COMPOSITES FOR MACHINE TOOL BEDS

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

MASTER OF TECHNOLOGY
IN
PRODUCTION ENGINEERING

By

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Department of Mechanical Engineering

NATIONAL INSTITUTE OF TECHNOLOGY
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Dedicated to my parents & guide

NATIONAL INSTITUTE OF TECHNOLOGY ROURKELA

CERTIFICATE

This is to certify that work in this project report entitled, **"COMPOSITES FOR MACHINE TOOL BEDS"** by **B.SRIKANTH** has been carried out under my supervision and guidance in partial fulfillment of the requirements for the award of **Master of Technology** Degree in ***Mechanical Engineering*** with "**Production Engineering**" specialization during session 2009-2011 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

To the best of my knowledge, this work has not been submitted to any other university/ institute for award of any Degree or Diploma.

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Date **B.SRIKANTH**

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ABSTRACT

In general, machine tool structures like lathe, milling, broaching, and grinding machines, etc. are subjected to regular unwanted vibrations. These machine tool vibrations or chatter are deleterious to machining operations. It results in degraded quality on the machined parts, shorter tool life, and unpleasant noise, hence are to be necessarily damped out. The important characteristics of the machine tool structures for metal cutting are high damping and static stiffness which ensure manufacture of work pieces of the required geometries with acceptable surface finish at the required rate of production in the most economical way. The unwanted vibrations must be arrested in ordet to ensure higher accuracy along with productivity.

In the present work, the chatter vibrations on a slotted table Horizontal Milling Machine have been damped out using composite structure as a substitute for the base of the work piece. Glass Fiber Polyester and Glass Fiber Epoxy plates are fixed on to the slotted table as a secondary bed material and the workpiece is mounted on this bed for feeding to the rotating milling cutter. Initially four holes are drilled on each plate of the composite and a set of five plates of each type of composite are mounted for conducting the experiments. A mild steel specimen of similar dimension of the composite plate is placed on the pile of the composites and the setup is fixed to the slotted table using bolts and nuts. An up milling operation is carried out and the vibration signal is recorded on the screen of the digital phosphorus storage oscilloscope. The signal and RMS amplitude, frequency and time period of vibrations are recorded. The experiment is repeated for different sets of composite plates by decreasing the number and the corresponding readings are recorded and tabulated. Moreover, experiments are also conducted without any composite material below the mild steel specimen. It is observed that the vibration amplitude decreases with increase in number of layers of sheets of composites and then increases with increase in number of plates. Moreover, the optimum number of composites are also experimentally determined. The design of the experimental setup has been modeled using CatiaV5R15.

Apart from total damping of the system, emphasis has also been focused to find out the material damping of the composite materials so as to select the same for effective damping of the structures. An energy balance approach has been used for calculating the material damping of the fiber reinforced composites used in the experiment.

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NOMENCLATURE

F_s	Oscillatory force
F_d	Damping force
F_{total}	Strain rate
m	Mass
a	Acceleration
$\dot{q}$	Relative displacement at the joint
C	Friction parameter
sgn	Signum function
ω	Frequency
η	Loss factor
η_1	Longitudinal loss factor
η_2	Transverse loss factor
η_{12}	Shear loss factor
W_f	Strain energy of fiber
W_m	Strain energy of fiber
E	Young's modulus
G	Rigidity modulus
S_{fict}	Static moments of different phases
ν	Poisson's ratio
mV	milli volts
ρ	Density

Chapter 1

INTRODUCTION

1.1 Machine tool vibrations and their adverse effects

Machining and measuring operations are invariably accompanied by relative vibrations between work piece and tool. These vibrations are due to one or more of the following causes:

(1) inhomogeneities in the work piece material;

(2) Variation of chip cross section;

(3) Disturbances in the work piece or tool drives;

(4) Dynamic loads generated by acceleration/deceleration of massive moving components;

(5) Vibration transmitted from the environment;

(6) Self-excited vibration generated by the cutting process or by friction (machine-tool chatter).

The adverse and undesirable effects of these vibrations include reduction in tool life, improper surface finish, unwanted noise and excessive load on the machine tool. A machine tool is expected to have high stiffness in order to avoid such effects. Hence the machines are to be made of robust structured materials through passive damping technology to suppress the chatter vibrations and thereby increasing the production rates.

1.2 Main objective of the research work

The main objective of the work is to study passive damping techniques in machine tool structures using composite materials and to reduce vibrations in the milling machine during cutting processes by using these materials as the base of the work piece which act like a bed absorbing vibration forces and record the vibration curves using digital storage phosphorous oscilloscope. Composites can be used in machine tool structures because of its inherent damping characteristics which reduces the undesirable effects of the vibrations. Passive damping technology has a wide variety of engineering applications, including bridges, engine mounts, and machine components such as rotating shafts, component vibration isolation, novel spring designs which incorporate damping without the use of traditional dashpots or shock absorbers, and structural supports.

Chapter 2

LITERATURE REVIEW

2.1 Introduction to machine tools

The function of machine tool is to produce a workpiece of the required geometric form with an acceptable surface finish at high rate of production in the most economic way [1]. In fact, general purpose machine tools, CNC lathes and machining centres are designed to cope with low cutting speeds with high cutting forces as well as high cutting speeds with low cutting forces. Machine Tool Structure must possess high damping, high static and dynamic stiffness. High cutting speeds and feeds are essential requirements of a machine tool structure to accomplish this basic function. Therefore, the material for the machine tool structure should have high static stiffness and damping in its property to improve both the static and dynamic performance..The static stiffness of a machine tool can be increased by using either higher modulus material or more material in the structure of a machine tool. However, it is difficult to increase the dynamic stiffness of a machine tool with these methods because the damping of the machine tool structure cannot be increased by increasing the static stiffness. Sometimes high specific stiffness is more important than stiffness to increase the natural frequency of the vibration of the machine tool structure in high speed machining [2]. Often the most economical way of improving a machine tool with high resonance peaks is to increase the damping rather than the static stiffness even though it is not easy to increase the damping of the machine tool structure. The chatter is a nuisance to the metal cutting process and can occur on any chip producing tool. Chatter or Self-excited vibrations occurs when the width of cut or cutting speed exceeds the stability limit of the machine tool [3, 4]. The effects of chatter are all adverse, affecting surface finish, dimensional accuracy, tool life and machine life [5].When the machine tool is operated without any vibration or chatter, the damping of the machine tool plays no important role in machining. However, the machine tool structure has several resonant frequencies because of its continuous structural elements. If the damping is too small to dissipate the vibrational energy of the machine tool, the resonant vibration occurs when the frequency of the machining operation approaches one of the natural frequencies of the machine tool structure. Therefore the material for the machine tool structure should have high static stiffness and damping in its property to improve both the static and dynamic performance.

2.2 Damping overview

The three essential parameters that determine the dynamic responses of a structure and its sound transmission characteristics are mass, stiffness and damping. Mass and stiffness are associated with storage of energy. Damping results in the dissipation of energy by a vibration system. For a linear system, if the forcing frequency is the same as the natural frequency of the system, the response is very large and can easily cause dangerous consequences. In the frequency domain, the response near the natural frequency is "damping controlled". Higher damping can help to reduce the amplitude at resonance of structures. Increased damping also results in faster decay of free vibration, reduced dynamic stresses, lower structural response to sound, and increased sound transmission loss above the critical frequency. A lot of literature have been published on vibration damping. ASME published a collection of papers on structural damping in 1959 [6]. Lazan's book published in 1968 gave a very good review on damping research work, discussed different mechanisms and forms of damping, and studied damping at both the microscopic and macroscopic levels [7]. Lazan conducted comprehensive studies into the general nature of material damping and presented damping results data for almost 2000 materials and test conditions. Lazan's results show that the logarithmic decrement values increase with dynamic stress, i.e., with vibration amplitude, where material damping is the dominant mechanism. This book is also valuable as a handbook because it contains more than 50 pages of data on damping properties of various materials, including metals, alloys, polymers, composites, glass, stone, natural crystals, particle-type materials, and fluids. About 20 years later, Nashif, Jones and Henderson published another comprehensive book on vibration damping [8]. Jones himself wrote a handbook especially on viscoelastic damping 15 years later [9]. Sun and Lu's book published in 1995 presents recent research accomplishments on vibration damping in beams, plates, rings, and shells [10]. Finite element models on damping treatment are also summarized in this book. There is also other good literature available on vibration damping [11-13].Damping in vibrating mechanical systems has been subdivided into two classes: Material damping and system damping, depending on the main routes of energy dissipation. Coulomb (1784) postulated that material damping arises due to interfacial friction between the grain boundaries of the material under dynamic condition. Further studies on material damping have been made by Robertson and Yorgiadis (1946), Demer (1956), Lazan (1968) and Birchak (1977). System damping arises from slip and other boundary shear effects at mating surfaces, interfaces or joints between distinguishable parts. Murty (1971) established that the energy dissipated at the support is very small compared to material damping.

2.3 Review on Research done in Damping of Composite materials

Bert [14] and Nashif et al.[15] had done survey on the damping capacity of fibre reinforced composites and found out that composite materials generally exhibit higher damping than structural metallic materials. Chandra et al. [16] has done research on damping in fiber-reinforced composite materials.

Composite damping mechanisms and methodology applicable to damping analysis is described and had presented damping studies involving macromechanical, micromechanical and Viscoelastic approaches. Gibson et al.[17] and Sun et al.[18,19] assumed viscoelasticity to describe the behavior of material damping of composites.

The concept of specific damping capacity (SDC) was adopted in the damped vibration analysis by Adams and his co workers [20-21], Morison [22] and Kinra et al [23].

The concept of damping in terms of strain energy was apparently first introduced by Ungar et.al [24] and was later applied to finite element analysis by Johnson et.al [25]. Gibson et.al [26] has developed a technique for measuring material damping in specimens under forced flexural vibration. Suarez et al [27] has used Random and Impulse Techniques for Measurement of Damping in Composite Materials. The random and impulse techniques utilize the frequency-domain transfer function of a material specimen under random and impulsive excitation. Gibson et al [28] used the modal vibration response measurements to characterize, quickly and accurately the mechanical properties of fiber-reinforced composite materials and structures.

Lin et al. [29] predicted SDC in composites under flexural vibration using finite element method based on modal strain energy (MSE) method considering only two interlaminar stresses and neglecting transverse stress.

Koo KN et al. [30] studied the effects of transverse shear deformation on the modal loss factors as well as the natural frequencies of composite laminated plates by using the finite element method based on the shear deformable plate theory.

SINGH S. P et al. [31] analyzed damped free vibrations of composite shells using a first order shear deformation theory in which one assumes a uniform distribution of the transverse shear across the thickness, compensated with a correction factor.

Polymeric materials are widely used for sound and vibration damping. One of the more notable properties of these materials, besides the high damping ability, is the strong frequency dependence of dynamic properties; both the dynamic modulus of elasticity and the damping characterized by the loss factor [30-35].

Mycklestad [32] was one of the pioneering scientists into the investigation of complex modulus behavior of viscoelastic materials (Jones, 2001, Sun, 1995). Viscoelastic material properties are generally modeled in the complex domain because of the nature of viscoelasticity. Viscoelastic materials possess both elastic and viscous properties. The typical behavior is that the dynamic modulus increases monotonically with the increase of frequency and the loss factor exhibits a wide peak [8, 33].

It is rare that the loss factor peak, plotted against logarithmic frequency, is symmetrical with respect to the peak maximum, especially if a wide frequency range is considered. The experiments usually reveal that the peak broadens at high frequencies. In addition to this, the experimental data on some polymeric damping materials at very high frequencies, far from the peak centre, show that the loss factor–frequency curve ‘‘flattens’’ and seems to approach a limit value, while the dynamic modulus exhibits a weak monotonic increase at these frequencies [34-38]. These phenomena can be seen in the experimental data published by Madigosky and Lee [34], Rogers [35] and Capps [36] for polyurethanes, and moreover by Fowler [37], Nashif and Lewis [38] for other polymeric damping materials.

The computerized methods of acoustical and vibration calculus require the mathematical form of frequency dependences of dynamic properties. A reasonable method of describing the frequency dependences is to find a good material model fitting the experimental data.

2.4 Review of the previous work done on the machine tool structures

In recent years many efforts have been made to increase the material damping of the machine tool structures.

Rahman et al. [37] have made attempts to review and summarize the key developments in the area of non-conventional materials for machine tool structures over the last decades. They have compared many beneficial properties of the machine tool structural materials with the conventional cast iron. For supporting the ever rising working speeds made possible by the development of tools and machining processes, the increasing requirements concerning the surface finish of the machined workpieces and the fabrication cost of the machine tool structures exerted the impetus to find alternatives to cast iron. Based on the results of previous studies they have stated that composite materials may be the choice to replace conventional materials.

Lee et al. [38] have improved the damping capacity of the column of a precision mirror surface grinding machine tool by manufacturing a hybrid column by adhesively bonding glass fiber reinforced epoxy composite plates to a cast iron column. For optimizing the damping capacity of the hybrid column they have calculated the damping capacity of the hybrid column with respect to the fiber orientation and thickness of the composite laminate plate and they have compared with the measured damping capacity. From experiments they have found out that the damping capacity of the hybrid column was 35% higher than that of the cast iron column.

Kegg et al. [39] have used composites for the massive slides for CNC milling machine in machining moulds and dies because presence of these massive slides do not allow rapid acceleration and deceleration during the frequent starts/stops encountered in machining moulds and dies. They have constructed the vertical and horizontal slides of a large CNC machine by bonding high modulus carbon-fiber epoxy composite sandwiches to welded steel structures using adhesives. These composites structures reduced the weight of the vertical and horizontal slides by 34% and 26%, respectively and increased damping by 1.5 to 5.7 times without sacrificing the stiffness.

Okuba et al. [40] have improved the dynamic rigidity of machine tool structures by studying the mode shape animation based on the results of modal analysis. This technique was successfully applied to a machining cell, an arm of automatic assembling machine and a conventional cylindrical grinder. The examples on a vertical milling machine, an NC lathe and a surface grinder show effectiveness of the software approach in suppressing the chatter and improving the surface finish.

Chowdhury [41] used epoxy resin as a bonding material between structural components of a milling machine to increase joint damping. It was reported that the bonded over arm of milling machine performed much better than those of welded and the cast iron.

Haranath et al. [42] have done attempts experimentally to establish that improvement can be attained by applied damping treatment using viscoelastic layers. They have done theoretical study on the vibrations of machine tool structures with applied damping treatment by using a conventional beam element. Models of milling machine, radial drilling machine and lathe have been analysed for their natural frequency and loss factors. They have found out the influence of layering treatment on the natural frequencies and loss factors.

Wakasawa et al. [43] have improved the damping capacity of machine tool structure by ball packing. In structures closely packed with balls, various damping characteristics appear in correspondence with the ball size and other conditions. The effect of ball size is the most significant factor in these structures. Excitation of structure is required to achieve an optimum packing ratio where the maximum damping capacity is obtained. For a 50% packing ratio, this excitation process is not necessary to obtain a stable damping capacity. Therefore, they have investigated the effects of magnitude of impulse, packed ball material, and structure size on the damping capacity at a 50% packing ratio. Finally, they have constructed actual machine tool structure models and the effectiveness of the balls packing for the damping capacity improvement has been investigated.

Chapter 3

COMPOSITES

3.1 Composite materials

Composite materials, often shortened to composites, are engineered or naturally occurring materials made from two or more constituent materials with significantly different physical or chemical properties which remain separate and distinct at the macroscopic or microscopic scale within the finished structure. The constituents are combined in such a way that they keep their individual physical phases and are neither soluble in each other nor form a new chemical compound. One constituent is called reinforcing phase which is embedded in another phase called matrix. The most visible applications is pavement in roadways in the form of either steel and aggregate reinforced Portland cement or asphalt concrete.

Mostly fibers are used as the reinforcing phase and are much stronger than the matrix and the matrix is used to hold the fibers intact. Examples of such composites are an aluminums matrix embedded with boron fibers and an epoxy matrix embedded with glass or carbon fibers. The fibers may be long or short, directionally aligned or randomly orientated, or 'some sort of mixture, depending on the intended use of the material. Commonly used materials for the matrix are polymers, metals, ceramics, carbon and fibers are carbon (graphite) fibers, aramid fibers and boron fibers.

Fiber-reinforced composite materials are further classified into the following
a) continuous fiber-reinforced
b) discontinuous aligned fiber-reinforced
c) discontinuous random-oriented fiber-reinforced.

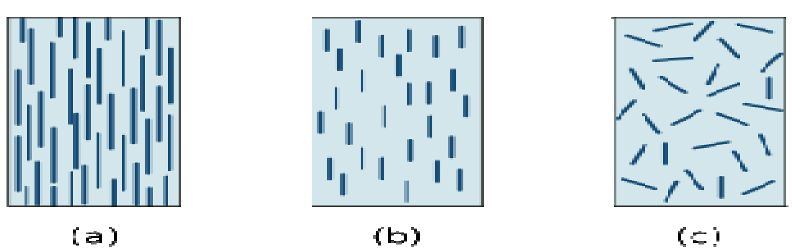

Fig 3.1 Types of fiber reinforced materials

Composites used in the work are Glass fiber epoxy and Glass fiber polyester.

Fiberglass is made from extremely fine fibers of glass. It is used as a reinforcing agent for many polymer products; the resulting composite material, properly known as fiber-reinforced polymer (FRP) or glass-reinforced plastic (GRP), is called "fiberglass" in popular usage. Uses for regular fiberglass include mats, thermal insulation, electrical insulation, reinforcement of various materials, tent poles, sound absorption, heat- and corrosion-resistant fabrics, high-strength fabrics, pole vault poles, arrows, bows and crossbows, translucent roofing panels, automobile bodies, hockey sticks, surfboards, boat hulls, and paper honeycomb.

Epoxy is a thermosetting polymer formed from reaction of an epoxide "resin" with polyamine "hardener". Epoxy has a wide range of applications, including fiber-reinforced plastic materials and general purpose adhesives. The applications for epoxy-based materials are extensive and include coatings, adhesives and composite materials such as those using carbon fiber and fiberglass reinforcements (although polyester, vinyl ester, and other thermosetting resins are also used for glass-reinforced plastic). The chemistry of epoxies and the range of commercially available variations allows cure polymers to be produced with a very broad range of properties. In general, epoxies are known for their excellent adhesion, chemical and heat resistance, good-to-excellent mechanical properties and very good electrical insulating properties. Many properties of epoxies can be modified (for example silver-filled epoxies with good electrical conductivity are available, although epoxies are typically electrically insulating). Variations offering high thermal insulation, or thermal conductivity combined with high electrical resistance for electronics applications, are available.

Polyester is a category of polymers which contain the ester functional group in their main chain. Although there are many types of polyester, the term "polyester" as a specific material most commonly refers to polyethylene terephthalate. Depending on the chemical structure polyester can be a thermoplastic or thermoset, however the most common polyesters are thermoplastics. Polyesters are used to make "plastic" bottles, films, tarpaulin, canoes, liquid crystal displays, holograms, filters, dielectric film for capacitors, film insulation for wire and insulating tapes.

Chapter 4

MILLING MACHINES

4.1 Milling machine

A milling machine is a machine tool used to machine various materials. Milling machines are often classed in two basic forms, horizontal and vertical, which refers to the orientation of the main spindle. Both types range in size from small, bench-mounted devices to room-sized machines. Unlike a drill press, which holds the workpiece stationary as the drill moves axially to penetrate the material, milling machines also move the workpiece radially against the rotating milling cutter, which cuts on its sides as well as its tip. Workpiece and cutter movement are precisely controlled to less than 0.001 in (0.025 mm), usually by means of precision ground slides and lead screws or analogous technology. Milling machines may be manually operated, mechanically automated, or digitally automated via computer numerical control (CNC). They can perform a vast number of operations, from simple to complex (slot and keyway cutting, planing, drilling to contouring, diesinking). Cutting fluid is often pumped to the cutting site to cool and lubricate the cut and to wash away the resulting swarf. The different types of milling machines are

1) Bed mill
2) Box mill
3) Gantry mill
4) Horizontal boring mill
5) Turrent mill
6) Knee and Column mill

4.1.1 Horizontal knee-and-column mill

The most distinguishing characteristic type of milling machine is the knee and column configuration. This type of milling machine is unique in that the table can be moved in all three directions. The table can be moved longitudinally in the X-axis as well as in and out on the Y-axis. Since the table rides on top of the knee, the table can be moved up and down on the Z-axis. There are several different types of knee and column type milling machines, but they all have the same characteristic. The knee slides up and down on the column face.

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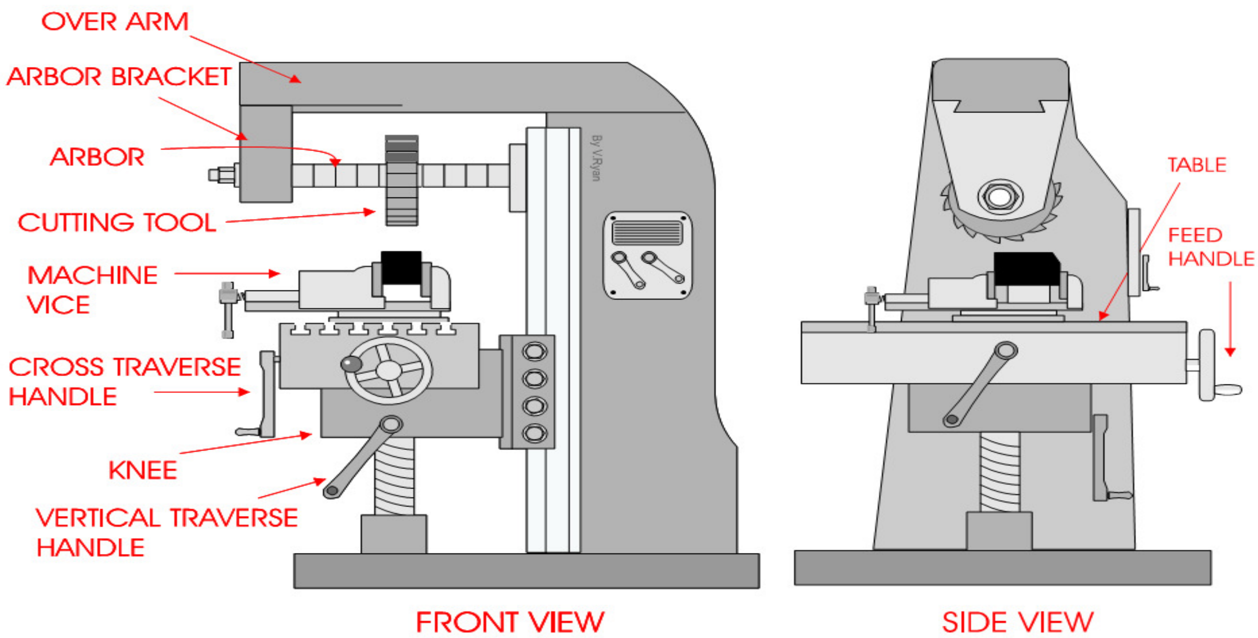


Fig 4.1 Horizontal Knee and Column Milling Machine

4.2 Milling Operation

Milling is the process of cutting away material by feeding a workpiece past a rotating multiple tooth cutter. The cutting action of the many teeth around the milling cutter provides a fast method of machining. The machined surface may be flat, angular, or curved. The surface may also be milled to any combination of shapes. The work piece is mounted on the table with the help of suitable fixtures. The desired contour, feed and depth of cut for the job are noted down. A suitable milling cutter for the specified job is selected and mounted on the arbor. The knee is raised till the cutter just touches the work piece. The machine is started. By moving the table, saddle and the knee, for the specified feed and depth of cut, the desired job may be finished. The machine may then be switched off. The different methods of milling are

1) Up milling
2) Down milling

4.2.1 Up Milling

Up milling is also referred to as conventional milling. The direction of the cutter rotation opposes the feed motion. For example, if the cutter rotates clockwise , the workpiece is fed to the right in up milling.

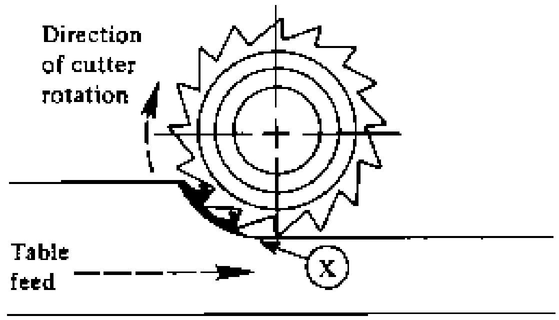


Fig 4.2 Up milling or Conventional milling

4.2.2 Down Milling

Down milling is also referred to as climb milling. The direction of cutter rotation is same as the feed motion. For example, if the cutter rotates counterclockwise , the workpiece is fed to the right in down milling.

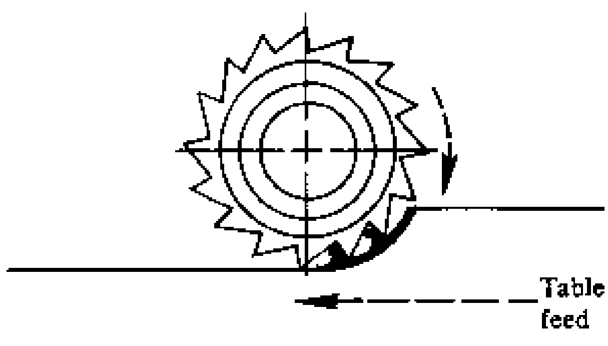


Fig 4.3 Down milling or climb milling

The chip formation in down milling is opposite to the chip formation in up milling. The figure for down milling shows that the cutter tooth is almost parallel to the top surface of the workpiece. The cutter tooth begins to mill the full chip thickness. Then the chip thickness gradually decreases.

4.3 Vibration in Machine Tools

The Machine, cutting tool, and workpiece from a structural system have complicated dynamic characteristics. Under certain condition vibrations of the structural system may occur, and as with all types of machinery, these vibrations may be divided into three basic types:

1. Free or Transient vibrations: resulting from impulses transferred to the structure through its foundation, from rapid reversals of reciprocating masses, such as machine tables, or from the initial engagement of cutting tools. The structure is deflected and oscillates in its natural modes of vibration until the damping present in the structure causes the motion to die away.
2. Forced vibration: resulting from periodic forces within the system, such as unbalanced rotating masses or the intermittent engagement of multitooth cutters (milling), or transmitted through the foundations from nearby machinery. The machine tool will oscillate at the forcing frequency, and if this frequency corresponds to one of the natural frequency of the structure, the machine will resonate in the corresponding natural mode of vibration.
3. Self-excited vibrations: usually resulting from a dynamic instability of the cutting process. This phenomenon is commonly referred to as machine tool chatter and, typically, if large tool-work engagements are attempted, oscillations suddenly build up in the structure, effectively limiting metal removal rates. The structure again oscillates in one of its natural modes of vibration.
4. The sources of vibration excitation in a machine tool structure are vibration due to in-homogeneities in the work piece , cross sectional variation of removed material, disturbances in the vibration of tool drives , rotation unbalanced members guide ways , gears, drive mechanisms and others .

4.4 Chatter in the Milling machine

The milling operation is a cutting process using a rotating cutter with one or more teeth. An important feature is that the action of each cutting edge is intermittent and cuts less than half of the cutter revolution, producing varying but periodic chip thickness and an impact when the edge touches the work piece. The tooth is heated and stressed during the cutting part of the cycle, followed by a period when it is unstressed and allowed to cool. The consequences are thermal and mechanical fatigue of the material and vibrations, which are of two kinds: forced vibrations, caused by the periodic cutting forces acting in the machine structure and chatter vibrations, which may be explained by two distinct mechanisms, called “mode coupling” and “regeneration waviness”, explained in Tobias (1965), Koenigsberger & Tlusty (1967) and Budak & Altintas (1995).

The mode coupling chatter occurs when forced vibrations are present in two directions in the plane of cut. The regenerative chatter is a self excitation mechanism associated with the phase shift between vibrations waves left on both sides of the chip and happens earlier than the mode coupling chatter in most machining cases, as explained by Altintas (2000). In milling, one of the machine tool work piece system structural modes is initially excited by cutting forces. The waved surface left by a previous tooth is removed during the succeeding revolution, which also leaves a wavy surface due to structural vibrations. The cutting forces become oscillatory whose magnitude depends on the instantaneous chip dynamic thickness, which is a function of the phase shift between inner and outer chip surface. The cutting forces can grow until the system becomes unstable and the chatter vibrations increase to a point when the cutter jumps out of the cut or cracks due the excessive forces involved. These vibrations produce poor surface finishing, noise and reduce the life of the cutter. In order to avoid these undesirable effects, the feed rate and the depth of cut are chosen at conservative values, reducing the productivity.

Chapter 5

DAMPING

5.1Definition of Damping

In physics, damping is any effect that tends to reduce the amplitude of oscillations in an oscillatory system, particularly the harmonic oscillator. In mechanics, friction is one such damping effect. In engineering terms, damping may be mathematically modeled as a force synchronous with the velocity of the object but opposite in direction to it. If such force is also proportional to the velocity, as for a simple mechanical viscous damper (dashpot), the force *F* may be related to the velocity *v* by F= -c*v* , where *c* is the viscous damping coefficient, given in units of newton-seconds per meter.

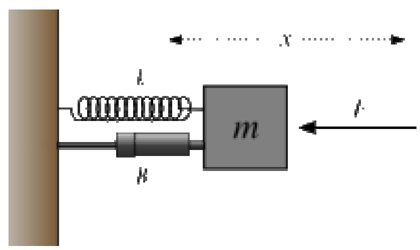


Fig 5.1 Mass spring damper system

An ideal mass-spring-damper system with mass *m* (kg), spring constant *k* (N/m) and viscous damper of damping coefficient *c* (in N-s/ m or kg/s) is subject to an oscillatory force and a damping force,

$$F_s = -kx \qquad\qquad F_d = -cv = -c\frac{dx}{dt} = -c\dot{x}.$$

Treating the mass as a free body and applying Newton's second law, the total force F_{tot} on the body

$$F_{tot} = ma = m\frac{d^2x}{dt^2} = m\ddot{x}.$$ Since $F_{tot} = F_s + F_d$, then => $\boxed{m\ddot{x} = -kx + -c\dot{x}.}$

This differential equation may be rearranged into

$$\ddot{x} + \frac{c}{m}\dot{x} + \frac{k}{m}x = 0. \qquad \omega_0 = \sqrt{\frac{k}{m}} \quad , \qquad \zeta = \frac{c}{2\sqrt{mk}}.$$

ω_0, is the (undamped) natural frequency of the system and ζ, is called the damping ratio.

5.2 Types of Damping

Three main types of damping are present in any mechanical system:

1) Internal damping (of material)

2) Structural damping (at joints and interfaces)

3) Fluid damping (through fluid-structure interactions)

5.2.1 Material (Internal) damping

Internal damping of materials originates from the energy dissipation associated with microstructure defects, such as grain boundaries and impurities; thermoelastic effects caused by local temperature gradients resulting from non uniform stresses, as in vibrating beams; eddy current effects in ferromagnetic materials; dislocation motion in metals; and chain motion in polymers. Several models have been employed to represent energy dissipation caused by internal damping. This variety of models is primarily a result of the vast range of engineering materials; no single model can satisfactorily represent the internal damping characteristics of all materials.

5.2.2 Structural damping

Rubbing friction or contact among different elements in a mechanical system causes structural damping[49]. Since the dissipation of energy depends on the particular characteristics of the mechanical system, it is very difficult to define a model that represents perfectly structural damping. The Coulomb-friction model is as a rule used to describe energy dissipation caused by rubbing friction. Regarding structural damping (caused by contact or impacts at joins), energy dissipation is determined by means of the coefficient of restitution of the two components that are in contact.

Assuming an ideal Coulomb friction, the damping force at a join can be expressed through the following expression:

$$f = c.\mathrm{sgn}(\dot{q})$$

where:

f = damping force, $\dot{q}$ = relative displacement at the joint, c= friction parameter

and the signum function is defined by:

$$\mathrm{sgn}(x) = 1 \text{ for } x \geq 0$$

$$\mathrm{sgn}(x) = -1 \text{ for } x < 0$$

5.2.3 Fluid damping

When a material is immersed in a fluid and there is relative motion between the fluid and the material, as a result the latter is subjected to a drag force. This force causes an energy dissipation that is known as fluid damping.

The damping phenomenon can be applied to the machine tool systems in two ways :

1. Passive damping
2. Active damping

Passive damping refers to energy dissipation within the structure by add on damping devices such as isolator, by structural joints and supports, or by structural member's internal damping. Active damping refers to energy dissipation from the system by external means, such as controlled actuator.

5.3 Damping mechanism in composite materials

Damping mechanisms in composite materials differ entirely from those in conventional metals and alloys [23]. The different sources of energy dissipation in fiber-reinforced composites are:

a) Viscoelastic nature of matrix and/or fiber materials

(b) Damping due to interphase

(c) Damping due to damage which is of two types :

(i) Frictional damping due to slip in the unbound regions between fiber and matrix .

(ii) Damping due to energy dissipation in the area of matrix cracks, broken fibers etc.

(d) Viscoplastic damping

(e) Thermoelastic damping

5.4 Damping in machine tools

Damping in machine tools basically is derived from two sources--material damping and interfacial slip damping. Material damping is the damping inherent in the materials of which the machine is constructed. The magnitude of material damping is small comparing to the total damping in machine tools. A typical damping ratio value for material damping in machine tools is **0.003**. It accounts for approximately **10%** of the total damping. The interfacial damping results from the contacting surfaces at bolted joints and sliding joints. This type of damping accounts for approximately **90%** of the total damping. Among the two types of joints, sliding joints contribute most of the damping[44].Welded joints usually provide very small damping which may be neglected when considering damping in joints.

Table 5.1 Typical damping values of different materials

Systems/Materials	Loss Factor
Welded Metal structure	0.0001 to 0.001
Bolted Metal structure	0.001 to 0.01
Aluminium	0.0001
Brass, Bronze	0.001
Beryllium	0.002
Lead	0.5 to 0.002
Glass	0.002
Steel	0.0001
Iron	0.0006
Tin	0.002
Copper	0.002
Plexiglas TM	0.03
Wood, Fiberboard	0.02

Chapter 6

MATERIAL DAMPING

6.1 Energy balance approach [50]

The loss factor η is commonly used to characterize energy dissipation, due to inelastic behaviour, in a material subjected to cyclic loading. Assuming linear damping behavior, η is defined by Vantomme[50] as;

$$\boxed{\eta = \frac{1}{2\pi}\ \frac{\Delta w}{w}}$$

where ΔW is the amount of energy dissipated during the loading cycle and W is the strain energy stored during the cycle.

Now considering η_1, η_2 and η_{12} :

η_1 – normal loading in fibre direction of UD lamina (longitudinal loss factor)

η_2 – normal loading perpendicular to fibres (transverse loss factor)

η_{12}—inplane shear loading (shear loss factor)

6.1.1 Two phase model

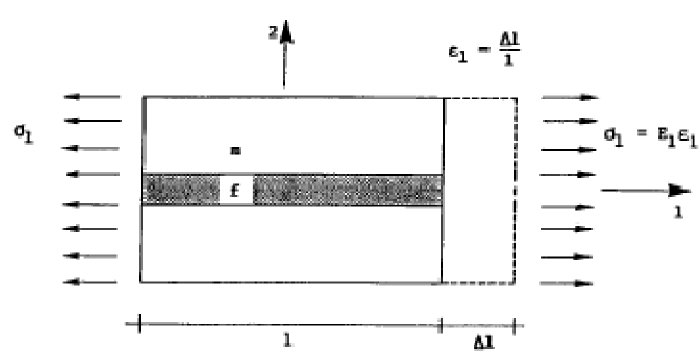


Fig.6.1 RVE loaded in 1-direction,Voigt model: matrix(m) and fibers(f) are connected in parallel

Longitudinal loss factor (η_1) is calculated by the following method : (loading in direction 1)

The total energy dissipated comprises the sum of that lost in the fibres and matrix. These amounts are proportional to the fractions of elastic strain energy stored in the fibres and matrix respectively; i.e.

$$\eta_1 = \eta_{fE}\frac{W_f}{W} + \eta_{mE}\frac{W_m}{W} \qquad \text{……….(1)}$$

η_{fE} and η_{mE} are the loss factors for fibres and matrix, associated with $\sigma - \varepsilon$ tensile loading.

Using the expressions for the strain energy,

$$W_f = 1/2 \int_{V_f} \sigma_{1f}\epsilon_{1f}\,dV$$

$$W_m = 1/2 \int_{V_m} \sigma_{1m}\epsilon_{1m}\,dV \qquad \text{…………………..(2)}$$

with

$$\sigma_{1f} = E_f\epsilon_{1f} = E_f\epsilon_1$$

$$\sigma_{1m} = E_m\epsilon_{1m} = E_m\epsilon_1 \qquad \text{………………….(3)}$$

gives:

$$W_f = 1/2\ E_f\epsilon_1^2\,V_f$$

$$W_m = 1/2\ E_m\epsilon_1^2\,V_m \qquad \text{………………………(4)}$$

Introduction of (4) into (2), with $W = W_f + W_m$, gives:

$$\boxed{\eta_1 = \frac{\eta_{fE}E_fV_f + \eta_{mE}E_mV_m}{E_fV_f + E_mV_m}} \qquad \text{………….(5)}$$

Transverse loss factor (η_2) is calculated: (loading in direction 2)

As before, η_2 is expressed as

$$\eta_2 = \eta_{fE}\frac{W_f}{W} + \eta_{mE}\frac{W_m}{W} \qquad \text{..................(6)}$$

The strain energy contributions are derived in an analogous manner as for η_1, but now with the assumption that the same transverse stress σ_2 is applied to both the fibres and the matrix. This development leads to

$$\boxed{\eta_2 = \frac{\eta_{fE}E_mV_f + \eta_{mE}E_fV_m}{E_mV_f + E_fV_m}} \qquad \text{............(7)}$$

Shear loss factor (η_{12}) is calculated: (loading in shear direction)

$$\eta_{12} = \eta_{fG}\frac{W_f}{W} + \eta_{mG}\frac{W_m}{W} \qquad \text{.........................(8)}$$

where η_{fG} and η_{mG} are the loss factors for fibres and matrix associated with shear loading. The strain energy fractions are worked out in the same way as for η_2, as it is assumed that the shear stresses on the fibres and matrix are the same. This leads to:

$$\boxed{\eta_{12} = \frac{\eta_{fG}G_mV_f + \eta_{mG}G_fV_m}{G_mV_f + G_fV_m}} \qquad \text{................(9)}$$

Equation (9) indicates that damping for a UD lamina, for shear loading, is again matrix-dominated, because the stiffness G_f is usually much larger than G_m. The similarity of equations (9) and (7), combined with the fact that $\eta_{mE} = \eta_{mG}$, leads to the conclusion that $\eta_2$and η_{12} should be very similar.

The coefficients η_{fE},η_{mE},η_{fG},η_{fG} are calculated from the graphs given in the book “Damping of materials and members in structural mechanics” by Benhamin J. Lazan . These graphs are plotted with E’ & E” , G’ & G” .

E' : Storage modulus of elasticity

E " : Loss modulus of elasticity

G' : Storage modulus of rigidity

G ": Loss modulus of rigidity

$\eta_E = E''/E'$, $\eta_G = G''/G'$, For each type of fibre phase and matrix phase material the respective coefficient values are taken from the graphs oriented as slopes of these lines .

6.1.2 Three phase model

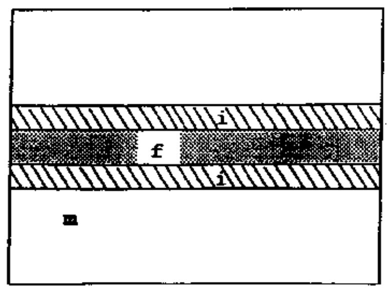


Fig6.2 RVE with three phases: the matrix(m) and fibers(f) and the interphase (i)

Using the energy balance approach, the elastic strain energy is now divided into three terms, giving the following expression for energy dissipation, analogous to equation mentioned before :

$$\eta = \eta_f \frac{W_f}{W} + \eta_m \frac{W_m}{W} + \eta_i \frac{W_i}{W} \qquad \text{.............(10)}$$

where η_i represents the loss factor of the interphase layer.

The elastic strain energies W_f, W_m and W_i may be evaluated as previously described, resulting in the following expressions for η_1, η_2 and η_{12}, for the UD laminae:

$$\eta_1 = \frac{\eta_{fE} E_f V_f + \eta_{mE} E_m V_m + \eta_{iE} E_i V_i}{E_f V_f + E_m V_m + E_i V_i} \qquad \text{.............(11)}$$

$$\eta_2 = \frac{\eta_{fE}\, E_m E_i V_f + \eta_{mE}\, E_i E_f V_m + \eta_{iE}\, E_m E_f V_i}{E_m E_i V_f + E_i E_f V_m + E_m E_f V_i} \quad \text{..........(12)}$$

$$\eta_{12} = \frac{\eta_{fG}\, G_m G_i V_f + \eta_{mG}\, G_i G_f V_m + \eta_{iG}\, G_m G_f V_i}{G_m G_i V_f + G_i G_f V_m + G_m G_f V_i} \quad \text{............(13)}$$

This model does not explain the experimental increase η_l as compared with the values from the graphs .In order to introduce the interphase effect into equation (11), it may be better to consider the elastic strain energy that is associated with the shear stress-strain cycle, when longitudinal bending is considered. Normally, this strain energy is negligible compared with the strain energy associated with the tension or compression cycle, but possibly the presence of a layer with very low stiffness properties may change the energy balance significantly[47].

6.1.3 Modified three phase model

Taking into account the strain energy associated with the shear cycle for the interphase layer only, and allowing for the fact that η_{iE} does not affect η_l, see equation (15), the following strain energy partition may be adopted:

$$\eta_l = \eta_{fE} \frac{W_{f_{\sigma-\epsilon}}}{W} + \eta_{mE} \frac{W_{m_{\sigma-\epsilon}}}{W} + \eta_{iG} \frac{W_{i_{\tau-\gamma}}}{W} \quad \text{.............(14)}$$

The strain energy fractions in equation (14) are developed for the three-phase model ,in which the strain and stress distributions along the cross-section are represented, assuming that a plane cross-section remains plane, and that the adhesion between interphase and matrix, and interphase and fibres, is perfect.

The strain energy fractions $W_{f\sigma-\varepsilon}$, and $W_{m\sigma-\varepsilon}$ are defined for the total volume of the beam specimen
as:

$$W_{f_{\sigma-\epsilon}} = 1/2 \int_{V_f} \sigma_f \epsilon_f \, dV \qquad\qquad W_{m_{\sigma-\epsilon}} = 1/2 \int_{V_m} \sigma_m \epsilon_m \, dV$$

The introduction of Hooke's law, and the relationships between the stresses for fibres and matrix at an arbitrary level and the maximum stress in the matrix σm max (which are based on strain linearity), lead to the expressions [48]:

$$W_{f_{\sigma-\epsilon}} = \tfrac{1}{2} \int_l \int_{\Omega_f} \frac{4\sigma^2_{m\,max} z^2 E_f}{h^2 E_m^2} \, d\Omega_f \, dx$$

$$W_{m_{\sigma-\epsilon}} = \tfrac{1}{2} \int_l \int_{\Omega_m} \frac{4\sigma^2_{m\,max} z^2}{h^2 E_m} \, d\Omega_m \, dx$$

where Ω represents the cross-section.

The relation between $\sigma_{m\,max}$ and the internal M is given by $\sigma_{m\,max} = \frac{Mh}{2I_{fict}}$
where I_{fict} is equal to a combination of the moments of inertia for the different phases:

$$I_{fict} = I_m + \frac{E_f}{E_m} I_f + \frac{E_i}{E_m} I_i$$

Finally we get

$$W_{f_{\sigma-\epsilon}} = \tfrac{1}{2} \frac{E_f I_f}{E_m^2 I_{fict}^2} \int_l M^2 \, dx$$

$$W_{m_{\sigma-\epsilon}} = \tfrac{1}{2} \frac{I_m}{E_m I_{fict}^2} \int_l M^2 \, dx \qquad \text{..........(15)}$$

The integrals in equation (15) may be evaluated for the first bending mode of the free beam specimen; this requires an equation for *M(x),* which can be developed from the first mode shape deflection equation for a prismatic beam with free end conditions, in transverse vibration given by

$$\int_l M^2 \, dx = 518.2 \frac{E^2 I^2 C_3^2}{l^3} \qquad \text{....................(16)}$$

where

L = the length of the beam specimen;

C3 = an arbitrary constant;

E = the Young's modulus in the 1-direction, for the three-phase compositematerial;

I = the moment of intertia for the rectangular crosssection of the beam

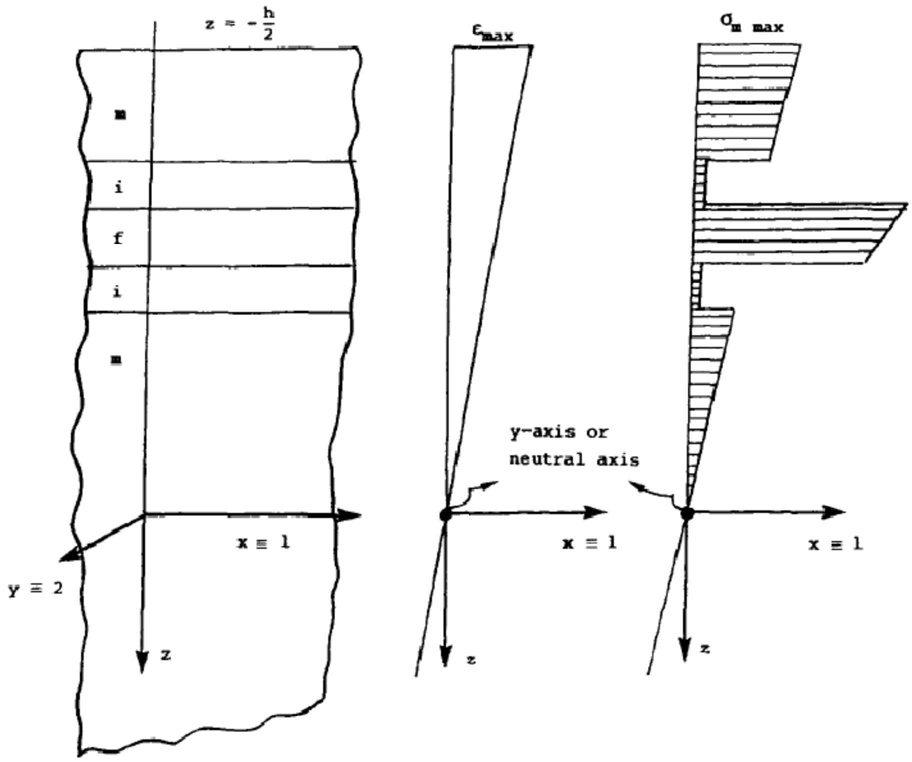


Fig6.3 Three phase model with strain and stress distributions for UD beam in bending vibration

Finally, after substituting the expressions obtained for the strain energy components into equation(16), and with simplifications, the following expression for η_1

$$\eta_1 = \eta_{fE}\frac{E_f I_f / E_m^2}{W} + \eta_{mE}\frac{I_m / E_m}{W} + \eta_{iG}\frac{21.6\int_{\Omega_i} S_{fict}^2 \, d\Omega_i \Big/ G_i b^2 l^2}{W}$$

where $$W = \frac{E_f I_f}{E_m^2} + \frac{I_m}{E_m} + \frac{21.6\int_{\Omega_i} S_{fict}^2 \, d\Omega_i}{G_i b^2 l^2}$$

'b' is the width of the rectangular cross-sectio and S_{fict} is a combination of the static moments of the different phases in the cross-section between the level on which shear is considered, z=*-h/2:*

$$S_{fict} = \int_{-h/2}^{-(h/2)-z} z \, d\Omega_m + \frac{E_i}{E_m}\int_{-h/2}^{-(h/2)-z} z \, d\Omega_i + \frac{E_f}{E_m}\int_{-h/2}^{-(h/2)-z} z \, d\Omega_f$$

6.2 Calculation of material damping of Glass fiber polyester

In this reinforced composite material there are different matrix and fibre phases :

Reinforcing material --- E-Glass fiber

Ef = 80Gpa
υf = 0.22
Gf = 32.7Gpa

Matrix ---polyester

Em = 3.5Gpa
υm = 0.25
Gm = 1.4Gpa

From graphs given in the book "Damping of material and members in structural mechanics" by LAZAN η_{mE} & η_{fE} are taken ; η_{mE} = 0.05 ,η_{fE} = 0.01

$$\eta_1 = \frac{\eta_{fE} E_f V_f + \eta_{mE} E_m V_m}{E_f V_f + E_m V_m}$$

$$\eta_1 = \frac{0.21975}{18.475} = 0.0118$$

$$\eta_2 = \frac{\eta_{fE} E_m V_f + \eta_{mE} E_f V_m}{E_m V_f + E_f V_m}$$

$$= \frac{1.0077}{18.475} = 0.0545$$

η_{mG} = 0.01 ,η_{fG} = 0.015

$$\eta_{12} = \frac{\eta_{fG} G_m V_f + \eta_{mG} G_f V_m}{G_m V_f + G_f V_m}$$

$$= \frac{0.08637}{8.483} = 0.0101$$

Applying all the stresses, assuming for a cycle of operations in a system of forces, the net material damping according to Koo & Lee [45] is given by

$$\eta_{av} = \frac{\eta 1 + \eta 2 + \eta 12}{60} = 0.0012$$

6.3 Calculating material damping of Glass fiber epoxy

Reinforcing material --- E-Glass fiber

$E_f = 80$ Gpa
$\upsilon_f = 0.22$
$G_f = 32.7$ Gpa

Matrix ---epoxy

$E_m = 3.5$ Gpa
$\upsilon_m = 0.25$
$G_m = 1.4$ Gpa

$\eta_{mE} = 0.03$, $\eta_{fE} = 0.01$;

$$\eta_1 = \frac{\eta_{fE} E_f V_f + \eta_{mE} E_m V_m}{E_f V_f + E_m V_m}$$

$$\eta_1 = \frac{0.2013}{18.475} = 0.0109$$

$$\eta_2 = \frac{\eta_{fE} E_m V_f + \eta_{mE} E_f V_m}{E_m V_f + E_f V_m}$$

$$= \frac{0.6077}{18.475} = 0.0328$$

$\eta_{mG} = 0.02$, $\eta_{fG} = 0.015$

$$\eta_{12} = \frac{\eta_{fG} G_m V_f + \eta_{mG} G_f V_m}{G_m V_f + G_f V_m}$$

$$= \frac{0.16812}{8.483} = 0.0198$$

The net material damping for a single fiber epoxy plate according to Koo & Lee is given by

$$\eta_{av} = \frac{\eta 1 + \eta 2 + \eta 12}{60} = 0.0010$$

For a single plate of glass fiber polyester and epoxy material damping is 0.0012 and 0.0010

For 'n' number of plates:

Loss factor for two plates $\eta_2 = \eta_1 \frac{W2}{W} + \eta_2 \frac{W1}{W}$, where W_1 and W_2 are the weights of the plates

Considering $W_1 = W_2$ (homogenous plates) $\eta_2 = 2\,\eta_1$ which implies $\eta_n = n\,\eta_1$

Table 6.1 Material damping of the composite layers

Number of plates	Glass fiber polyester	Glass fiber epoxy
1	0.0012	0.0010
2	0.0024	0.0020
3	0.0036	0.0030
4	0.0048	0.0040
5	0.0060	0.0050

6.4 Material damping of sandwich plates Glass fiber epoxy and polyester

In the sandwich plates the weights of the epoxy and polyester are different which means the plates are non-homogenous [46] ;

$$\eta_2 = \eta_1\frac{W2}{W} + \eta_2\frac{W1}{W} \quad \text{, (GF Polyester) } W_1 = 0.43 \text{ kg, (GF epoxy) } W_2 = 0.37 \text{ kg}$$

$$\eta_4 = \eta_1\frac{W4}{W} + \eta_2\frac{W3}{W} + \eta_3\frac{W2}{W} + \eta_4\frac{W1}{W} \quad , W_1 = W_3,\ W_2 = W_4$$

$$\eta_4 = 2\left[\,\eta_1\frac{W2}{W} + \eta_2\frac{W1}{W}\,\right] \qquad \eta_6 = 3\left[\,\eta_1\frac{W2}{W} + \eta_2\frac{W1}{W}\,\right]$$

$$\eta_8 = 4\left[\,\eta_1\frac{W2}{W} + \eta_2\frac{W1}{W}\,\right] \qquad \eta_{10} = 5\left[\,\eta_1\frac{W2}{W} + \eta_2\frac{W1}{W}\,\right]$$

Table 6.2 Material damping of the Sandwich plates

Number of plates	Sandwich plates
2	0.0011
4	0.0022
6	0.0033
8	0.0044
10	0.0055

Chapter 7

EXPERIMENTATION

7.1 Experimental set-up

Table 7.1 Specimen Details

Material No	Name of the Material	Cross section (mm)
1	Glass Fiber Polyester	210x210x6
2	Glass Fiber Epoxy	210x210x5
3	Mild steel	210x210x10

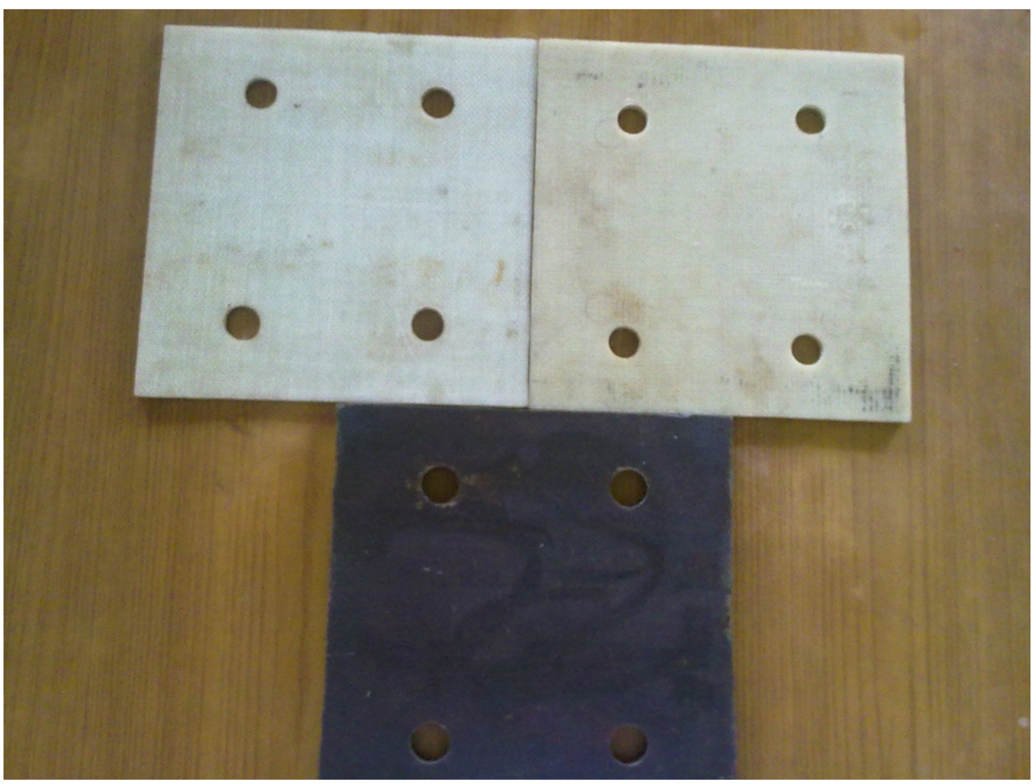


Fig 7.1 Glass fiber epoxy , Glass fiber polyester and Mild steel (clock wise starting from top left)

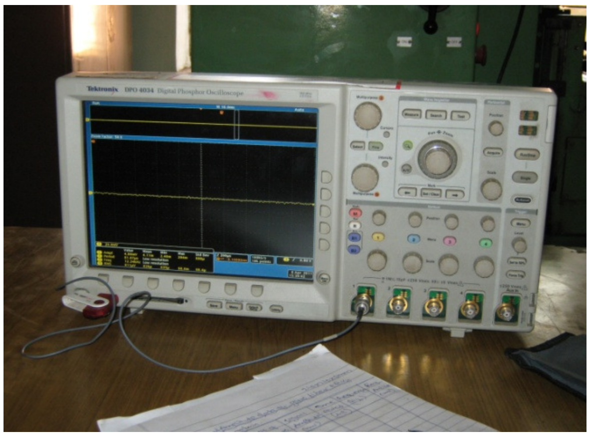
7.2 Instrumentation

The following equipment is needed in recording the amplitude, frequency, period of the vibrations during the machining operation

(1) Power supply unit

(2) Vibration pick-up

(3) Digital Storage Oscilloscope



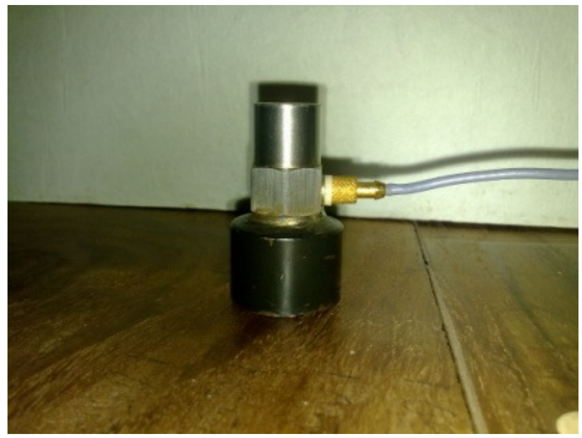


Fig 7.2 Digital Storage Oscilloscope of Tektronix 4000 series and Vibration pickup

Digital Storage Oscilloscope Tektronix 4000 series

Display: - 8x10 cm. rectangular mono-accelerator c.r.o. at 2KV e.h.t. Trace rotation by front panel present. Vertical Deflection: - Four identical input channels ch1, ch2, ch3, ch4.

Band-width:- (-3 db) d.c. to 20 MHz (2 Hz to 20 MHz on a.c.)

Sensitivity: - 2 mV/cm to 10 V/cm in 1-2-5 sequence.

Accuracy: - ± 3 %

Variable Sensitivity:- > 2.5% 1 range allows continuous adjustment of sensitivity from 2-1(mV/cm).

Input impedance: - 1M/28 PF appx.

Input coupling: - D.C. and A.C.

Input protection: - 400 V d.c.

Display modes: - Single trace ch1 or ch2 or ch3 or ch4. Dual trace chopped or alternate modes automatically selected by the T.B. switch.

7.3 Experimental procedure

The work specimen of 210mm x 210mm x10mm is a mild steel square plate. Four holes of 18mm diameter are drilled on the specimen at the corners. The glass fiber epoxy and polyester composite plates are thoroughly cleaned and polished. Plates are fixed on to a bench vice and the edges are filed to clear off the irregularities. All the plates are made to the exact dimensions for the ease of the further operations. Four holes are drilled on each plate and these holes are needed to be coaxial when the plates are placed upon one another and also with the mild steel. A right hand cut two-flutes drill bit of size 18mm is used to make holes. All the plates are carefully made homogenously similar to avoid interfacial vibrations and slipping. The work piece is then mounted onto the layered sheets of composites and tightly bolted to slotted table of the milling machine using square head bolts.

Initially five glass fiber polyester plates each of 6mm thickness are placed upon the bed along with mild steel. A contact type magnetic base vibration pickup connected to a digital phosphor storage oscilloscope of Tektronix 4000 series is placed on the mild steel during the machining operation. The response signals with respect to amplitude, time period, RMS amplitude and frequency are recorded and stored on the screen of the storage oscilloscope. Then the numbers of layers are reduced to four layers and the observations are recorded. In this way, the experiments are repeated by decreasing the number of layers of various composites. The experiments are conducted for 5,4,3,2,1 number of layers respectively. The whole process is again repeated using glass fiber epoxy plates each of 5mm thickness and also with the Sandwich plates (both fiber epoxy and polyester) combination of 10,8,6,4,2 layers respectively. Finally mild steel plate alone is machined with no layer under it and the readings are noted and compared.

An Upmilling cutting operation with constant feed of 16mm/min and depth of cut of 0.02mm is performed during all the experiments. An oil-water emulsion made from animal fat is used as a cutting fluid.

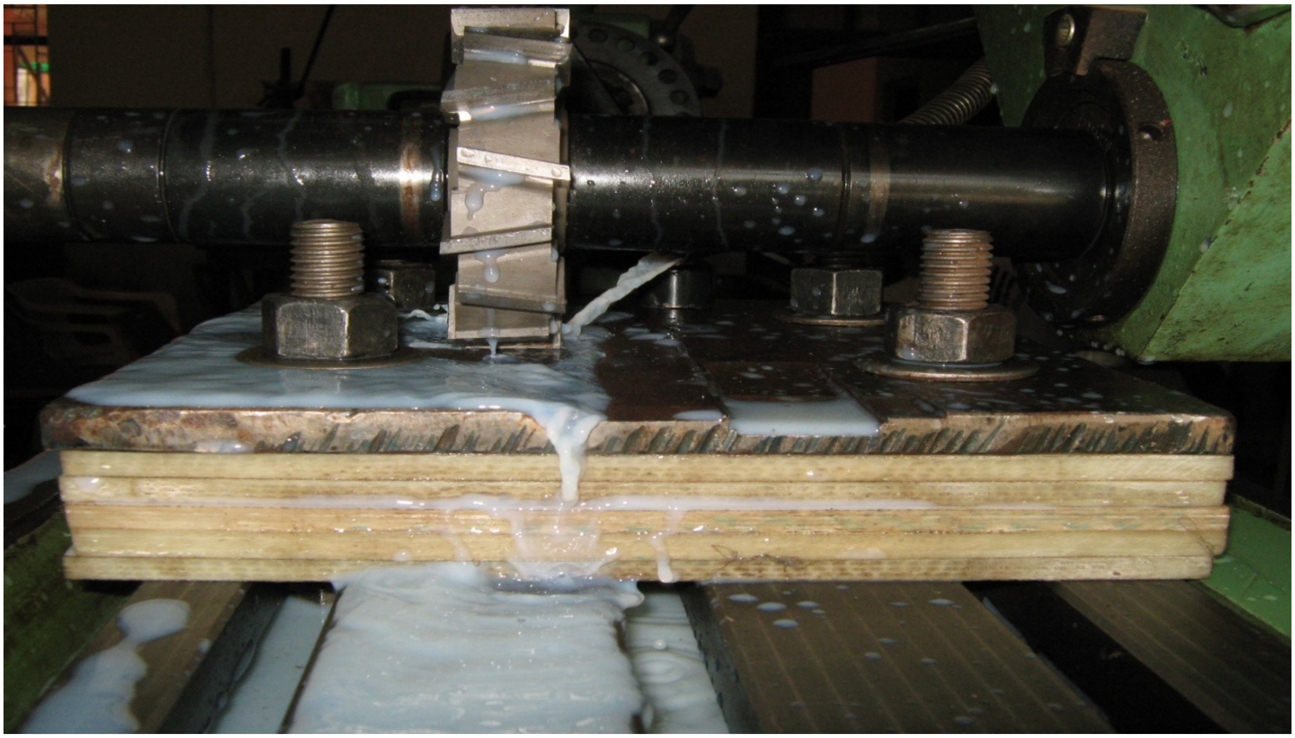


Fig 7.3 Five layers of Glass fiber polyester bolted to the slotted table milling machine

Fig 7.4 Three layers of Glass fiber polyester bolted to the slotted table milling machine

Fig 7.5 Four layers of Glass fiber epoxy bolted to the slotted table milling machine

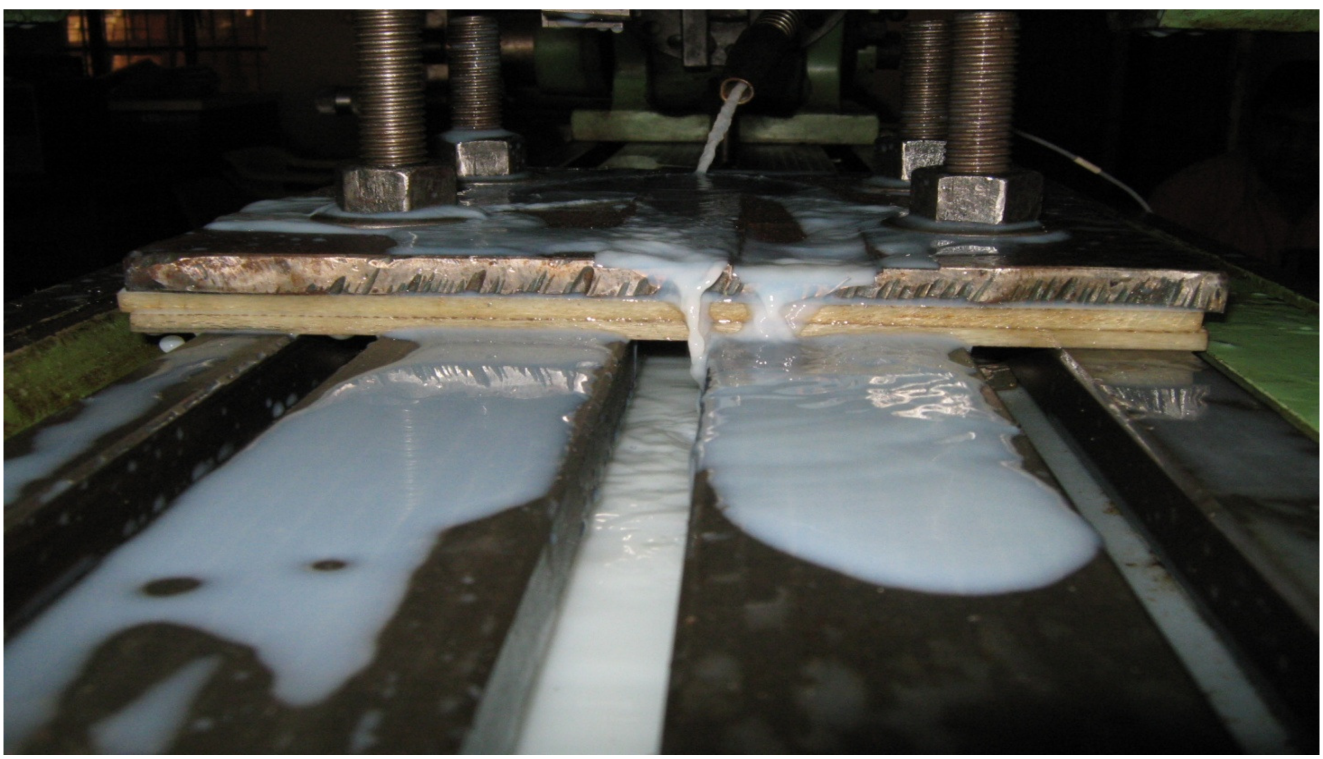


Fig 7.6 Two layers of Glass fiber epoxy bolted to the slotted table milling machine

Fig 7.7 Ten layered sandwich plates of Glass fiber epoxy and polyester bolted to the slotted table

Fig 7.8 Six layered sandwich plates of Glass fiber epoxy and polyester bolted to the slotted table

Fig 7.9 Six layered sandwich plates of Glass fiber epoxy and polyester bolted to the slotted table

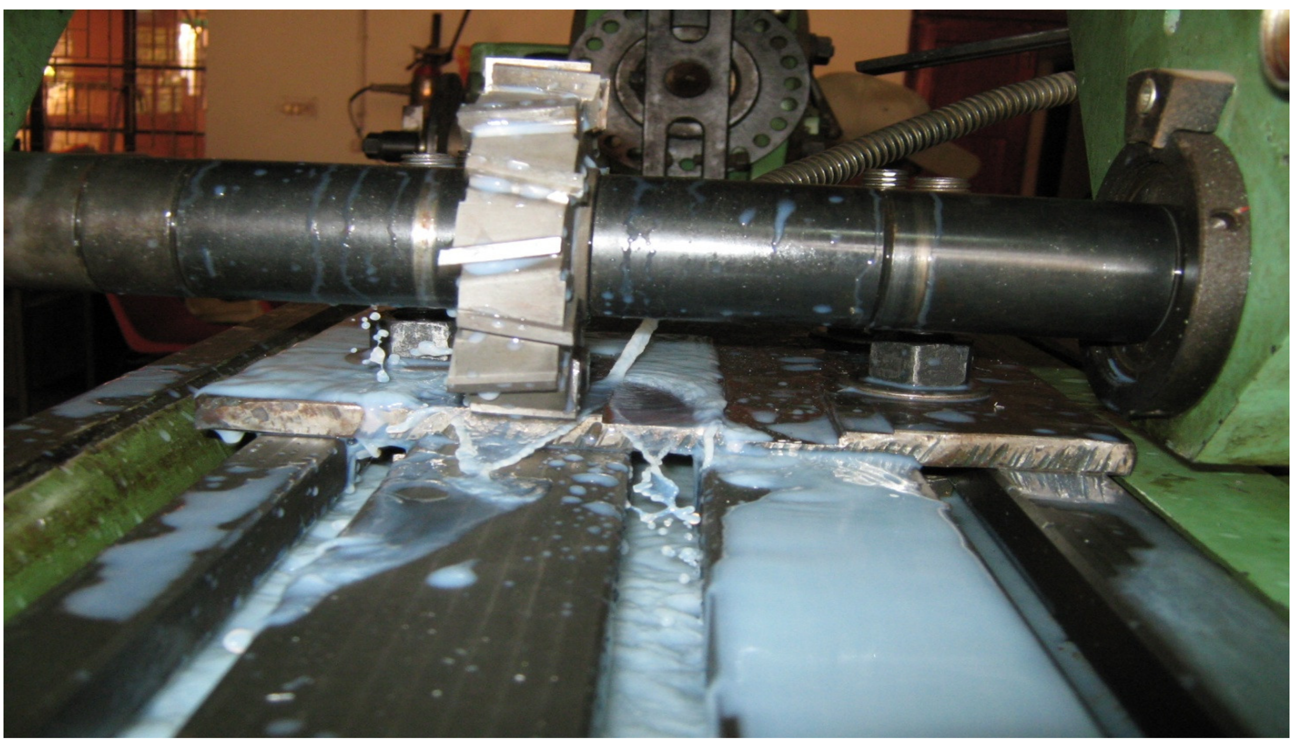


Fig7.10 Mild steel bolted to the slotted table milling machine

Chapter 8

Results and Discussion

8.1 Experimental Results

Table 8.1 Experimental Frequency and Amplitude data for Glass fiber polyester

Sl.no	Depth of cut (mm)	Feedrate (mm/min)	Number of layers	Signal Amplitude(mV)	Time Period(µs)	Frequency (KHz)	RMS Amplitude(mV)
1	0.02	16	5	49.6	292.0	3.425	9.99
2	0.02	16	4	46.4	339.0	2.786	10.2
3	0.02	16	3	23.2	978.7	1.022	5.10
4	0.02	16	2	30.4	510.0	1.961	6.75
5	0.02	16	1	52.4	902.5	1.108	13.4

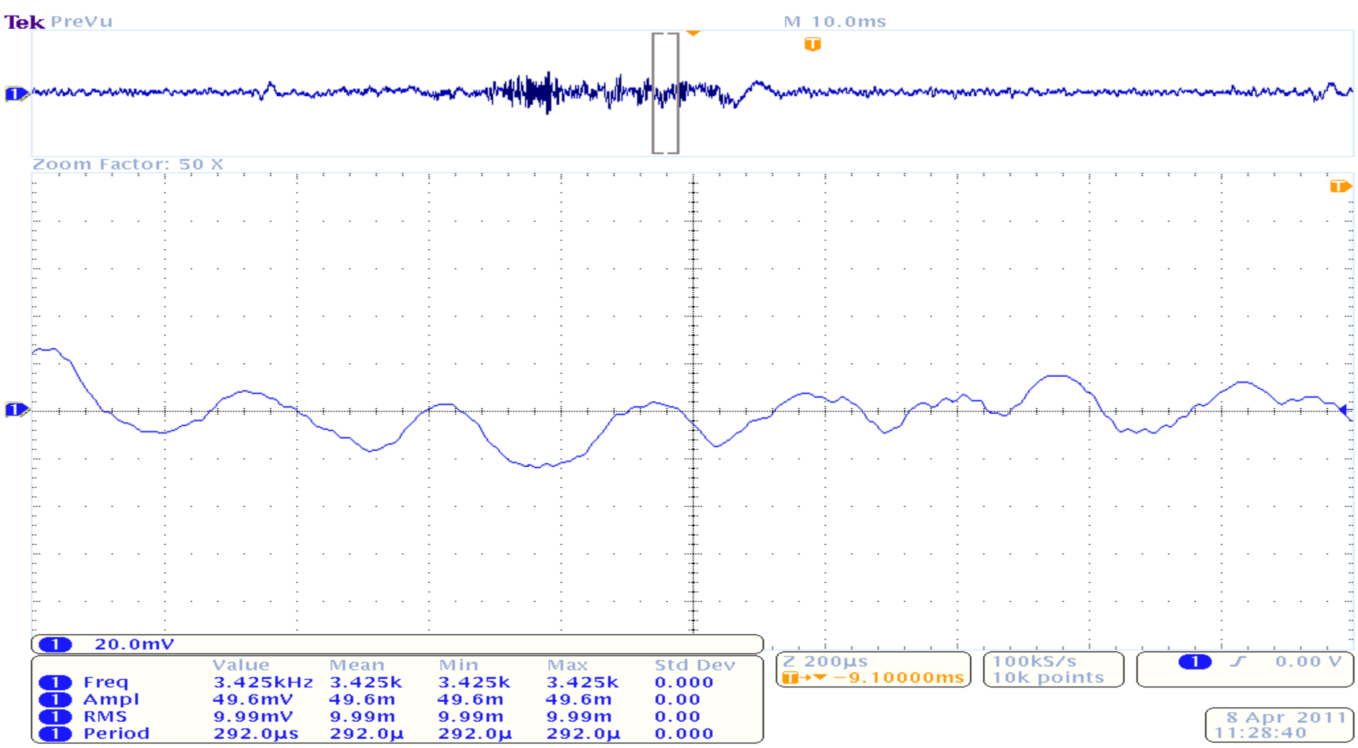


Fig 8.1 Vibration signal for five layers of Glass fiber polyester plates

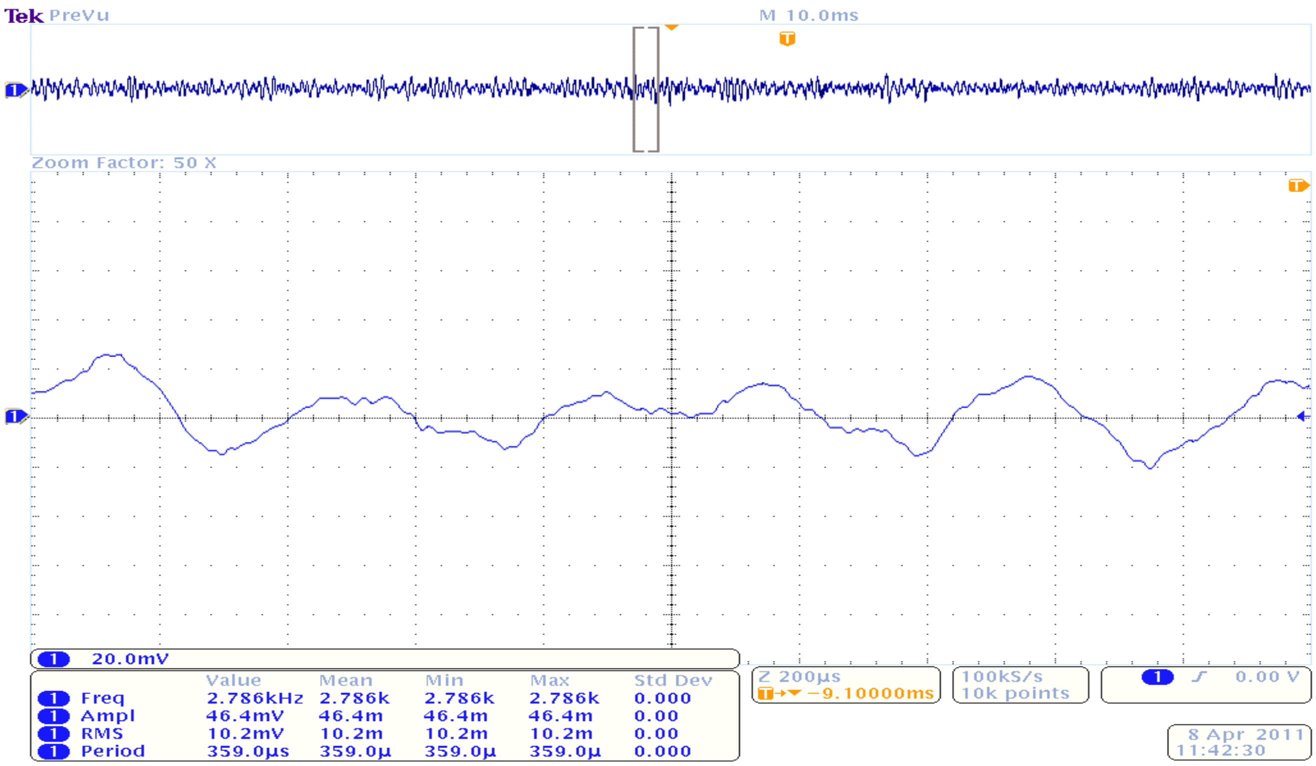


Fig 8.2 Vibration signal for four layers of Glass fiber polyester plates

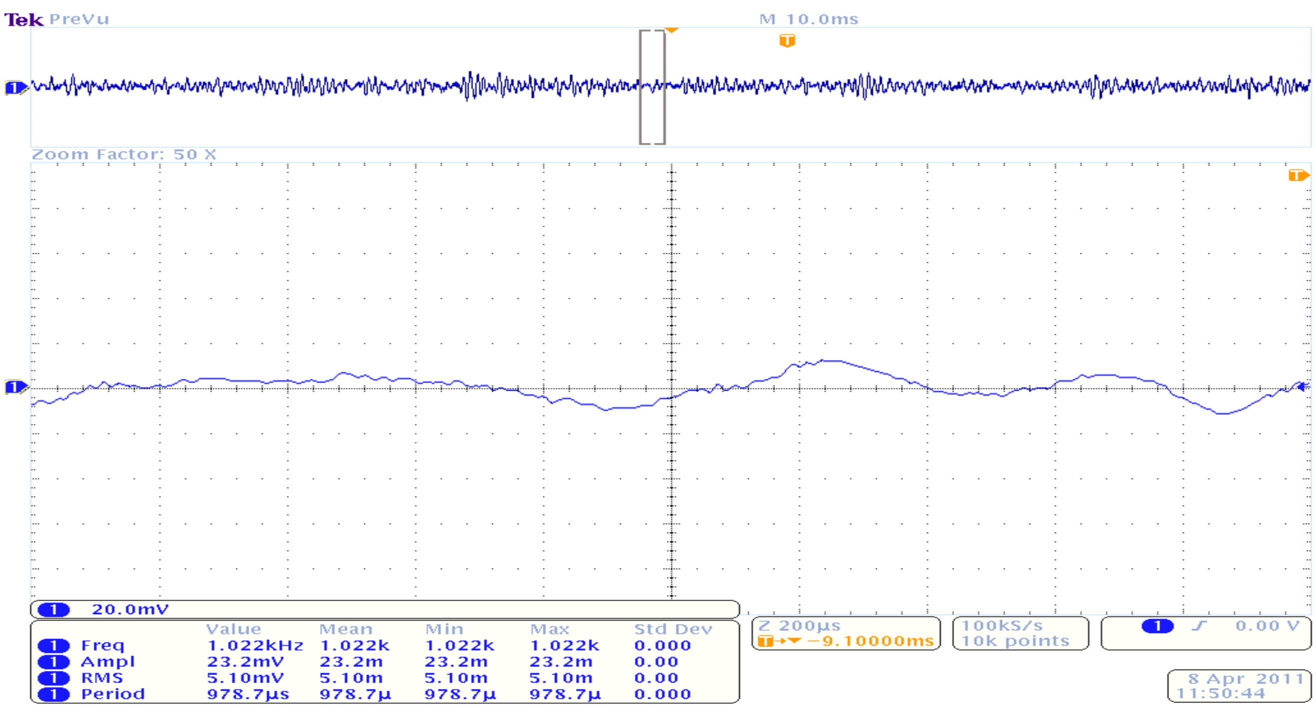


Fig 8.3 Vibration signal for three layers of Glass fiber polyester plates

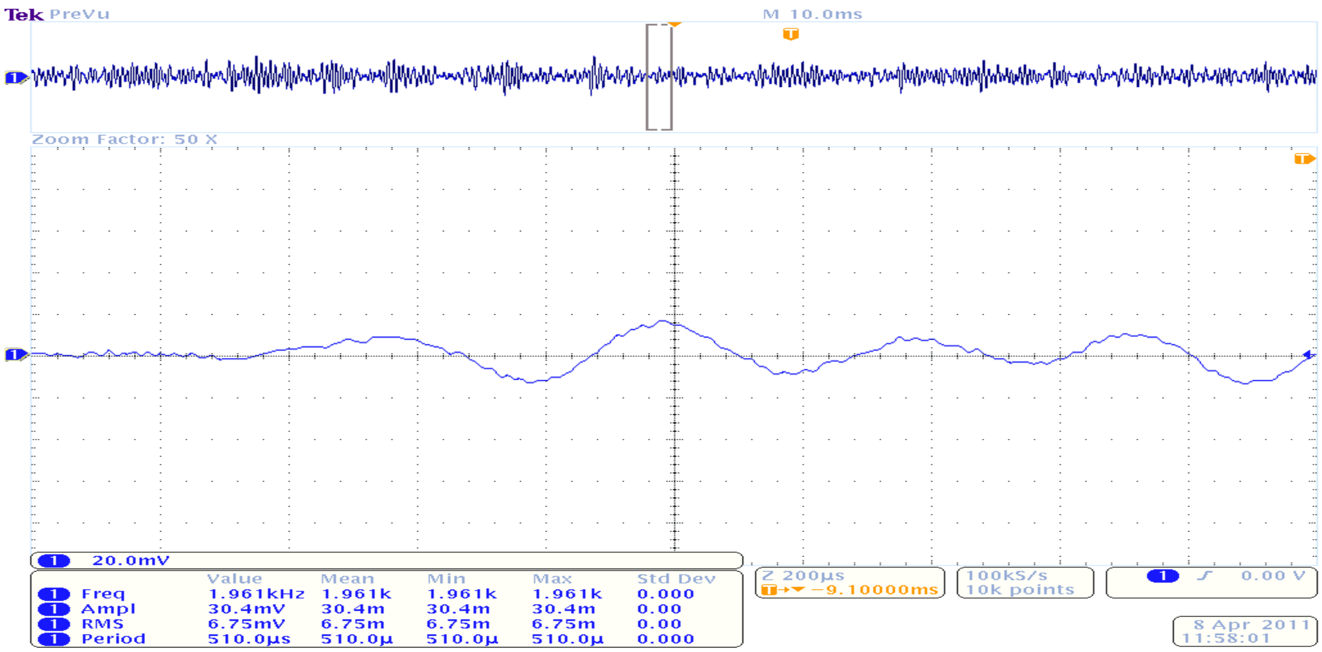


Fig 8.4 Vibration signal for two layers of Glass fiber polyester plates

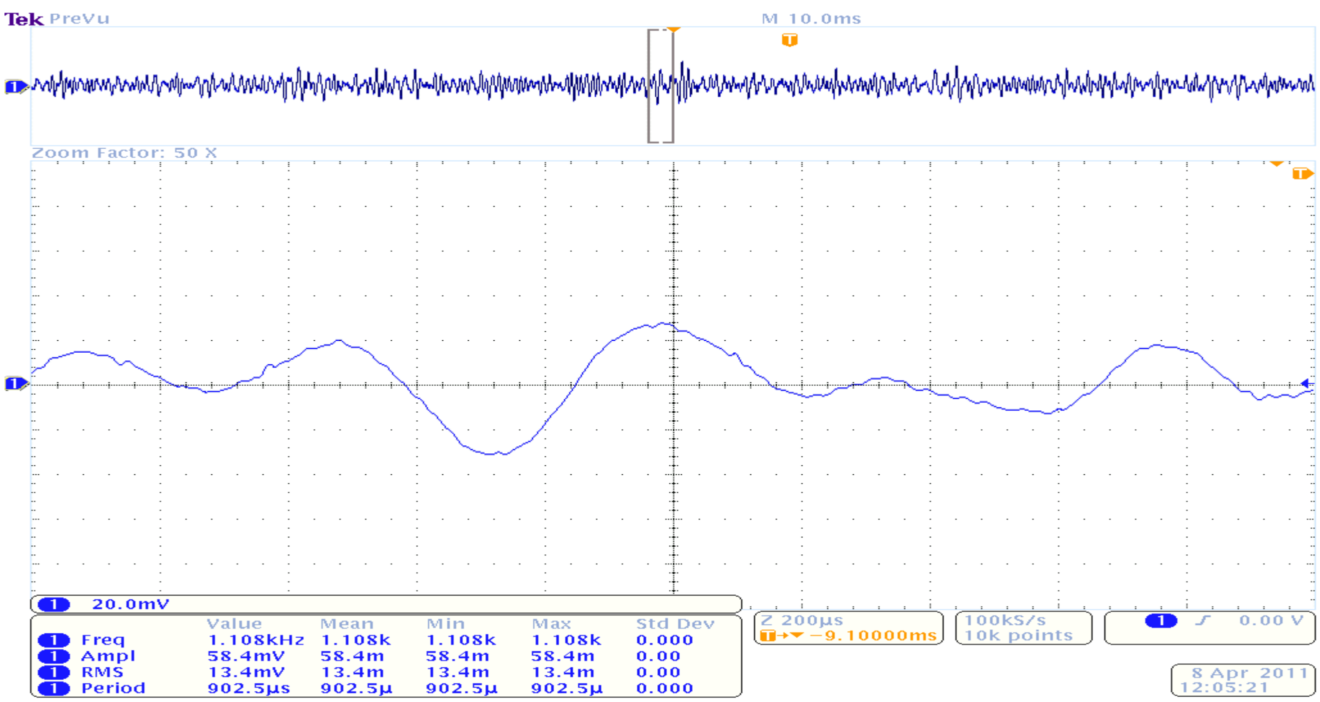


Fig 8.5 Vibration signal for single layer of Glass fiber polyester plate

Table 8.2 Experimental Frequency and Amplitude data for Glass fiber epoxy

Sl.no	Depth of cut (mm)	Feedrate (mm/min)	Number of layers	Signal Amplitude(mV)	Time Period(μs)	Frequency (KHz)	RMS Amplitude(mV)
1	0.02	16	5	67.2	421.9	2.37	16.8
2	0.02	16	4	51.2	558.1	1.792	13.2
3	0.02	16	3	40.0	537.5	1.860	9.04
4	0.02	16	2	28.8	441.7	2.264	7.61
5	0.02	16	1	20.8	845.0	1.183	5.26

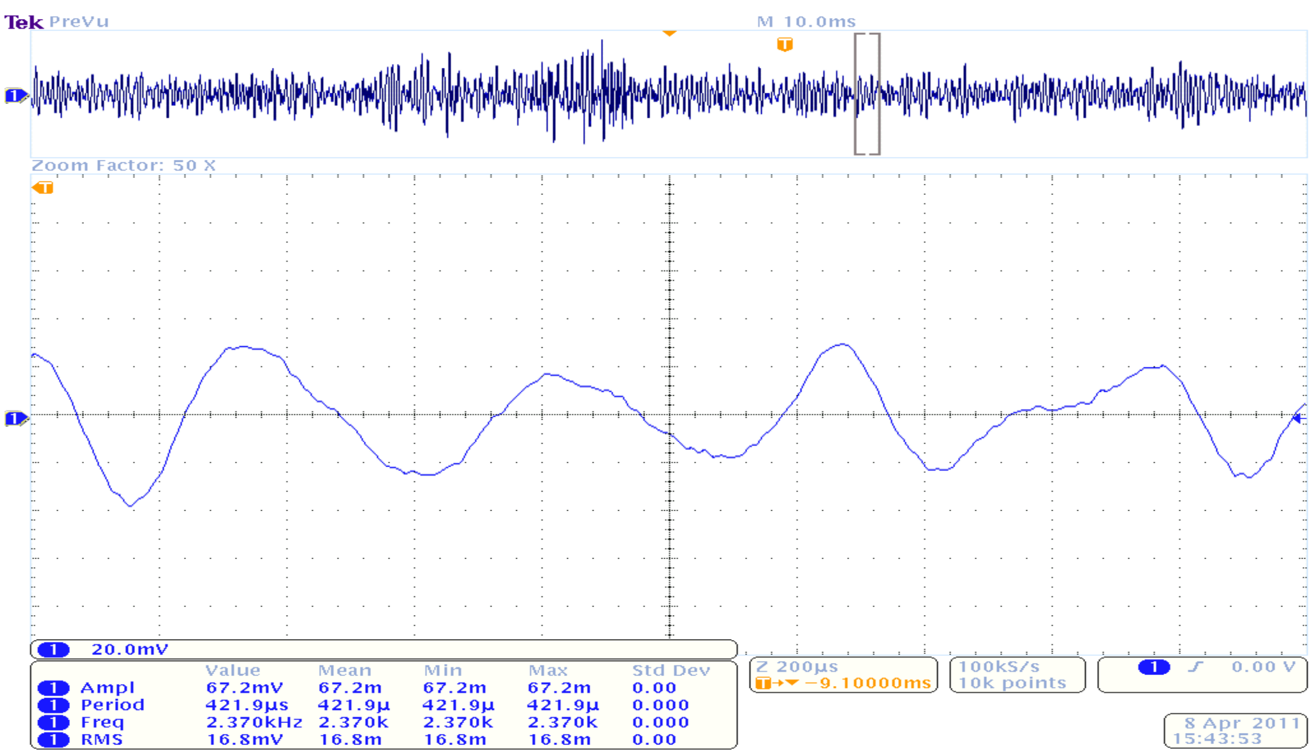


Fig 8.6 Vibration signal for five layers of Glass fiber epoxy plates

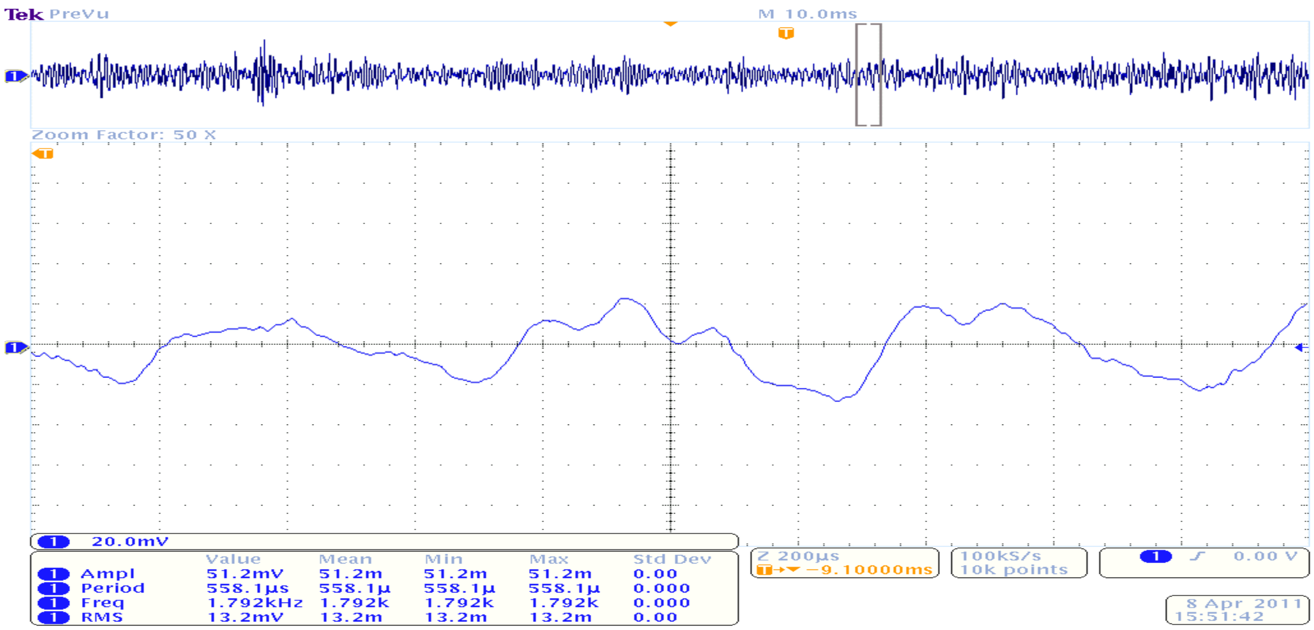


Fig 8.7 Vibration signal for four layers of Glass fiber epoxy plates

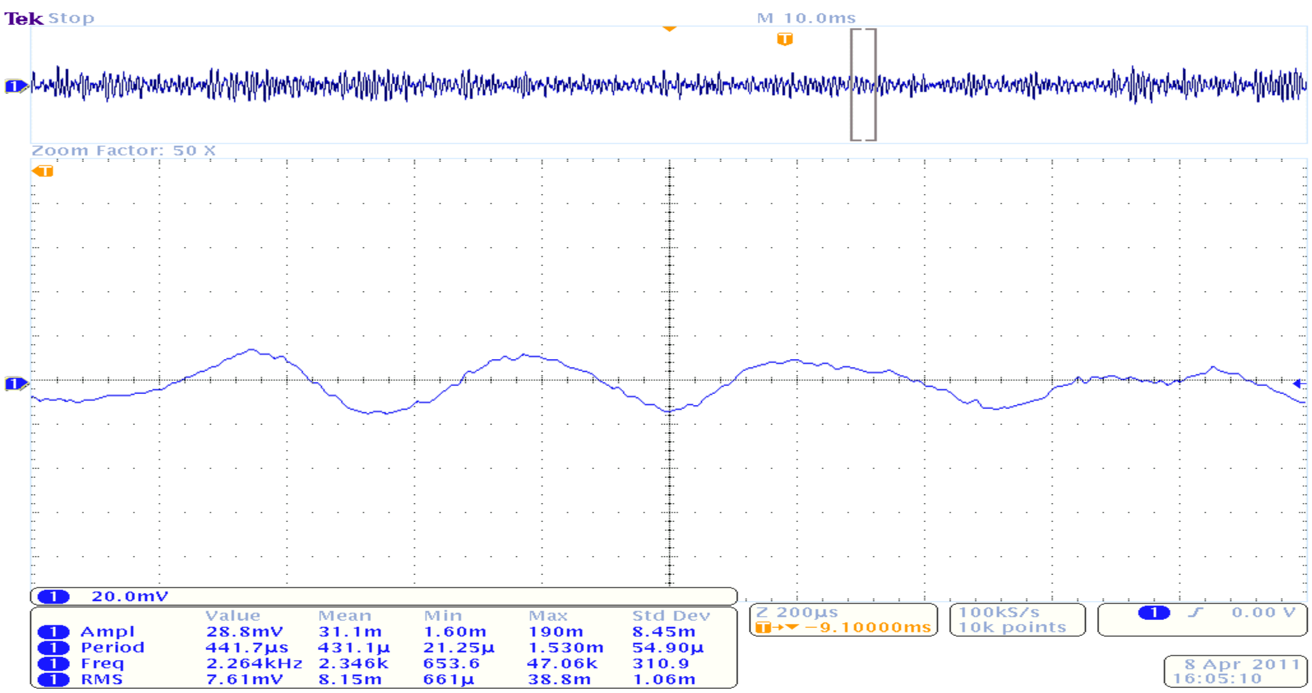


Fig 8.8 Vibration signal for two layers of Glass fiber epoxy plates

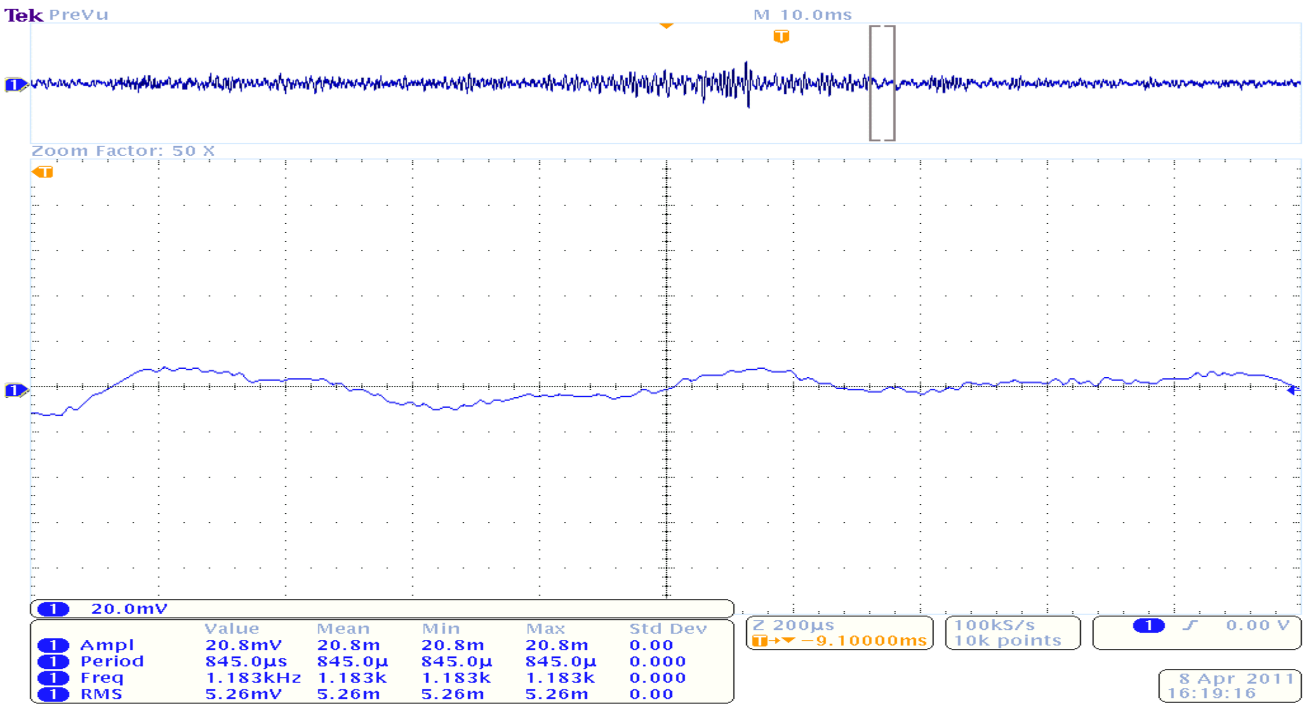


Fig 8.9 Vibration signal for single layer of Glass fiber epoxy plate

Table 8.3 Experimental data for the sandwich plates of Glass fiber epoxy and polyester

Sl.no	Depth of cut (mm)	Feedrate (mm/min)	Number of layers	Signal Amplitude(mV)	Time Period(µs)	Frequency (KHz)	RMS Amplitude(mV)
1	0.02	16	10	39.2	740.0	1.351	9.56
2	0.02	16	8	49.6	716.7	1.395	11.7
3	0.02	16	6	33.6	692.5	1.444	8.06
4	0.02	16	4	59.2	502.3	1.99	13.5
5	0.02	16	2	65.6	437.5	2.286	14.1

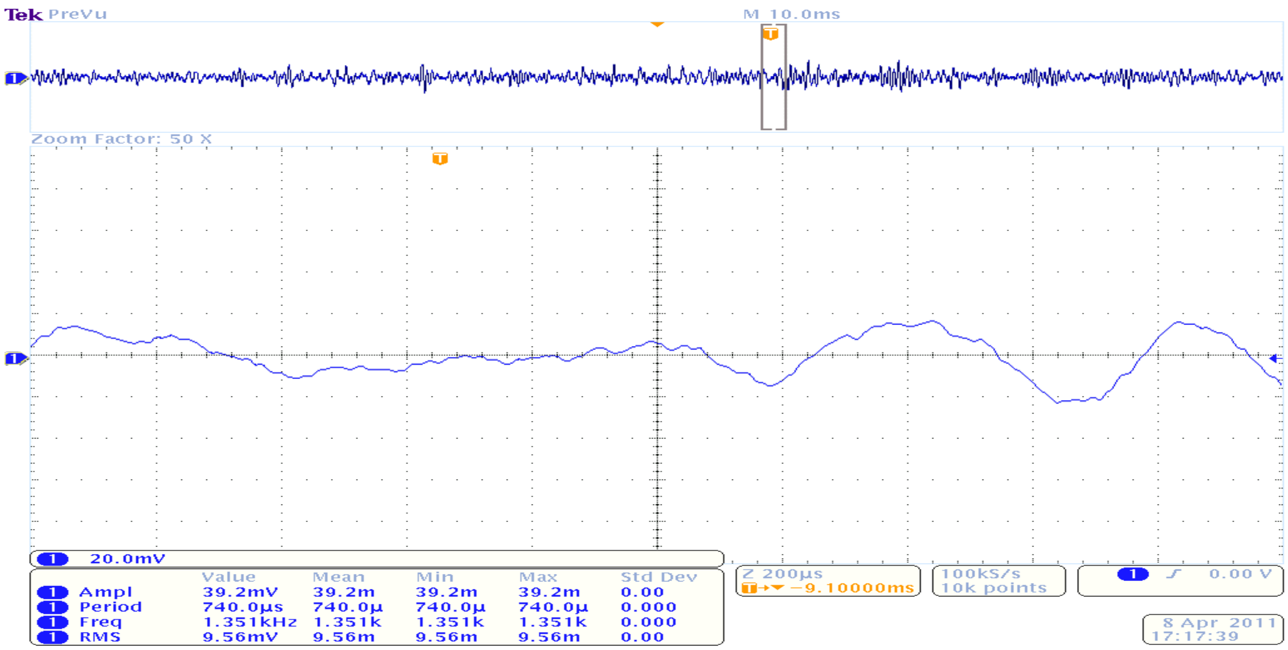


Fig 8.10 Vibration signal for ten layers sandwich plates of Glass fiber epoxy and polyester

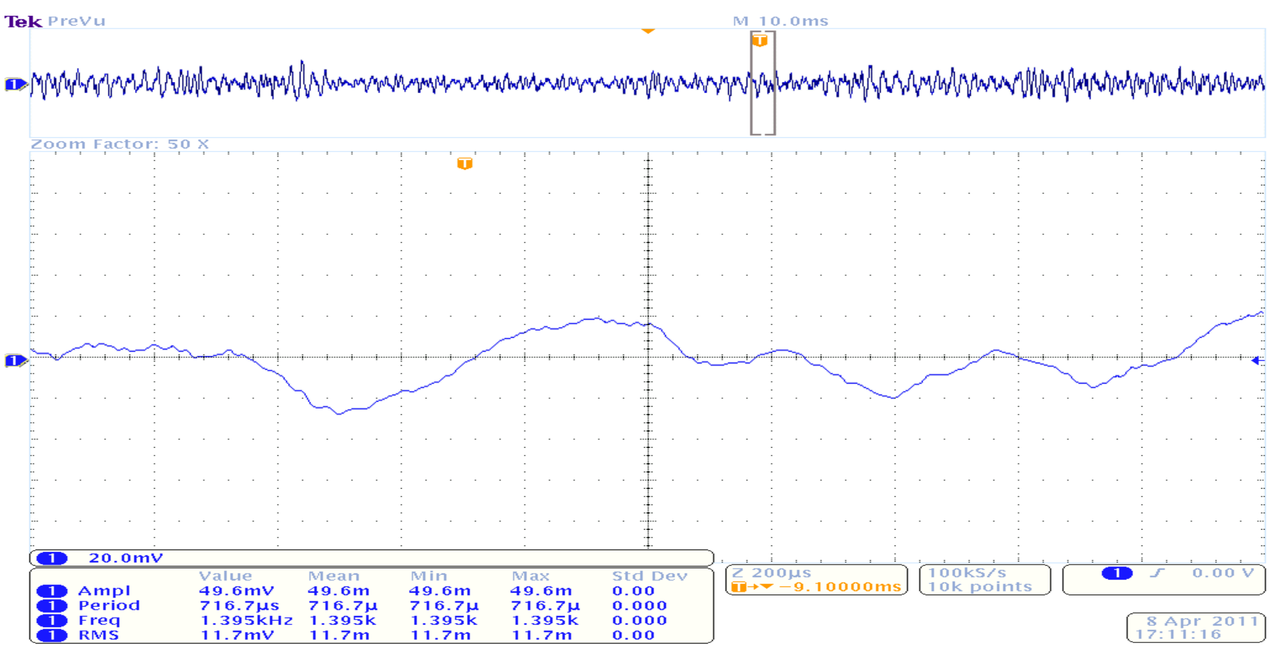


Fig 8.11 Vibration signal for eight layers sandwich plates of Glass fiber epoxy and polyester

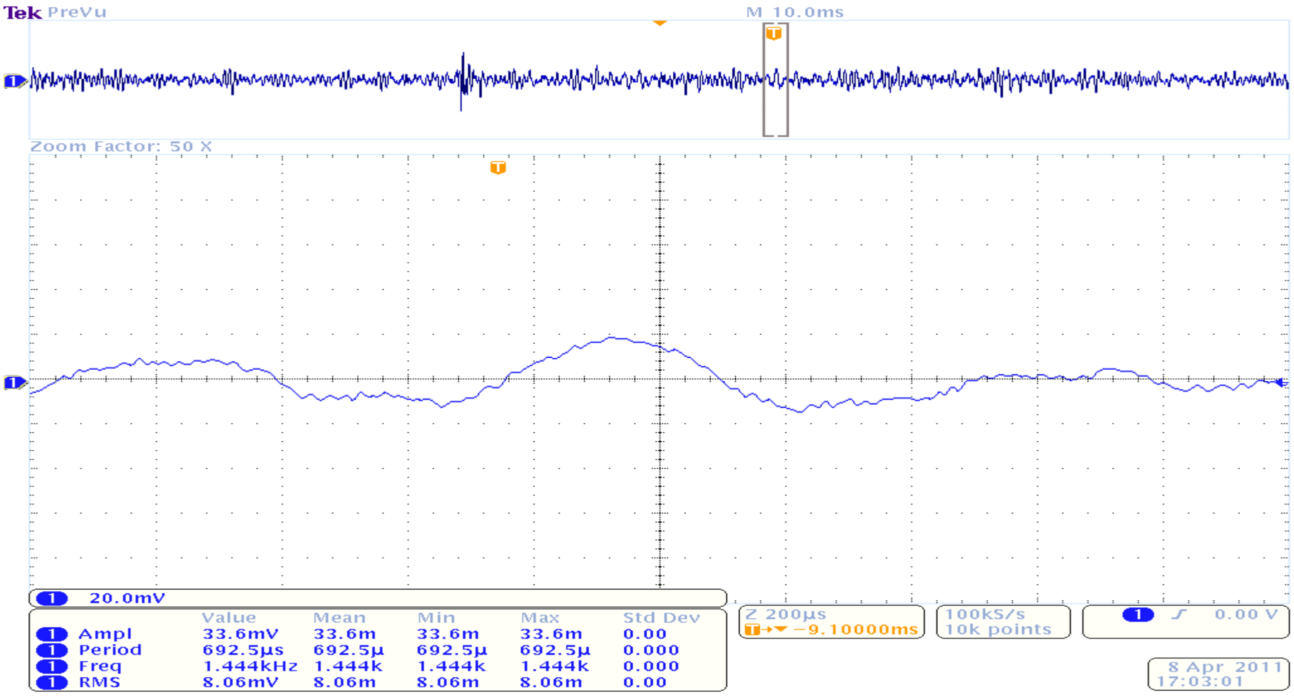


Fig 8.12 Vibration signal for six layers sandwich plates of Glass fiber epoxy and polyester

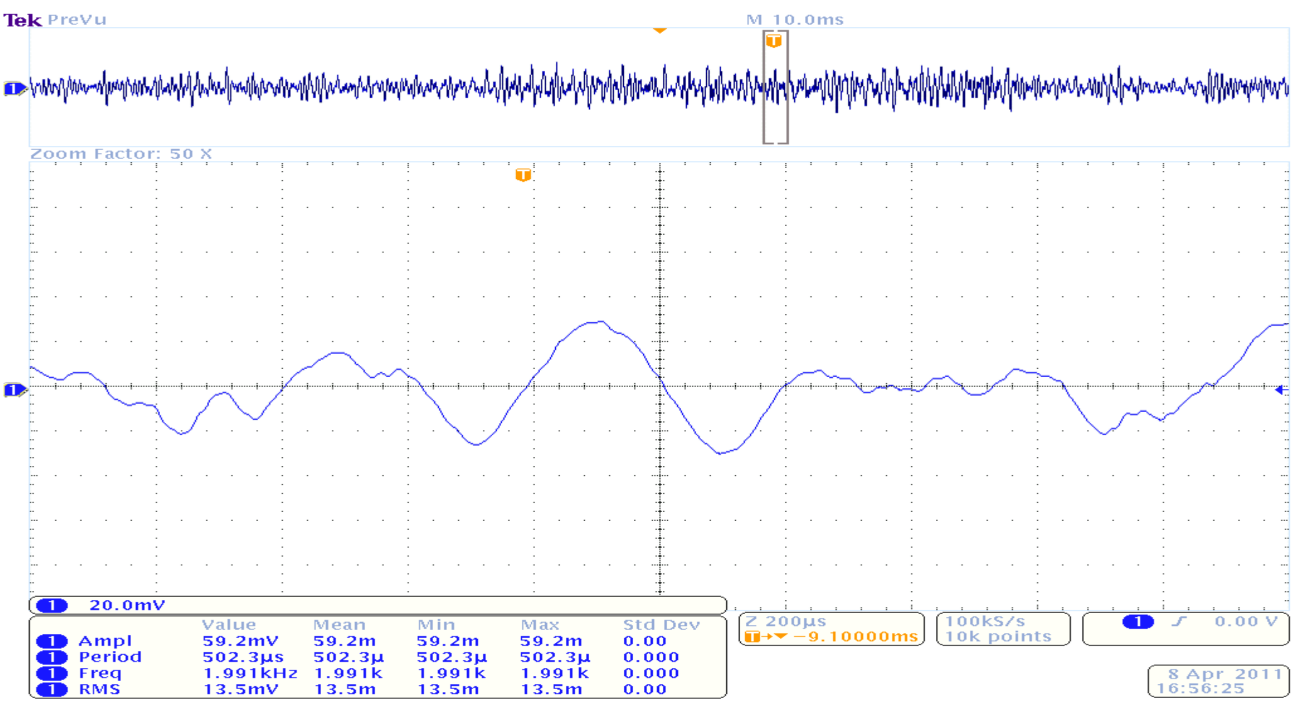


Fig 8.13 Vibration signal for four layers sandwich plates of Glass fiber epoxy and polyester

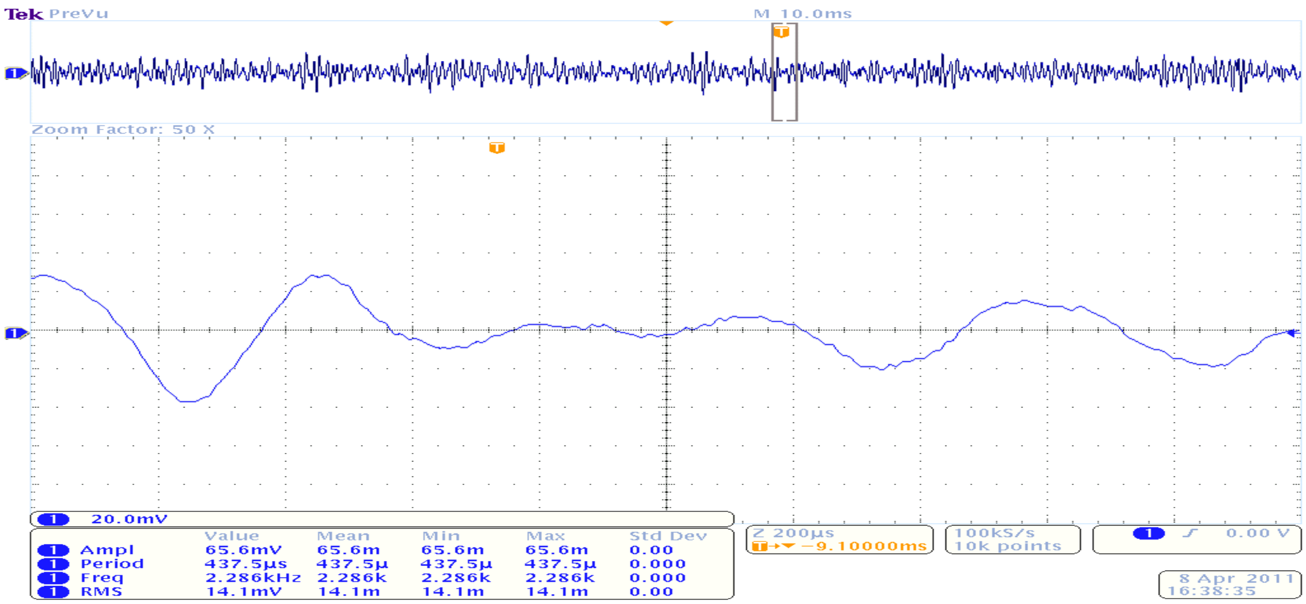


Fig 8.14 Vibration signal two layers sandwich plates of Glass fiber epoxy and polyester

Table 8.4 Experimental data for the Mild steel plate

Sl.no	Depth of cut (mm)	Feedrate (mm/min)	Number of layers	Signal Amplitude(mV)	Time Period(µs)	Frequency (KHz)	RMS Amplitude(mV)
1	0.02	16	1	59.2	618.6	1.617	15.6

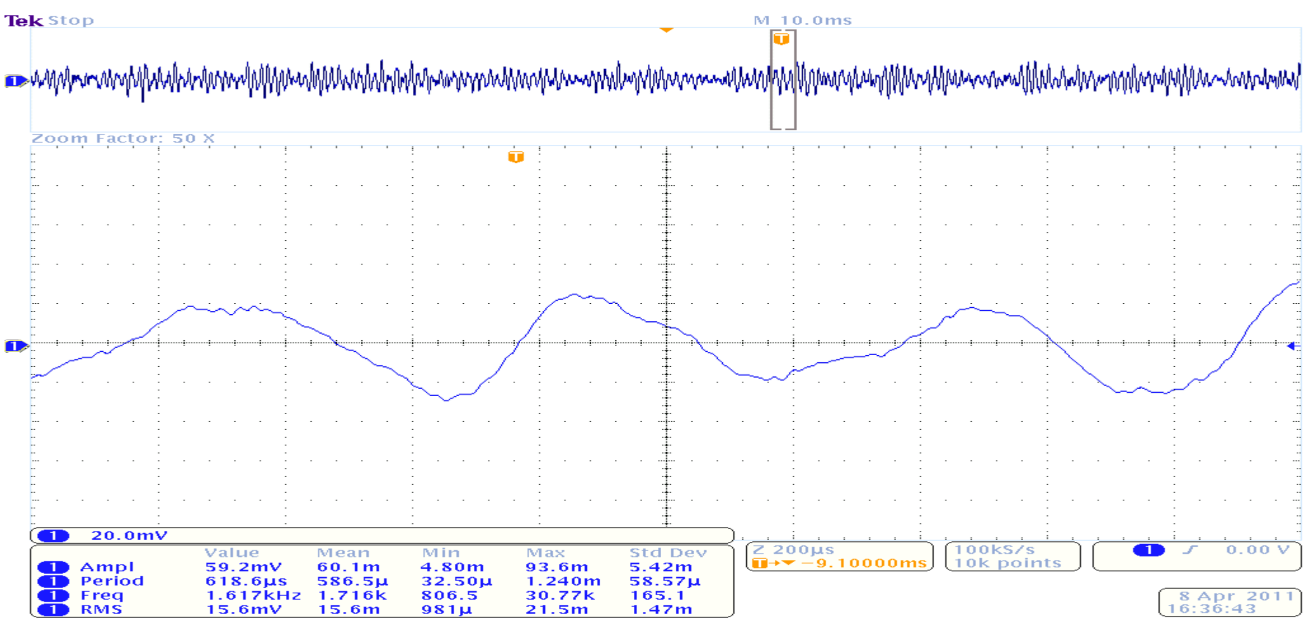


Fig 8.15 Vibration signal for the mild steel plate

8.2 Discussions

The above graphs show the variation of signal amplitude with respect to number of layers for different combinations of composites. It is observed that when the numbers of layers are increased, the signal amplitude has decreased for both the composites to a certain extent and then increased abruptly. The maximum amplitude is obtained when no composite material was used indicating that the presence of composite materials decreases the vibration amplitude and increases the counter vibration characteristics of the system. This shows that with increase in the plates the damping can be increased but only to a certain limit and it would have a negative effect with much of progress. Hence optimum level of plates is to be decided to profitably damp out the vibrations. This optimum number of plates is different for glass fiber polyester and glass fiber epoxy.

Table 8.5 Optimum no. of plates and Height of machine too bed

Type of composite	**Optimum no. of plates**	**Height of the machine bed**
Glass fiber polyester	Three	18 mm
Glass fiber epoxy	Two	10mm
Sandwich plates	Six	33mm

Chapter 9

CONCLUSION

1) Use of composite materials reduces the vibrations of the system as desired which is justified from the experimental observations. With increase in number of layers of composites at an optimum level the vibrations are decreased considerably.
2) Effective damping can be obtained only by proper fixation of the composites to the bed and the work piece. With improper nut and bolt joint there is a danger of additional slip vibrations between the plates. Hence a proper and intact joint is preferably necessary. On the contrary the optimum number of plates is decided and a single plate of optimum thickness is used as the bed material.
3) Extensive experiments with different layer combinations along with sandwich plates are carried out to determine vibration response of work specimen with specified machining parameter, i. e., depth of cut and feed rate. The experiments are duly carried out at small feed, low depth of cut and low cutter speed to primarily investigate the scope of damping phenomenon in composite materials.
4) Abrupt increase in vibration amplitude has also been observed with increase in number of layers of composites above an optimum limit interposed between the table and work piece
5) The results obtained are compared with respect to each other. Out of the two materials, signal amplitudes obtained are less for Glass fiber epoxy material. Therefore, it can be concluded that Glass fiber epoxy material can be used for machine tool structures to reduce the undesirable effects of vibrations.
6) The density of the matrix phase plays an important role in damping the vibrations. With same fiber phase, lower the matrix phase density more is the damping ability. Though both the thermosets polyester and epoxy have same material properties like Young’s modulus, Rigidity modulus and Poisson’s ratio, epoxy has more damping ability than polyester because of its low density.

Chapter 10

SCOPE OF FUTURE WORK

As the forces in the milling machine are bidirectional the total damping (in terms of loss factor) cannot be calculated from the oscilloscope single amplitude readings alone. In addition to oscilloscope, dynamometer and FFT analyzer has to be used. In each experimental step for each vibration curve, the single amplitude and frequency at various disturbance zones are to be taken and total damping (in terms of loss factor) is calculated from the graph plotted between frequency and amplitude which on whole is a tedious method. Hence, FFT analyzer is connected to oscilloscope on a circuit basis to get the final FFT curve for each experimental step. Dynamometer is used to calculate the cutting forces during the experiments which has reasonable effects on the vibration curves.

By testing suitable damping materials for structures according to the design requirements one can use the findings of the present work in various vibration problems. Experimental technique used in this report can be applied to achieve vibration isolation in different machine tool structures.

The present work can be extended for other types of polymer and metal matrix composites with fibers like carbon, boron with different types of fiber orientation.

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