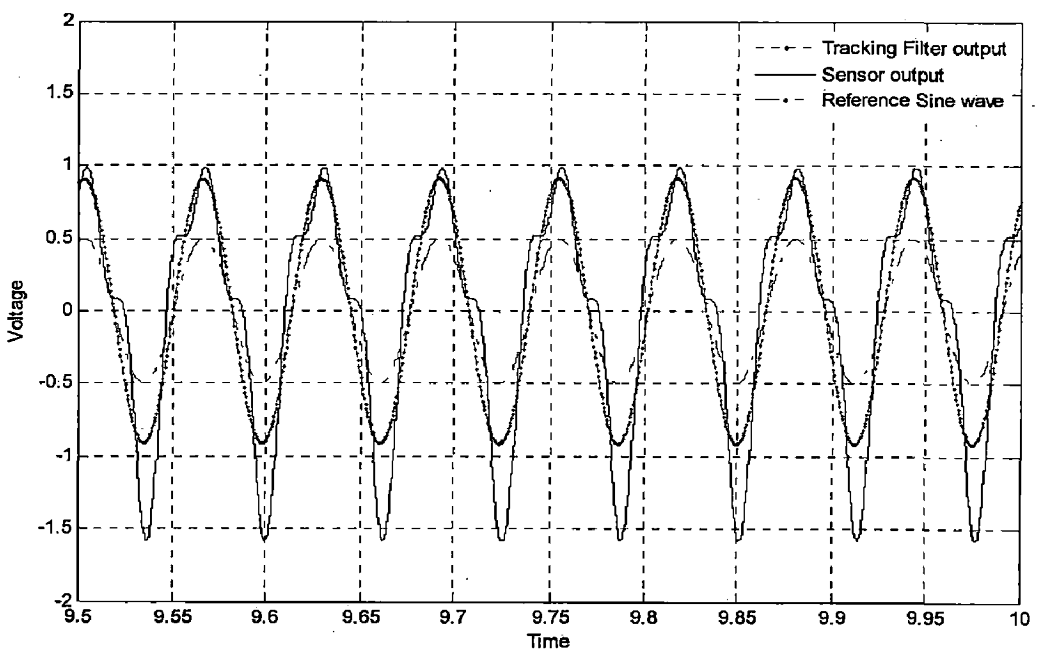


Fig. 5.5 SIMULINK diagram of modified tracking filter with its output

It is clearly seen from the output waveforms that the tracking filter is able to track the fundamental component in the input waveform very precisely and is also able to extract out the phase of this component. Since the control algorithm is implemented in M-file, which does not have parallel processing, hence the modification from the original algorithm to speed up the processing..

5.2.2. RMS value and error evaluation

Once the output of the tracking filter is obtained, the rms value of this signal is evaluated assuming it is a discrete set of data for a cycle. This value of rms is compared with reference value of rms. The difference between these two rms values gives the error signal. This error signal is given as the input to the PI controller and the corresponding control signal is generated.

The rms value of a discrete set of data points is given by

$$x_{rms} = \sqrt{\frac{(x_1^2 + x_2^2 + \ldots\ldots\ldots x_N^2)}{N}} \tag{5.1}$$

5.2.3. PI controller

Proportional Control Action

This type of control action involves a proportional constant, namely K_p, which is defined as

$$\frac{U(s)}{E(s)} = K_p \tag{5.2}$$

where $U(s)$ is the output signal and $E(s)$ is the error signal, measured in frequency domain. It is essentially an adjustable amplifier that helps reducing the effects of disturbances. This relatively simple type of controller is also useful in reducing the system sensitivity to changes in the plant's parameter. However, if the proportional gain is set too high, system instability (closed-loop) and signal distortion may result. Proportional control action is generally not useful for controlling steady state errors. This shortcoming of proportional control action gives rise to the need of another type of control action: the integral control action.

Integral Control Action

The relationship of the integral control action with the output and error signals, in frequency domain, is defined as

$$\frac{U(s)}{E(s)} = \frac{K_i}{s} \tag{5.3}$$

where K_i is the integral gain, U(s) is output, and E(s) relates to the error signal. In the time domain, the rate of change of the output is proportional to the error signal by the constant K_i. This means when the error changes, the rate at which the output signal change depends on the constant K_i. If error is zero, the output signal remains unchanged (slope = 0). This control action is also known as the reset control.

Proportional-plus-Integral Control Action (PI)

Combining Eq (5.2) and Eq (5.3), we have

$$\frac{U(s)}{E(s)} = K_P + \frac{K_i}{s} \tag{5.4}$$

Rearranging (5.4) we obtain

$$\frac{U(s)}{E(s)} = K_P\left(1 + \frac{1}{Ts}\right) \tag{5.5}$$

where $T = K_P/K_i$ is known as the integral time or reset rate (times per minute), the rate at which K_P is repeated (duplicated). Note that K_P affects both proportional and integral parts of the controller. For a system of type 1 with a ramp input, K_P controls the ramp-error constant and consequently affects the magnitude of the non-zero steady-state error. For a type 0 system, the value of K_i affects the ramp-error constant while K_P affects phase margin, gain margin, resonant peak, and bandwidth. In general, a PI control action can improve steady-state error, but at the expense of stability. In addition, if the location of the zero of the controller is correctly placed, the damping can also be improved. In the design of a PI controller, it's good to keep the zero close to the origin and far away from the most significant poles, while keeping K_P and K_i small. When properly designed, a PI controller can improve damping but extend rise time. In frequency domain, it can reduce bandwidth and improve gain and phase margins as well as resonant peak. The general effects of a properly designed controller are summarized in table 5.1

The PI control for implementation in PC converted to difference equations and was implemented. For this purpose bilinear (TUSTIN) transformation was used which is as given below:

$$s \rightarrow \frac{2}{T}\left(\frac{1-z^{-1}}{1+z^{-1}}\right) \tag{5.6}$$

Table 5.1 Characteristics of a PI controller and the effect of K_i on its performance

Characteristics	PI controller	Effects of increasing K_i
Maximum Overshoot	reduced	increased
Damping	improved	-
Rise Time	increased	decreased
Settling time	increased	increased
Steady-state error	improved	-
Phase margin	improved	decreased
Gain margin	improved	-
Resonant peak	improved	increased
Bandwidth	decreased	increased
Filter	low-pass	-

which gives the difference equation as

$$y(n) = K_1 * e(n) + K_1 * K_2 * e(n-1) + y(n-1) \tag{5.7}$$

where $K_1 = \frac{T}{2}\left(K_p + \frac{K_i * T}{2}\right)$

$$K_2 = \frac{\frac{K_i * T}{2} - K_p}{\frac{K_i * T}{2} + K_p}$$

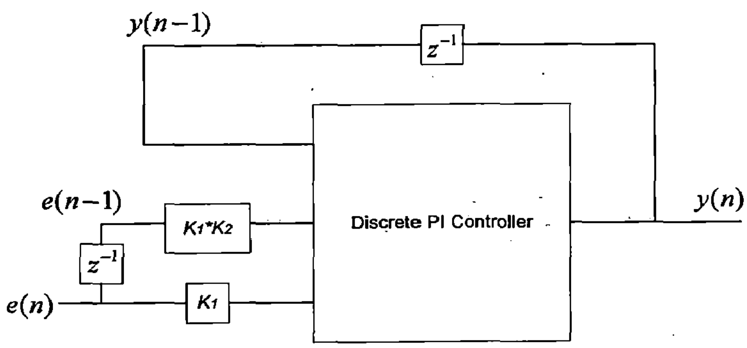


Fig. 5.6 State flow diagram representation of discrete PI controller

The output of the PI controller is augmented along with the Reference Input Sine Wave to adjust the output waveform in case of an error between the reference and actual signals.

5.3. RESULTS AND OBSERVATIONS

A model of the whole system in open-loop and closed-loop under no-load condition was simulated in SIMULINK, the model and output is shown in fig.5.8 and fig.5.9.

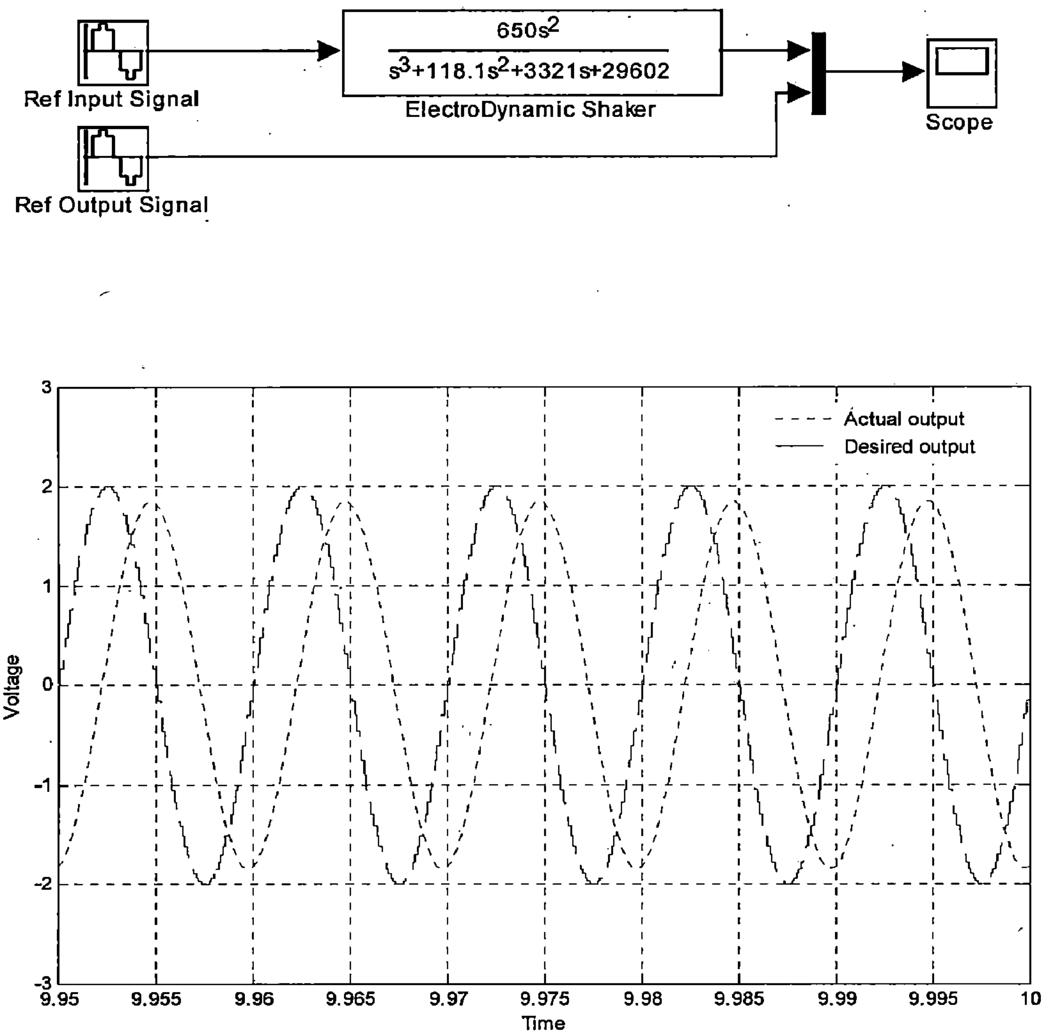
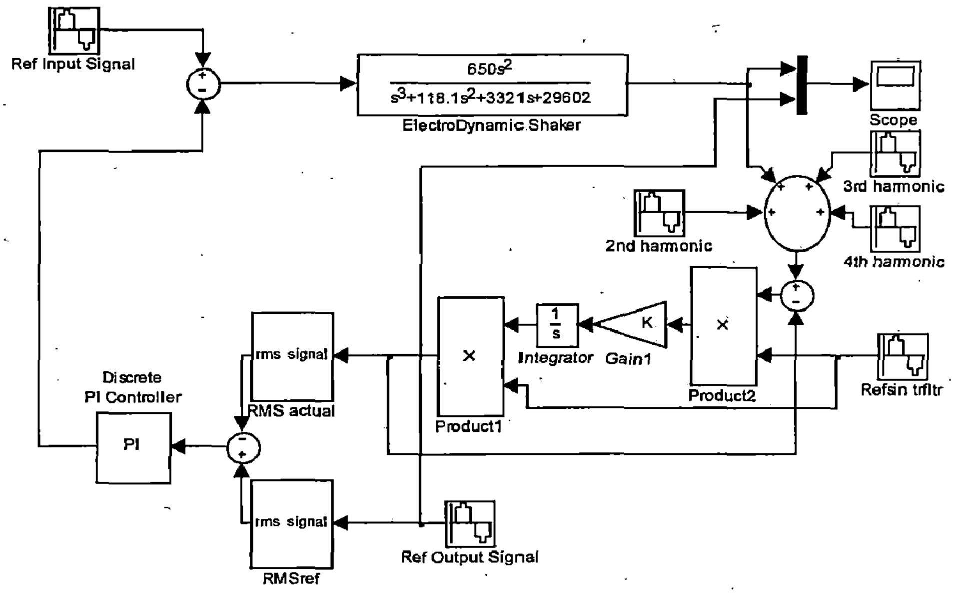


Fig. 5.8 Simulation of system under open-loop, no-load condition with output response



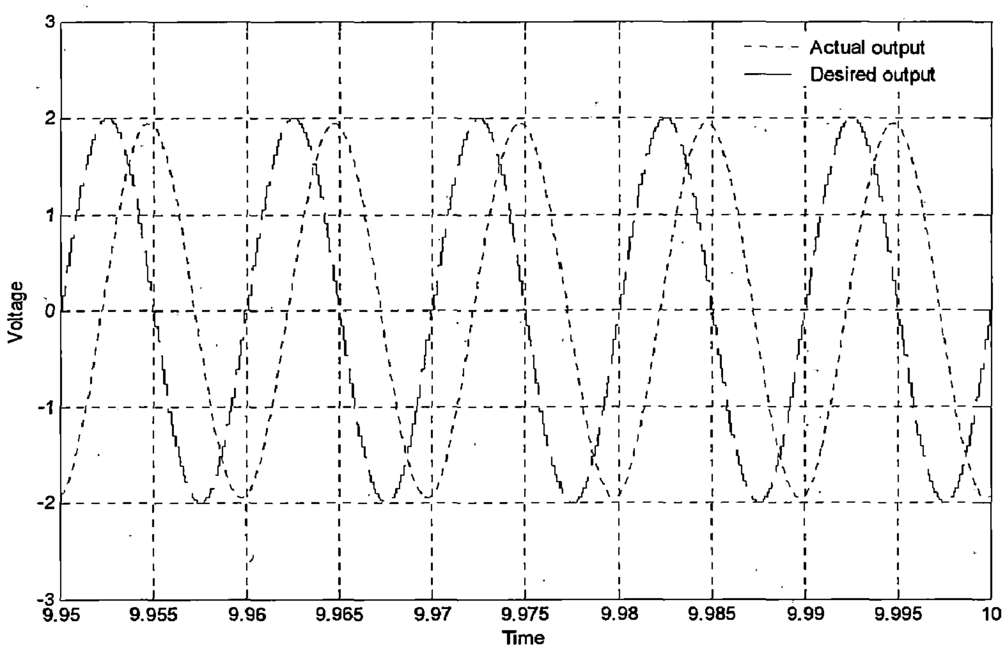


Fig. 5.9 Simulation of the system under closed-loop, no-load condition with output response

We see that the output waveform has a smaller magnitude because of the transfer function assumed. It is also further seen that the error in case of open loop system the error is higher than in case of closed loop system. Once the simulation results were proved to be satisfactory, the same was implemented on the available shaker for practical testing and validation of the algorithm.

For the experimental validation of the control scheme acceleration vs. frequency profile as shown below is defined for the shaker.

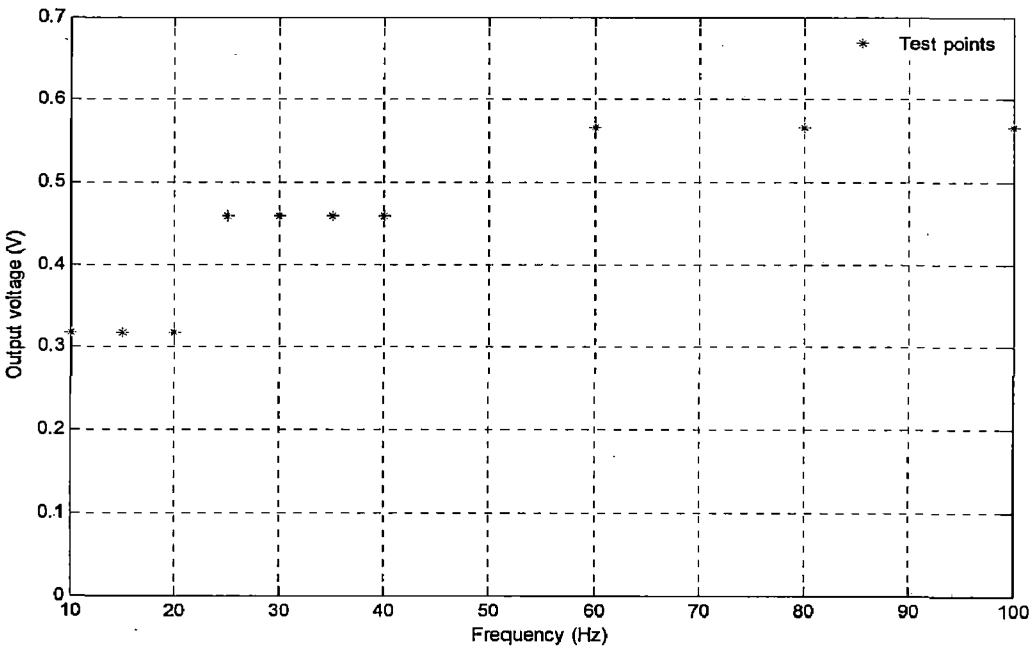


Fig. 5.10 Test profile for the shaker

The above test profile was chosen so as not to force the shaker in a non-linear range of operation. Using the proposed control scheme the shaker was tested under both no-load and loaded with a mass of 2Kg and tested. The harmonic analysis of the output waveform under open-loop and closed-loop performance was also evaluated. The results of the experiments are shown in subsequent pages.

In the time-response column, the representations of the various channels are given below:

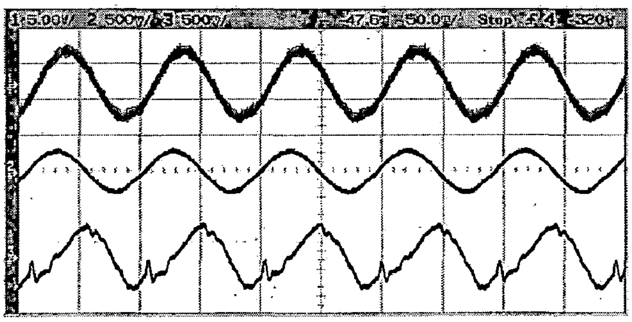
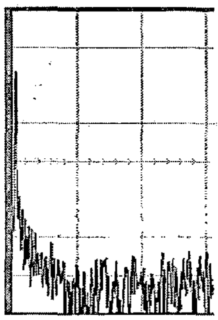
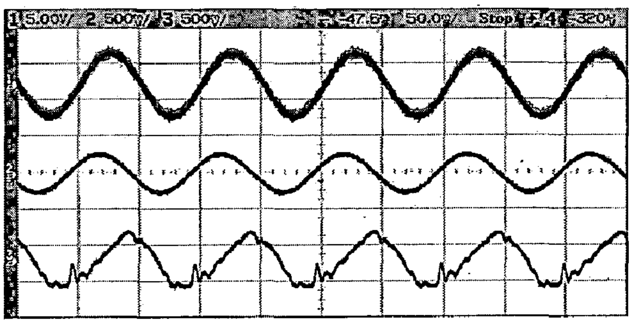
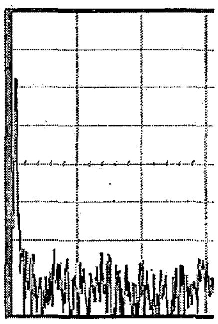
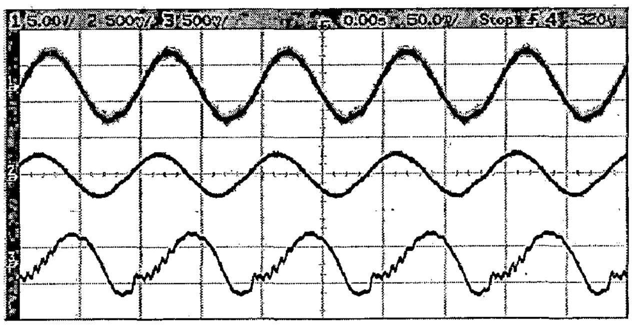
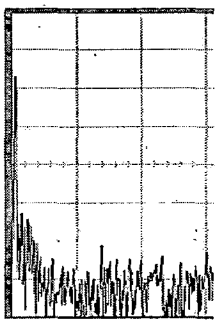
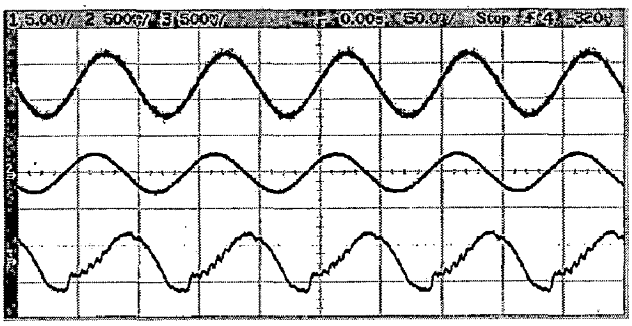
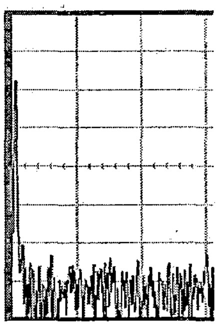
CH1: Output of the DAC in PCI-9112

CH2: Output of the buffer amplifier (voltage follower)

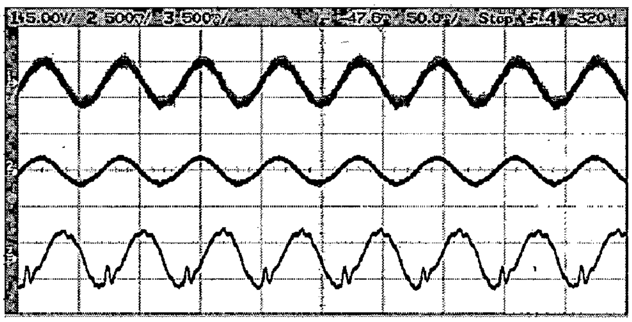
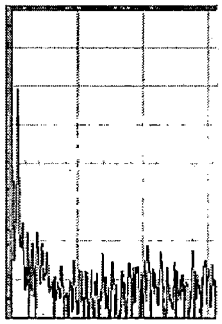
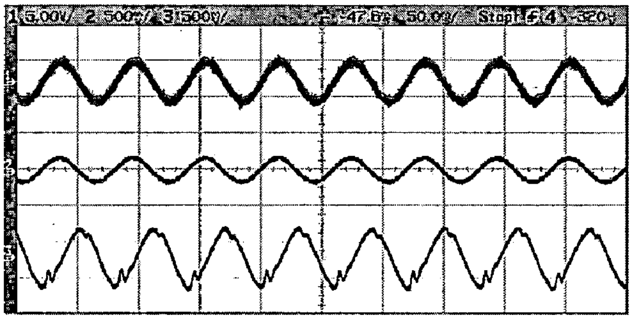
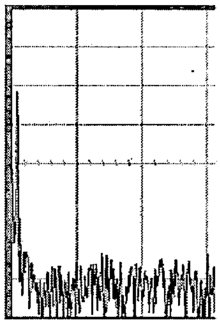
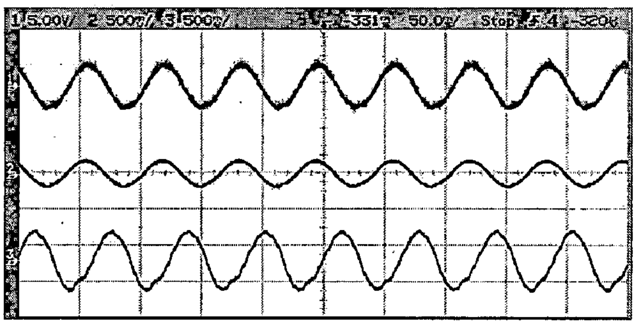
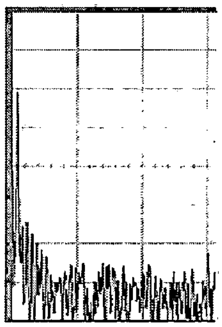
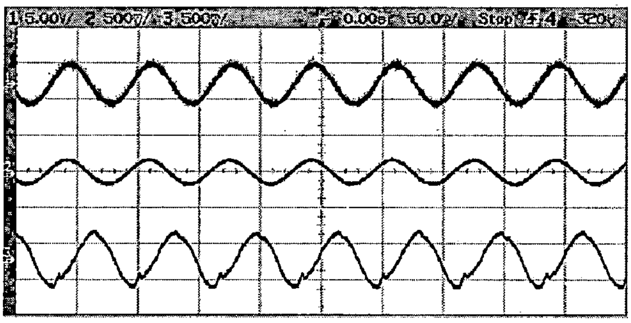
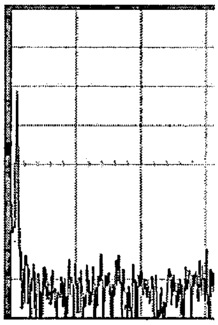
CH3: Output response of the shaker

The second column namely harmonic analysis gives the harmonic contents in each case

Frequency of Operation: 10 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

Frequency of Operation: 15 Hz

<table>
<tr><th colspan="2">Condition</th><th>Time Response</th><th>Harmonic Analysis</th></tr>
<tr><td rowspan="2">No Load</td><td>Open Loop</td><td></td><td></td></tr>
<tr><td>Closed Loop</td><td></td><td></td></tr>
<tr><td rowspan="2">Load (m_L= 2 Kg)</td><td>Open Loop</td><td></td><td></td></tr>
<tr><td>Closed Loop</td><td></td><td></td></tr>
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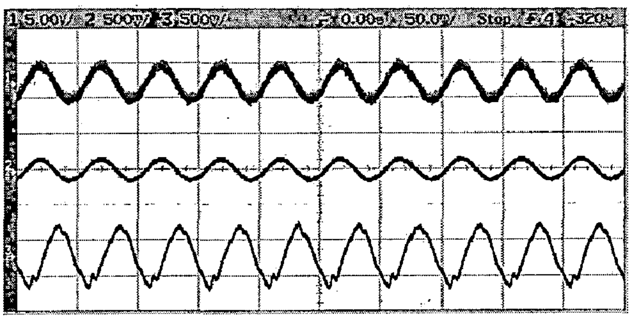
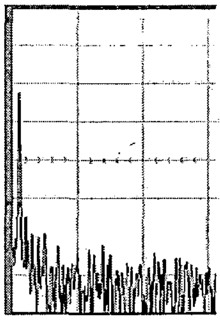
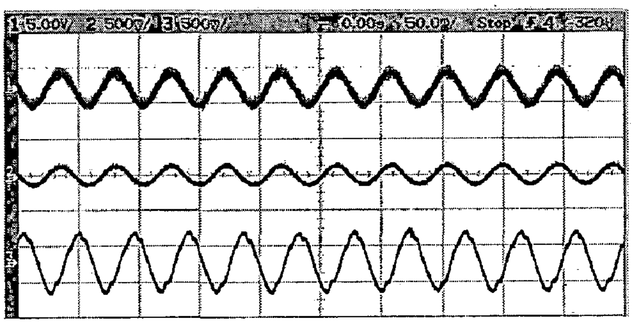
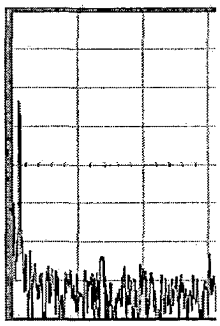
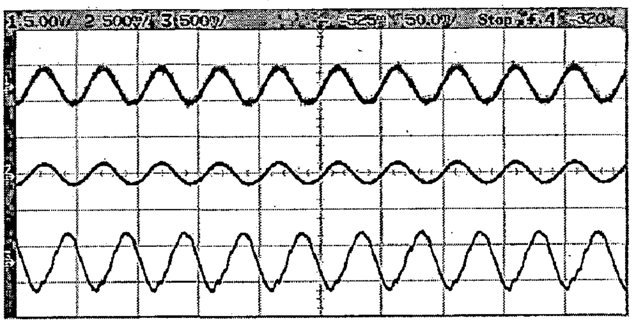
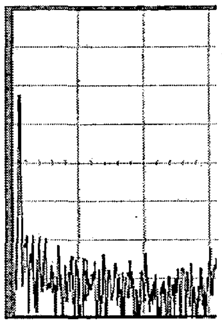
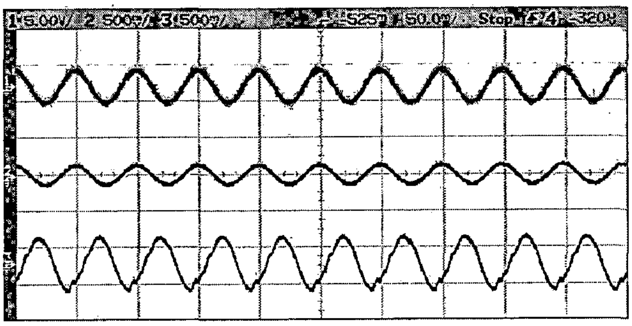
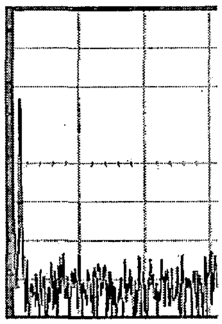
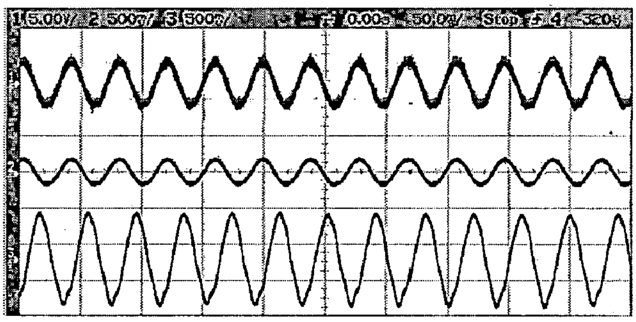
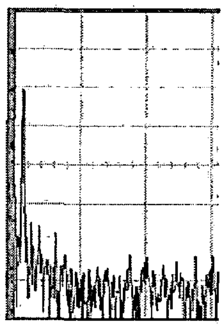
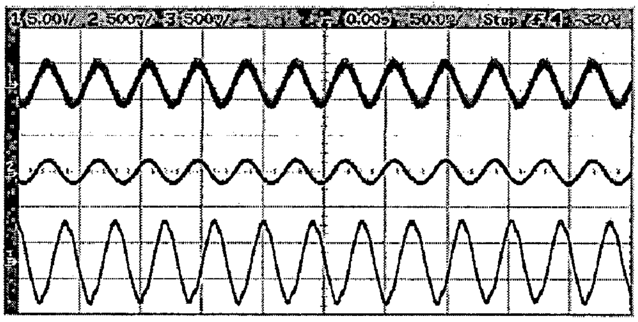
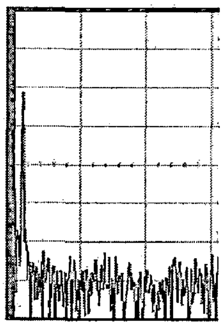
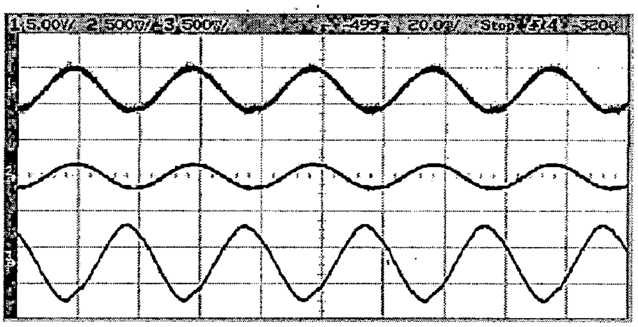
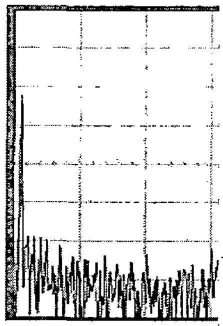
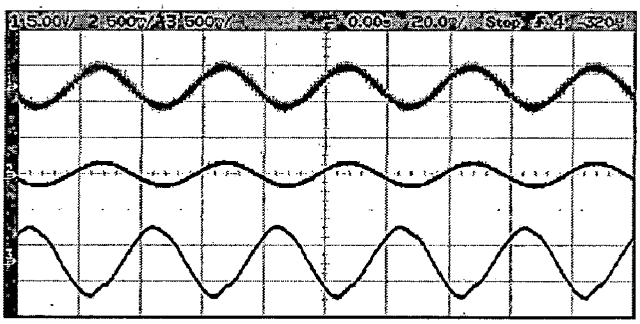
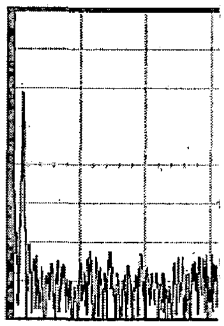
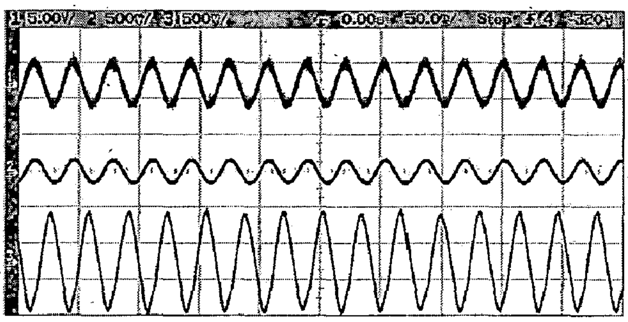
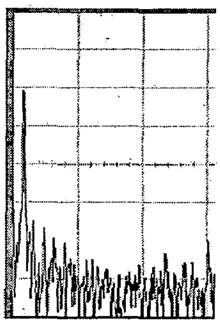
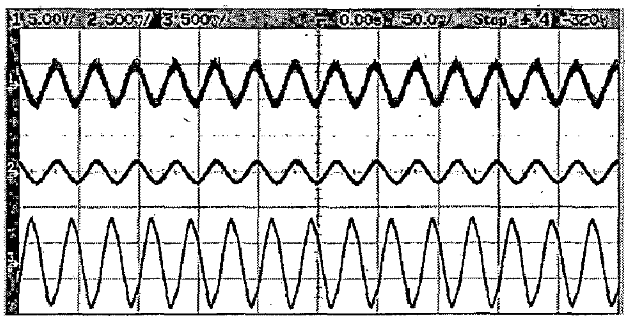
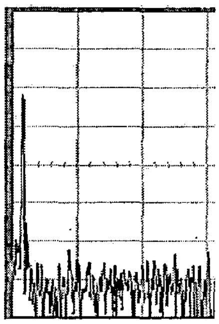
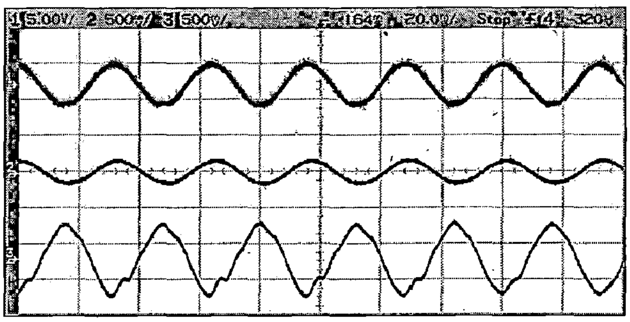
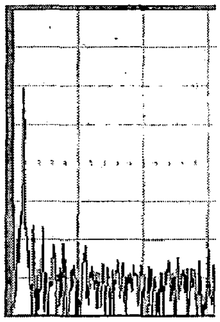
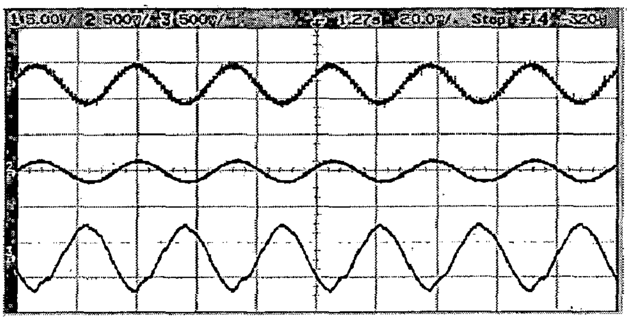
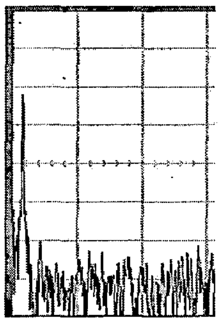
Frequency of Operation: 20 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

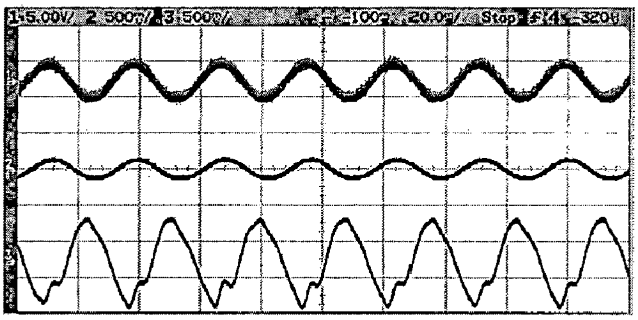
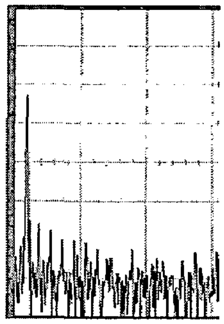
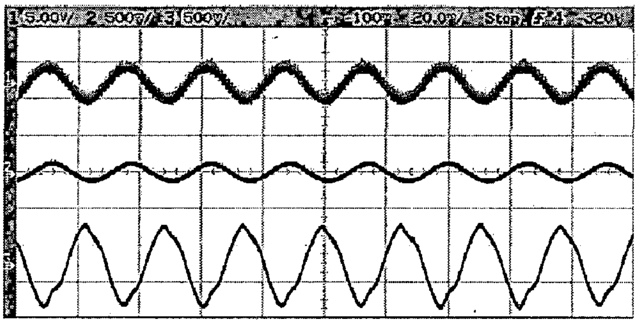
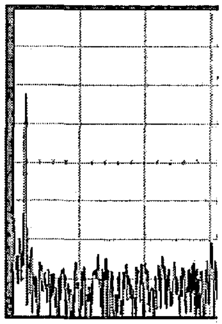
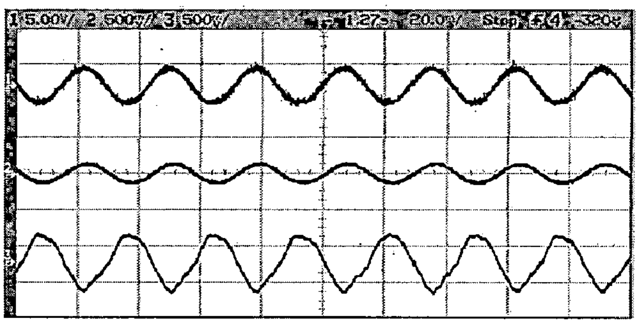
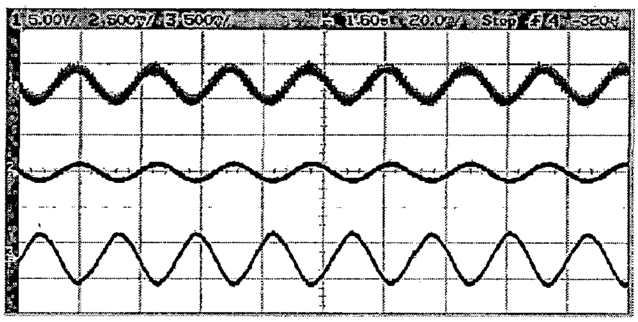
Frequency of Operation: 25 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

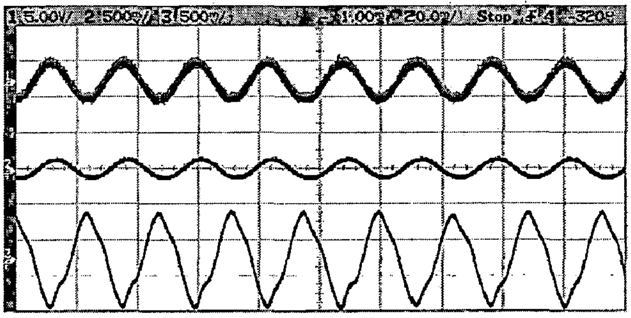
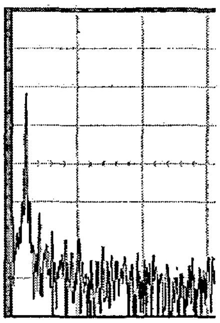
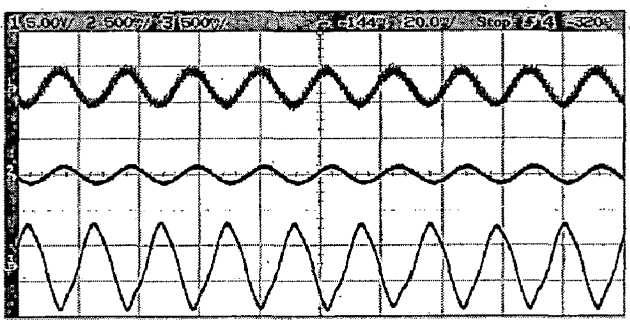
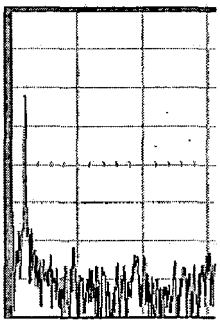
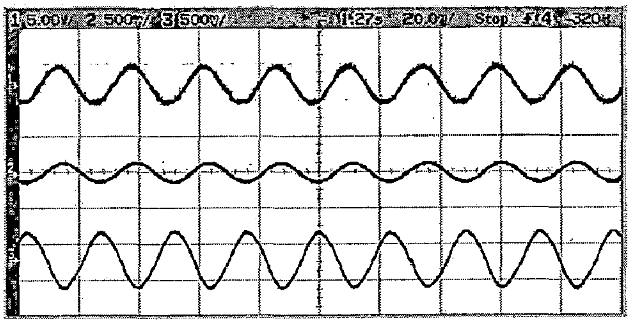
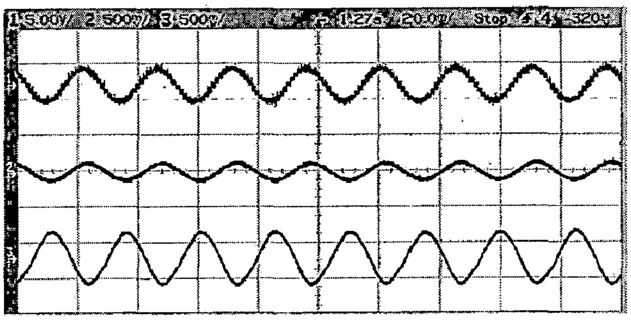
Frequency of Operation: 30 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

Frequency of Operation: 35 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

Frequency of Operation: 40 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
No Load	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
Load (m_L= 2 Kg)	Closed Loop		

Frequency of Operation: 60 Hz

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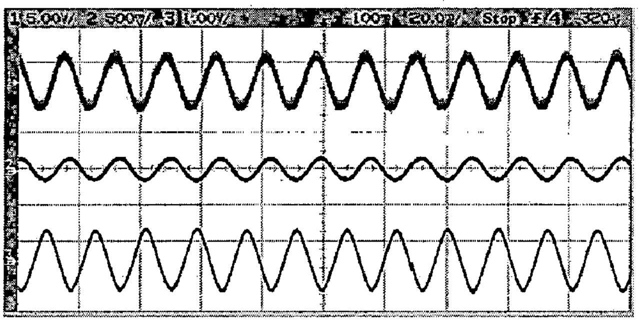
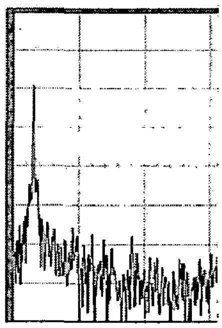
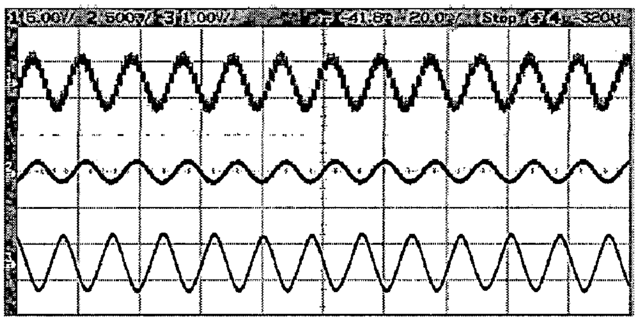
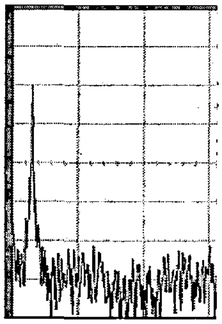
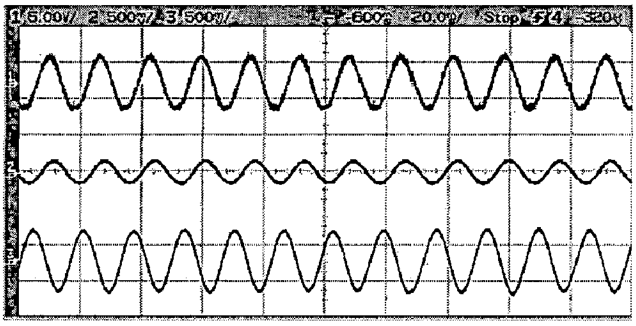
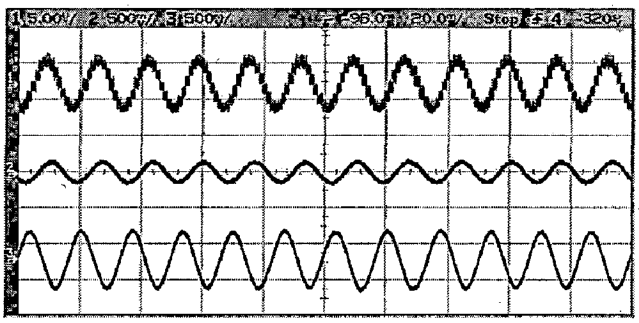
<tr><td rowspan="2">No Load</td><td>Open Loop</td><td></td><td></td></tr>
<tr><td>Closed Loop</td><td></td><td></td></tr>
<tr><td rowspan="2">Load (m_L= 2 Kg)</td><td>Open Loop</td><td></td><td></td></tr>
<tr><td>Closed Loop</td><td></td><td></td></tr>
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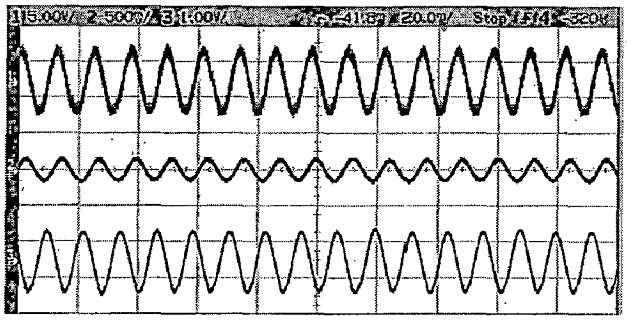
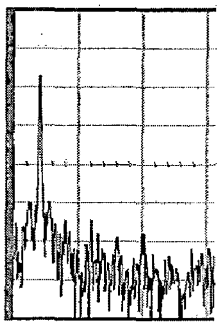
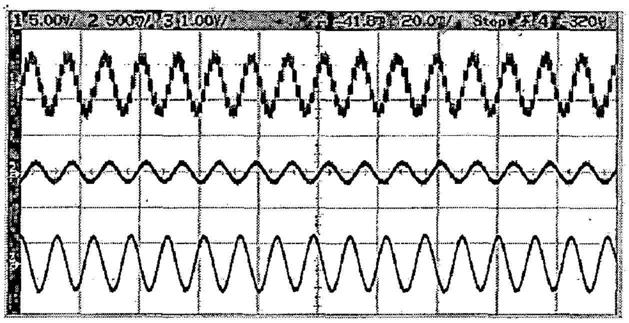
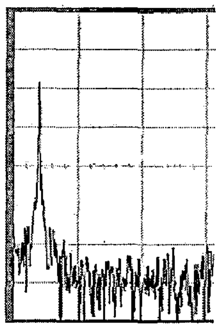
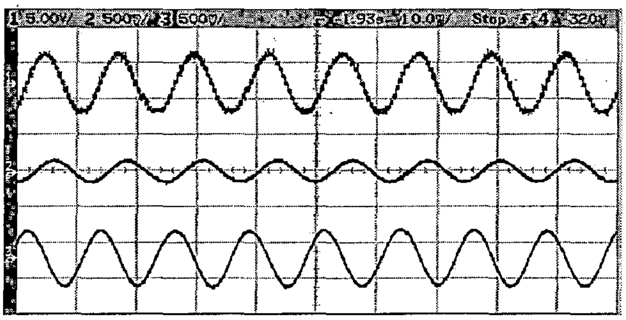
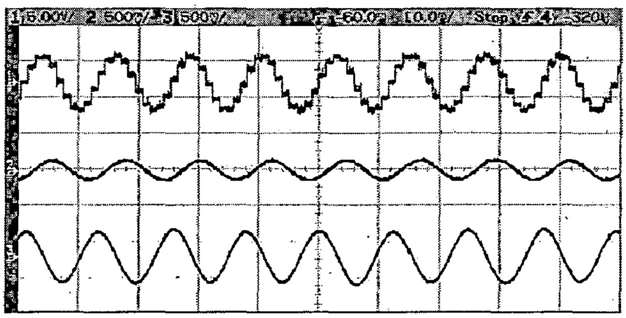
Frequency of Operation: 80 Hz

Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

Frequency of Operation: 100 Hz

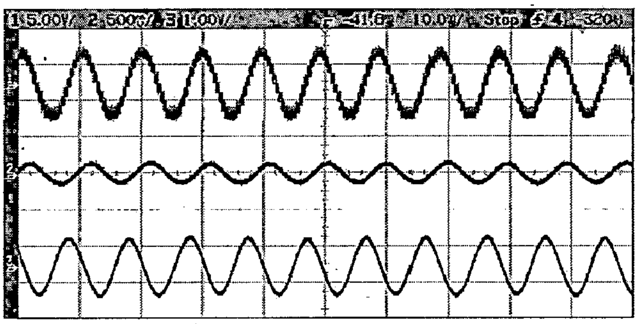
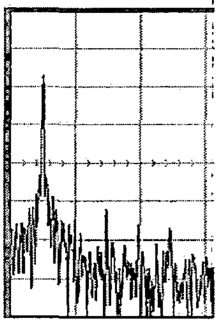
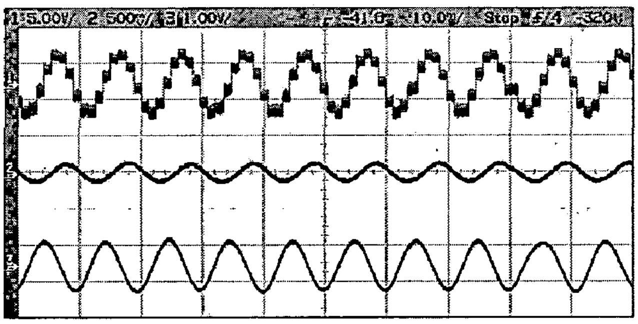
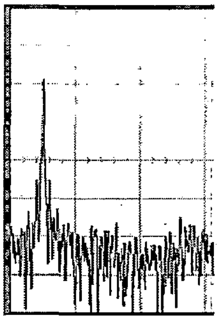
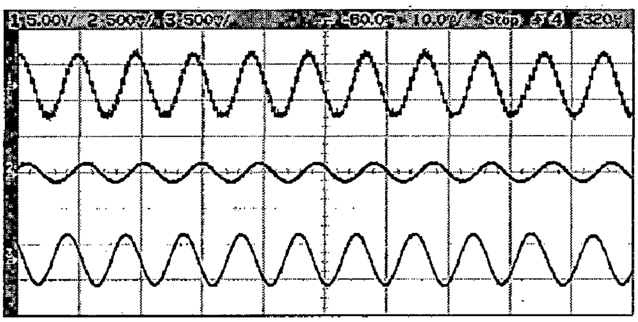
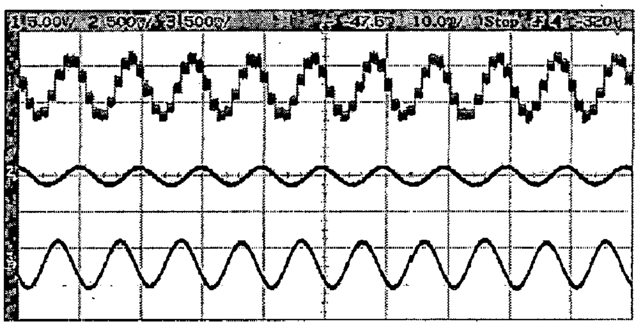
Condition		Time Response	Harmonic Analysis
No Load	Open Loop		
	Closed Loop		
Load (m_L= 2 Kg)	Open Loop		
	Closed Loop		

Table 5.2 No-Load condition experimental results with % error values

Frequency (Hz)	Reference value (V)	Actual output open-loop (V)	Actual output closed-loop (V)	Error open loop (%)	Error closed loop (%)
10	0.3181	0.2783	0.2932	-12.51	-7.83
15	0.3181	0.2856	0.3050	-10.22	-4.12
20	0.3181	0.2863	0.3082	-10.00	-3.11
25	0.4596	0.4065	0.4218	-11.55	-8.22
30	0.4596	0.4183	0.4258	-8.99	-7.35
35	0.4596	0.3965	0.4143	-13.73	-9.86
40	0.4596	0.4173	0.4367	-9.20	-4.98
60	0.5656	0.5469	0.5473	-3.31	-3.24
80	0.5656	0.5500	0.5416	-2.76	-4.24
100	0.5656	0.5490	0.5232	-2.93	-7.50

Table 5.3 Load condition experimental results with % error values

Frequency (Hz)	Reference value (V)	Actual output open-loop (V)	Actual output closed-loop (V)	Error open loop (%)	Error closed loop (%)
10	0.3181	0.2752	0.2872	-13.49	-9.71
15	0.3181	0.2829	0.2967	-11.07	-6.73
20	0.3181	0.2830	0.3005	-11.03	-5.53
25	0.4596	0.4012	0.4100	-12.71	-10.79
30	0.4596	0.4145	0.4224	-9.81	-8.09
35	0.4596	0.3937	0.4079	-14.34	-11.25
40	0.4596	0.4122	0.4256	-10.31	-7.40
60	0.5656	0.5428	0.5395	-4.03	-4.61
80	0.5656	0.5472	0.5409	-3.25	-4.37
100	0.5656	0.5443	0.5128	-3.77	-9.34

From the results obtained above the following conclusions have been drawn

- There is always an error in the system due to the error in the mathematical model calculated for the shaker system.
- The error increases for the case of open-loop system as expected and implementation of the controller reduces this error.
- Smoothening of the waveform is readily seen in the harmonic analysis.
- The performance of the closed loop system at higher frequencies is not as good as its performance in the lower frequency range. This is due to the limitation of the card which leads to reduction in the peak value of generated sinusoidal input wave.

Thus a controller for sinusoidal acceleration control of the electrodynamic shaker was implemented in real-time. The control scheme was tested under no-load and loaded condition. The results showed an improvement in the response of the system in terms of magnitude improvement and suppression of harmonics after the implementation of the controller.

CRITICAL ANALYSIS OF THE WORK

This section deals with the limitations of this work

- The mathematical model of the ElectroDynamic Shaker has been obtained after the practical experimentation. The model so obtained doesn't exactly coincide with the practical data. Also the model has been obtained for a frequency range of 250 Hz. The exact mathematical model of the ElectroDynamic Shaker has been explained in [2, 3]. The effect of the other poles and zeros has been effectively neglected, suggesting a change in the dynamics of the system. This error poses a limitation to determine the exact input sinusoidal reference waveform magnitude. Since this magnitude is essential, the output of the system is not exactly the desired value.
- Tuning of PI controller is not very accurate.
- A linear power amplifier has been used in the experimentation, which has the inherent drawbacks of low conversion efficiency, higher distortion, larger size and weight.
- ADLINK PCI-9112 DAQ card which has been used in this work has its own limitations like output frequency does not coincide with the exact desired frequency, stepped sine wave output of DAC with increase in frequency.
- The whole control algorithm has been implemented in PC/MATLAB, use of DSP kits would have speeded up the execution time and will be more cost-effective.
- The band-pass filter circuit implemented does not have the exact desired cut-off frequencies due to variability of the various passive circuit elements used in the circuit.
- Unwanted noise signals are always picked by the DSO which leads to inherent measurement errors.
- The fixturing of load is an important factor. There should be no relative velocity between the moving table and the mass. Though all efforts were made to eliminate this, still a small error would have crept.
- The available accelerometer has been designed to operate only till 240 Hz at a room temperature of 23°C.

FUTURE SCOPE

Some of the thrust areas on which focus can be given are listed below

- The phase difference factor has not been taken into consideration; if it is taken into consideration and control action is taken a better output response can be obtained.
- The same algorithm can be implemented in DSP kits for cost-effectiveness and also complex algorithms can be implemented for speedier operation.
- The PI controller can be replaced using other types of controllers (PID, fuzzy, neural) and the output response of the system should be checked under these conditions.
- Acceleration control schemes can also be implemented for random and shock vibration testing schemes.

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