Study of Vibration Transmissibility of Operational Industrial Machines

Sindhura Chilakapati
Sri Lakshmi Jyothirmai Mamidala

Department of Applied Signal Processing
Blekinge Institute of Technology
SE–371 79 Karlskrona, Sweden
This thesis is submitted to the Department of Applied Signal Processing at Blekinge Institute of Technology in partial fulfilment of the requirements for the degree of Master of Sciences in Electrical Engineering with emphasis on Signal Processing.

Contact Information:
Authors:
Sindhura Chilakapati
E-mail: sich15@student.bth.se
Sri Lakshmi Jyothirmai Mamidala
E-mail: srma15@student.bth.se

University advisor:
Imran Khan
Department of Signal Processing

University Examiner:
Sven Johansson
Department of Signal Processing

Department of Applied Signal Processing
Blekinge Institute of Technology
SE–371 79 Karlskrona, Sweden
Internet : www.bth.se
Phone : +46 455 38 50 00
Fax : +46 455 38 50 57
Abstract

Industrial machines during their operation generate vibration due to dynamic forces acting on the machines. This vibration may create noise, abrasion in the machine parts, mechanical fatigue, degrade performance, transfer to other machines via floor or walls and may cause complete shutdown of the machine. To limit the vibration pre-installation, vibration isolation measures are usually employed in workshops and industrial units. However, such vibration isolation may not be sufficient due to varying operating and physical conditions, such as machine ageing, structural changes and new installations etc. Therefore, it is important to assess the quantity of vibration generated and transmitted during true operating conditions.

The thesis work is aimed at the estimation of vibrational transmissibility or transfer from industrial machines to floor and to other adjacent installed machines. This study of transmissibility is based on the measurement and analysis of various spectral estimation tools such as Power Spectral Density (PSD), Frequency Response Function (FRF) and Coherence Function. The overall study is divided into three major steps. Firstly, the initial measurements are carried in BTH on simple Single Degree of Freedom (SDOF) systems to gain confidence in measurement and analysis. Then the measurements are performed on a Lathe machine “Quick Turn Nexus 300-II” in a laboratory at BTH. Finally, the measurements are taken on the machines of an Industrial workshop (KOSAB). The analysis results revealed that vibration measurements in industry are challenging and not easy as measurement in labs. Measurements are contaminated by noise from other machines, which degrade the coherence function. However, vibration transferred from one machine to the floor or other machines may be studied using FRF and PSD. Appropriate further isolations may be employed based on the spectral analysis.

Keywords: Noise and Vibration, Spectral estimation, Vibration isolation, Vibration transfer.
On the very outset of this report, we would like to express our sincere gratitude to our supervisor Mr. Imran Khan for introducing us to the topic and for the valuable expert advice throughout the work. Furthermore, we would like to thank our examiner, Prof. Sven Johansson for his useful comments and remarks through the learning process of this master thesis. We also extend our sincere gratitude towards Prof. Lars Hakansson for his valuable guidance and suggestions. The Department of Signal Processing has provided the support and equipment which made our thesis work complete and productive.

We would also extend our gratitude to “KOSAB”, a largest manufacturer of electrodes and wear materials, Olofström, Sweden for their cooperation in letting us experiment on their machines and gain practical experience.

We are ineffably indebted to our family members and relatives for their everlasting love and support throughout the journey of our studies. Finally, we would like to thank one and all, who might have their direct or indirect contribution in completion of our thesis.

Thank you all.
## Contents

Abstract i  
Acknowledgments ii  

1 Introduction 1  
1.1 Background .................................. 1  
1.1.1 Vibration .................................. 1  
1.1.2 Vibration Transfer ......................... 2  
1.1.3 Vibration Isolation Techniques ............... 2  
1.1.4 Vibration Monitoring ....................... 2  
1.2 Motivation and Scope ......................... 3  
1.3 Requirements for the methodology ............... 3  
1.4 Research Questions .......................... 4  
1.5 Measurement Methodology ..................... 4  
1.6 Applications of vibration measurements ........... 5  
1.7 Thesis Organization .......................... 5  

2 Theoretical Framework 6  
2.1 Literature Review ................................ 6  
2.2 Vibrations ................................... 8  
2.2.1 Classification ................................ 8  
2.2.2 Models of systems ........................... 8  
2.2.3 Industrial Vibration Sensors ................. 9  
2.3 Functions used in Spectral Analysis .......... .... 11  
2.3.1 Spectral Density Estimation ................. 11  
2.3.2 Frequency Response Function ............... 13  
2.3.3 Coherence .................................. 15  
2.3.4 Errors ..................................... 17  

3 Methodology 18  
3.1 Set-up 1: ..................................... 18  
3.1.1 Hammer Excitation: .......................... 19  
3.1.2 Shaker Setup ................................ 20  
3.2 Set-up 2: ..................................... 21
3.3 Set-up 3: ................................ 23

4 Results and Analysis 25
4.1 Analysis of Set-up 1: .......................... 25
4.2 Analysis of Set-up 2: .......................... 31
  4.2.1 Measurement 1 ........................... 32
  4.2.2 Measurement 2 ........................... 35
  4.2.3 Measurement 3 ........................... 39
  4.2.4 Measurement 4 ........................... 43
  4.2.5 Power Spectral Densities for different measurements .... 47
  4.2.6 Frequency Response Function ................. 49
  4.2.7 Coherence ................................ 50
4.3 Set-up 3 .................................. 52
  4.3.1 State 1 ................................... 52
  4.3.2 State 2 ................................... 59
  4.3.3 State 3 ................................... 66
  4.3.4 State 4 ................................... 72
  4.3.5 State 5: ................................... 78

5 Conclusions and Future Work 85
  5.1 Set-up 1: .................................. 85
  5.2 Set-up 2: .................................. 85
  5.3 Set-up 3: .................................. 86
  5.4 Other factors affecting Coherence .................. 86
  5.5 Summary ................................... 87
  5.6 Future works ................................ 88

References 89

Appendices 92

A Machine used in Set-up 2 93
B Machine used in Set-up 2 95

APPENDICES 96
List of Figures

1.1 Schematic representation of thesis work .......... 4
2.1 Dependency Factors of Errors .................... 17
3.1 Different Set-ups in the Methodology ............... 18
3.2 Block diagram of the Hammer Excitation System .... 19
3.3 Block diagram of the Shaker Excitation System ..... 20
3.4 Block diagram of Set-up 2 ...................... 22
3.5 Block diagram of Set-up 3 ...................... 23
4.1 Force signal and Acceleration signal from hammer .... 26
4.2 Force signal and Acceleration signal from Shaker .... 27
4.3 Energy Spectral Density of force signal (upper plot) and acceleration signal (lower plot) for Hammer .... 28
4.4 Power Spectral Density of force signal (upper plot) and acceleration signal (lower plot) for shaker .......... 29
4.5 Frequency Response Function for hammer and shaker excitation systems ................................ 30
4.6 Comparison of coherence between Hammer and Shaker .... 31
4.7 Acceleration signal from the foot 4 in BTH lab .......... 32
4.8 Acceleration signal from the floor near foot 4 in BTH lab .... 33
4.9 Acceleration signal from the foot 7 in BTH lab .......... 34
4.10 Acceleration signal from the floor near foot 7 in BTH lab .... 35
4.11 Acceleration signal from the foot 4 in BTH lab .......... 36
4.12 Acceleration signal from the floor 4 in BTH lab .......... 37
4.13 Acceleration signal from the foot 6 in BTH lab .......... 38
4.14 Acceleration signal from the floor near foot 6 in BTH lab .... 39
4.15 Acceleration signal from the foot 4 in BTH lab .......... 40
4.16 Acceleration signal from the floor near foot 4 in BTH lab .... 41
4.17 Acceleration signal from the foot 5 in BTH lab .......... 42
4.18 Acceleration signal from the floor near foot 5 in BTH lab .... 43
4.19 Acceleration signal from the foot 3 in BTH lab .......... 44
4.20 Acceleration signal from the floor near foot 3 in BTH lab .... 45
4.21 Acceleration signal from the foot 6 in BTH lab .......... 46
4.22 Acceleration signal from the floor near foot 6 in BTH lab ........ 47
4.23 Power Spectral Densities of acceleration signals from different feet of machine ............................................ 48
4.24 Power Spectral Densities of acceleration signals from floor near different feet of machine .................................. 49
4.25 FRF at different feet of machine .................................... 50
4.26 coherence at different feet of machine ............................... 51
4.27 Acceleration signal at foot 1 of machine 1 ........................ 53
4.28 Acceleration signal from floor near foot 1 of machine 1 .......... 53
4.29 Acceleration signal from foot of machine 2 ....................... 54
4.30 Acceleration signal from the floor near foot of machine 2 .......... 54
4.31 PSD of acceleration signals from foot 1 of machine 1 and foot of machine 2 .................................................. 55
4.32 PSD of acceleration signals from floor near foot 1 of machine 1 and floor near foot of machine 2 .............................. 56
4.33 Transmissibilities of acceleration signals from both machines .. 57
4.34 Coherence of acceleration signals from floor near feet of both machines ......................................................... 58
4.35 Acceleration signal from foot of machine 1 ......................... 59
4.36 Acceleration signal from the floor near foot of machine 1 .......... 60
4.37 Acceleration signal from foot of machine 2 ........................ 60
4.38 Acceleration signal from the floor near foot of machine 2 .......... 61
4.39 Power Spectral Densities of Acceleration signals at feet of both machines ......................................................... 62
4.40 Power Spectral Densities of Acceleration signals at floor near feet of both machines ............................................. 63
4.41 Transmissibilities of Acceleration signals from both machines .. 64
4.42 Coherence plot of acceleration signals from two machines ........ 65
4.43 Acceleration signal from foot 1 of machine 1 ....................... 66
4.44 Acceleration signal from floor near foot 1 of machine 1 .......... 67
4.45 Acceleration signal from foot of machine 2 ........................ 67
4.46 Acceleration signal from floor near foot of machine 2 .......... 68
4.47 Power Spectral densities of acceleration signals at foot 1 of machine 1 and foot of machine 2 ....................................... 69
4.48 Power Spectral densities of acceleration signals at floor near foot 1 of machine 1 and foot of machine 2 ............................ 69
4.49 Transmissibility acceleration signals from both machines ......... 70
4.50 Coherence plots of acceleration signals from both machines .......... 71
4.51 Acceleration signal from foot 2 of machine 1 ........................ 72
4.52 Acceleration signal from floor near foot 2 of machine 1 .......... 73
4.53 Acceleration signal from foot of machine 2 ........................ 73
4.54 Acceleration signal from floor from foot of machine 2 ............ 74
4.55 Power Spectral Density of acceleration signals at foot 2 of machine 1 and foot of machine 2 ........................................ 75
4.56 Power Spectral Density of acceleration signals from floor near foot 1 of machine 1 and foot of machine 2 .......................... 75
4.57 Transmissibility of acceleration signals from both machines ........................................ 76
4.58 Coherence function of acceleration signals from both machines) ........................................ 77
4.59 Acceleration signal from foot 2 of machine 1 ........................................ 78
4.60 Acceleration signal from floor near foot 2 of machine 1 ........................................ 79
4.61 Acceleration signal from foot of machine 2 ........................................ 79
4.62 Acceleration signal from floor near foot of machine 2 ........................................ 80
4.63 PSD of acceleration signals from foot 2 of machine 1 and foot of machine 2 ........................................ 81
4.64 PSD of acceleration signals from floor near foot 2 of machine 1 and foot of machine 2 ........................................ 81
4.65 Transmissibility of acceleration signals from both machines ........................................ 82
4.66 Coherence of acceleration signals from both machines ........................................ 83
5.1 Schematic representation of Factors affecting Coherence .................... 87
5.2 Summary of the Conclusions from various models ........................................ 87
A.1 Mazak Quick Turn Nexus 300-II machine ........................................ 93
A.2 Picture of the machine in SvarLab ........................................ 94
B.1 MAZAK Intetrex 200-III S (Machine 1) ........................................ 95
B.2 MAZAK Quick Turn 10 (Machine 2) ........................................ 96

vii
List of Tables

3.1.1 Specifications of Hammer Set-up .......................................... 19
3.1.2 Specifications of Shaker set-up ........................................... 21
3.2.1 Specifications of set-up 2 .................................................. 23
3.3.1 Specifications of Set-up 3 .................................................. 24
3.3.2 Different states of machine .................................................. 24
4.1.1 Specifications used in Spectral Analysis ................................ 28
4.2.1 Specifications used in Spectral Analysis of Set-up 2 .................. 32
4.3.1 Specifications used in spectral analysis of Set-up 3 ................. 52
A.0.1 Specifications of machine ................................................... 94
Chapter 1

Introduction

1.1 Background

1.1.1 Vibration

Vibration is the movement produced when a body describes an oscillatory motion about a reference position [1]. It can be periodic like motion of a pendulum or random like a movement of tires on a road. The motion might comprise of a single component occurring at a particular frequency or of several components occurring at several frequencies. Vibration components at different frequencies may be revealed by plotting the vibration amplitude against frequency [2]. Vibrations in machines can be a result of combination of different conditions. A centrifugal force created due to unbalanced rotation of weights around the machine’s axis results in vibrations. They are also caused due to angular misalignment occurred when machine shafts are out of line. Wear and tear of the components like ball bearings, drive belts, gears etc. might also result in vibrations. Loose attachments of bearings to its mounts also results in large vibrations[3].

Noise and Vibration contribute a great part in the present academic scenario. They can be found in different disciplines like mechanics, civil engineering, industrial heave-duty process pumps etc. Noise and Vibrations can be analyzed in several ways. Analytical analysis can be commonly done using Finite Element Method (FEM) [4]. For successful model of vibrations, greater detailed models are to be used. Finite Element Method with dynamic analysis needs information about the boundary conditions of the system. Acoustic analysis can be done using acoustic FEM, as long as the cavity is comprised of the noise. Boundary Element Method (BEM) can be used when radiation problems come into light [4]. Acoustic field is built up and the sound is radiated using the existing and known vibration patterns[5].
1.1.2 Vibration Transfer

For rotating motors, the unbalanced force produced results in displacement of the motor. Since the rotating speed of the motor is very high, the displacement caused is very small. As the mass rotates, the direction of force changes. When the vibrating motor is mounted to another object, it tries to move that object too. This means when the motor is mounted to an object, it acts as a single system. This occurs only in the case of rigid materials with secure mounting.

If the motor is mounted on a flexible material like foam, displacement is relatively less. If a material is compressible, a part of the vibration produced in the motor is absorbed. This means that the entire vibration from the motor is not transferred to the object.

1.1.3 Vibration Isolation Techniques

Vibration isolation is the technique of isolating the equipment from the vibrating object or the source of vibrations. Since vibration is undesirable in many disciplines, certain methods have been evolved to avert the transfer of vibrations to such systems. Vibration propagation is efficiently occurred via certain mechanical waves i.e longitudinal, lateral or flexural waves. To absorb these mechanical waves, Passive vibration isolation techniques like mechanical spring dampers, negative-stiffness isolators, tuned mass dampers etc. are utilized. Physical factors such as dimensions, weight, movement of the object, operating environment, nature of vibrations and cost of providing the isolation, influence the selection of isolation technique [5]. Active vibration isolation technique contains a feedback or feed-forward mechanism involving sensors that creates a destructive interference which can cancel out the incoming vibrations [6].

1.1.4 Vibration Monitoring

Industrial machines such as computer and numerical controlled (CNC) Lathe machine, power generators, compressors, turning and milling machines etc. produce vibration during operation. The vibration may transfer to the floor and other parts of the building and also disturb nearby machines. Hence, vibration monitoring and analysis of such machines during operation in any industry is very obvious [7].

In modern technology, Vibrations monitoring and analysis is one of the important tool to safeguard machines. It is a widely used technique in condition
Chapter 1. Introduction

Based maintenance. It is based on the information content provided by the machine vibration signals that stands as an indicator of machine condition used for diagnosis of machinery faults. This technique has been widely used for detecting and monitoring incipient and severe machinery faults in the parts like bearings, shafts, couplings, motors etc. The prime notion of Vibration measurement is predictive maintenance i.e. to help estimate the condition of in-service equipment. This tool can detect vibration levels, thereby assess their condition and gives a lot of information with relevance to fault conditions in different types of machines. This enables to reduce unnecessary downtime and failures, predicts the lifetime of the equipment and helps to plan repairs. The optimal notion of vibration analysis is to examine the rotating machinery to detect the problems and to suppress the large-scale problems. This practice is commonly used in strategic maintenance systems. Its main objective is to develop and sustain a highly productive and safe working environment. The maintenance practices and strategies vary for different enterprises [8].

1.2 Motivation and Scope

Vibration reduction and isolation is an essential part of machine operation. In this Master Thesis work vibration transferred from operational industrial machines to floor and other machines will be studied. The main aim is to characterize the amount of vibration transferred and see whether the existing vibration isolation is sufficient. Based on the study the industry may take adequate measures to ensure the desired vibration isolation.

Isolating the machinery reduces the amount of noise and vibration transmitted to the structure in which machine is sheltered or to the surrounding machines in that structure. For an isolation material to be effective, it gives the machine relatively more freedom of motion than the machine installed with no isolation [9].

1.3 Requirements for the methodology

- NI Signal Express 2014
- MATLAB 2013a or higher version with Signal Processing Tool Box, Image Processing toolbox and Parallel Processing Toolbox.
1.4 Research Questions

1. How to perform measurement of Vibration and analysis of a machine?

2. What is the effect of measured vibrations on the floor and other nearby installed machine(s)?

1.5 Measurement Methodology

The vibration will be measured as acceleration using piezo electric ICP type accelerometers. The data acquisition is performed via NI DAQ and National instruments Signal Express 2014. The analysis is performed in MATLAB [10]. The overall study is divided into three steps i.e. simple models, with initial measurements carried in BTH to gain confidence in measurement.

![Figure 1.1: Schematic representation of thesis work](image)

In the basic model, the machine and floor will be excited by other excitation sources such as Impulse hammer or Shaker to study the vibration transfer. After studying the simple systems, the vibration transferred from a CNC machine (MAZAK Quick Turn Nexus 300-II) in BTH is studied. Finally, vibration measurements are performed in a workshop "KOSAB" where the measurements are taken from two CNC machines (MAZAK Intetrex 200-III S and MAZAK Quick Turn 10) are taken simultaneously.
1.6 Applications of vibration measurements

• Estimation of the level of existing isolation
• Estimation of the remaining lifetime of the equipment
• Estimation of amount of vibration transferred
• Estimation of faults in the machinery
• Improving the overall ability of the system
• Increasing the reliability and minimizing the need for maintenance
• Guaranteeing the safety of working personnel
• Making sure that it has tolerable impact on the environment

1.7 Thesis Organization

• Chapter 1 gives brief introduction about the thesis, its methodology and its applications.
• Chapter 2 gives the knowledge about the theoretical framework involved with the methodology.
• Chapter 3 shows the various models implemented in this thesis. Analysis of various measurements are detailed in the Chapter 4.
• The respective conclusions drawn are then specified in Chapter 5.
Chapter 2

Theoretical Framework

2.1 Literature Review

The measurement and analysis of dynamic vibration implicates the sensors called accelerometers to measure the vibration. A data collector or a dynamic signal analyzer is involved to collect the data. This Dynamic Vibration Analysis provided a proven technology to detect failures and estimate the reliability of the machine [11]. When high vibrations are measured, the cause of vibration can be detected and terminated. In this methodology, the damage and failure can be prevented by shutting down the machine.

The measurement of velocity of vibrations and its analysis has got importance in diagnosing the faults in the machines, bearings, and gears etc. The periodic vibration analysis is also useful for monitoring the overall conditions of the machines with time. Although the faults may be diagnosed by analysing acoustic waveforms, temperature variations, oil and stress wave forms [12]. But due to ease of measurement and analysis, vibration measurement is the most commonly used method in diagnosing faults of machines.

Numerous methods for the vibration measurement of rotating machines are reported[12]. Firstly, the frequency domain approach is one of the fundamental approaches for vibration measurement, but the method is not suitable for analysing the fast transient signals of vibrations. In the methods based on time-domain analysis, the vibration measurement consists of some statistical indexes like, rms value and peak factor, but most of these indexes are sensitive to operating conditions and noise. Later on, an improved method for vibration measurement and monitoring with smart sensing unit is used but the instrumentation is very complex and costly. Also, synchro and a fast rotating magnetic field (RMF) are used to generate an emf in the rotor circuit of a synchro to measure vibration. The prediction of the vibration behaviour, and the sound radiated from electrical machines require the accurate determination of the resonant frequencies and exciting radial-forces. So, laboratory-techniques for the measurement of resonant frequencies, vibrations and noise based on digital processing of sig-
nals are introduced [13]. The method uses processing of different signals required from several transducers with the help of a data acquisition system and a personal computer. The measurement set-up is also used to determine the electromagnetic excitation-forces, which produce vibrations in a machine.

Later on, a kind of nine accelerometers allocation scheme containing redundant information is proposed and the corresponding formulae are presented [14]. Then, with the method of using linear accelerometers to measure six degrees-of-freedom (DOF) acceleration, the Kalman filter algorithm is applied for the processing of the acceleration signal [15].

The experiment consisted in measuring the accelerations of the floor is done by taking force induced by the test persons and measurement with the help of accelerometers. This data was processed by the computer code Accelero and the overall weighted accelerations were obtained. These were evaluated according to the former ISO 2631-2:1989 and a method proposed by the researchers Toratti and Talja [16]. Using the direct measurements, the natural frequencies of the floor were obtained.

Also, to understand the vibration generation characteristics of floor structures, FFT analysis is conducted to measure the structures of surroundings [17].

Machines with high accuracy that are sensitive to ground vibrations are generally designed using crude assumptions on the dynamic properties of the floor where they are placed. The effect of dynamic coupling between floor dynamics and machine dynamics is considered here. A new Transfer Path Analysis is demonstrated based on the Frequency Based Sub-structuring technique for the case that ground vibration levels are measured for free interface conditions. In this method, the disturbance vibrations have been measured in fixed interface conditions (so-called blocked forces or equivalent forces). After proper coupling of the machine model with the experimental characteristics of the floor dynamics, these ground vibrations are translated into machine vibrations[18].

Generally, A large range of transducers such as inductive transducers and piezo-electric transducers are used to measure vibrations. An electric signal proportional to oscillating velocity of motion is generated by Inductive transducers. A signal proportional to motion of acceleration is generated by Piezo-electric transducers [3].
2.2 Vibrations

Any motion that repeats itself after an interval of time is called vibration or oscillation [1]. The theory of vibration deals with the study of oscillatory motions of bodies and the forces associated with them. The vibrations produced can be desirable, as in certain types of machine tools or production lines. But in most cases, the vibrations of mechanical systems are undesirable as it reduces efficiency, wastes energy and may be harmful or even dangerous. The vibration dynamics of point mass and other basic simple models represent some of the real life mechanical systems. In case of manufacturing products, improvement of productivity is required in which we need to run big manufacturing machines fast. And this fast movement of big machines usually generates more vibrations.

2.2.1 Classification

Depending on whether there is an external force or not, the classification of mechanical vibrations is done as free vibration and forced vibration.

**Free Vibration:**
If a system, after an initial disturbance, is left to vibrate on its own, the ensuing vibration is known as free vibration [19]. Here, no external force acts on the system. One of the example is oscillation of simple pendulum. In free vibration, the mechanical system will oscillate with its natural frequency and eventually go down to zero due to damping effects.

**Forced Vibration:**
If a system is subjected to an external force, the resulting vibration can be called as forced vibration. These oscillations can be observed in machines such as diesel engines. If the frequency of external force coincides with one of natural frequencies of system, then resonance occurs and systems undergoes large oscillations[19].

2.2.2 Models of systems

To gain a complete understanding of the vibration produced or transferred a study of the dynamic and structural properties of the underlying system is necessary. The system can be modeled as lumped (discrete) or distributed (continuous). Thereby, the lumped systems are focused where complex structures or systems are represented as a number of interconnected simple systems[5].

The minimum number of independent coordinates required to determine com-
pletely the positions of all parts of a system at any instant of time defines the number of degrees of freedom of the system\[13\]. Most often the number of masses are also referred to as DOF (Degree Of Freedom) when measurements are done in a single coordinate. Depending on the ‘Degree Of Freedom’, the system can be single degree of freedom of system [SDOF] or Two degree of freedom systems or Multi degree of freedom of system[MDOF] \[20\].

2.2.3 Industrial Vibration Sensors

The vibration analysis requires the measurement and analysis of rotating machines utilizing different vibration sensors such as accelerometers, velocity transducers, or displacement probes. The mostly used sensor in industry is accelerometer. The accelerometer, cable, connector, and mounting method are chosen differently for each application so that the quality measurements and accurate vibration data are obtained for the further analysis. The other sensor which is used is displacement transducer which is similar to accelerometer, but outputs an electric signal proportional to its displacement. Displacement transducers behave better at low frequencies \[21\].

Advantages of accelerometers:

Accelerometers are full-contact transducers typically mounted directly on high-frequency elements, such as rolling-element bearings, gearboxes, or spinning blades. These versatile sensors can also be used in shock measurements such as in explosions and failure tests. Also used in slower, low-frequency vibration measurements. The other benefits of an accelerometer include linearity over a wide frequency range and a large dynamic range.

Types of accelerometers:

Classification of accelerometers is done accordingly whether the accelerometer is charge type or voltage type. Charge amplifier is a charge/voltage converter which converts charge output of accelerometer to voltage with low impedance. It operates being supplied current by measurement equipment with a constant current source. The output signal is obtained superposed on the current supply \[22\].

Another type of classification is done depending on purpose. Most manufacturers have a wide range of accelerometers. A small group of "general purpose" types will satisfy most of the needs. These are available with either top or side mounted connectors and have sensitivities in the range 1 to 10 mV or pC per m/s². The remaining accelerometers have their characteristics slanted towards a particular application. For example, small size accelerometers that are intended
for high level or high frequency measurements and for use on delicate structures, panels, etc. and which weigh only 0.5 to 2 grams.

Other special purpose types are optimized for: simultaneous measurement in three mutually perpendicular planes; high temperatures; very low vibration levels; high level shocks; calibration of other accelerometers by comparison; and for permanent monitoring on industrial machines.

**Piezo-electric sensor:**

Generally, vibration is measured using a ceramic piezoelectric sensor or accelerometer. Mostly the accelerometers work on the principle of the piezoelectric effect, which occurs when a voltage is generated across certain types of crystals as they are stressed. The acceleration of the test structure is transmitted to a seismic mass inside the accelerometer that generates a proportional force on the piezoelectric crystal. This external stress on the crystal then generates a high-impedance, electrical charge proportional to the applied force and, thus, proportional to the acceleration.

Piezoelectric or charge mode accelerometers require an external amplifier or in-line charge converter to amplify the generated charge, lower the output impedance for compatibility with measurement devices. They also require to minimize susceptibility to external noise sources and crosstalk. Other accelerometers have a charge-sensitive amplifier built inside them. This amplifier accepts a constant current source and varies its impedance with respect to a varying charge on the piezoelectric crystal. These sensors are referred to as Integrated Electronic Piezoelectric (IEPE) sensors. Measurement hardware for these types of accelerometers provide built in current excitation for the amplifier. It exhibits better all-round characteristics than any other type of vibration transducer. It has very wide frequency and dynamic ranges with good linearity throughout the ranges. It is relatively robust and reliable so that its characteristics remain stable over a long period of time. Additionally, the piezoelectric accelerometer is self-generating, so that it doesn’t need a power supply. There are no moving parts to wear out, and finally, its acceleration proportional output can be integrated to give velocity and displacement proportional signals [23].

**Signal Conditioning Requirements:**

When preparing an accelerometer to be measured properly by a DAQ device, the following conditions are seen to meet our signal conditioning requirements:

- Amplification to increase measurement resolution and improve signal to noise ratio.
Chapter 2. Theoretical Framework

- Current excitation to power the charge amplifier in IEPE sensors
- AC coupling to remove DC offset, increase resolution, and take advantage of the full range of the input device.
- Filtering to remove external, high-frequency noise.
- Proper grounding to eliminate noise from current flow between different ground potentials
- Dynamic range to measure the full amplitude range of the accelerometer.

2.3 Functions used in Spectral Analysis

One of the most widely used methods for data analysis is spectral analysis. For the analysis of vibration phenomena, which is used in characterizing the nature of the mechanical systems, these mathematical functions are used.

2.3.1 Spectral Density Estimation

Spectral density estimation (SDE) is a function which is used to estimate the spectral density of a random signal from the sequence of time samples of the signal. Spectral density characterizes the frequency content of the considered signal. It is to detect any periodicities in the data, by observing peaks at the frequencies corresponding to these periodicities.

Power Spectrum:
For periodic signals e.g. vibration in rotating machinery, Power spectrum is most commonly used. In periodic signals the power of the signal is concentrated at discrete frequencies. Power Spectrum has several applications in noise and vibration. It is ideally suited for detecting the periodic effects such as harmonic patterns in machine vibration spectra. [24].

Mathematically the power spectrum can be represented as [25],

\[
\hat{p}_{xx}^{PS}(f_k) = \frac{2}{NLU_{PS}} \sum_{l=0}^{L-1} \left| \sum_{n=0}^{N-1} x_1(n)w(n)e^{-j2\pi n f_k} \right|^2, f_k = \frac{k}{N}F_S, 0 < k \leq N/2
\]  

(2.1)

\[
\hat{p}_{xx}^{PS}(f_k) = \frac{1}{NLU_{PS}} \sum_{l=0}^{L-1} \left| \sum_{n=0}^{N-1} x_1(n)w(n)e^{-j2\pi n f_k} \right|^2, f_k = \frac{k}{N}F_S, k = 0
\]  

(2.2)

$L$ is number of periodograms, $N$ is length of periodogram,
Chapter 2. Theoretical Framework

$F_S$ is sampling frequency.

The window-dependent magnitude normalization factor is [25],

$$U_{PS} = \frac{1}{N} [\sum_{n=0}^{N-1} w(n)]^2 \quad (2.3)$$

**Power Spectral Density (PSD):**
For continuous signals like random signals where the power of the signal is distributed over all time, the power spectral density (PSD) is considered. It describes how power of a signal or time series is distributed over frequency [24]. Power spectral density function (PSD) shows the strength of the variations (energy) as a function of frequency. It shows at which frequencies variations are strong and at which frequencies variations are weak.

Considering a factory with many machines where some unwanted vibrations are present, the locating of offending machines by analyzing at PSD is possible (since it would give the frequencies of vibrations). Mathematically the power spectral density can be represented as [25],

$$\hat{p}^{PSD}_{xx}(f_k) = \frac{2}{F_s N U_{PSD}} \sum_{l=0}^{L-1} \sum_{n=0}^{N-1} x_1(n)w(n)e^{-j2\pi n \frac{k}{N}} |^2, f_k = \frac{k}{N} F_S, 0 < k \leq N/2 \quad (2.4)$$

$$\hat{p}^{PSD}_{xx}(f_k) = \frac{1}{F_s N U_{PSD}} \sum_{l=0}^{L-1} \sum_{n=0}^{N-1} x_1(n)w(n)e^{-j2\pi n \frac{k}{N}} |^2, f_k = \frac{k}{N} F_S, k = 0 \quad (2.5)$$

$L$ is number of periodograms,
$N$ is length of periodogram,
$F_S$ is sampling frequency.

The window-dependent magnitude normalization factor is [25],

$$U_{PSD} = \frac{1}{N} \sum_{n=0}^{N-1} (w(n))^2 \quad (2.6)$$

**Cross Spectral Density (CSD):**
The cross-spectral density (CSD) between any two signals $x(t)$ and $y(t)$, $S_{xy}(f)$ is given by [26],

$$S_{xy}(f) = \lim_{T \to \infty} \frac{1}{T} E[X_k^*(f,T)Y_k(f,T)] \quad (2.7)$$
where, $X_k(f, T)$ and $Y_k(f, T)$ are the Fourier Transforms of $x(t)$ and $y(t)$ over $k^{th}$ record of length $T$.

The cross-spectral density (also called as cross power spectrum) is the Fourier transform of the cross-correlation function [25] given as,

$$p_{xy}^{CPSD}(f_k) = \frac{1}{F_s N L U_{PSD}} \sum_{l=0}^{L-1} \left| \sum_{n=0}^{N-1} (x^*_l(n)y_l(n))w(n)e^{-j2\pi n k/N} \right|^2, f_k = \frac{k}{N} F_S$$

(2.8)

where $R_{xy}$ is the cross-correlation $x(t)$ of $y(t)$.

From an extension of the Wiener–Khinchin theorem, the Fourier transform of the cross-spectral density is the cross-covariance function. For discrete signals $x(n)$ and $y(n)$, the relationship between the cross-spectral density and the cross-covariance is:

$$S_{xy}(w) = \frac{1}{2\pi} \sum_{n=-\infty}^{\infty} R_{xy}(n)e^{-jwn}$$

(2.9)

**Energy Spectral Density (ESD):**

Energy spectral density is a function which describes how the energy of a signal or a time series is distributed with frequency [27]. Here, the term energy is used and thus the energy of a signal is:

$$E = \int_{-\infty}^{\infty} |x(t)|^2 dt$$

(2.10)

The energy spectral density is calculated for transients like pulse signal which have a finite total energy. In this case, Parseval’s theorem gives us an alternate expression for the energy of the signal in terms of its Fourier transform,

$$\int_{-\infty}^{\infty} |x(t)|^2 dt = \int_{-\infty}^{\infty} |X(f)|^2 df$$

(2.11)

Here $f$ is the frequency in Hz, i.e., cycles per second.

### 2.3.2 Frequency Response Function

The frequency response of a system is a frequency dependent function which expresses how a sinusoidal signal of a given frequency on the system input is transferred through the system. Frequency response function can be further defined as a mathematical relation between the input and the output of a system.
There are many tools available for performing vibration analysis and testing. The frequency response function is one of them.

In experimental modal analysis frequency response function is a frequency based measurement function which is used to identify the resonant frequencies, damping, mode shapes of a physical structure. It is the structural response to an applied force as a function of frequency. The response may be given in terms of displacement, velocity, or acceleration. It can be obtained from either measured data or analytical functions. It is an important tool for analysis and design of signal filters such as low pass filters and high pass filters, and in control systems.

Frequency response functions are complex functions, with real and imaginary components. They can be represented in terms of magnitude and phase. The concept of frequency response function is the foundation of modern experimental system analysis. A linear system such as an SDOF or an MDOF, when subjected to sinusoidal excitation, will respond sinusoidal at the same frequency and at specific amplitude that is characteristic to the frequency of excitation. The phase of the response in general case, will be different than that of the excitation. The phase difference between the response and the excitation will vary with frequency. The system does not need to be excited at one frequency at the time. The same applies if the system is subjected to a broadband excitation comprising a blend of many sinusoids at any given time, such as in the white noise from Gaussian random excitation or an impulse. To study the system response at various frequencies, the excitation and the response signals must be subjected to the DFT.

The frequency response can found experimentally or from a transfer function model. It can be presented graphically or as a mathematical function. For example, considering the frequency response function between two points on a structure. It would be possible to attach an accelerometer at a particular point and excite the structure at another point with a force gauge instrumented hammer. Then by measuring the excitation force and the response acceleration the resulting frequency response function would describe as a function of frequency the relationship between those two points on the structure.

**Mathematical description:**
The basic formula for a frequency response function is:

\[
H(f) = \frac{Y(f)}{X(f)}
\]  

(2.12)

Where: \(H(f)\) is the frequency response function, 
\(Y(f)\) is the output of the system in the frequency domain,
Chapter 2. Theoretical Framework

$X(f)$ is the input to the system in the frequency domain.

Frequency response functions are used for single input and single output analysis, and for the calculation of the $H_1(f)$ or $H_2(f)$ which are the two types of frequency response functions. These are used extensively for hammer impact analysis or resonance analysis. The $H_1(f)$ frequency response function is used in situations where the output to the system is expected to be noisy when compared to the input [28]. The $H_2(f)$ frequency response function is used in situations where the input to the system is expected to be noisy when compared to the output. $H_1(f)$ or $H_2(f)$ can be used for resonance analysis or hammer impact analysis, $H_2(f)$ is most commonly used with random excitation.

The equation of $H_1(f)$ is [5],

$$H_1(f) = \frac{\hat{G}_{yx}(f)}{\hat{G}_{xx}(f)}$$

(2.13)

Where $H_1(f)$ is the frequency response function, $\hat{G}_{yx}(f)$ is the Cross Spectral Density in the frequency domain of $x(t)$ and $y(t)$, $\hat{G}_{xx}(f)$ is the Auto Spectral Density in the frequency domain of $x(t)$.

The equation of $H_2(f)$ is [5],

$$H_2(f) = \frac{\hat{G}_{yy}(f)}{\hat{G}_{xy}(f)}$$

(2.14)

Where $H_2(f)$ is the frequency response function, $\hat{G}_{yy}(f)$ is the Cross Spectral Density in the frequency domain of $y(t)$ and $x(t)$, $\hat{G}_{xy}(f)$ is the Auto Spectral Density in the frequency domain of $y(t)$.

The frequency response function is a frequency domain analysis, therefore the input and the output to the system should be in frequency spectra. So the force and acceleration are first converted into spectra. Matlab function "tfestimate" is used for this purpose, which impalements the $H_1$ estimator.

2.3.3 Coherence

Theory:

The spectral coherence can be defined as a statistic which is used to examine the relation between two signals or data sets. It is used to estimate the power
transfer between input and output of a linear system. If the signals are ergodic, and the system function is linear, it can be used to estimate the causality between the input and output.

Mathematical Description:

The coherence (sometimes called magnitude-squared coherence) between two signals $x(t)$ and $y(t)$ ($\hat{Y}_{yx}$) is a real-valued function that is defined as [5],

$$\hat{Y}_{yx}(f) = \frac{\hat{H}_1(f)}{\hat{H}_2(f)} = \frac{|\hat{G}_{yx}(f)|^2}{\hat{G}_{yy}(f) \ast \hat{G}_{xx}(f)}$$ (2.15)

If $\hat{Y}_{yx} = 1$, then $\hat{H}_1 = \hat{H}_2$

- This implies that we have no extraneous noise, and also the measured output, $y(t)$, derives solely from the measured input, $x(t)$.

- If $C_{xy}$ is less than one but greater than zero it is an indication that either: noise is entering the measurements, that the assumed function relating $x(t)$ and $y(t)$ is not linear, or that $y(t)$ is producing output due to input $x(t)$ as well as other inputs. If the coherence is equal to zero, it is an indication that $x(t)$ and $y(t)$ are completely unrelated, considering above constraints.

- On the other hand, when $x$ and $y$ are uncorrelated, the sample coherence converges to zero at all frequencies, as the number of blocks in the average goes to infinity.

- In all the three cases, there will be a bias error in the determination of the frequency response in at least one of the estimators [5].

- Coherence function is the quality measure of estimated frequency response, regardless of estimator type. A common use for the coherence function is in the validation of input/output data collected in an acoustics experiment for purposes of system identification. For example, let us have known signal which is input to an unknown system, such as a reverberant room, say, and is the recorded response of the room. Ideally, the coherence should be one at all frequencies. If the microphone is situated at a null in the room response for some frequency, it may record mostly noise at that frequency. The coherence of a linear system therefore represents the fractional part of the output signal power that is produced by the input at that frequency. This quantity is also an estimate of the fractional power of the output that is not contributed by the input at a particular frequency.
2.3.4 Errors

Two kinds of errors come into light during analysis the vibrations. They are Random Error and the Bias Error [29].

- Constant deviation from the desired output is called Bias Error. The bias occurs due to limited frequency resolution. Practically, this error should be minimized by selecting a block size for the FFT, and there by gradually increasing the block size until peaks do not increase in height, when the block size is increased further. It turns out that the bias error, essentially depends on the ratio of the resonance bandwidth and the frequency increment [30]. Bias Error can be subtracted, but Random Error cannot be reduced.

- Random Error is the error produced when the signal is changed rapidly. The total random error therefore depends on the time window as well as number of averages and the overlap percentage.

- A trade-off can be established between the random error and bias error.

![Figure 2.1: Dependency Factors of Errors](image)

- There are two completely different errors involved in the FRF estimates. During estimation of frequency response functions using H1 and H2 estimator, these errors come into light.: Spectral analysis errors, and Model errors. The spectral errors in the FRF estimates are further divided into two parts: the errors caused by the estimator itself, without any extraneous noise and then errors caused by the extraneous noise [31].
Chapter 3

Methodology

This Chapter deals with different system set-ups that are implemented in the Experimental work. Each Set-up is discussed in Figure 3.1. The set-up of the machines are further described in Appendix A and Appendix B.

**Figure 3.1: Different Set-ups in the Methodology**

### 3.1 Set-up 1:

In this set-up, a Single Degree of Freedom (SDOF) system is excited by hammer and a shaker in the same environment. Thereby, the corresponding vibrations
are analyzed.

3.1.1 Hammer Excitation:

Block Diagram:

The block diagram of the set-up is shown in Figure 3.2:

![Block diagram of the Hammer Excitation System](image)

Figure 3.2: Block diagram of the Hammer Excitation System

Specifications:

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>Mass</td>
<td>547 grams</td>
</tr>
<tr>
<td>Accelerometer (Voltage Type)</td>
<td>Sensitivity</td>
<td>5.10 m/s^2</td>
</tr>
<tr>
<td>Hammer</td>
<td>Sensitivity</td>
<td>3.3 mV/N</td>
</tr>
</tbody>
</table>

Table 3.1.1: Specifications of Hammer Set-up
**System Setup:**

- The mass is mounted on one side of the cantilever beam and an accelerometer is mounted on the other side of cantilever beam as shown in the figure 3.2.

- The impulse hammer and the accelerometer are connected to the two channels of Data Acquisition System which is further connected to the computer.

- The system is excited on the mass with a hammer strike.

- The input data (force) and output data (acceleration) are collected at the respective channels of data acquisition unit using MATLAB.

### 3.1.2 Shaker Setup

**Block diagram:**

![Block diagram of the Shaker Excitation System](image)

*Figure 3.3: Block diagram of the Shaker Excitation System*
 Specifications:

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>Mass</td>
<td>547 grams</td>
</tr>
<tr>
<td>Accelerometer (voltage type)</td>
<td>Sensitivity</td>
<td>5.10 m/s(^2)</td>
</tr>
<tr>
<td>Impedance Head</td>
<td>Sensitivity of Force</td>
<td>22.4 mV/N</td>
</tr>
<tr>
<td></td>
<td>Sensitivity of acceleration</td>
<td>10.2 mV/(m/s(^2))</td>
</tr>
</tbody>
</table>

Table 3.1.2: Specifications of Shaker set-up

System Setup:

- The mass is mounted on one side of the beam and an accelerometer is mounted on the other side of cantilever beam.
- The shaker and mass are connected along a spring. Thereby, the motion is transferred along one translational axis as shown in Figure 3.3.
- The input force from impedance head and the output from accelerometer are connected to the two channels of Data Acquisition System which is further connected to the computer.
- The shaker is driven by a Data Signal Analyser from which a random noise of level 200 mV is generated and is further amplified using an “Amplifier”.
- The data from accelerometer and impedance head is collected at the respective channels of data acquisition unit using MATLAB.

3.2 Set-up 2:

In this model, the experimental setup is implemented on a practical machine where the measurements are taken from the machine and the floor near the foot of the machine.
Block Diagram:

![Block Diagram of Set-up 2](image)

**Figure 3.4: Block diagram of Set-up 2**

**System System:**

- The system consists of a Machine with 8 feet as shown in Figure 3.4.

- Two accelerometers are placed on the foot of the machine and the floor near the foot respectively. This is repeated at two feet of the machine. (Say 3, 7).

- The data is collected from the four accelerometers using a Data Acquisition System when the machine is in running state.

- The Data Acquisition System is further connected to the Personal Computer.
Specifications:

<table>
<thead>
<tr>
<th>Amplifiers</th>
<th>Channel 1</th>
<th>Channel 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplifier-1</td>
<td>Amplitude Sensitivity 31.6 mV/(m/s^2)</td>
<td>Amplitude Sensitivity 1 V/(m/s^2)</td>
</tr>
<tr>
<td></td>
<td>Transducer Sensitivity 3.125 Pc/(m/s^2)</td>
<td>Transducer Sensitivity 3.114 Pc/(m/s^2)</td>
</tr>
<tr>
<td>Amplifier-2</td>
<td>Amplitude Sensitivity 31.6 mV/(m/s^2)</td>
<td>Amplitude Sensitivity 1 V/(m/s^2)</td>
</tr>
<tr>
<td></td>
<td>Transducer Sensitivity 1 pC/(m/s^2)</td>
<td>Transducer Sensitivity 3.091 pC/(m/s^2)</td>
</tr>
</tbody>
</table>

Table 3.2.1: Specifications of set-up 2

3.3 Set-up 3:

In this model, the experimental setup is implemented on two practical machines in a workshop. The measurements are taken from both the machines and the floor near the foot of the machines.

Block Diagram

Figure 3.5: Block diagram of Set-up 3
Chapter 3. Methodology

Specifications:

<table>
<thead>
<tr>
<th>Amplifiers</th>
<th>Channel 1</th>
<th>Channel 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplifier-1</td>
<td>Amplifier Sensitivity</td>
<td>1 V/(m/s²)</td>
</tr>
<tr>
<td>Transducer Sensitivity</td>
<td>10.2* (10^-3) V/(m/s²)</td>
<td></td>
</tr>
<tr>
<td>Amplifier-2</td>
<td>Amplifier Sensitivity</td>
<td>1 V/(m/s²)</td>
</tr>
<tr>
<td>Transducer Sensitivity</td>
<td>0.996 pC/(m/s²)</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3.1: Specifications of Set-up 3

<table>
<thead>
<tr>
<th>Type</th>
<th>Machine 1</th>
<th>Machine 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>State</td>
<td>State</td>
</tr>
<tr>
<td>1</td>
<td>Not running</td>
<td>Not running</td>
</tr>
<tr>
<td>2</td>
<td>Not running</td>
<td>Running</td>
</tr>
<tr>
<td>3</td>
<td>Running</td>
<td>2</td>
</tr>
<tr>
<td>4</td>
<td>Running</td>
<td>1</td>
</tr>
<tr>
<td>5</td>
<td>Running</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 3.3.2: Different states of machine

System Setup

- The system consists of one machine with two feet and another machine with one foot.
- Two accelerometers are placed on the foot and the floor respectively of one machine. This is repeated at the foot of the other machine.
- The charger type amplifiers used are driven using charge amplifiers.
- The data is collected from the four accelerometers using a Data Acquisition System when both the machines are in different states as shown in Table 3 4. 
- The Data Acquisition System is further connected to the Personal Computer.
Chapter 4
Results and Analysis

This Chapter deals with plotting and analysis of results obtained. Each sub-section corresponds to a particular set-up. The obtained signals are plotted and the transfer of vibration is studied accordingly.

4.1 Analysis of Set-up 1:

In this section, the plots obtained by exciting a Single Degree of Freedom (SDOF) system are studied and the vibrations transferred from the system are discussed.

Force and Acceleration signals:

The Input signal(force) and the output signal(acceleration) for the hammer and shaker excitation are respectively plotted in Figure 4.1 and Figure 4.2 respectively:
Chapter 4. Results and Analysis

Figure 4.1: Force signal and Acceleration signal from hammer

- The force and the acceleration signal can be calculated and plotted by dividing the obtained voltage signals with the sensitivity of the hammer and the sensitivity of the accelerometer.

- Force signal shown in Figure 4.1 is an impulse signal. Generally, a Force window is used for analyzing an impulse signal. In this experiment, we use a “Rectangular window” in Spectral Analysis for the purpose of averaging since the signal is transient.

- From Figure 4.1, it can be observed that the exponential decay of the response is started at the instant of hammer excitation.
Chapter 4. Results and Analysis

Figure 4.2: Force signal and Acceleration signal from Shaker

- From Figure 4.2, for the random noise as input, the response of shaker is also random in nature.

- In this experiment, we use a “Hanning window” in Spectral Analysis since the signal is random in nature.

Spectral Densities
The type of Spectral Density for the analysis can be decided based on the nature of signal.

- Since, the acceleration signal shown in Figure 4.1 is an “Exponential Decay”, the “Energy Spectral Density” is estimated for Spectral Analysis and is plotted in Figure 4.3. Since, the input signal and acceleration signal shown in Figure 4.2 is “Random” in nature, the “Power Spectral Density” is estimated for Spectral Analysis and is plotted in Figure 4.4.
Specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling Frequency ((F_s))</td>
<td>4096 Hz</td>
</tr>
<tr>
<td>Block Length ((N))</td>
<td>(2^{14})</td>
</tr>
<tr>
<td>Overlap Percentage</td>
<td>50</td>
</tr>
<tr>
<td>No. of Averages</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 4.1.1: Specifications used in Spectral Analysis

To calculate the Power Spectral Density, equation in (2.3) is used. The Energy Spectral Density can be calculated by multiplying the Power Spectral Density with the time which can be calculated as:

\[
T = \frac{\text{length of the window}}{\text{Sampling frequency (Hz)}}
\]

\[
T = 4 \text{ sec}
\]

Figure 4.3: Energy Spectral Density of force signal (upper plot) and acceleration signal (lower plot) for Hammer
Figure 4.4: Power Spectral Density of force signal (upper plot) and acceleration signal (lower plot) for shaker

- Since, the decay is approximately up to 6 dB in Figure 4.3 (upper plot), therefore the frequency range 0 Hz to 230 Hz is used.

- Similarly, since shaker excitation is present in the range 0 Hz to 300 Hz, the Power Spectra Densities are analyzed in that frequency range.

**Frequency Response Function**

The Frequency response functions of Hammer and Shaker are plotted together as follows:
Chapter 4. Results and Analysis

Figure 4.5: Frequency Response Function for hammer and shaker excitation systems

- The Resonance Frequency is the frequency at highest peak. From Figure 4.5, the Resonance Frequency can be observed at,

\[ f_r = 62 \text{Hz} \] (4.1)

- From Figure 4.5, it can be observed that the responses of Hammer and Shaker are almost the same.

- Since the system considered imitates Single Degree Of Freedom (SDOF) system, the obtained FRF plot should have one peak value. But in the set-up, the mass is attached to a cantilever beam. This might affect the behavior of the system due to which a second peak is obtained.

- Also it can be observed that, between 500-100 Hz the vibration transferred is higher and approx. 32 dB at the resonance frequency. After 100 Hz, the level of the transferred vibration is between 0 and -10 dB. Further more, both hammer and shaker results are identical.
Coherence
The coherence plots of Hammer and Shaker are plotted together and compared:

![Graph showing Coherence](image)

Figure 4.6: Comparison of coherence between Hammer and Shaker.

- The responses of Shaker and Hammer are approximately equal.
- The Coherence of hammer excitation system is slightly better when compared to shaker excitation system.
- The lower coherence are the coherence and anti-coherence. For the rest of the frequency range the coherence is close to 1, which shows high quality signal and output linearly derived from the input.

4.2 Analysis of Set-up 2:
In this section, the vibration transmissibility of a machine to the floor is studied and the respective plots are analyzed using the specifications shown below. It was not possible to excite the system with hammer or shaker. Therefore, operational forces are taken as input and output to the system. The vibration signal
measured at the foot is considered as input, while the vibration signal measured at floor position is considered as response. Instead of FRF (Response/Force), the system properties are studied from Response/Response ratio or transmissibility.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling Frequency (Fs)</td>
<td>4096 Hz</td>
</tr>
<tr>
<td>Block Length (N)</td>
<td>$2^{14}$</td>
</tr>
<tr>
<td>Overlap Percentage</td>
<td>50</td>
</tr>
<tr>
<td>No. of Averages</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 4.2.1: Specifications used in Spectral Analysis of Set-up 2

### 4.2.1 Measurement 1

In this type of measurement, the accelerometers are placed at feet 4 and 7 and the machine is in working state running at constant rpm.

**Acceleration signals**  
The acceleration signal shown in Figure 4.7 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine.

![Figure 4.7: Acceleration signal from the foot 4 in BTH lab](image)
The acceleration signal shown in Figure 4.8 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed near the floor of the foot 4 of the Machine.

The signal level of acceleration signal from foot 4 is lower when compared to acceleration signal from floor near foot 4. This might be due to many environmental factors that come into light since the measurements are taken on an real time implementation. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.
Chapter 4. Results and Analysis

34

Figure 4.9: Acceleration signal from the foot 7 in BTH lab

The acceleration signal shown in Figure 4.9 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 7 of the Machine. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.
Figure 4.10: Acceleration signal from the floor near foot 7 in BTH lab

The acceleration signal shown in Figure 4.10 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 7 of the Machine. The peaks in the acceleration signals indicates that accelerometer is at Resonance. The sudden peak might also be due to a sudden change in functionality of the machine.

4.2.2 Measurement 2

In this type of measurement, the accelerometers are placed at feet 4 and 6.
Chapter 4. Results and Analysis

Acceleration signals

Figure 4.11: Acceleration signal from the foot 4 in BTH lab

The acceleration signal shown in Figure 4.11 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine.
Chapter 4. Results and Analysis

Figure 4.12: Acceleration signal from the floor foot 4 in BTH lab

The acceleration signal shown in Figure 4.12 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.

The lower level of the acceleration signal from foot 4 when compared to signal from floor neat foot 4 might be due to the same reason as in mThe peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.easurement 1.
The acceleration signal shown in Figure 4.13 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 6 of the Machine. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.
The acceleration signal shown in Figure 4.14 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine.

The signal level of acceleration signal from foot 6 is higher when compared to acceleration signal from floor near foot 6. Since foot 6 is at the backside of the machine, this shows that the behaviour of the back part is different from front part of the machine. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.

4.2.3 Measurement 3

In this type of measurement, the accelerometers are placed at feet 4 and 5.
Chapter 4. Results and Analysis

Acceleration Signals

Figure 4.15: Acceleration signal from the foot 4 in BTH lab

The acceleration signal shown in Figure 4.15 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine.

The lower level of the acceleration signal from foot 4 when compared to signal from floor neat foot 4 might be due to the same reason as in measurement 1 and 2.
The acceleration signal shown in Figure 4.16 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 4 of the Machine.

The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.
The acceleration signal shown in Figure 4.17 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 5 of the Machine.

The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.
Figure 4.18: Acceleration signal from the floor near foot 5 in BTH lab

The acceleration signal shown in Figure 4.18 are periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 5 of the Machine.

The signal level of acceleration signal from foot 5 is higher when compared to acceleration signal from floor near foot 5. Since foot 5 is at the backside of the machine, this shows that the behaviour of the back part is different from front part of the machine. The peak in acceleration signal indicates that accelerometer is at resonance. The sudden peak might also be due to a sudden change in functionality of the machine.

4.2.4 Measurement 4

In this type of measurement, the accelerometers are placed at feet 3 and 6.
Chapter 4. Results and Analysis

Acceleration signals

Figure 4.19: Acceleration signal from the foot 3 in BTH lab

The acceleration signal shown in Figure 4.19 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 3 of the Machine.
The acceleration signal shown in Figure 4.20 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the floor near foot 3 of the Machine.

The signal level of acceleration signal from floor near foot 3 is little higher when compared to acceleration signal from foot 3.
The acceleration signal shown in Figure 4.21 is periodic in nature since the machine is rotating. This signal is collected from an accelerometer placed on the foot 6 of the Machine.
Chapter 4. Results and Analysis

4.2.5 Power Spectral Densities for different measurements

Any rotating machine produces periodic signals. Since the measurements are done in an industry, the signal is interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis for averaging. Rectangular window is chosen for making the analysis similar as Set-up 1.

The Power Spectral Densities for different acceleration signals collected from feet of the machine are shown in the Figure 4.23
Figure 4.23: Power Spectral Densities of acceleration signals from different feet of machine

The Power Spectral Densities for different acceleration signals collected from floor near the feet of machine are shown in the figure 4.24
Chapter 4. Results and Analysis

4.2.6 Frequency Response Function

The Frequency Response Functions for acceleration signals at different feet are shown in figure 4.25.
4.2.7 Coherence

The Coherence Functions for acceleration signals at different feet are shown in figure 4.26.
Conclusions from Set-up 2:

- First the coherence is not good i.e. for most of the frequency spectra the values is less than 1. This indicates the presence of noise in the measurements and the output (floor) signal is not linearly derived from the Foot signal. Nevertheless some important observations may be made on the basis of PSD and Transmissibility plots.

- From the PSD plots of acceleration signals from feet of the machine, some clear and distinct peaks at frequencies 62 Hz, 100 Hz, 160 Hz, 320 Hz and 340 Hz are visible. The acceleration spectra is similar for all the feet except Foot4. From the PSD plots of acceleration signals from floor of the machine, some clear and distinct peaks at frequencies 50 Hz, 110 Hz, 160 Hz, 320 Hz and 400 Hz are visible. The acceleration spectra is almost similar near all feet.

- From the transmissibility plot, it may be seen that the vibration transfer is higher from Foot3 to Floor3 as compared to other Feet and Floor. This may
be due to the compressor location for the machine. In this plot, MDOFs are seen but the system may be approximated as 2 DOF with resonances at 100 Hz and 320 Hz. Furthermore it may also be concluded that the transmissibility (vibration transfer) is below 0 dB or around -20 dB for most of the spectrum. However at 320 Hz and 340 Hz the transmissibility is 22 dB and 21 dB respectively.

- The behaviour of the front part (foot 3, foot 4) of the machine is different from the back part (foot 5, foot 6). This is might be due to difference in intensity of the machine vibration. There is an exceptional behavior at foot 7 which might be due to other environmental factors, noisy chords, faulty accelerometer etc.

4.3 Set-up 3

The vibration transmissibility of one machine to the floor and another machine is studied and the respective plots are analyzed with the following specifications. Here, the machine 1 has two feet and machine 2 has only one foot. Since the system was not excited by any external force a similar approach as used in setup-2 is used in this case also.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling Frequency (Fs)</td>
<td>4096 Hz</td>
</tr>
<tr>
<td>Block Length (N)</td>
<td>$2^{14}$</td>
</tr>
<tr>
<td>Overlap Percentage</td>
<td>50</td>
</tr>
<tr>
<td>No. of Averages</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 4.3.1: Specifications used in spectral analysis of Set-up 3

4.3.1 State 1

In this state, both the machines are not running. This is done to measure the background noise present when the system is not excited.

Acceleration Signals

The acceleration signals are collected from accelerometers placed at different feet and floor near the feet of both machines

The acceleration signals shown in Figure 4.27, figure 4.28, figure 4.29 and figure 4.30 are collected from an accelerometer placed on the foot 1 of Machine 1, floor near foot 1 of machine 1, foot of machine 2 and floor near foot of machine 2 respectively (machine 2 has only one foot).
Figure 4.27: Acceleration signal at foot 1 of machine 1

Figure 4.28: Acceleration signal from floor near foot 1 of machine 1
Chapter 4. Results and Analysis

Figure 4.29: Acceleration signal from foot of machine 2

Figure 4.30: Acceleration signal from the floor near foot of machine 2
Spectral Densities

Any rotating machine produces periodic signal. Since the measurements are done in an industry, the acceleration signals shown above are interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis.

The Power Spectral Densities of the acceleration signal from an accelerometer placed on the feet of both machines are shown in Figure 4.31.

![Figure 4.31: PSD of acceleration signals from foot 1 of machine 1 and foot of machine 2](image)

The Power Spectral Densities of the acceleration signal from an accelerometer placed on the floor near the feet of both machines are shown in Figure 4.32.
Figure 4.32: PSD of acceleration signals from floor near foot 1 of machine 1 and floor near foot of machine 2

The signal level is acceleration signal from floor near feet of both machines is higher when compared to the level of signal from feet since the environmental factors in the real time implementation come into picture.

Transmissibility

The transmissibility for the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.33. This is done for both the machines.
Figure 4.33: Transmissibilities of acceleration signals from both machines

Coherence

The coherence functions between the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.34. This is done for both the machines.
For both the machines, the coherence is very low (level is nearly zero). However, certain observations can be drawn from the PSD and transmissibility plots.

Since both the machines are not running, PSD of acceleration signals from the feet of the machine have approximately constant magnitude. The magnitude of PSD of machine 1 is around -60 dB and machine 2 is around -90 dB. The magnitude of PSD of machine 1 is higher when compared to machine 2. This may be due to the compressor or motor running for the machine.

From PSD plot of acceleration signals from the floor near the feet of machine, the magnitude of FRF of machine 1 and machine 2 are approximately equal around -78 dB. Slight variation in PSD can be due to the presence of background noise.

Also the PSD on the respective floors is high as compared to the feet for both the machines. This shows that some background noise is coming from other nearby working machines, which is affecting the floor more when compared to the feet.

The magnitude of transmissibility for the machine 2 is less than 0 dB except at certain frequencies. The magnitude of transmissibility for the machine 1 is far less than 0 dB. This means that the magnitude of transfer of vibration is
less. It can also be observed that the average magnitude of transmissibility is constant at most of the frequencies.

4.3.2 State 2

In this state, machine 1 is not running and machine 2 is running.

Acceleration signals

The acceleration signals are collected from accelerometers placed at different feet and floor near the feet of both machines.

The acceleration signals shown in Figure 4.35, figure 4.36, figure 4.37 and figure 4.38 are collected from an accelerometer placed on the foot 1 of Machine 1, floor near foot 1 of machine 1, foot of machine 2 and floor near foot of machine 2 respectively.

![Figure 4.35: Acceleration signal from foot of machine 1](image)

The acceleration signal in Figure 4.35 is not symmetric about the axis since the data acquisition system gives the initial signal with a bias in it.
Chapter 4. Results and Analysis

Figure 4.36: Acceleration signal from the floor near foot of machine 1

Figure 4.37: Acceleration signal from foot of machine 2
Chapter 4. Results and Analysis

Figure 4.38: Acceleration signal from the floor near foot of machine 2

Spectral Densities

Any rotating machine produces periodic signal. Since the measurements are done in an industry, the acceleration signals shown above are interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis for the purpose of averaging.

Figure 4.39 represents the Power Spectral Densities of Acceleration signals at feet of both machines. Figure 4.40 represents the Power Spectral Densities of Acceleration signals at floor near the feet of both machines.
Figure 4.39: Power Spectral Densities of Acceleration signals at feet of both machines
Figure 4.40: Power Spectral Densities of Acceleration signals at floor near feet of both machines

Transmissibility

The Transmissibility for the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.41. This is done for both the machines.
Figure 4.41: Transmissibilities of Acceleration signals from both machines

Coherence Function

The Figure 4.42 shows the coherence functions accelerations signals from both the machines.
Figure 4.42: Coherence plot of acceleration signals from two machines

- The machine 1 has very low coherence since its magnitude is nearly 0. This might be since the machine 1 is not running. It has a peak at 26 Hz. The machine 2 has relatively good coherence (near to 1) in the range 80-340 Hz. The coherence is bad (less than 0.8) in the range 350-500 Hz. From the coherence plot, it can be seen that the system has peaks at 26 Hz, 52 Hz.

- Since the machine 1 is not running, the PSD of acceleration signal from machine 1 has constant magnitude i.e. around -78 dB for the acceleration signal from the foot of machine 1 and around -65 dB for the acceleration signal near floor of machine 1. However there is a clear peak at 25 Hz. This peak is possibly coming from the machine 2 operation. The magnitude of PSD of machine 2 is decreasing in the range 150-350 Hz. The PSD at floor near foot of machine 2 is decreasing in the range 150-350 Hz and increasing in the range 350-500 Hz.

- The magnitude of transmissibility for machine 1 is far greater than 0 dB i.e. around -30 dB. This means that the magnitude of transfer of vibration is relatively low. It can also be observed that the average magnitude of
transmissibility is constant at most of the frequencies.

4.3.3 State 3

In this state, machine 1 is running and machine 2 is not running.

Acceleration Signals

The acceleration signal shown in figure 4.43, figure 4.44, figure 4.45 and figure 4.46 as collected from accelerometers placed on the foot 1 of Machine 1, floor near foot 1 of machine 1, foot of Machine 2, floor near foot of machine 2 respectively.

![Acceleration signal from foot 1 of machine 1](image)

Figure 4.43: Acceleration signal from foot 1 of machine 1
Figure 4.44: Acceleration signal from floor near foot 1 of machine 1

Figure 4.45: Acceleration signal from foot of machine 2
Spectral Densities

Any rotating machine produces periodic signal. Since the measurements are done in an industry, the acceleration signals shown above are interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis for the purpose of averaging.

The Power Spectral Densities of the acceleration signals from an accelerometers placed of the feet of both machines are shown in figure 4.47. The Power Spectral Densities of the acceleration signals from an accelerometers placed of the feet of both machines are shown in figure 4.48.
Figure 4.47: Power Spectral densities of acceleration signals at foot 1 of machine 1 and foot of machine 2

Figure 4.48: Power Spectral densities of acceleration signals at floor near foot 1 of machine 1 and foot of machine 2
Transmissibility

The transmissibility plots for the acceleration signals from the foot of the machine and from the floor near the foot of the machine are shown in figure 4.49. This is done for both the machines.

![Transmissibility plot]

Figure 4.49: Transmissibility acceleration signals from both machines

Coherence

Figure 4.50 shows the coherence plot between acceleration signals from both the machines.
Chapter 4. Results and Analysis

Figure 4.50: Coherence plots of acceleration signals from both machines

**Observations from State 3:**

- It can be observed from the Coherence plot that machine 1 has good coherence (near to 1) in the frequency ranges 0 Hz to 100 Hz and 250 Hz to 300 Hz. The coherence is bad (almost 0) in other frequency ranges. It has a peak at 18 Hz. It can be observed that machine 2 has good coherence only in the range 10 Hz to 50 Hz. From the coherence plot shown in figure 4.46, there is a peak at 18 Hz. The coherence is bad at remaining frequencies. This might be since the machine 2 is not running.

- From the transmissibility plot of machine 1, it can be observed that the magnitude is constant till 100 Hz, and then it gradually decreases until 154 Hz. From 154 Hz, it increases until 258 Hz and then decreases. The magnitude of transmissibility varies between -50 dB and -20 dB. This means that the magnitude of vibration transferred is very less. It can be observed from Figure 4.49 that the magnitude of transmissibility of machine 2 increased till 33 Hz and then decreased. The magnitude of FRF is less than 0 dB except in the range 15 Hz to 38 Hz.
• The magnitudes PSD of acceleration signals from feet of machine 1 and floor near machine 1 are around -65 dB and -60 dB respectively. They have peaks at 18 Hz and 255 Hz. Since the machine 2 is not running, the peak in the PSDs of machine 2 at 18 Hz can be due to the vibration of machine 1.

4.3.4 State 4

In this state, both the machines are running and foot 2 of the machine 1 is considered.

Acceleration Signals

The acceleration signal shown in figure 4.51, figure 4.52, figure 4.53 and figure 4.54 as collected from accelerometers placed on the foot 2 of Machine 1, floor near foot 2 of machine 1, foot of Machine 2, floor near foot of machine 2 respectively.

![Image](image.png)

Figure 4.51: Acceleration signal from foot 2 of machine 1
Chapter 4. Results and Analysis

Figure 4.52: Acceleration signal from floor near foot 2 of machine 1

Figure 4.53: Acceleration signal from foot of machine 2
Chapter 4. Results and Analysis

Figure 4.54: Acceleration signal from floor from foot of machine 2

Spectral Densities

Any rotating machine produces periodic signal. Since the measurements are done in an industry, the acceleration signals shown above are interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis for the purpose of averaging.

Figure 4.55 shows the Power Spectral Densities of acceleration signals from feet of both the machines. Figure 4.56 shows the Power Spectral Densities of acceleration signals from the floor near the feet of machines.
Figure 4.55: Power Spectral Density of acceleration signals at foot 2 of machine 1 and foot of machine 2

Figure 4.56: Power Spectral Density of acceleration signals from floor near foot 1 of machine 1 and foot of machine 2
Transmissibility

The Transmissibilities for the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.57. This is done for both the machines.

Figure 4.57: Transmissibility of acceleration signals from both machines
Coherence

The Coherence plot between acceleration signals from both the machines is shown in Figure 4.58.

Figure 4.58: Coherence function of acceleration signals from both machines.

Observations from State 4:

- The machine 1 has slightly good coherence in the range 10-115 Hz. The coherence is bad at remaining frequencies since the level is almost near to 0.
  The machine 2 has good coherence (near to 1) in the ranges 10-70 Hz and 80-340 Hz. The system has bad coherence (near to 0) in the range 400-500 Hz. Since both the machines are running, coherence for both the systems can be compared and observed that machine 2 has relatively better coherence.

- For machine 1, the magnitude of PSD of acceleration signals at foot of machine 1 is around -65 dB and magnitude of PSD of acceleration signals from floor near foot of machine 1 is around -60 dB.
  For machine 2, the magnitude of PSD of acceleration signals at feet is around -60 dB and the magnitude of PSD of acceleration signals from the floor is
around -40 dB which is decreasing in the frequency range of 200 Hz - 350 Hz.

- From the transmissibility plot of machine 1, the magnitude of transmissibility is relatively higher in the frequency range 0 Hz-124 Hz i.e. the transfer is higher. The magnitude is then decreasing. The level of transmissibility is almost constant in the range 200-350 Hz. From the transmissibility plot of machine 2, it can be observed that the magnitude of transmissibility is mostly greater than 0 dB. This means that the magnitude of the transfer of vibration is larger. The magnitude of FRF is increasing in the range 50-110 Hz. It can be observed that the level of FRF of machine 2 is higher than machine 1.

4.3.5 State 5:

In this state, both the machines are running and foot 2 of machine 1 is considered.

Acceleration signals:
The acceleration signal shown in figure 4.59, figure 4.60, figure 4.61 and figure 4.62 as collected from accelerometers placed on the foot 2 of Machine 1, floor near foot of machine 1, foot of Machine 2, floor near foot of machine 2 respectively.

![Acceleration signal from foot 2 of machine 1](image-url)
Chapter 4. Results and Analysis

Figure 4.60: Acceleration signal from floor near foot 2 of machine 1

Figure 4.61: Acceleration signal from foot of machine 2
Chapter 4. Results and Analysis

Figure 4.62: Acceleration signal from floor near foot of machine 2

Power Spectral Densities:

Any rotating machine produces periodic signal. Since the measurements are done in an industry, the acceleration signals shown above are interrupted with noise. In our experiment, a “Rectangular window” is used in Spectral Analysis. Figure 4.63 shows the Power Spectral Densities of acceleration signals from feet of both the machines. Figure 4.64 shows the Power Spectral Densities of acceleration signals from floor near feet of both the machines.
Figure 4.63: PSD of acceleration signals from foot 2 of machine 1 and foot of machine 2

Figure 4.64: PSD of acceleration signals from floor near foot 2 of machine 1 and foot of machine 2
Transmissibility

The frequency response functions for the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.65. This is done for both the machines.

![Graph showing transmissibility](image)

Figure 4.65: Transmissibility of acceleration signals from both machines

Coherence

The coherence functions between the acceleration signals from the foot of the machine and from the floor near the foot of the machine respectively is shown in Figure 4.66. This is done for both the machines.
Chapter 4. Results and Analysis

Figure 4.66: Coherence of acceleration signals from both machines

- It can be observed that the system of machine 1 had bad coherence since the magnitude of coherence is nearly 0 at most of the frequencies. The system might have resonance frequency at 78 Hz, 213 Hz, 380 Hz, 460 Hz etc since there are dips at these frequencies. For machine 2, the system has good coherence (near to 1) in the ranges 120-215 Hz and 260-340 Hz. The system has bad coherence in the range 400-500 Hz. There are certain dips at frequencies 75 Hz, 220 Hz etc which means there is a possibility of resonance at these frequencies.

- For machine 1 the magnitude of PSD of acceleration signals from the feet is around -68 dB and the magnitude of PSD of acceleration signals from the floor is around -60 dB. For machine 2 the magnitude of PSD of acceleration signals from the feet is around -60 dB and the magnitude of PSD of acceleration signals from the floor is around -45 dB. This is decreasing in the range of 250 Hz - 400 Hz.

- From the transmissibility plot of machine 1 in Figure 4.65, the magnitude is increasing in the range 0-100 Hz and thereby decreasing. The average level of transmissibility of machine 1 is around -30 dB which means that the magnitude of vibration transferred is less. From the transmissibility plot of machine 2 in Figure 4.65, it can be observed that the magnitude is mostly greater than 0 dB. This means that the
vibration transferred from the machine to the floor has higher magnitude. The magnitude of transmissibility is increasing in the range 70-335 Hz. The average of level of transmissibility of acceleration signals from machine 2 is higher than machine 1. This means that vibration transfer from machine 2 to floor is higher.
When the different set-ups discussed are implemented, studied and analyzed, the below conclusions are drawn based on the observations.

5.1 Set-up 1:

- The system setup of the model imitates SDOF and can be extended further to larger masses.

- However, the excitation sources may not possible to be used. Most of the machines do not have appropriate excitation points in case of hammer excitation. Furthermore the use of shaker to excite heavy machinery is also not feasible both from excitation point of view and the complexity involved during shaker setup, such as isolation or suspension. Therefore, the PSD and transmissibility may be used to study the vibration transferred from machine to floors.

- The responses of Shaker and Hammer excitation are observed to be approximately equal.

- The coherence of Hammer excitation obtained is slightly better when compared to the coherence of Shaker excitation.

5.2 Set-up 2:

- The magnitude of Vibration transferred is relatively high at frequencies between 300 and 350 Hz. For the rest of the frequency spectrum, less vibration is transferred.

- The behavior of front part of the machine is different from the behavior of the back.

- In this Setup, MDOFs are observed during the analysis. But the system maybe approximated as a 2 DOF. The magnitude of vibration transfer is
above 0 dB near the 2nd resonance frequency, which is somewhere between 300 and 350 Hz. The behavior is almost similar for the SDOF case.

- It can also be inferred that the transmissibility is below 0 dB for most of the spectrum except at 300 Hz and 350 Hz where the transfer is relatively higher.

- Generally field measurements are contaminated by noise and it may be difficult to obtain higher coherence as expected in laboratory environments. Furthermore, signals originating from other machines may also influence the coherence.

5.3 Set-up 3:

- Only in certain frequency range (100-350) Hz, the magnitude of vibration transferred is higher. This means that the system acts as a Band Pass Filter.

- When the magnitude of transmissibility increases, the magnitude of vibration transferred also increases and vice versa.

- When one of the machine is not running, its behavior can be effected by the vibration from the machine which is in running state. This can be observed due to the peaks in the machine 1 (not running) when the machine 2 is running and vice versa.

- In all the states, it is observed that the vibration of one machine is transferred to the floor and also effects the behavior of other machine.

- The vibration transferred from machine 2 foot to floor is higher as compared to machine 1.

5.4 Other factors affecting Coherence

- Apart from the behavior of Frequency Response Function, linearity of the system also affects the Coherence. If the System is Linear, the magnitude of the Coherence is likely to be neared to 1.

- There is a dip in Coherence at the Resonant Frequency of the system.

- The Coherence is also reduced by other factors. These can be represented as shown below in the Figure 5.1.
Chapter 5. Conclusions and Future Work

5.5 Summary

The above conclusions drawn from different set-ups can be summarized and represented as shown in the Figure 5.2.

- **Set-up 1**: Coherence of hammer excitation system is better than shaker excitation system.
- **Set-up 2**: Machine has relatively good coherence at the front feet than the coherence at the back feet.
- **Set-up 3**: The vibration of one machine is transferred to floor and also effects the behavior of other machine.

Figure 5.1: Schematic representation of Factors affecting Coherence.

Figure 5.2: Summary of the Conclusions from various models.
5.6 Future works

The work can further be extended in many ways.

• Amount of vibration transferred from one machine to the floor and other machines can be estimated. The effect of the vibration transferred on the performance of other machine can also be estimated.

• The efficiency of isolators can be estimated based on the amount of vibration transferred.


89


[17] K.-w. KIM, J.-o. YEON, H.-k. SHIN, and K.-s. YANG, “Floor impact sound and vibration propagation characteristics in adjacent houses,”


References


Appendices
Appendix A

Machine used in Set-up 2

The Set-up 2 is implemented on a “Mazak Quick Turn Nexus 300-II” machine which is shown in Figure 1. This machine is a CNC turning center that integrates advanced technology and productivity to get exceptional performance.

Figure A.1: Mazak Quick Turn Nexus 300-II machine

- It has 2 axes (X and Z) and processes different sizes of workpieces very efficiently.
- This can be re-configured to match any requirement of workpiece length.
Appendix A. Machine used in Set-up 2

- The CNC turning centers have improved machine stability
- We performed our experiment on this Lathe machine present in our "BTH SvarLab". The machine is in run state and measurements are taken from different feet present at the front and back of the machine.

<table>
<thead>
<tr>
<th>Main Spindle</th>
<th>Rotating Speed (Maximum)</th>
<th>4000/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feed axes</td>
<td>Travel (X axis)</td>
<td>225 mm</td>
</tr>
<tr>
<td></td>
<td>Travel (Y axis)</td>
<td>680 mm</td>
</tr>
<tr>
<td>Dimensions</td>
<td>Length</td>
<td>3005 mm</td>
</tr>
<tr>
<td></td>
<td>Width</td>
<td>2050 mm</td>
</tr>
</tbody>
</table>

Table A.0.1: Specifications of machine

The machines in the BTH SvarLab are captured and shown below.

Figure A.2: Picture of the machine in SvarLab
Appendix B

Machine used in Set-up 2

The Set-up 3 is implemented on two CNC Lathe machines, “MAZAK Integrex 200-III S” and “MAZAK Quick Turn 10”. The experiment is done in a workshop “KOSAB”, located in Olofström, Sweden. KOSAB is a largest manufacture of electrodes and wear materials for spot welds in Sweden. It is specialized in production of copper, but manufactures in all materials like steel, brass, plastic etc.

Machine 1:

Figure B.1: MAZAK Intetrex 200-III S (Machine 1)
Appendix B. Machine used in Set-up 2

Specifications:

- It is a 5-axes machine which contains two spindles rotating with speed 5000 rpm.
- It also contains an automatic rod handling with feeder table.
- It has 65 mm spindle passage.

Machine 2:

![Machine 2: MAZAK Quick Turn 10](image)

Figure B.2: MAZAK Quick Turn 10 (Machine 2)

Specifications:

- It has one spindle with speed 6000 rpm.
- It has 8 tool positions.
- It has an automatic rod handling with feeder table.
- It has 40 mm spindle passage.