Vibration Analysis of the Platter and the Spindle Assembly of Hard Disk Drive

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By

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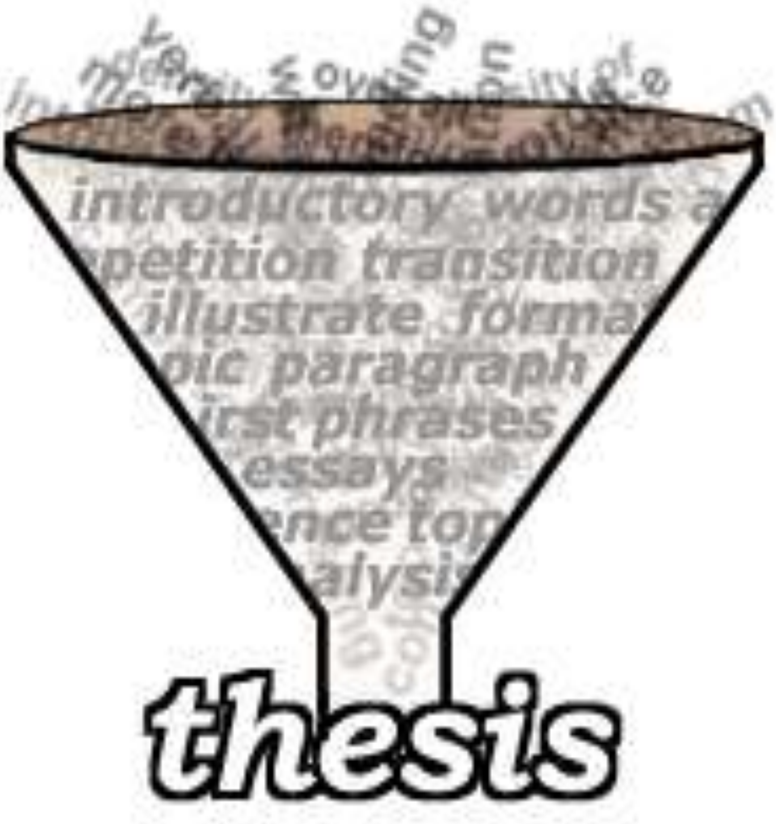

I would like to start my thesis with the following quote;

"Knowing is not enough; we must apply. Willing is not enough; we must do."(Johann Wolfgang von Goethe)

Before starting my research I had knowledge but did not know when and where to apply. After going through all the tough times throughout my research I have learnt how to use your knowledge. Even if you have little knowledge you must know how and where to apply.

Acknowledgment

First and foremost, I would like to offer my deepest gratitude to the supervisor of this research: Dr. Jen-Yuan (James) Chang, who, with his guidance and tuition allowed for the completion of this dissertation. Without his help and support throughout the research it would have been impossible to complete.

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Working towards the Master’s Degree in Massey University was the most important, amazing and astonishing experience in my life. This research and training has completely changed the way of my thinking toward problem solving.

Abstract

As the demand for higher data transfer rate in a hard disk drive (HDD) has increased, faster rotational disk speed has been imposed. Unfortunately, higher rotational disk speed leads to higher vibration and that leads to high frequent track misregistration (TMR). In order to solve this problem, a new damping system has been suggested that will help to reduce vibration in high speed hard disk drives.

The thesis contains a detailed study of vibration and its types. Causes of vibrations have been discussed too.

The thesis also contains a study of vibration and its effects on rotating disc such as HDD Platter. Detailed vibration analysis was performed on the HDD platter and spindle-assembly so that natural frequencies of all vibrational modes can be calculated.

Research includes, the thermal analysis of spindle disk assembly used in computer hard disk with new design approach of spindle to minimize the repeatable run-out (RRO) of track following position error signal (PES) in high track per inch (TPI) disk drives.

The thesis proposed a simple but analytical approach for the design of HDD enclosure with different sector shapes and for the spindle motor with different slot/notch angles, based on the analysis and simulation results.

Based on the simulation and experimental results, the proposed method can be introduced as a very promising tool for the design of HDDs and various other high performance computer disk drives such as floppy disk drives, CD ROM drives, and their variations having spindle mechanisms similar to those of HDDs.

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CHAPTER 1: Introduction

1.1 Problem Description

The hard disk drive or HDD plays an important role in the modern era of digital technology. The HDD industry began its journey in 1956, and since then, it has travelled through a history of extra-ordinary achievements. Storage capacity of the HDD has grown from mere 5 MB (Mega Bytes) in 1956 on fifty 24-inch disks to more than 1000GB (Giga Bytes) stored on one disk of 2 ½ inch diameter. During this relatively short period, the HDD industry has fostered excellent innovations in various scientific and technological disciplines related to the design and manufacturing of HDD.

As the demand for bigger storage capacity and higher data transfer in a hard disk drive (HDD) has increased, higher rotational disk speed and disk with many more tracks per inch (TPI) has been imposed. Unfortunately, higher rotational disk speed leads to high frequent track misregistration (TMR). Due to the increase in the rotational speed of the disk, disk vibration has become very important factor in the performance of HDDs.

Aim of the research was to analyse the vibration generated by computer hard disk drives at different spinning speeds and also with different case designs and then find the best suitable running speed of HDD motor and then develop a damping system that will reduce or eliminate the vibration from platter of the disk.

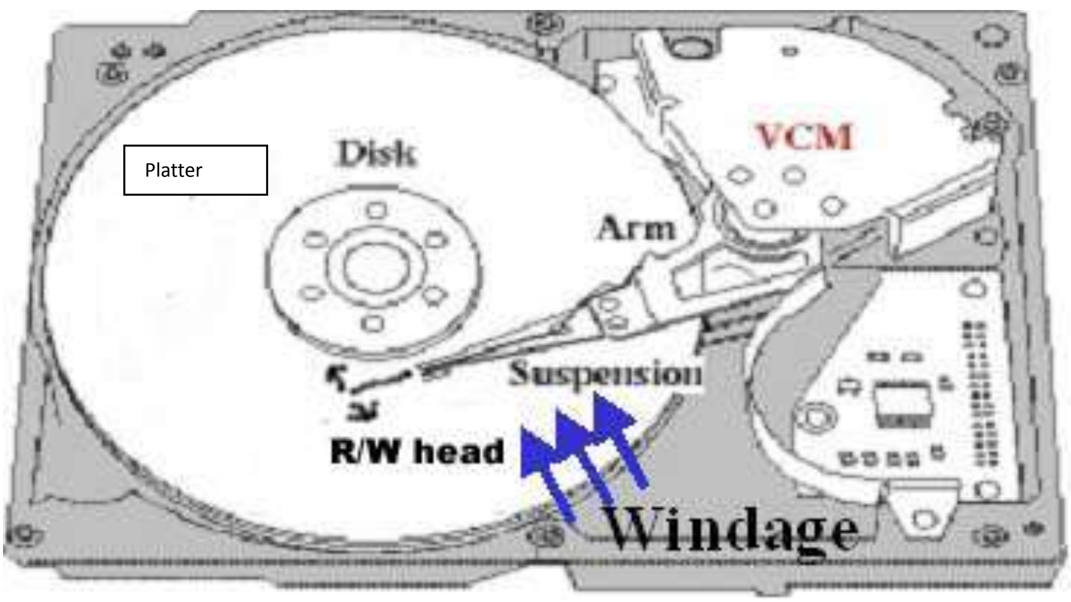
To analyze the vibrations of the HDDs, the vibration natural frequencies of disk platters are computed theoretically and by FEM in COSMOS (Solid Works). Initially, a vibration analysis was carried on 2.5inch Hard Disk Drives running at 7200RPM and 5400RPM speed. These natural frequencies will also be verified experimentally by modal testing.

1.2 Motivation

> "The limiting factor of data access speed in the modern day computers is no longer the speed of the Central Processing Unit (CPU), commonly known as the processor, but the speed of the HDD". *Professor Lasse Natvig*

There are many ways of facilitating higher data transfer in a hard disk drive (HDD). Software can optimize to meet specific restrictions for particular applications. Concerning hardware, an increase of the rotational speed has proven to be an efficient solution (Deeyiengyang & Ono, 2001). Unfortunately, a higher rotational disk speed leads to more mechanical vibrations on the disk platter due to the induced wind in the drive. Because of the mechanical vibrations, the read/write (R/W) head is affected with a stochastic offset, called a position error. A schematic overview of the system is given in Figure 2.1.

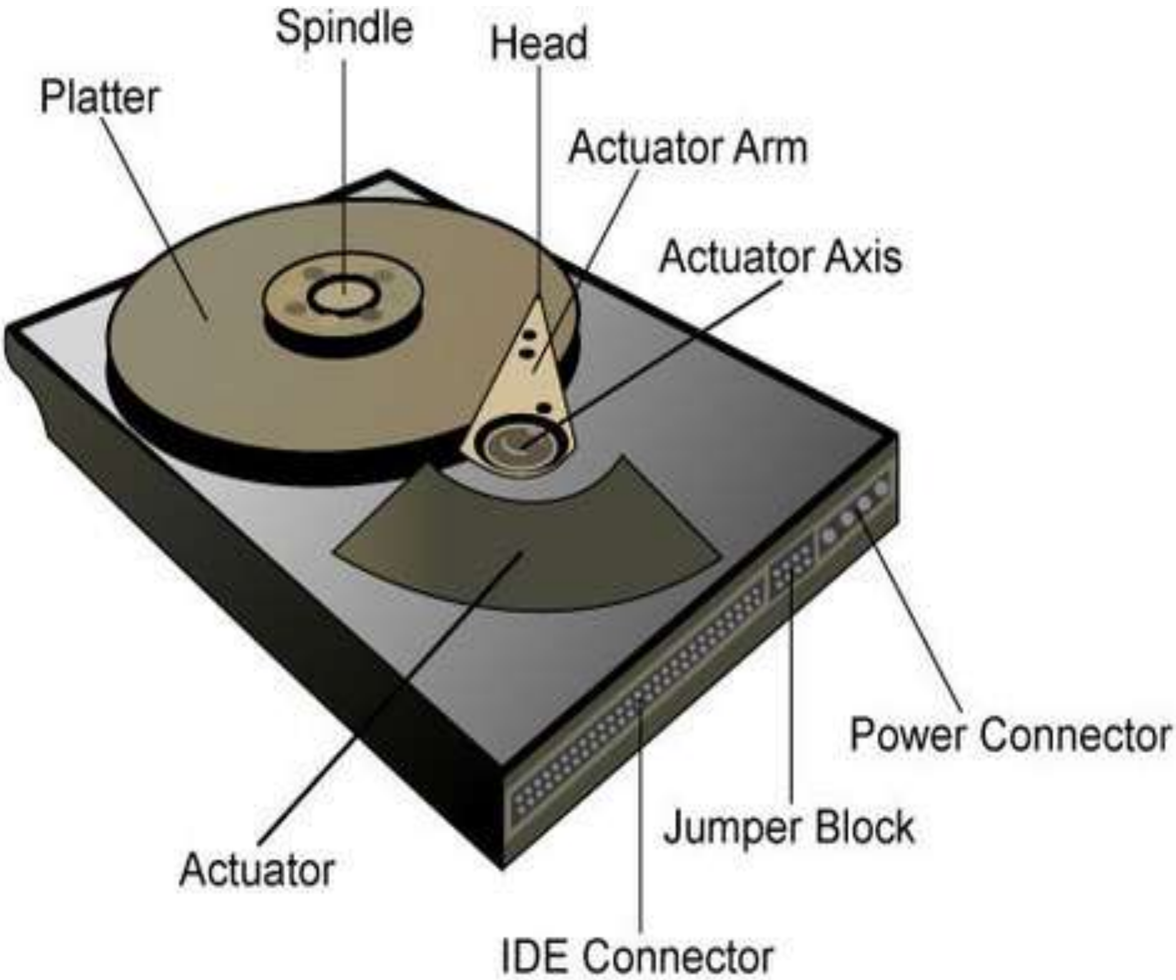
There are two ways of suppressing the wind effect on the platter. An aero-elastic model of the flow pattern in the drive can be obtained, and the housing of the drive can be redesigned. This technique was done in, and is commented later in the thesis. Another approach is to design a feedback control algorithm for the servo to actively dampen the wind induced vibrations which is left for future research. (Deeyiengyang & Ono, 2001)



i Figure 2: Windage Effect (Deeyiengyang & Ono, 2001)

1.3 Introduction to Hard Disk Drives

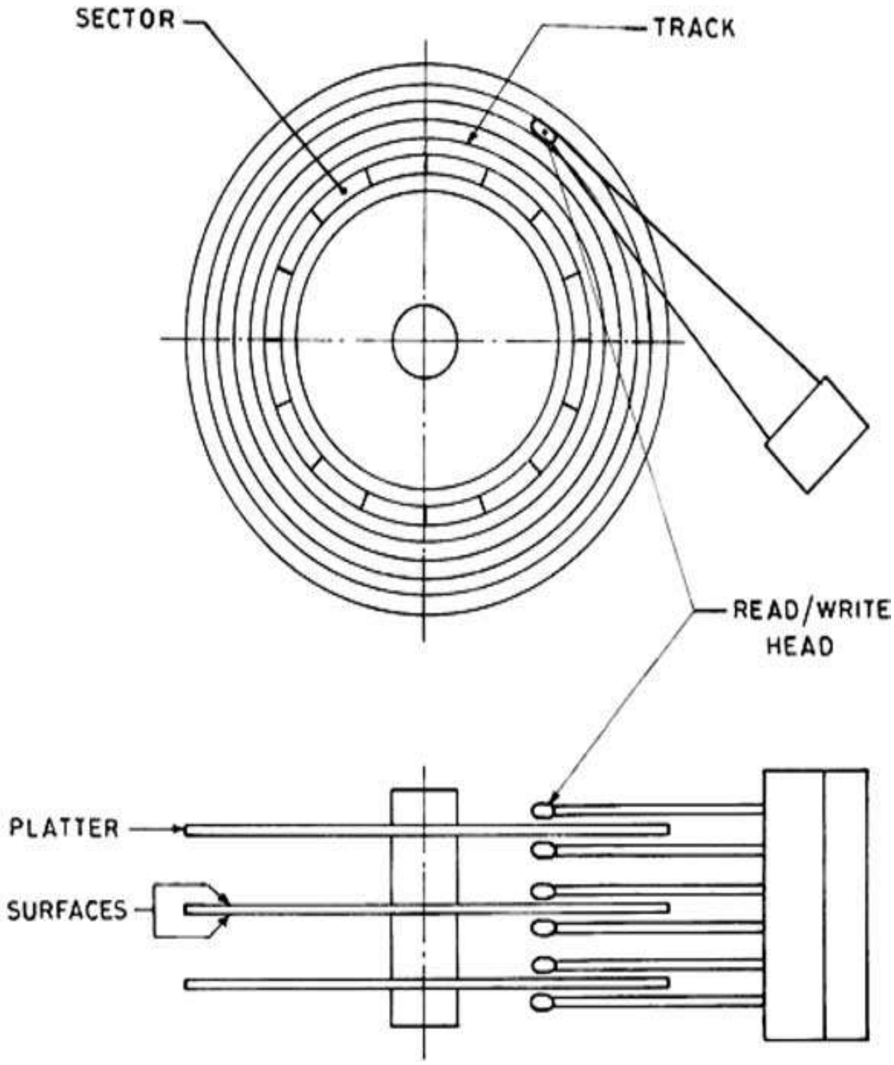
Computer hard disk drives (HDDs) are used to store a large quantity of information for retrieval as and when required. Components used in HDD can be broadly classified into 4 categories – magnetic components, mechanical components, electro-mechanical components, and electronics. Figure 1 shows the main components of hard disk drive. The spindle and disk assembly consist of a number of disks mounted on a spindle motor.



ii Figure-1 Components of Hard Disk Drive

1.4 History of Hard Disk Drives

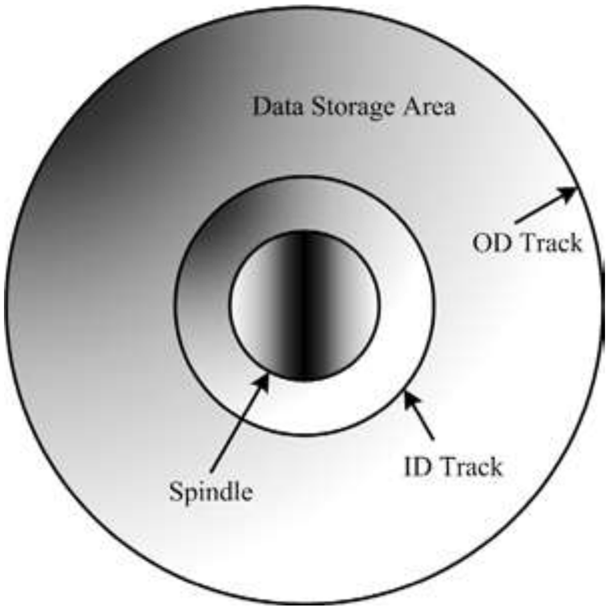
In Last four decades it evolved from a fifty 24-inch diameter disks storing only 5 MB of data to today's drives storing close to 1TB (1000GB) of data on one surface of a disk in a 31/2 inch drive and up to 500GB in 21/2 inch drive. There has been a significant increase in the data storage size of the HDDs; it is made possible by the increase in track density, which is measured as tracks per inch (TPI). Some of the current disk drives have track densities in excess of 15,000 TPI, or 66 micro inches between tracks. (Al Mamun, Guo, & Bi, 2007)



iii Figure 1.1 Above picture showing tracks and sectors on the HDD and also showing the Read/Write Head of the HDD

1.5 How does Hard Disk Drive (HDD) work?

In HDD data is recorded on a continuously spinning disk (platter) made of aluminium or glass and coated on both sides with a thin layer of magnetic material. Disk is coated with several layers of other materials too. The platter is mounted through a hole at the centre on the shaft (spindle) of a motor that spins the disks. In desktop application, the disks are spun usually at 5400 or 7200 RPM. The spinning speed can be 10,000 RPM or beyond in high performance HDD.



iv Figure 1.2 Platter of Hard Disk Drive

Two separate elements, the write and read heads, are used for writing data to or reading data from the disks. These two heads are fabricated together on a larger structure called the slider that serves several important purposes. The slider provides electrical connectivity to both heads, and helps to place the read and write heads in close proximity to the magnetized bits by flying over the surface of the spinning disk. Well defined aerodynamic surface is created on the surface of the slider facing the disk to achieve the desired flying characteristic. The air moving along with the spinning disk and entrained between the disk and the slider’s aerodynamic surface produces an air bearing that makes the slider float. (Al Mamun, Guo, & Bi, 2007)

The designed distance between the head and platter is called the flying height. It can literally measure to a few millionths of an inch. A good analogy is, try to imagine flying a Boeing 747 with about 6 inches above ground level. It is therefore easy to understand that if the RW head happens to "knock" on the spinning platter out of design specification, a Read Write head crash occurs and that could lead to problems such as data loss and severe damage to the surface of platter. So that is why it is very important to have no (or very less) vibration on platter of HDD. *Vibration* has become a serious obstacle to performance of Hard Disk Drives.

1.6 Introduction to Vibration

In its simplest form, vibration can be considered to be the oscillation or repetitive motion of an object around an equilibrium position.

Vibration refers to mechanical oscillation about an equilibrium point is called vibration. The oscillations may be periodic such as the motion of a pendulum or it could be random such as the movement of a tire on a gravel road.

Vibration is occasionally "desirable". For example the motion of a tuning fork, the reed in a woodwind instrument or harmonica, or the cone of a loudspeaker is desirable vibration, necessary for the correct functioning of the various devices.

More often, vibration is undesirable, wasting energy and creating unwanted sound – noise. For example, the vibrational motions of engines, electric motors, or any mechanical device in operation are typically unwanted. Such vibrations can be caused by imbalances in the rotating parts, uneven friction, the meshing of gear teeth, etc. Careful designs usually minimize unwanted vibrations.

The study of sound and vibration are closely related. Sound, or pressure waves are generated by vibrating structures (e.g. vocal cords); these pressure waves can also induce the vibration of structures (e.g. ear drum). Hence, when trying to reduce noise it is often a problem in trying to reduce vibration. (De Silva, 2000)

1.7 Types of Vibration

- **Free vibration** occurs when a mechanical system is set off with an initial input and then allowed to vibrate freely. Examples of this type of vibration are pulling a child back on a swing and then letting go or hitting a tuning fork and letting it ring. The mechanical system will then vibrate at one or more of its "natural frequency" and damp down to zero. (Inman D. J., 2006)

- **Forced vibration** is when an alternating force or motion is applied to a mechanical system. Examples of this type of vibration include a shaking washing machine due to an imbalance, transportation vibration (caused by truck engine, springs, road, etc.), or the vibration of a building during an earthquake. In forced vibration the frequency of the vibration is the frequency of the force or motion applied, with order of magnitude being dependent on the actual mechanical system. (Inman D. J., 2006)

CHAPTER 2: Vibration Analysis

The fundamentals of vibration analysis can be understood by studying the simple mass–spring–damper model. Indeed, even a complex structure such as an automobile body can be modelled as a "summation" of simple mass–spring–damper models. The mass–spring–damper model is an example of a simple harmonic oscillator. The mathematics used to describe its behaviour is identical to other simple harmonic oscillators such as the RLC circuit. (De Silva, 2000)

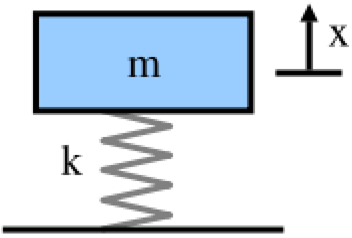
Note: In this section of thesis the step by step mathematical derivations will not be included, but will focus on the major equations and concepts in vibration analysis. Please refer to the references at the end of the thesis for detailed derivations.

What causes the system to vibrate: from conservation of energy point of view

Vibrational motion could be understood in terms of conservation of energy. In the below example we have extended the spring by a value of *x* and therefore have stored some potential energy ($\frac{1}{2}kx^2$) in the spring. Once we let go of the spring, the spring tries to return to its un-stretched state (which is the minimum potential energy state) and in the process accelerates the mass. At the point where the spring has reached its un-stretched state all the potential energy that we supplied by stretching it has been transformed into kinetic energy ($\frac{1}{2}mv^2$). The mass then begins to decelerate because it is now compressing the spring and in the process transferring the kinetic energy back to its potential. Thus oscillation of the spring amounts to the transferring back and forth of the kinetic energy into potential energy. (Inman D. J., 2008)

In our simple model the mass will continue to oscillate forever at the same magnitude, but in a real system there is always something called **damping** that dissipates the energy and therefore the system eventually bringing it to rest.

2.1 Free Vibration without Damping



v Figure 2.0 Mass-Spring vibration without damping

To start the investigation of the mass–spring–damper we will assume the damping is negligible and that there is no external force applied to the mass (i.e. free vibration).

The force applied to the mass by the spring is proportional to the amount the spring is stretched "x" (we will assume the spring is already compressed due to the weight of the mass). The proportionality constant, k, is the rigidity of the spring and has units of force/distance (e.g. lbf/in or N/m)

$$F_s = -kx.$$

The force generated by the mass is proportional to the acceleration of the mass as given by Newton's second law of motion.

$$\Sigma\, F = ma = m\ddot{x} = m\frac{d^2x}{dt^2}.$$

The sum of the forces on the mass then generates this ordinary differential equation:

$$m\ddot{x} + kx = 0.$$

If we assume that we start the system to vibrate by stretching the spring by the distance of *A* and letting go, the solution to the above equation that describes the motion of mass is:

$$x(t) = A\cos(2\pi f_n t).$$

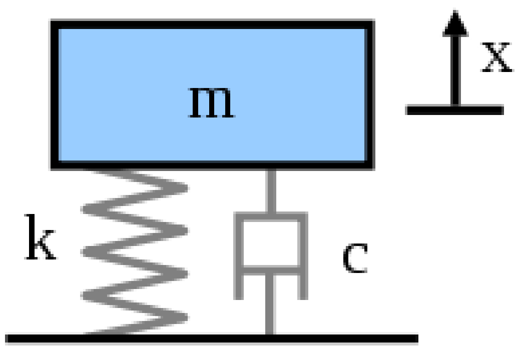
This solution says that it will oscillate with simple harmonic motion that has an amplitude of *A* and a frequency of f_n. The number f_n is one of the most important quantities in vibration analysis and is called the **undamped natural frequency**. For the simple mass–spring system, f_n is defined as:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}.$$

Note: Angular frequency ω (ω = 2π*f*) with the units of radians per second is often used in equations because it simplifies the equations, but is normally converted to "standard" frequency (units of Hz or equivalently cycles per second) when stating the frequency of a system.

If you know the mass and stiffness of the system you can determine the frequency at which the system will vibrate once it is set in motion by an initial disturbance using the above stated formula. Every vibrating system has one or more natural frequencies that it will vibrate at once it is disturbed. This simple relation can be used to understand in general what will happen to a more complex system once we add mass or stiffness. For example, the above formula explains why when a car or truck is fully loaded the suspension will feel "softer" than unloaded because the mass has increased and therefore reduced the natural frequency of the system. (Inman D. J., 2008)

2.2 Free Vibration with Damping



vi Figure: 2.2 Mass-Spring vibration with damping

We now add a "viscous" damper to the model that outputs a force that is proportional to the velocity of the mass. The damping is called viscous because it models the effects of an object within a fluid. The proportionality constant *c* is called the damping coefficient and has units of Force over velocity (lbf s/in or N s/m).

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

By summing the forces on the mass we get the following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = 0.$$

The solution to this equation depends on the amount of damping. If the damping is small enough the system will still vibrate, but eventually, over time, will stop vibrating. This case is called underdamping – this case is of most interest in vibration analysis. If we increase the damping just to the point where the system no longer oscillates that means we reach the point of **critical damping** (if the damping is increased past critical damping the system is called overdamped).

The value that the damping coefficient needs to reach for critical damping in the mass spring damper model is:

$$c_c = 2\sqrt{km}.$$

To characterize the amount of damping in a system a ratio called the damping ratio (also known as damping factor and % critical damping) is used. This damping ratio is just a ratio of the actual damping over the amount of damping required to reach critical damping. The formula for the damping ratio (ζ) of the mass spring damper model is:

$$\zeta = \frac{c}{2\sqrt{km}}.$$

For example, metal structures (e.g. airplane fuselage, engine crankshaft) will have damping factors less than 0.05 while automotive suspensions in the range of 0.2–0.3.

The solution to the underdamped system for the mass spring damper model is the following:

$$x(t) = Xe^{-\zeta\omega_n t}\cos(\sqrt{1-\zeta^2}\omega_n t - \phi), \qquad \omega_n = 2\pi f_n.$$

The value of *X*, the initial magnitude, and ϕ, the phase shift, are determined by the amount the spring is stretched. The formulas for these values can be found in the references. (Inman D. J., 2008)

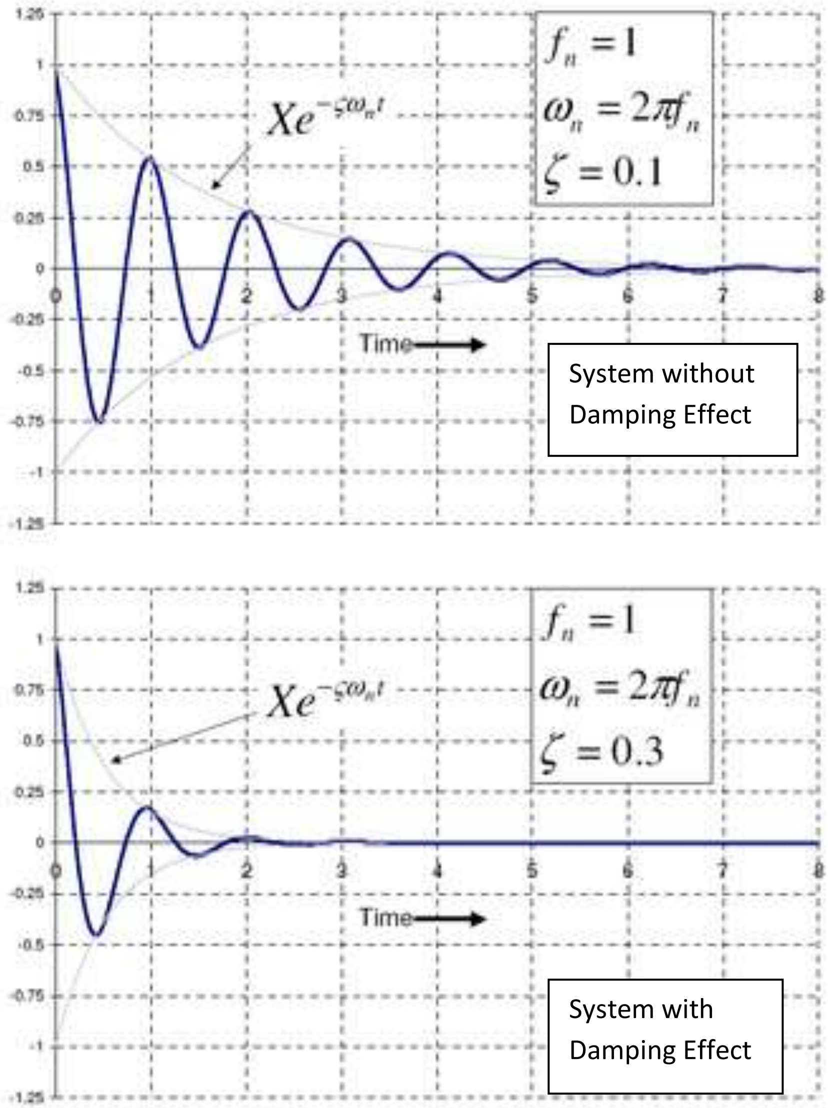
2.3 Damped and Undamped Natural Frequencies

The major points to note from the solution are the exponential term and the cosine function. The exponential term defines how quickly the system "damps" down – the larger the damping ratio, the quicker it damps to zero. The cosine function is the oscillating portion of the solution, but the frequency of the oscillations is different from the undamped case.

The frequency in this case is called the "damped natural frequency", f_d, and is related to the undamped natural frequency by the following formula:

$$f_d = \sqrt{1-\zeta^2} f_n.$$

The damped natural frequency is less than the undamped natural frequency, but for many practical cases the damping ratio is relatively small and hence the difference is negligible. Therefore the damped and undamped description are often dropped when stating the natural frequency (e.g. with 0.1 damping ratio, the damped natural frequency is only 1% less than the undamped). (Inman D. J., 2008)

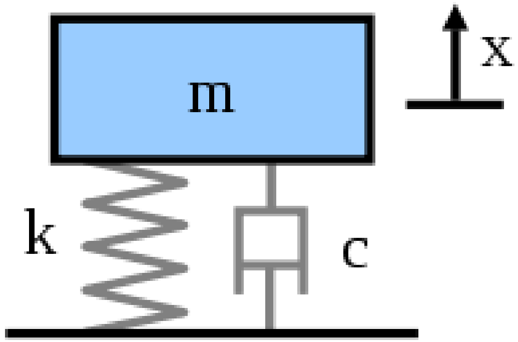


vii Figure: 2.3 Damping Ratios Effect (De Silva, 2000)

The above plots present how 0.1 and 0.3 damping ratios effect how the system will "damp" down over time.

What is often done in practice is to experimentally measure the free vibration after an impact (for example by a hammer) and then determine the natural frequency of the system by measuring the rate of oscillation as well as the damping ratio by measuring the rate of decay. The natural frequency and damping ratio are not only important in free vibration, but also characterize how a system will behave under forced vibration. (Inman D. J., 2008)

2.4 Forced Vibration with Damping



viii Figure 2.4 Mass-Spring vibration without damping

In this section we will see the behaviour of the spring mass damper model when we add a harmonic force in the form below. A force of this type could, for example, be generated by a rotating imbalance.

$$F = F_0 \cos(2\pi f t).$$

If we again sum the forces on the mass we get the following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(2\pi f t).$$

The steady state solution of this problem can be written as:

$$x(t) = X \cos(2\pi f t - \phi).$$

The result states that the mass will oscillate at the same frequency, f, of the applied force, but with a phase shift ϕ.

The amplitude of the vibration "X" is defined by the following formula.

$$X = \frac{F_0}{k} \frac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}.$$

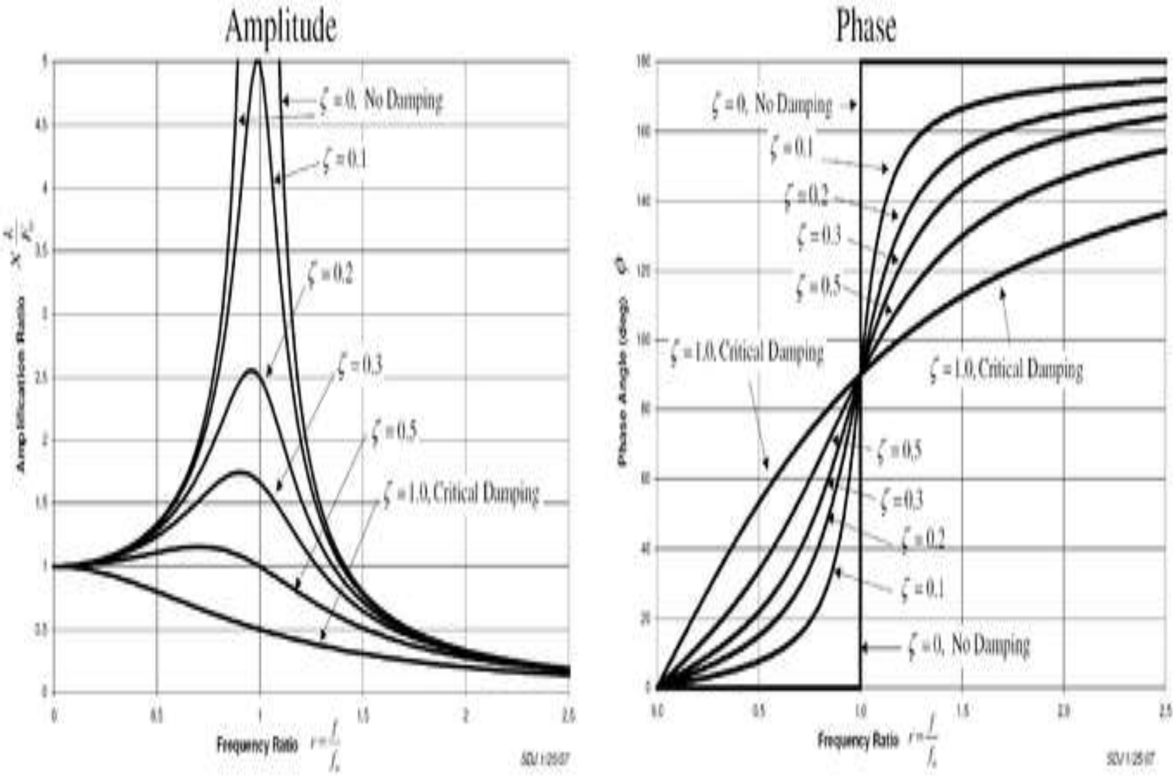
Where "r" is defined as the ratio of the harmonic force frequency over the undamped natural frequency of the mass–spring–damper model

$$r = \frac{f}{f_n}.$$

The phase shift, ϕ, is defined by the following formula.

$$\phi = \arctan\left(\frac{2\zeta r}{1 - r^2}\right).$$

(Inman D. J., 2008)



ix Figure: 2.41 Frequency Response of The System (De Silva, 2000)

The plot of these functions, called "the frequency response of the system", presents one of the most important features in forced vibration. In a lightly damped system when the forcing frequency nears the natural frequency ($r \approx 1$) the amplitude of the vibration can get extremely high. This phenomenon is called **resonance** (subsequently the natural frequency of a system is often referred to as the resonant frequency). In rotor bearing systems any rotational speed that excites a resonant frequency is referred to as a critical speed. (De Silva, 2000)

If resonance occurs in a mechanical system it can be very harmful – leading to eventual failure of the system. Consequently, one of the major reasons for vibration analysis is to predict when this type of resonance may occur and then to determine what steps to take to prevent it from occurring. As the amplitude plot shows, adding damping can significantly reduce the magnitude of the vibration. Also, the magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted (for example, changing the speed of the machine generating the force).

The following are some other points in regards to the forced vibration shown in the frequency response plots.

- At a given frequency ratio, the amplitude of the vibration, X, is directly proportional to the amplitude of the force F_0 (e.g. if you double the force, the vibration doubles)

- With little or no damping, the vibration is in phase with the forcing frequency when the frequency ratio $r < 1$ and 180 degrees out of phase when the frequency ratio $r > 1$

- When $r \ll 1$ the amplitude is just the deflection of the spring under the static force F_0. This deflection is called the static deflection δ_{st}. Hence, when $r \ll 1$ the effects of the damper and the mass are minimal.

- When $r \gg 1$ the amplitude of the vibration is actually less than the static deflection δ_{st}. In this region the force generated by the mass ($F = ma$) is dominating because the acceleration seen by the mass increases with the frequency. Since the deflection seen in the spring, X, is reduced in this region, the force transmitted by the spring ($F = kx$) to the base is reduced. Therefore the mass–spring–damper system is isolating the harmonic force from the mounting base – referred to as vibration isolation. Interestingly, more damping actually reduces the effects of vibration isolation when $r \gg 1$ because the damping force ($F = cv$) is also transmitted to the base.

2.5 What Causes Resonance?

Resonance is simple to understand if you view the spring and mass as energy storage elements – with the mass storing kinetic energy and the spring storing potential energy. As discussed earlier, when the mass and spring have no force acting on them they transfer energy back and forth at a rate equal to the natural frequency. In other words, if energy is to be efficiently pumped into both the mass and spring the energy source needs to feed the energy in at a rate equal to the natural frequency. Applying a force to the mass and spring is similar to pushing a child on swing, you need to push at the correct moment if you want the swing to get higher and higher. As in the case of the swing, the force applied does not necessarily have to be high to get large motions; the pushes just need to keep adding energy into the system.

The damper, instead of storing energy, dissipates energy. Since the damping force is proportional to the velocity, the more the motion, the more the damper dissipates the energy. Therefore a point will come when the energy dissipated by the damper will equal the energy being fed in by the force. At this point, the system has reached its maximum amplitude and will continue to vibrate at this level as long as the force applied stays the same. If no damping exists, there is nothing to dissipate the energy and therefore theoretically the motion will continue to grow on into infinity. (De Silva, 2000)

CHAPTER 3: Literature Review

3.1 Vibration and Vibration Analysis

Article 1: Vibration of hard disk drive spindle systems

I have reviewed the paper on the analysis on vibration of hard disk drive spindle systems with distributed journal bearing forces.

It discusses the model that is developed to predict the transverse vibration of the disk spindle systems in HDDs. It is found that the aspect ratio between the bearing width to the shaft length was significant hence has a tendency to be slightly more flexible.

A model of distributed linear spring and damping forces through distribution functions of dynamic coefficients represents the distributed restoring and damping forces which occur in spindles with a low aspect ratio between the bearing width to the shaft length.

When a comparison is made to the values predicted by an expected model with discrete forces it is found that the spindle model in which the distributed bearing forces predicts the same natural frequencies for all transverse modes - but higher modal damping of the rocking modes. This has all been found and revealed by vibration analysis.

A flexible shaft spindle that has a larger ratio of the bearing width to the shaft length, gives a more clear result of the difference in damping prediction. (Jintanawan, 2005)

Article 2: Vibration and noise analysis of computer hard disk drives

Another article I have reviewed has discussed the analysis of the vibration and noise generated by the computer Hard disk drives of different speeds. It also discusses natural frequencies of disk platters that were calculated theoretically and by FEM. Later in the article, it is showed that calculated natural frequencies and natural frequencies obtained by FEM are compared with the experimentally obtained natural frequencies. It was then verified that all of these frequency are fairly similar.

With the help of capacitance type of probe, overall vibration amplitude of rotating disk platter was measured. Noise analyses of the hard disk drives without covers were performed in seek and idle mode. Analysis of the total noise within the drive and different sources of noise has been done to identify the main source of noise. Thorough analysis shows that in the lower speed drives the major source of noise was the actuator arm flying over the platter and in the high speed drives, disk platter air flutter noise was main source. (N, V, & V, 2006)

Article 3: A Study of Characteristics of Disk Vibration and Rotating Airflow in Magneto Optical Disk Drives

In this article a study on dynamic characteristics of rotating disks in magneto optical disk drives was presented. Natural frequencies of the rotating disks are investigated experimentally and theoretically/numerically. The frequency response and critical speeds of ASMO (Airflow in Magnetic Optical) disk are discussed. The characteristics of airflow around the disk and their effects on disk vibrations are also discussed. It is found that the numerical calculation of the natural frequencies of rotating disks agrees well with the experimental results. The airflow around the disk in cartridge affects the characteristics of the disk vibrations to reduce the modal frequencies of the disk. The experiment shows that negative vertical offsets of the disk in the cartridge possibly increase the vibration amplitudes. Being influenced by the geometry of the cartridge, the rotation of the disk causes an asymmetric airflow in the presence of window. (Kim, Han, & Son, 2006)

Article 4: Suppression of Resonance Amplitude of Disk Vibrations by Squeeze Air Bearing Plate

Another article I have reviewed has investigated the effect of suppressing the vibration of a spinning disk/spindle system by applying a squeeze film damping to a commercially available HDD with five platters. The out-of-plane and in-plane vibrations of the top disk were measured with and without a squeeze air bearing and illustrated by the Campbell diagram. When the squeeze air bearing plate with angular length of 180^0 and radial length of 20 mm was used at 120 *u*m clearance, the largest resonance amplitude of the out-of-plane disk vibration decreases to less than one third. It was found that the disk flutter caused by the surrounding air flow could be significantly suppressed. It was also found that the suppressing effect of the out-of-plane vibration can decrease proportionally to the in-plane vibration. (Deeyiengyang & Ono, 2001)

3.2 Finite Element Analysis (FEM)

Article 5: Finite Element Analysis of Flexural Vibrations In Hard Disk Drive Spindle Systems

This paper is concerned with the flexural vibration analysis of the hard disk drive (HDD) spindle system by means of the finite element method. In contrast to previous research, every system component is here analytically modelled taking into account its structural flexibility and also the centrifugal elect particularly on the disk. To prove the electiveness and accuracy of the formulated models, commercial HDD systems with two and three identical disks are selected as examples. Then their major natural modes are computed with only a small number of element meshes as the shaft rotational speed is varied, and subsequently compared with the existing numerical results obtained using other methods and newly acquired experimental ones. (LIM, 2000)

CHAPTER 4: Finite Element Method (FEM)

The Finite element method (FEM) is a powerful technique to analyse structural vibrations of a system. Originally it was developed for numerical solution of complex problems in structural mechanics, and it remains the method of choice for complex systems. In the FEM, the structural system is modelled by a set of appropriate finite elements interconnected at points called nodes. Elements may have physical properties such as thickness, coefficient of thermal expansion, density, Young's modulus, shear modulus and Poisson's ratio. (Seungchul, 1999)

Material properties of main components

	Disk platter	Hub, shaft, and yoke	Magnetic
Young's modulus	75×10^9 pa	204×10^9 pa	72×10^9 pa
Poisson's ratio	0.36	0.28	0.3
Density	$2750\,kg/m^3$	$7800\,kg/m^3$	$5600\,kg/m^3$
Thickness	0.64 mm	–	–
Inner radius	11.82 mm	–	–
Outer radius	32.5 mm	–	

a Table 4.0 Material Properties

Each component of the HDD must be modelled accurately so that we can analyse the vibration interaction behaviours of the whole disk spindle system of HDD. So, the finite element modelling of disk platter is performed first in this section.

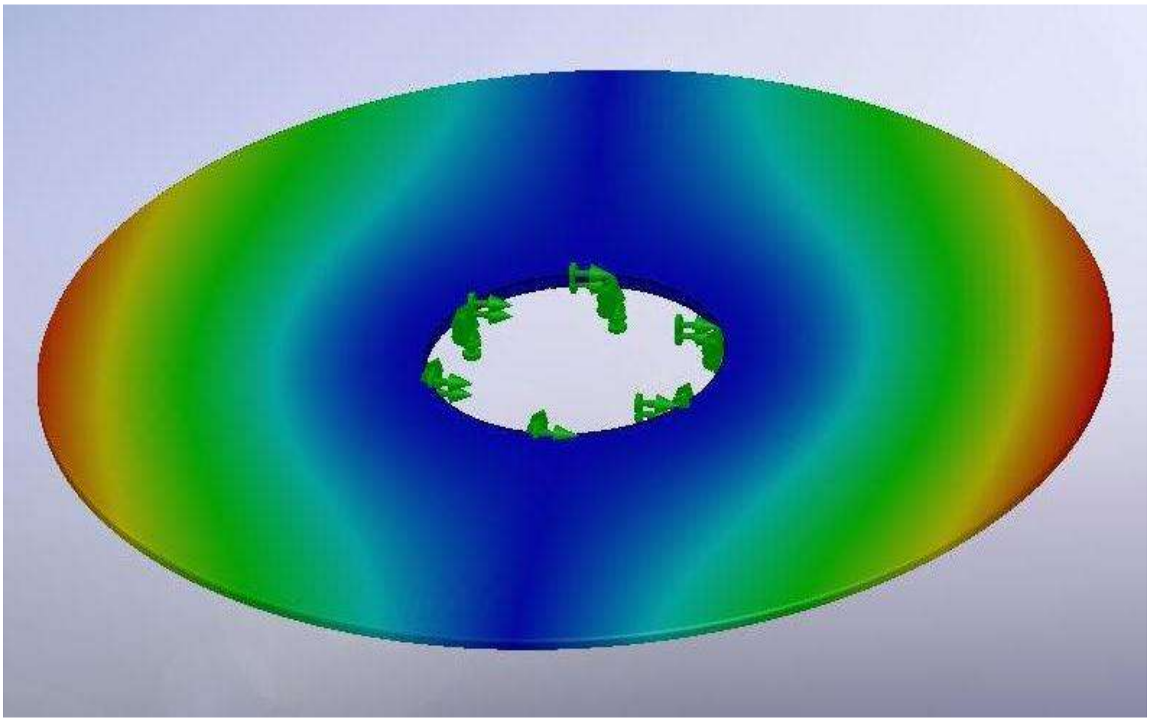
4.1 FEM of Disk Platter

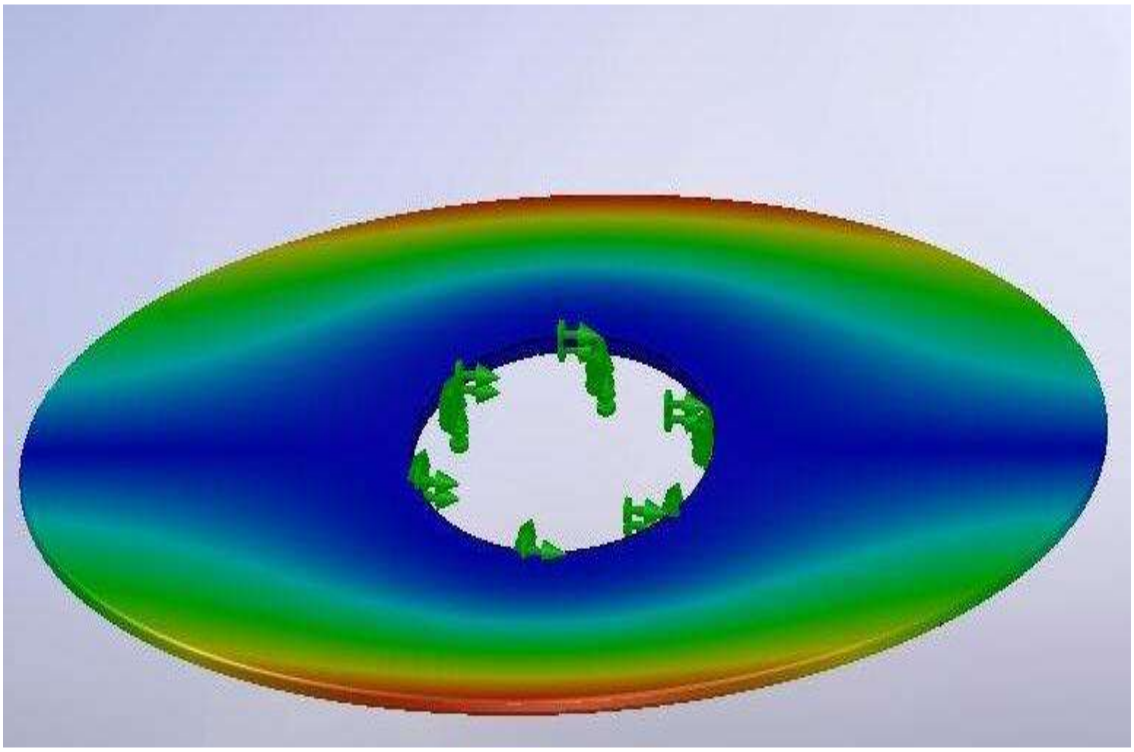
In my case;

- The HDD platter was modelled in Solid Works to find out the natural frequencies and mode shapes.
- HDD platter solid model was prepared; mesh was generated.
- Boundary conditions were applied i.e. inner diameter was fixed because it is more close to the real conditions as the platter is fixed on a hub.
- Material properties were specified and then solution for modal analysis, mode shapes and natural frequencies was obtained.

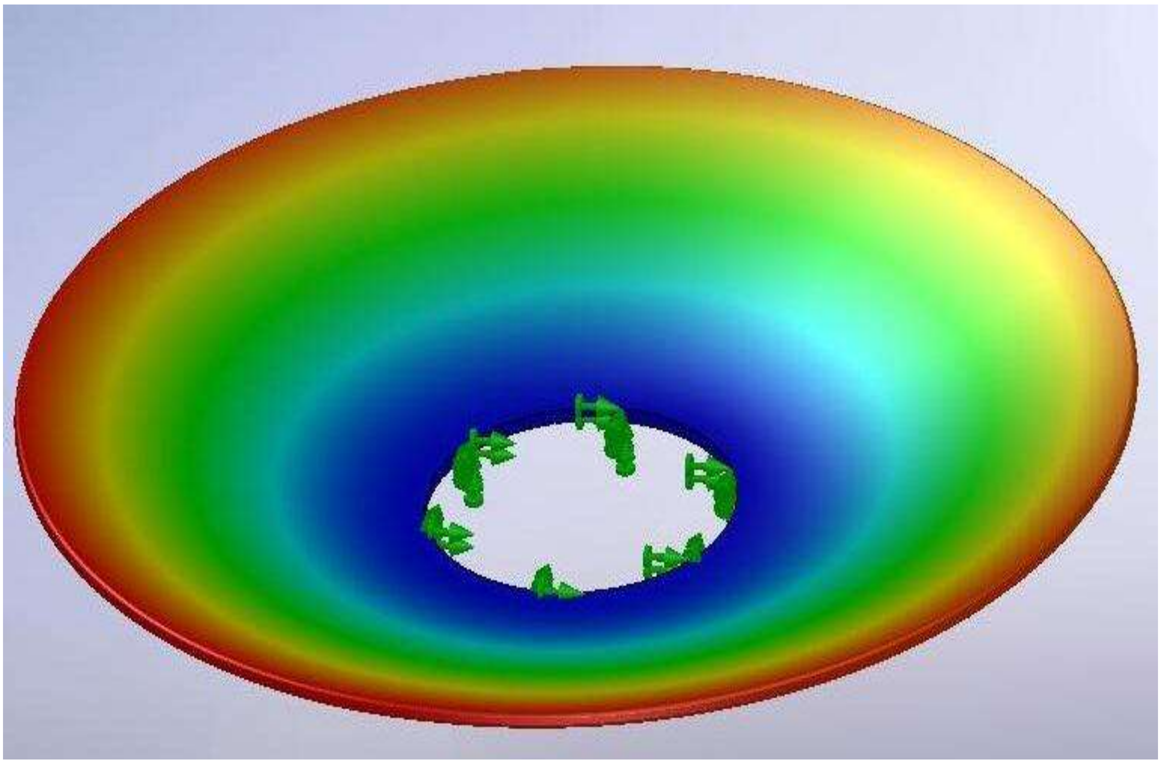
4.2 Natural Frequency and Modes In 2.5inch HDD When Stationary

Hard disk platter FEM mode shapes when disk is stationary are shown below:

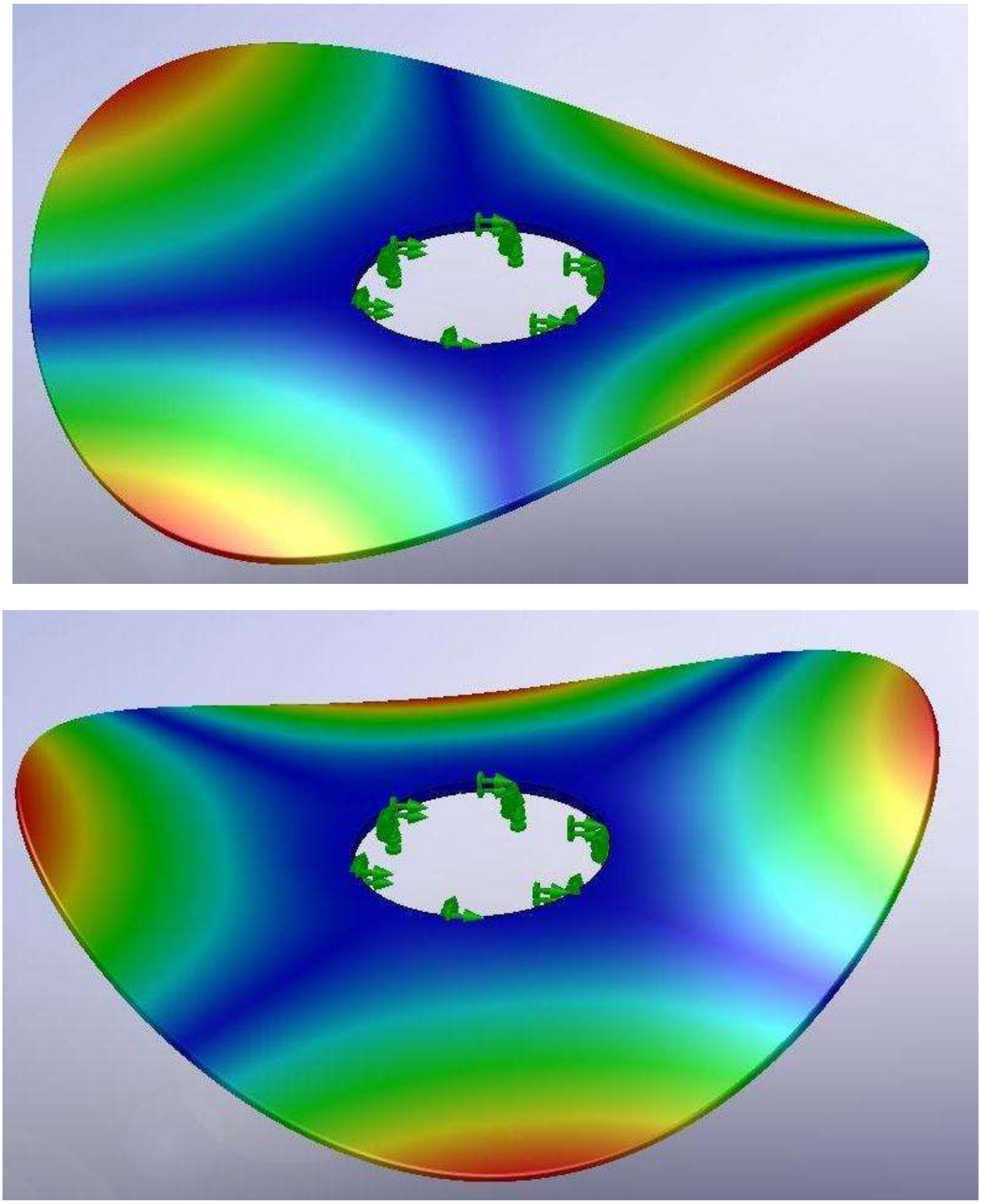




x Figure 4.20: Above pictures shows Mode1,1 (Top one is sine and bottom one cosine)



xi Figure 4.21: Above picture shows Mode0,1



xii Figure 4.22 Above pictures shows Mode2,1

From the mode shapes it is observed that first $mode_{1,1}$ repeat itself and is same as second mode but with opposite deformation direction; we can say that first mode is sine version of $Mode_{1,0}$ and second mode is cosine version. Third mode, $mode_{0,1}$, is umbrella mode also known as axis mode induced by shaft and it is not repeated. It is noted that platter vibration modes split except third mode. The last two modes are $\text{Mode}_{2,1}$ and also repeats itself.

The first five natural frequencies obtained through FEM are shown in following table:

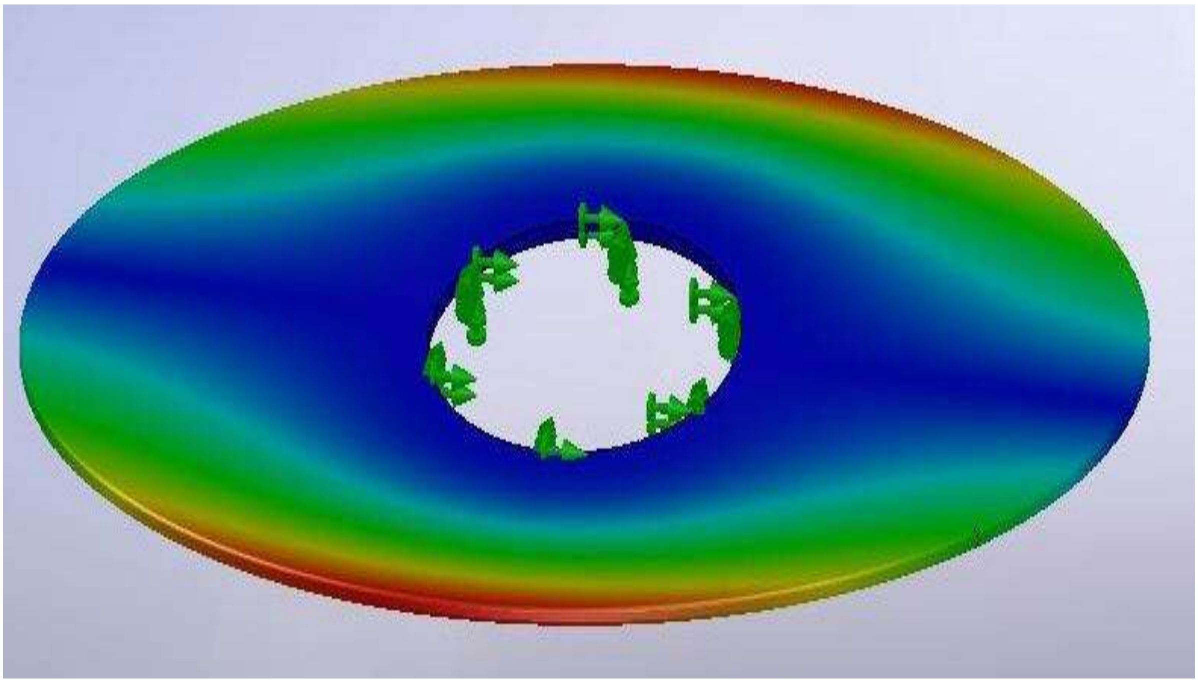
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	6496.1	1033.9	0.00096723
2	6504.1	1035.2	0.00096603
3	6692.9	1065.2	0.00093879
4	7875.1	1253.4	0.00079786
5	7888.9	1255.6	0.00079646

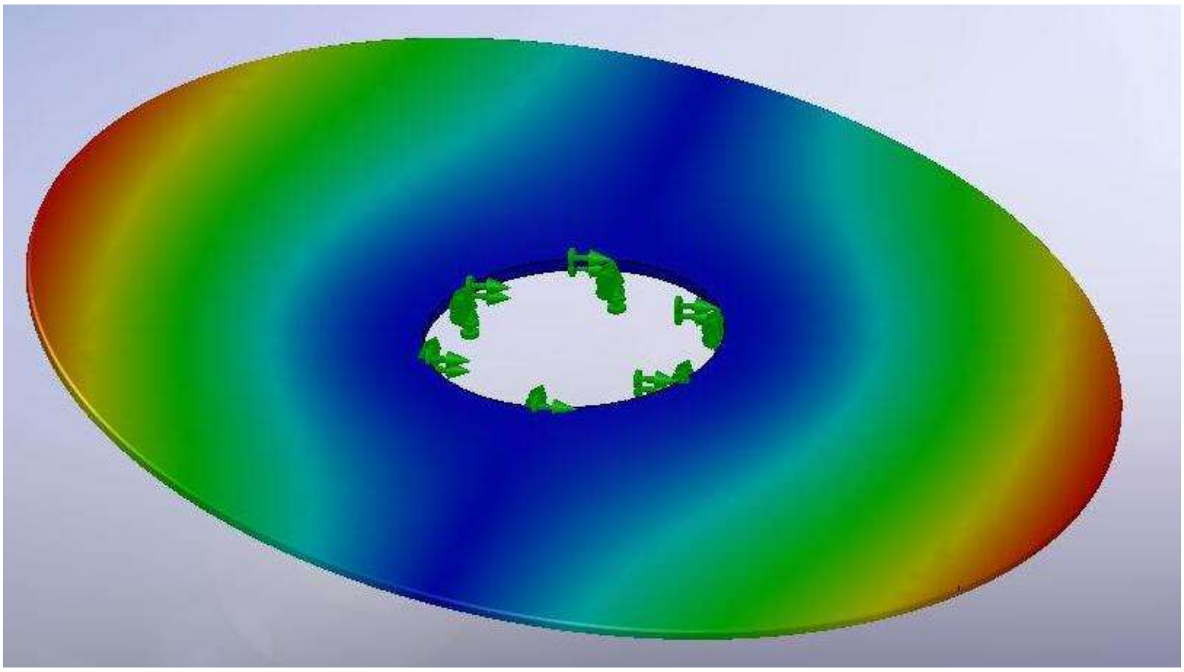
b Table 4.1 Natural Frequencies of Platter at Stationary

After modelling each component accurately, results for disk platter have been obtained. The first five natural frequencies of the disk without rotation were calculated by using FEM.

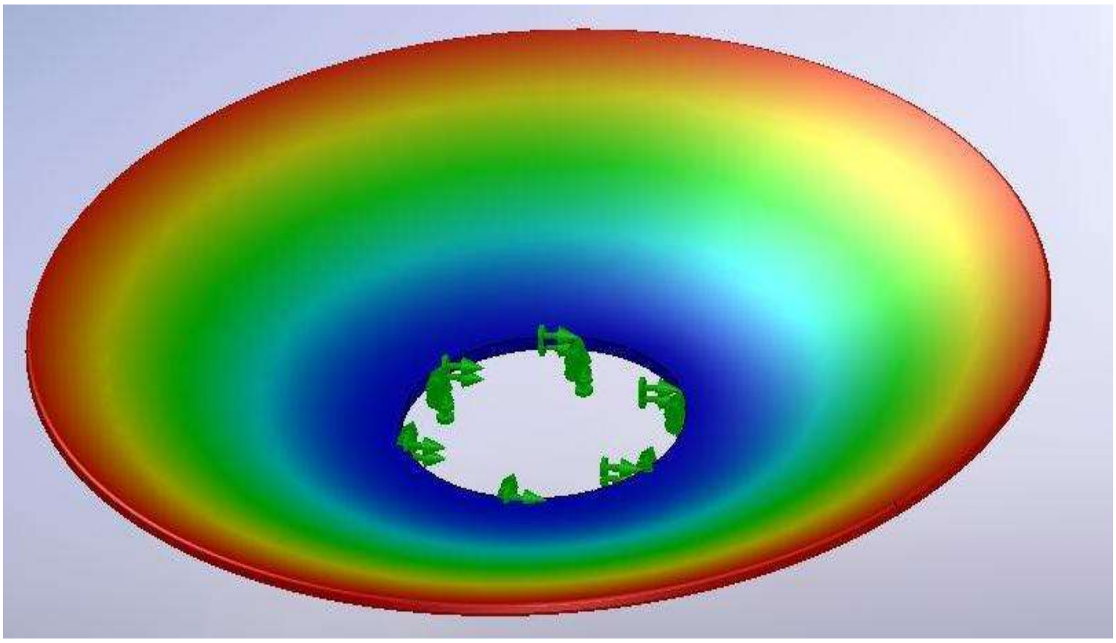
4.3 Natural Frequency and Modes in 2.5inch HDD at 5400RPM

Hard disk platter FEM mode shapes when disk is stationary are shown below:

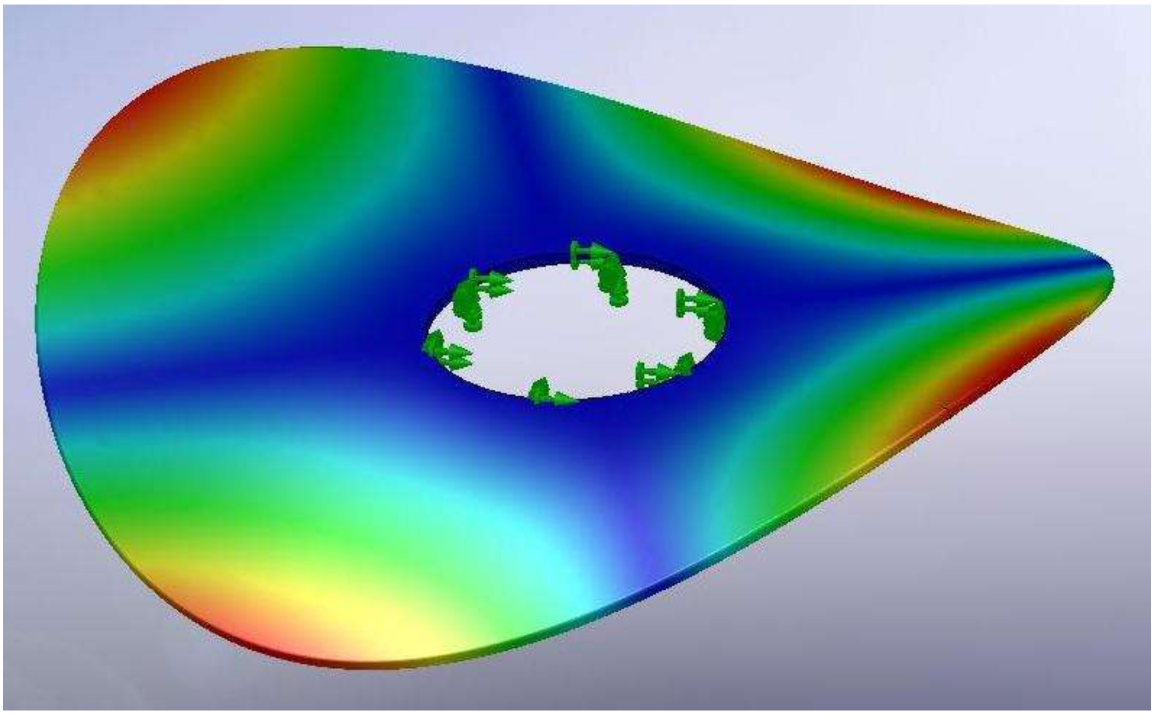




xiii Figure 4.30 Above pictures shows Mode1,1 of platter at 5400RRPM



xiv Figure 4.31 Above picture shows Mode0,1 of platter at 5400RRPM



xv Figure 4.32 Above picture shows Mode2,1 of platter at 5400RRPM

From the above mode shapes of platter at 5400RPM it is observed that first mode$_{1,1}$ repeat itself and is same as second mode but with opposite deformation direction; we can say that first mode is sine version of Mode$_{1,0}$ and second mode is cosine version. Third mode, mode$_{0,1}$, is umbrella mode and it is not repeated. It is noted that platter vibration modes split except third mode. The last two modes are Mode$_{2,1}$ and also repeats itself.

The first five natural frequencies of platter at 5400RPM obtained through FEM are shown in following table:

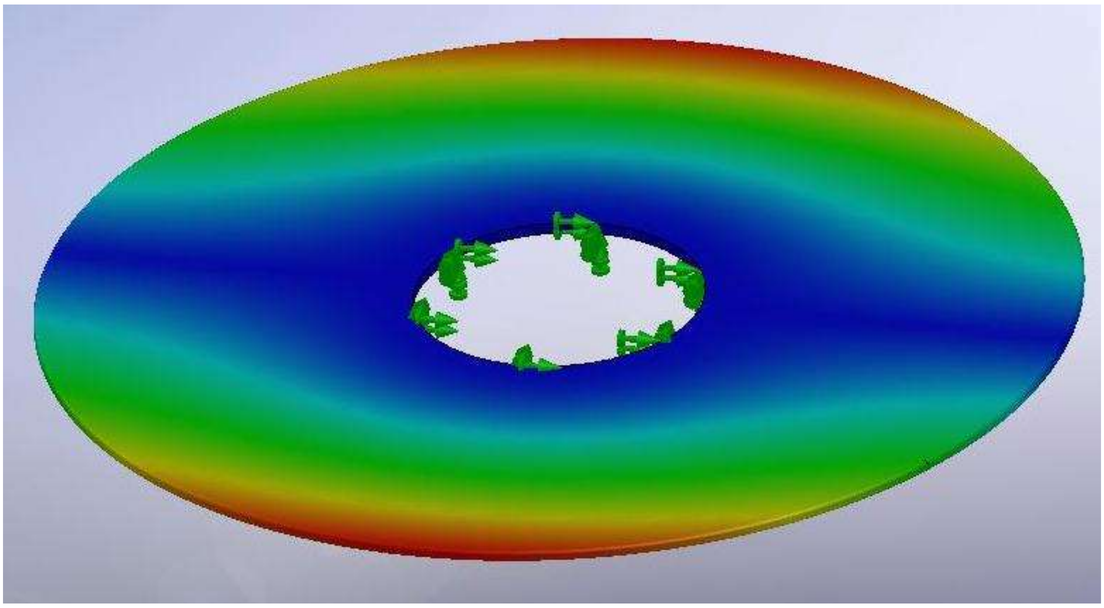
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	6589.7	1048.8	0.00095349
2	6597.7	1050.1	0.00095233
3	6768.7	1077.3	0.00092827
4	7996	1272.6	0.00078579
5	7999.2	1273.1	0.00078547

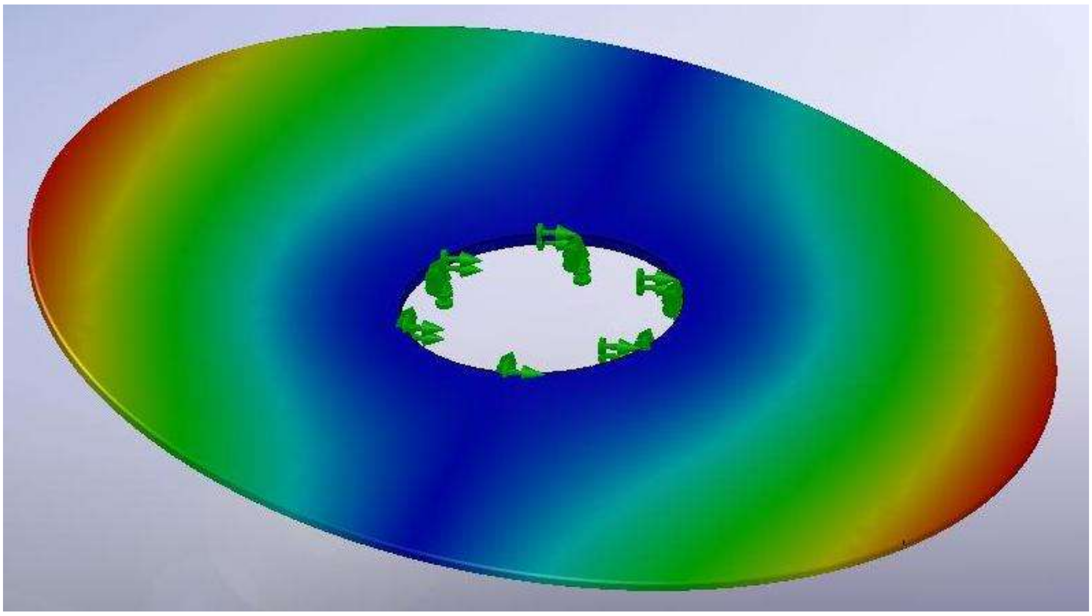
c Table 4.2 Natural Frequencies of Platter at 5400RPM

Above table shows that natural frequencies of the disk at the speed of 5400RPM is higher than stationary disk natural frequencies.

4.4 Natural Frequency and Modes in 2.5inch HDD at 7200RPM

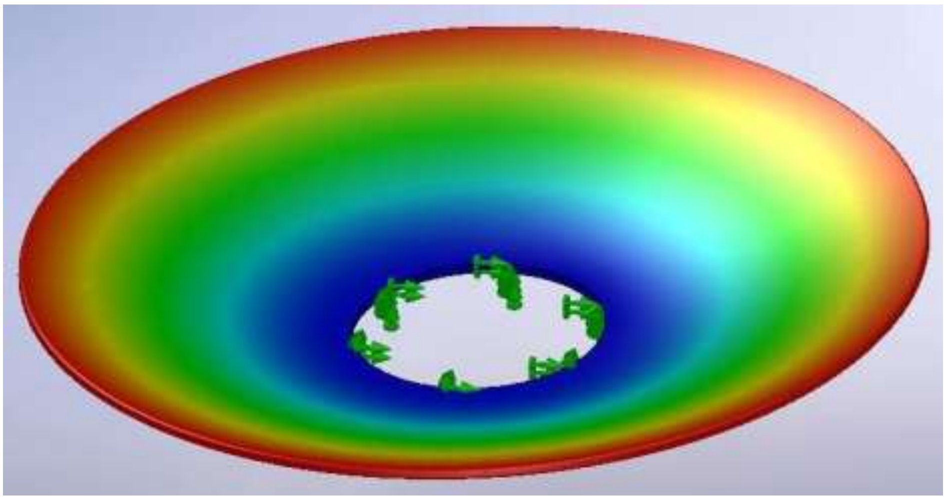
Hard disk platter FEM mode shapes when disk is stationary are shown below:





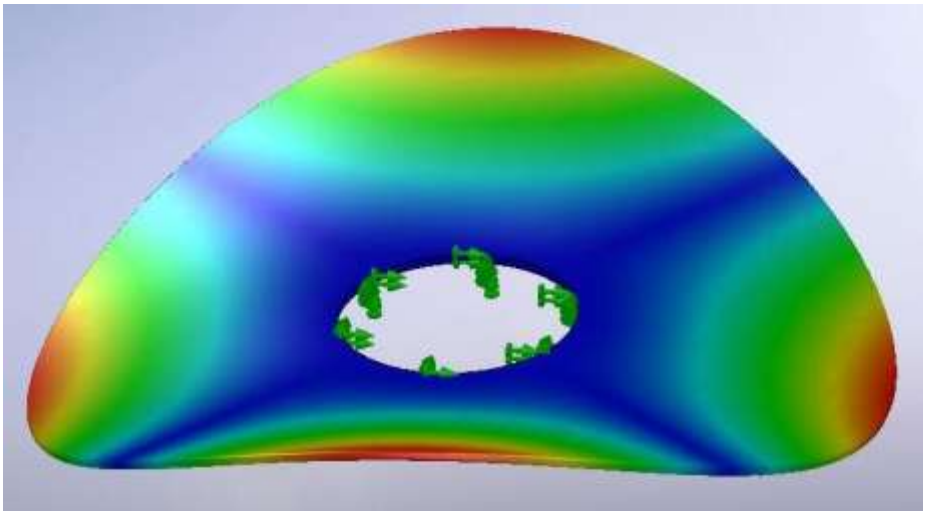
xvi Figure 4.40 Above pictures shows Mode1,1 at 7200RPM

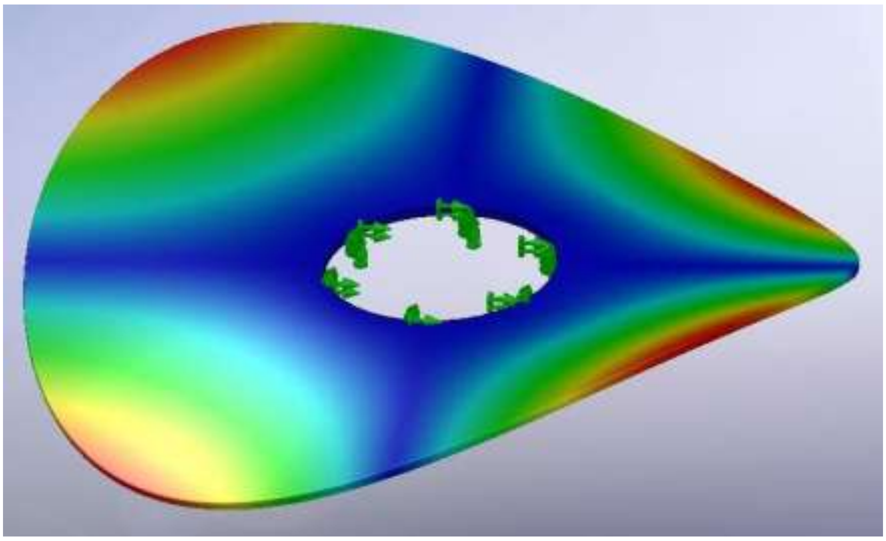
The modes shown in above pictures are (1,1) modes with one nodal diameter and one circular node. When vibrating in the (1,1) mode a circular platter acts much like a *dipole* source; instead of pushing air away from the membrane like the (0,1) mode does, in the (1,1) mode one half of the disk pushes air up while the other half sucks air down resulting in air being pushed back and forth from side to side.



xvii Figure 4.41 Above pictures shows Mode0,1 at 7200RPM

This mode number is designated as **(0, 1)** because there are no nodal diameters, but one circular node.





xviii Figure 4.42 Above pictures shows Mode2,1 at 7200RPM

The fourth and fifth mode of a platter at the speed of 7200RPM are the **(2,1)** modes, which has two nodal diameters (at right angles to each other) and one nodal circle. When vibrating in the (2,1) mode a disk acts much like a quadruple source which is worse at radiating sound than the (1,1) dipole mode and much less effective at radiating sound than the (0,1) monopole mode.

The first five natural frequencies of platter at 7200RPM obtained through FEM are shown in following table:

Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	6623.3	1054.1	0.00094864
2	6631.4	1055.4	0.00094749
3	6795.7	1081.6	0.00092459
4	8036.2	1279	0.00078186
5	8044	1280.2	0.0007811

d Table 4.3 Natural Frequencies of Platter at 7200

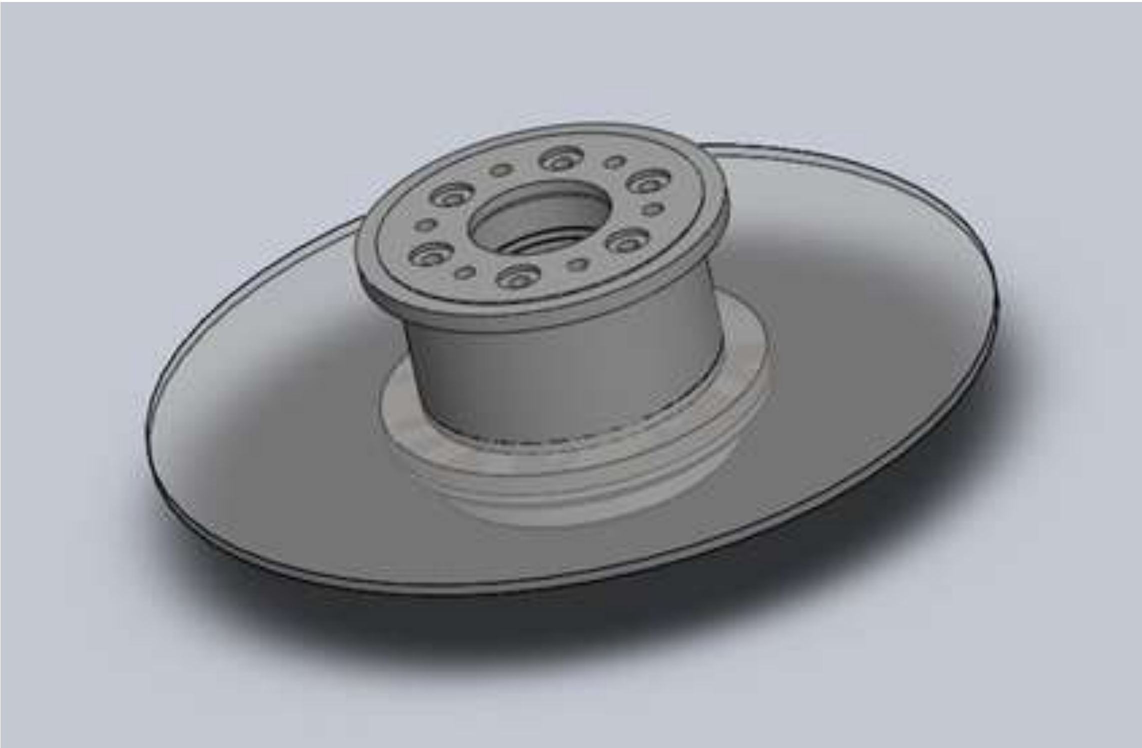
Above table shows that natural frequencies of the disk at the speed of 7200RPM is higher than 5400RPM and stationary disk's natural frequencies.

4.5 FEM of Complete Hard Disk Drive Assembly

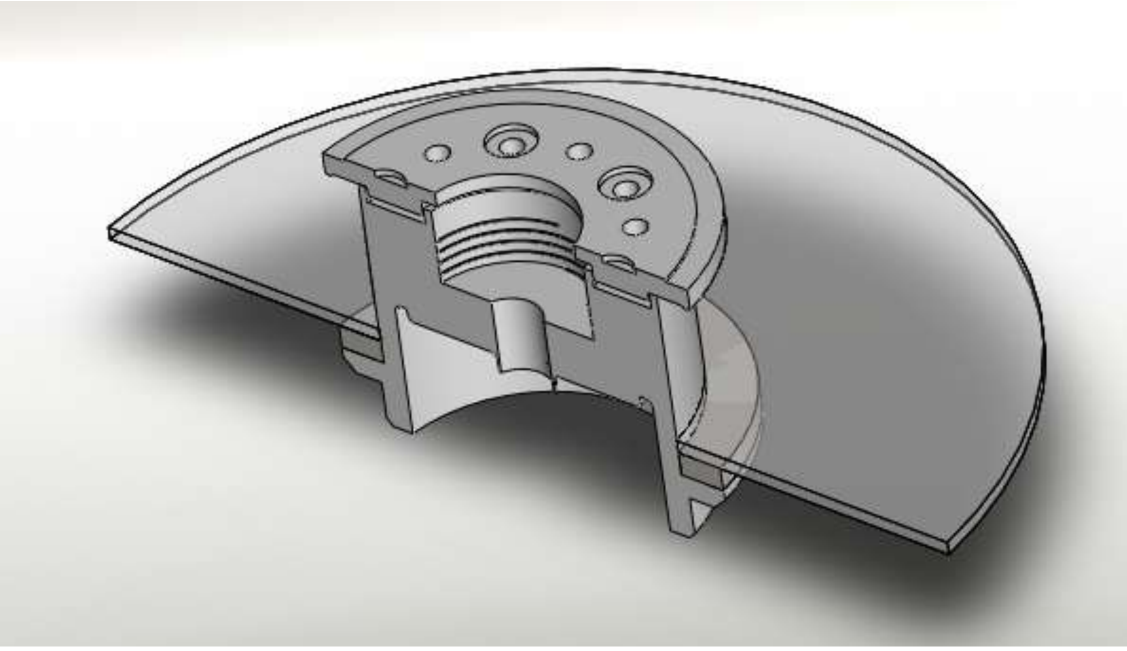
In order to obtain a precise finite element model, FEM of the complete system was done so that accurate model can be used to:

1. obtain the structural modes of the complete system find their natural frequencies
2. Develop subsequent design by refining the dynamics properties.

Some components and interfaces in the system needed to simplified so that we can achieve the meaningful results from analysis. After simplifications, finite element model was constructed that takes into consideration all the flexibilities of all the parts in the disk/platter - spindle system, as shown in figure below.



xix Figure 4.50 Model of Spindle Assembly



xx Figure 4.51 Cross-Section View of Spindle Assembly

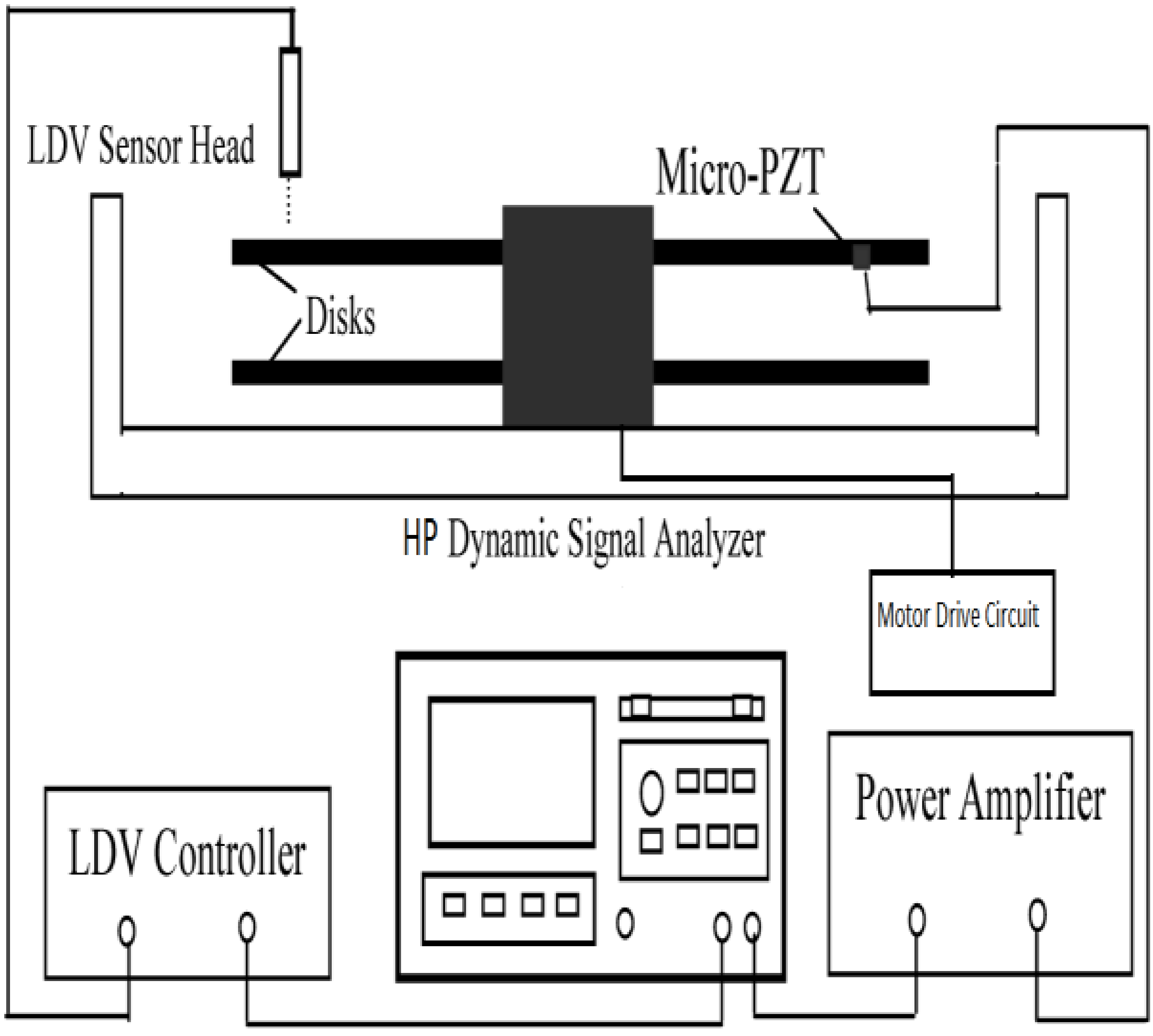
Below table shows the natural frequencies of the disk-spindle systems.

Mode No.	Frequency(Hertz)
1,1(sine)	1038.2
1,1(cosine)	1039.5
0,1	1058.2
2,1(sine)	1273.4
2,1(cosine)	1275.6

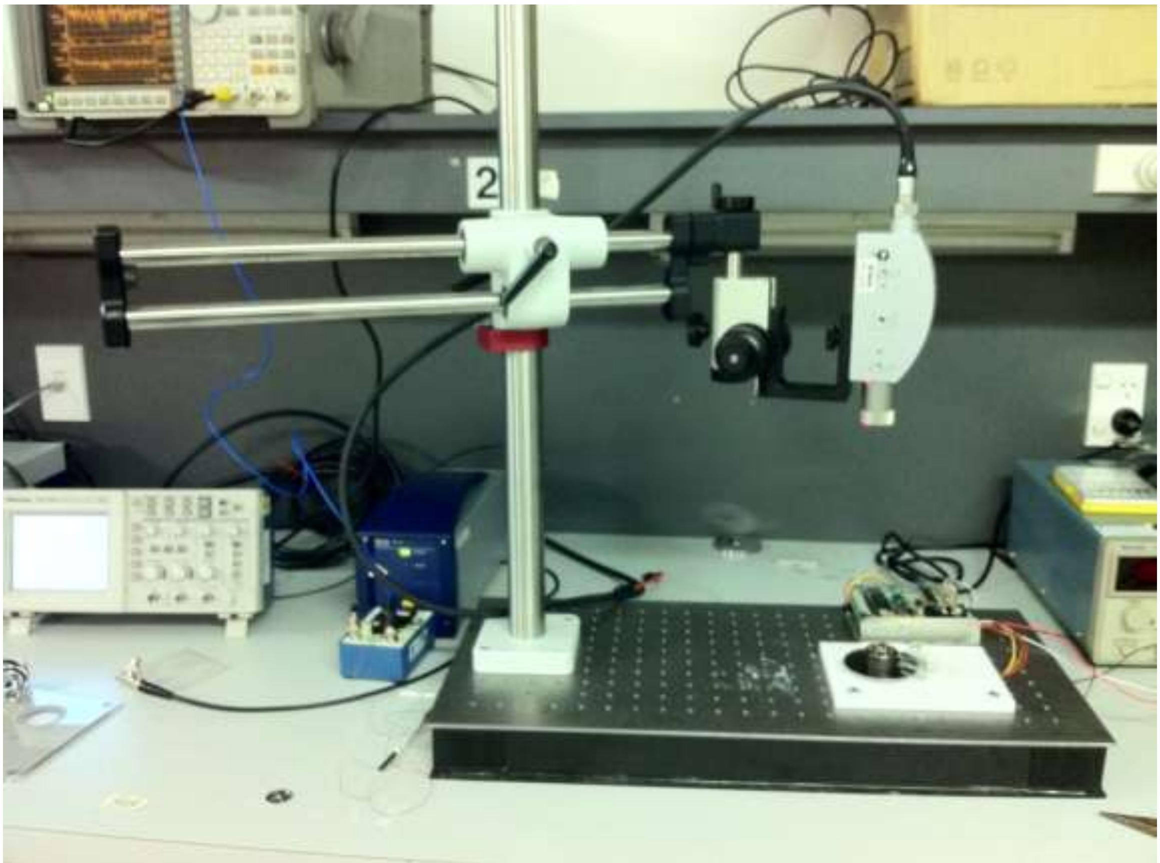
e Table 4.4 Natural Frequencies of Disk-Spindle Assembly

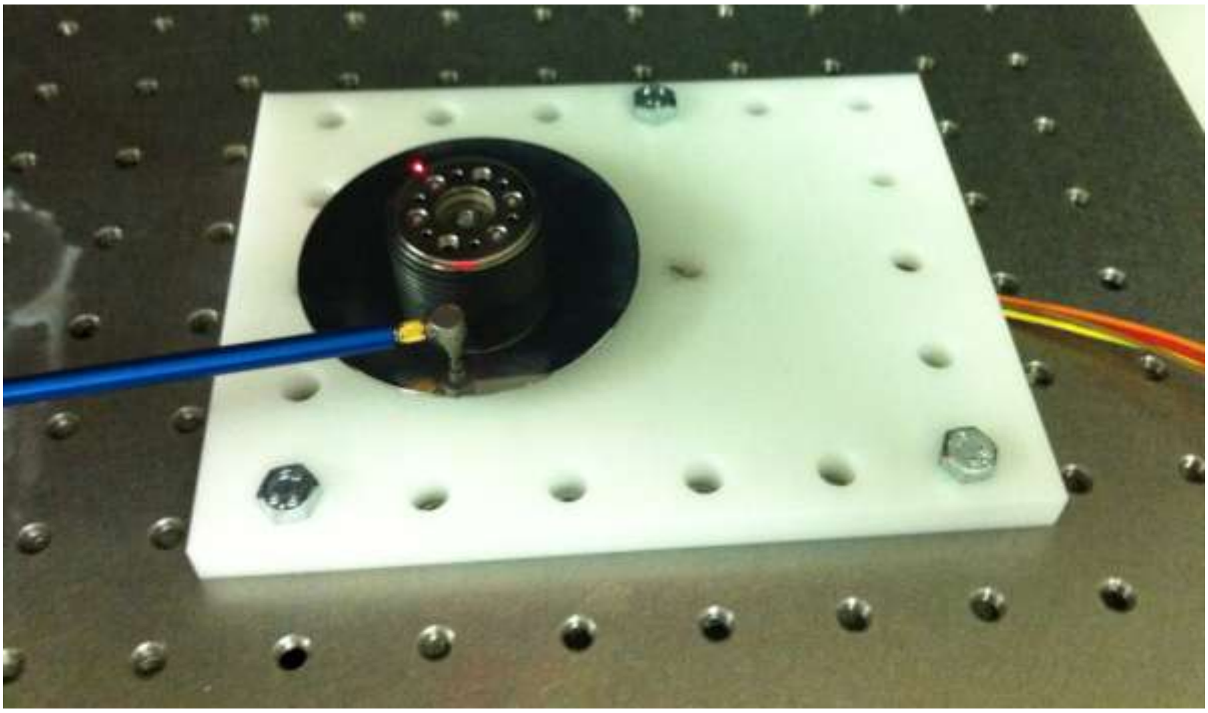
Similar mode shapes were obtained with slightly different natural frequencies. It shows that different parts such as the ball bearings, the shaft and the hub have been responsible for slight changes in natural frequencies.

CHAPPTER 5: Experimental Illustration



xxi Figure 5.0 Illustration of Experimental setup





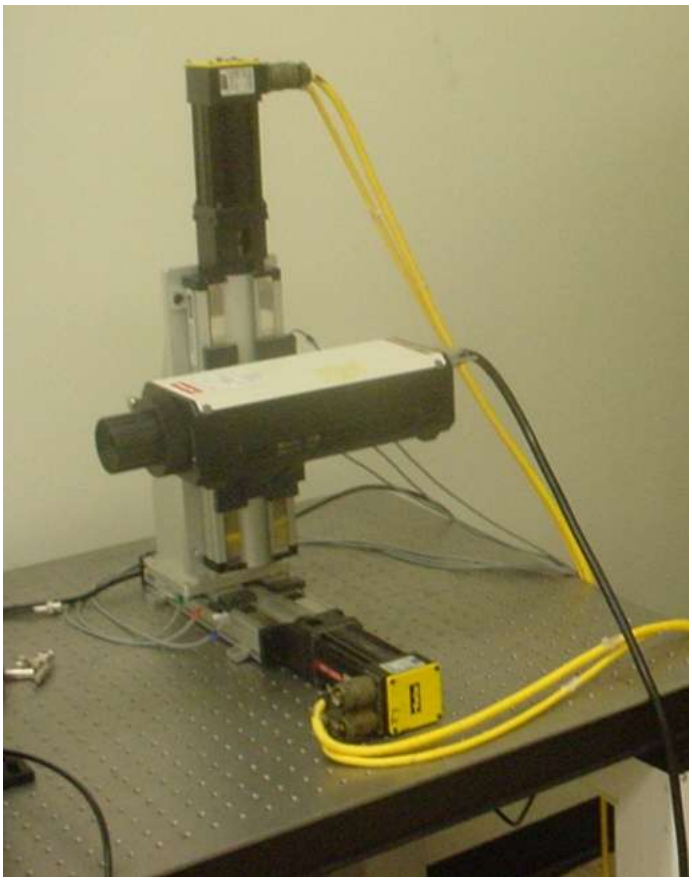
xxii Figure 5.01 Experimental Setup

5.1 Experimental Setup and Apparatus

For experimental testing, I have designed a test stand that should cancel out all the other structural vibration so the accurate results can be achieved.

Apparatus used for testing:

- Laser Vibrometer (LDV sensor head and controller): it was used to measure vibration, in term of distance travel in Y-Direction
- HP DYNAMIC SIGNAL ANALYSER: it's a versatile two- or four-channel high-performance FFT-based spectrum analyser, it was used to analyse the measurements taken by LDV sensor
- Micro-PZT hammer and Power Amplifier: It was used to excite the system
- MOTOR DRIVE CIRCUIT HC6250B-PT: was used to run the HDD spindle motor and also control the speed of it.



xxiii Figure 5.10: Laser Vibrometer

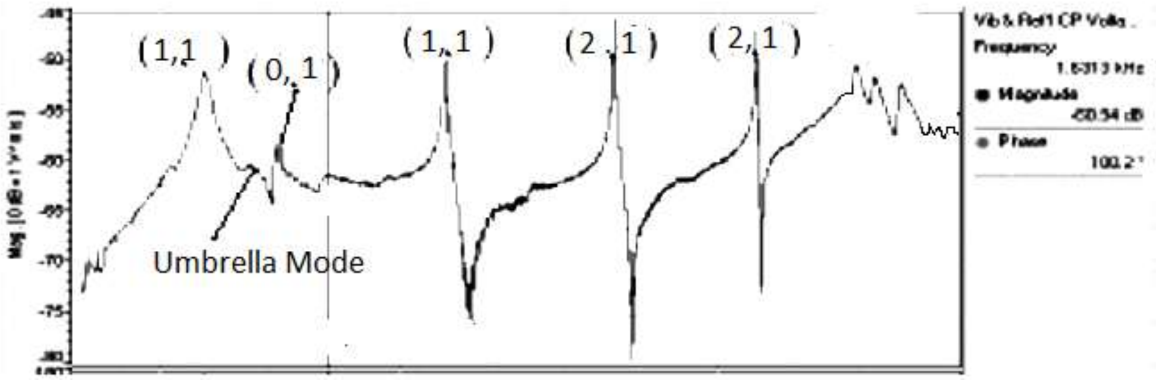
xxiv Figure 5.11: HP Dynamic Signal Analyzer

5.2 Experimental Results

To calculate the vibration interaction behaviour of the whole disk-spindle system, modal testing of the HDD spindle assembly has been performed. For modal testing, impact hammer was used to provide an excitation to measure the frequency response functions.

Sometime all modes of the vibration do not get excited because impulse produced by an impact hammer is usually inadequate to excite all the modes, and to overcome this issue, a special servo-motor system should have been developed to automatically drive the hammer but here under actual test condition, platter without rotation, I have used another simple and convenient method. An impact hammer was used manually.

Figure below show different modes of vibration obtained by using the proposed method.

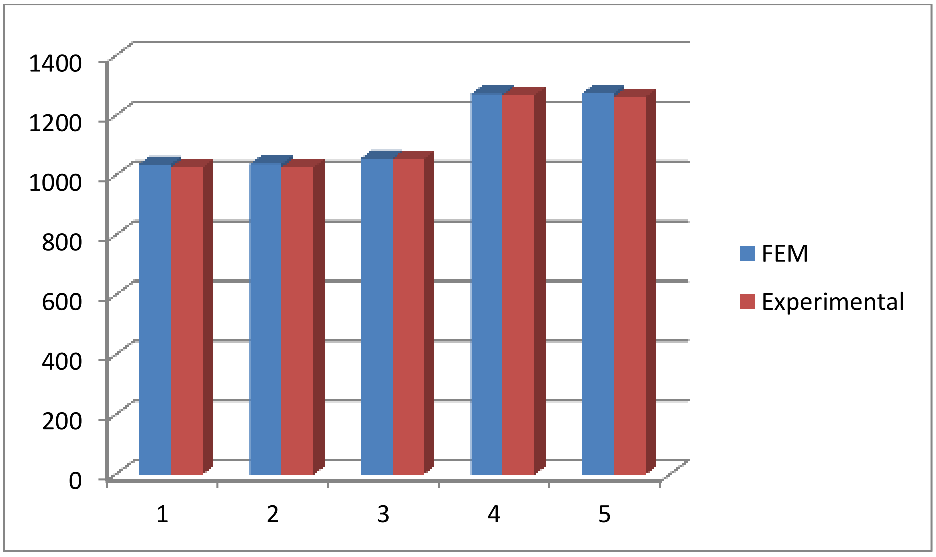


xxv Figure 5.20 Modes of vibration

As it can be seen in above figure, all the modes of the disk platters have been excited by such method of using impact hammer. Therefore; it can be said that no modes had been missed out by this proposed experimental method.

Table below shows the comparison between FEM analysis and experimental testing, it can be seen that both results are quite similar with very less error percentage. It can also be seen that modelling for the HDD spindle system is effective. Modelling method used in this research can help engineers for their future research on HDD designs. It is also much quicker to use FEM simulation rather than real test.

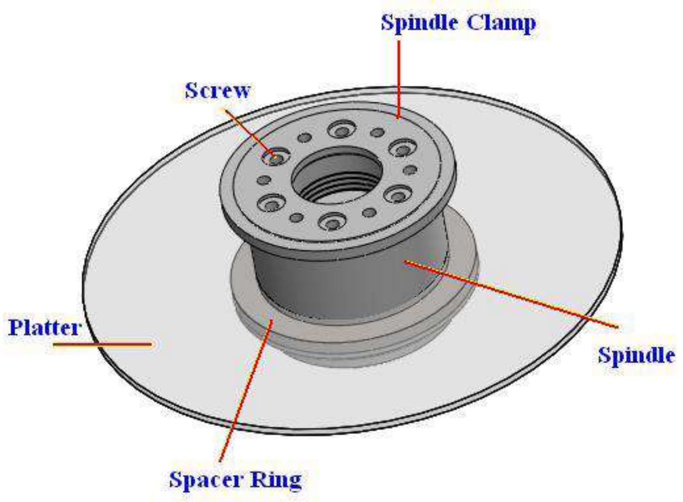
Mode No.	**FEM Frequency(Hz)**	**Experimental Frequency(Hz)**	**Error Percentage**
1,1(sine)	1038.2	1027.6	1.03%
1,1(cosine)	1039.5	1028.1	1.1%
0,1	1058.2	1055.6	0.25%
2,1(sine)	1273.4	1269.2	0.33%
2,1(cosine)	1275.6	1263.3	1.7%



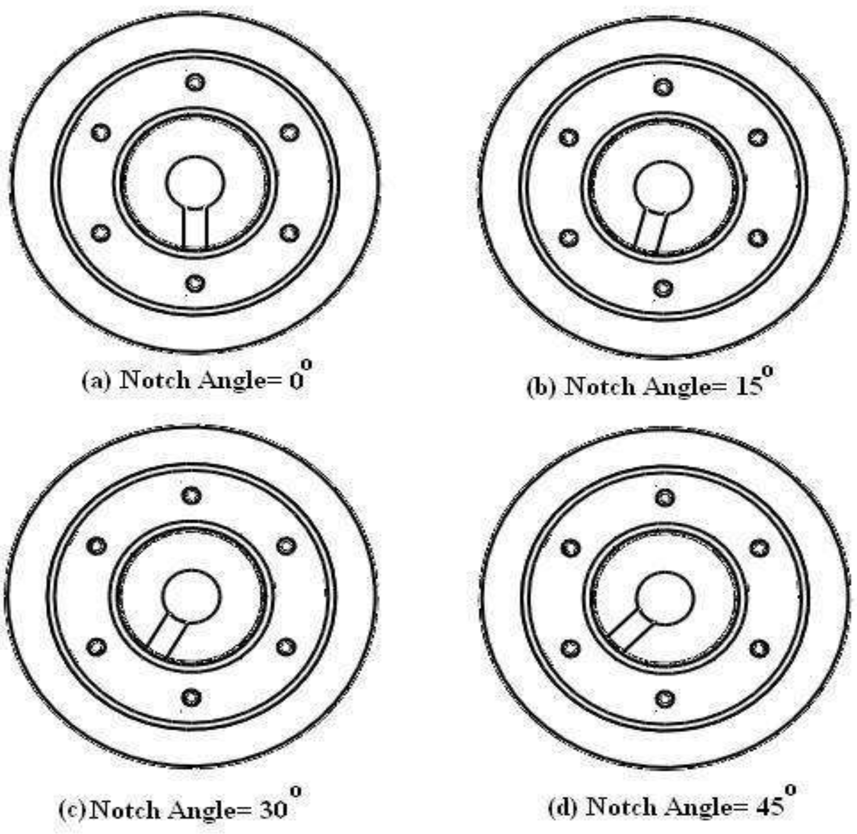
f Table 5.1 Natural Frequencies Comparison

CHAPTER 6: Thermal Analysis of Disk Spindle Assembly on RRO in Computer Hard Disk Drives

Magnetic disk-spindle assembly (DSA) is one of the most important mechanical systems in a computer hard disk drive (HDD). Its dynamic characteristics have great influence on HDD's performance. Because of various applications of HDD, wide operating temperature of the HDD has been carefully engineered. The corresponding thermal effect is found to be a critical effect on the performance and lifetime of HDD. This paper emphasizes primary on management of disk's repeatable run-outs caused by thermal expansion mismatch between components in a DSA. The RRO errors defined as the radial deviation from perfectly circle are commonly found to be caused by dynamics of a DSA, including thermal dynamic case of thermal shocks and operational temperature changes. To minimize these off-track errors and to provide performance improvement, mechanical design of the aforementioned components such as magnetic disk, spindle hub, and disk clamp in a DSA to sustain thermal excitations so as to allow the HDD to operate within allowable performance



xxvi Fig. 6.1 Schematic of a disk-spindle assembly

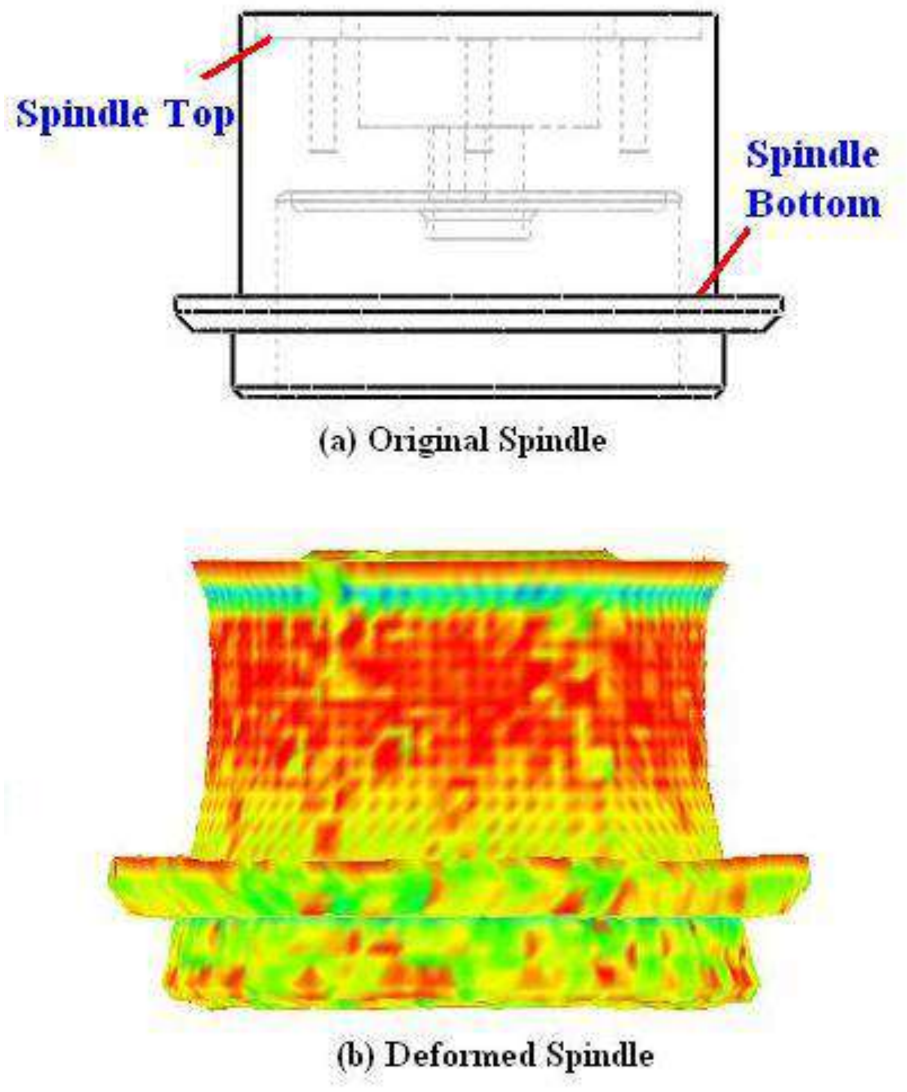


xxvii Fig. 6.2 Spindle hub designs with different slot angles.

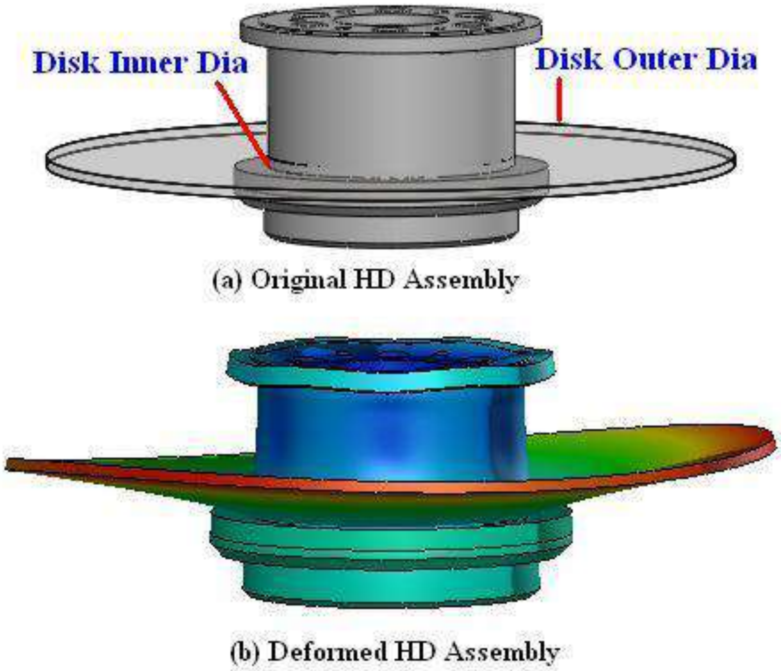
targets at various temperature points is found to be critically important. This paper presents a disk-spindle assembly design approach that would be able to reduce certain RRO components caused by thermal expansion mismatches in a DSA. (Chen Y. , 2002)

6.1 Problem Setup

Fig. 6.1 shows the schematic of the disk-spindle assembly model. The spindle clamp is constrained to spindle hub at six evenly spaced screw locations. Material properties of the spindle and the clamp, and the disk have been taken as stainless steel and glass, respectively. The assembled model was generated by using commercial CAD software and the corresponding thermal finite element model was conducted with hinge boundary condition placed between spindle shaft and the inner most circular through hole as shown in Fig. 6.1. Different spindle notch/slot designs were developed to represent bearing insertion feature or any imperfection caused during manufacturing processes to the DSA at notch angles of 0°, 15°, 30° and 45° with respect to the centre of the spindle screw as illustrated in Fig. 6.2.



xxviii Fig. 6.3. Cone shaped deformation of hard disk spindle on 80°C operational temperature and 0° notch angle.

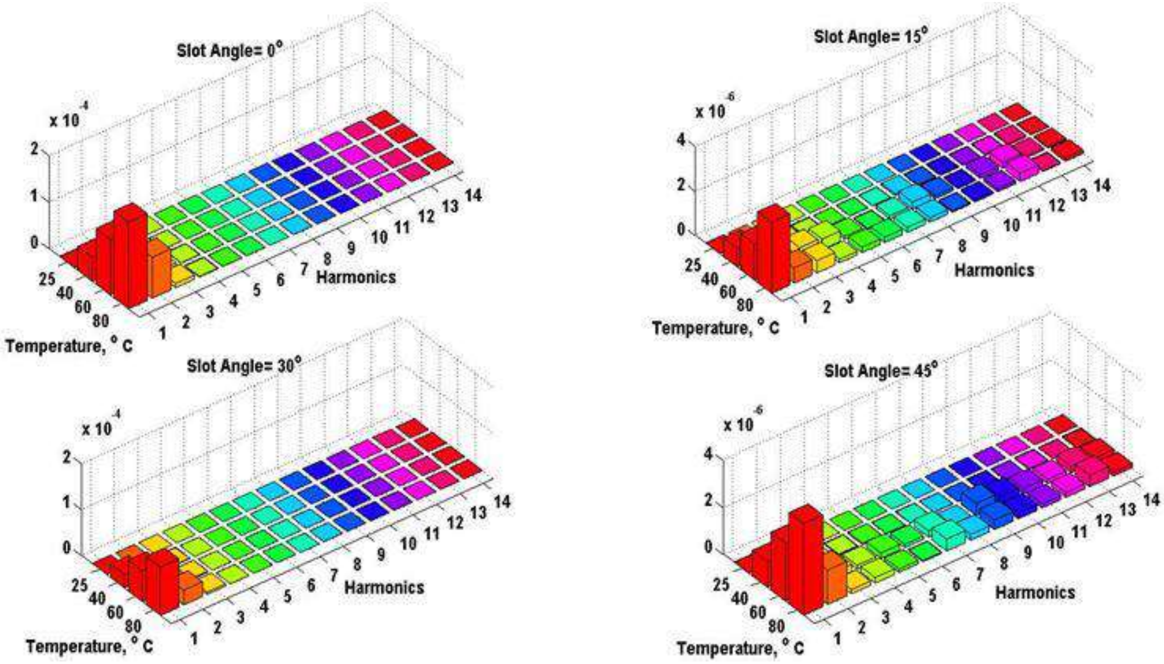


xxix Figure 6.4: Potato Chip formation of Platter in hard disk spindle assembly at 80°C operational temperature and 0° spindle notch angle.

6.2 Experimental Investigation

Thermal stress observations have been taken at different operational temperatures between 25°C – 80°C. Because of the six fasteners at spindle top, it is found as expected that 6th harmonic exist on the disk inner diameter (ID). Because it is constrained by the screws, less thermal expansion occurs at spindle hub top portion as that at bottom, resulting in cone shaped deformation as shown in Fig. 6.3. This uneven thermal stress distribution also causes the disk to deform as a potato chip as shown in Fig. 6.4. Fourier analysis has been performed on radial displacement of the disk ID and OD, the outer diameter, at different temperatures and slot designs. Fig. 6.5 displays RRO harmonic components at disk OD radial displacement as functions of ambient temperature and slot orientation.

Large amplitudes for the RRO at 1th and 2nd harmonics are observed at the high temperature region at 80°C comparatively to room temperature at 25°C. It can also be observed that amplitudes of the 1st harmonic, caused by imperfection, and 2nd harmonic,-



xxx Fig. 6.5. RRO harmonics of OD radial displacement (mm) at different temperature and slot angle designs.

*Matlab codes for this are available are availabe on CD

-caused by spindle hub potato-chipping, increase monotonically with temperature. On the other hand, by changing the slot angle from 0 to 45^{o}, a major reduction in harmonic amplitudes has been observed. Slot angle at 15^{o} was reported as the best position for spindle notch as it reduced RRO more significantly comparative to other notch designs.

6.3 Conclusion

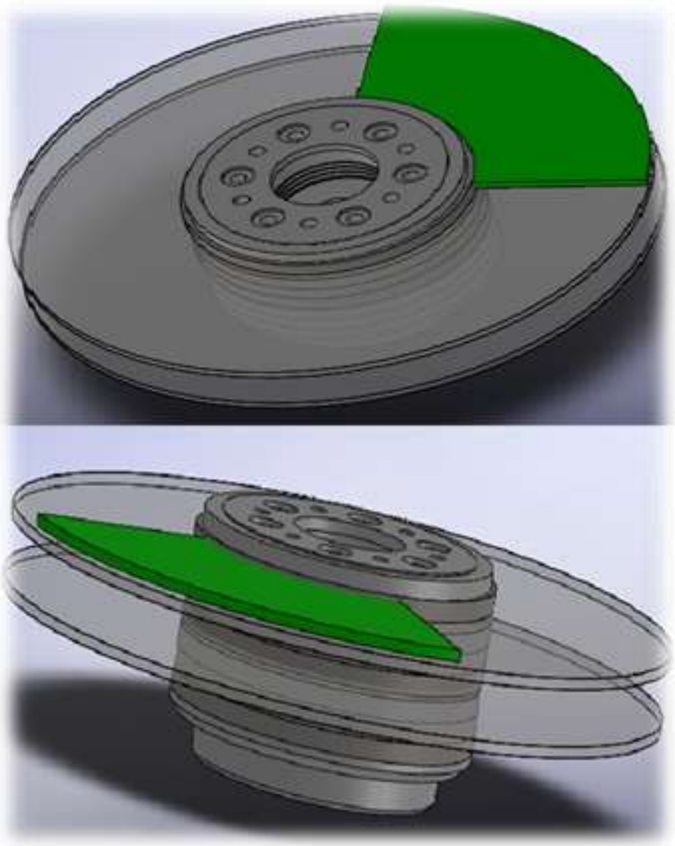
This part examines radial repeatable run-out triggered by component thermal expansion mismatches in a disk-spindle assembly. Fourier analysis using finite element result at spindle top, spindle bottom, disk ID and OD suggests that spindle hub can deform as a potato-chip, which results in 2nd harmonic commonly seen in RRO position error signal measurements. It is reported that one can intentionally design an insertion feature, the notch, for spindle bearing to significantly reduce amplitudes of the 1st and 2nd RRO harmonics, which provides mechanical solution in resolving low frequency RRO in computer hard disk drives.

CHAPTER 7: Summary and Future Work

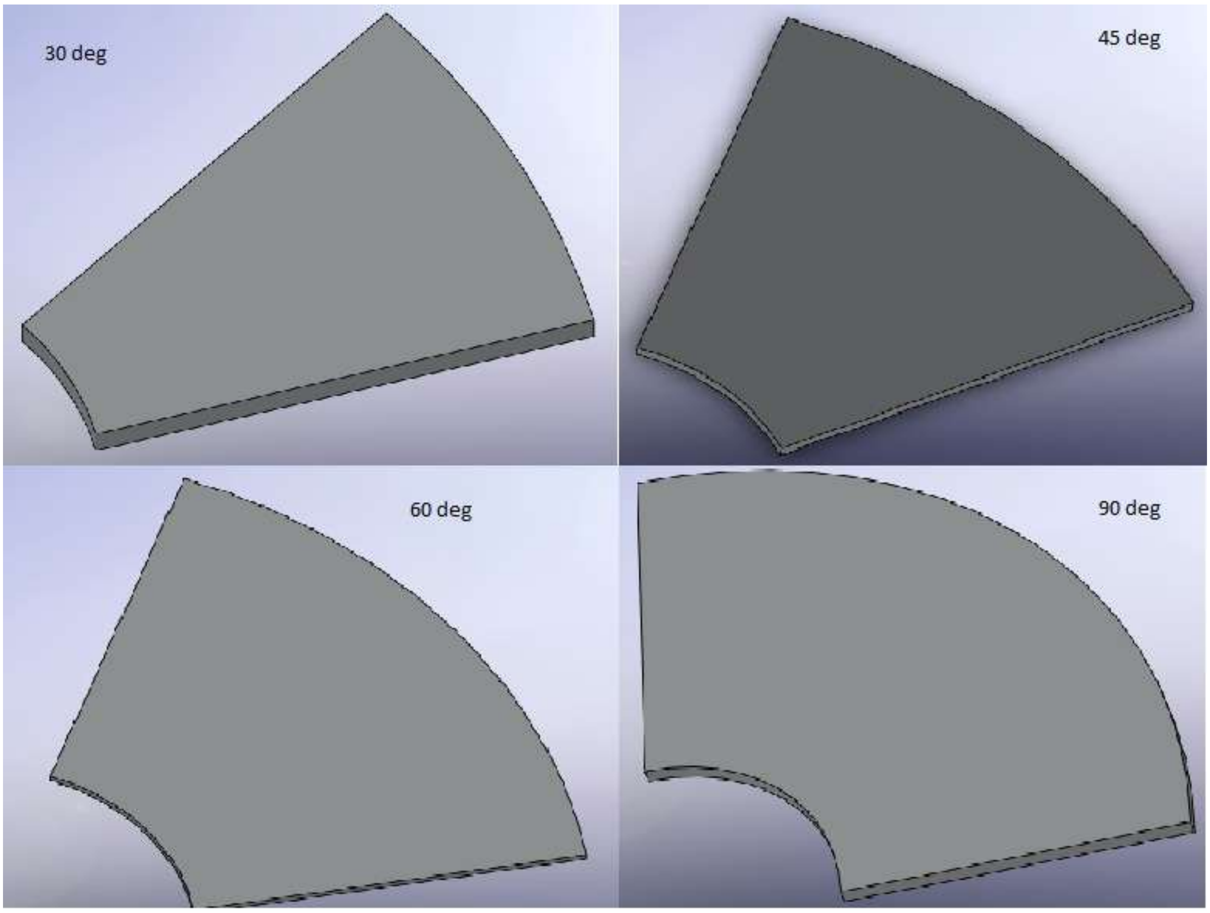
Vibration in any system can be reduced or even it can be eliminated with the help of some kind of damping system. One way of reducing vibration HDD platter is to use air damping system. In the thesis, a detailed vibrational and thermal analysis of HDD platter and HDD Spindle Assembly was performed. The thesis proposed a simple but analytical approach for the design of HDD enclosure with different sector shapes and for the spindle motor with different slot/notch angles, based on the analysis and simulation results.

The thesis contains a literature review of related general articles. A historic guide through the vibration control history of the HDD was described. The necessary equations of motion for the dynamic modelling of the system were presented.

7.1 Sector Plates



xxxi Figure 7.1 HDD Spindle-Platter Assembly Model with Sector Plate



xxxii Figure 7.11 Sector Plates of Different Angles

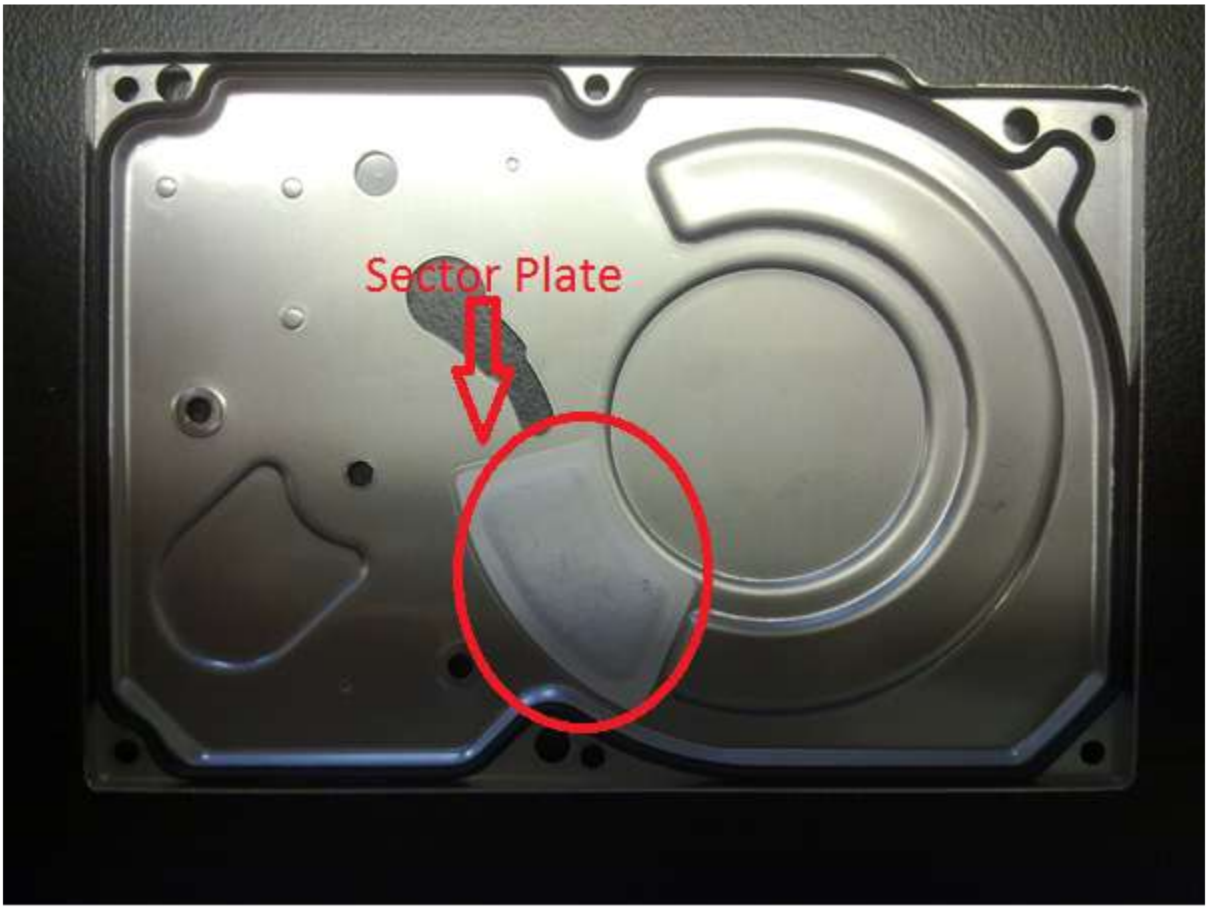
7.2 Future Work

The Design concept may be manufactured to verify the simulation results. The design concept can be planned to expand further for the control of vibration in other devices.

Based on such a series of studies, the proposed method can be concluded as a very promising tool for the design of HDDs and various other high performance computer disk drives such as floppy disk drives, CD ROM drives, and their variations having spindle mechanisms similar to those of HDDs

7.3 Production design for sector plate:

Following picture shows one possible way of implementing those sector plates into HDD enclosure. From the manufacturing perspective it can be easily made with by a punching process.



xxxiii Figure 7.2 Possible design of HDD enclosure with sector plate

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Appendix A: Abbreviations

ABS -- Air-Bearing Surface
ADB -- Aerodynamic Bearing
ADC -- Analog-to-Digital Converter
AFC -- Adaptive Feedforward Control
AGC -- Automatic Gain Control
BLDC -- Brushless DC
BPI -- Bits per Inch
CD-ROM -- Compact Disk - Read Only Memory
DAC -- Digital-to-Analog Converter
DASD -- Direct Access Storage Device
DFT -- Discrete Fourier Transform
DISO -- Dual Input Single Output
DMS -- Decoupled Master Slave
DSP -- Digital Signal Processor
DTR -- Discrete Track Recording
DVD -- Digital Versatile Disk
EMF -- Electromotive Force
EMI -- Electro-magnetic Interference
FDB -- Fluid Dynamic Bearing
FIR -- Finite Impulse Response
FOH -- First-Order Hold
GA -- Genetic Algorithm
GB – Gigabytes
GM -- Gain Margin
GMR -- Giant Magneto-Resistive
HAMR -- Heat Assisted Magnetic Recording
HDD -- Hard Disk Drive
ID -- Inner Diameter
IDE -- Integrated Device Electronics
ITAE -- Integral of Time multiplied by Absolute value of Error
IVC -- Initial Value Compensation
LBA -- Logical Block Address
LDV -- Laser Doppler Vibrometer
LMI -- Linear Matrix Inequality
LQG -- Linear Quadratic Gaussian
LQR -- Linear Quadratic Regulator
LTI -- Linear Time Invariant
LTR -- Loop Transfer Recovery
MASSC -- Maximum Allowable Stable Step Change
MEMS -- Micro Electro-Mechanical Systems
MIMO -- Multi Input Multi Output
MMF -- Magnetomotive Force

MR -- Magneto-resistive
NRRO -- Non-repeatable Runout
OD -- Outer Diameter
PCB -- Printed Circuit Board
PES -- Position Error Sensing
PID -- Proportional-Integral-Derivative
PMACM -- Permanent Magnet AC Motor
PMSM -- Permanent Magnet Synchronous Motor
PTOS -- Proximate Time Optimal Servomechanism
RAM Random Access Memory
RAMAC Random Access Method of Accounting and Control
ROM -- Read Only Memory
RPM -- Revolutions per Minute
RRO -- Repeatable Runout
SISO -- Single Input Single Output
SNR -- Signal-to-Noise Ratio
SQP -- Sequential Quadratic Programming
SSTW -- Self-Servo Track Writing
STM -- Servo Timing Mark
STW -- Servo Track Writer
TFI -- Thin Film Inductive
TMR -- Track Mis-Registration
TPI -- Tracks per Inch
UMP -- Unbalanced Magnetic Pull
VCM -- Voice Coil Motor

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