TRIBHUVAN UNIVERSITY

INSTITUTE OF ENGINEERING

PULCHOWK CAMPUS

THESIS NO: M-41-MSMDE-2018-2022

Modal Analysis of a Spindle of Computer Numeric Controlled Milling Machine

by

Anup Adhikari

A THESIS

SUBMITTED TO THE DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

MASTERS OF SCIENCE IN MECHANICAL SYSTEM'S DESIGN AND ENGINEERING

DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING

LALITPUR, NEPAL

MARCH, 2022

COPYRIGHT

The author has agreed that the library, Department of Mechanical and Aerospace Engineering, Pulchowk Campus, Institute of Engineering may make this report freely available for inspection. Moreover, the authors have agreed that permission for extensive copying of this project report for scholarly purpose may be granted by the professor(s) who supervised the project work recorded herein or, in their absence, by the Head of the Department wherein the project report was done. It is understood that the recognition will be given to the author of this report and to the Department of Mechanical and Aerospace Engineering, Pulchowk Campus, Institute of Engineering in any use of the material of this project report. Copying or publication or other use of this report for financial gain without approval of the Department of Mechanical and Aerospace Engineering, Pulchowk Campus, Institute of Engineering and authors' written permission is prohibited.

Request for permission to copy or to make any other use of the material in this report in whole or in part should be addressed to:

Head

Department of Mechanical and Aerospace Engineering

Pulchowk Campus, Institute of Engineering

Lalitpur, Nepal.

TRIBHUVAN UNIVERSITY

INSTITUTE OF ENGINEERING

PULCHOWK CAMPUS

DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING

The undersigned hereby certify that they have read, and recommended to the Institute of Engineering for acceptance, thesis entitled "Modal Analysis of a Spindle of Computer Numeric Controlled Milling Machine" submitted by Anup Adhikari in partial fulfillment of the requirements for the degree of Master of Science, Mechanical System Design and Engineering.

Supervisor, Dr. Sanjeev Maharjan
Assistant Professor
Department of Mechanical and Aerospace Engineering,
Pulchowk Campus, Institute of Engineering, TU

External Examiner, Er. Jahir Ahamad Jibran
Assistant Professor
Department of Mechanical Engineering,
School of Engineering, Kathmandu University

Committee Chairman, Dr. Surya Prasad Adhikari
Head
Department of Mechanical and Aerospace Engineering
Pulchowk Campus, Institute of Engineering, TU

Date: March 20, 2022

ABSTRACT

Computer Numerical Controlled (CNC) machine for milling can accomplice the purposes of drilling and turning. Machine tools is one of many aspects that can possibly impact the quality of the finishing product of a machining. Exploration on the stability and identification of the vibration that occurred in the spindle of CNC machine is carried out. A spindle comprises of a motor, a taper for holding tools, and a shaft that holds together all the different components. Spindles rotate on an axis, which gets input on movement from the associated CNC controller.

Modal analysis was carried out for the model of spindle. Rest of the cases assessment were done against this model. Total of eight cases formed by additional stiffeners varying position and figures. The first case was of spindle model only. This study uses the outcome of vibration obtained to support by performing modal analysis while varying the stiffeners position and shape.

The frequency analysis helps us to explore effect of stiffeners in vibration. The input parameters were kept same so that any result observed would be because of the addition of stiffeners only. One case each for rectangular and circular shaped stiffeners applied near tool holder was created. The first natural frequency for spindle only was obtained as 290.28 Hz. Maximum deformation was 4.04. Application of stiffeners showed reduced modal amplitude.

Combination of rectangular and circular shaped stiffeners were applied together and in addition with the stiffeners in groove to create four more cases. The modal analysis for all these cases was performed to obtain mode frequency and deformation shapes for the spindle.

The deformation shape was found similar in eight cases. Addition of stiffeners has no significant effect on shifting of mode shape and. Addition of stiffeners however, lowers amplitude of modal displacement to reduce vibration on tool holder.

The shape of stiffeners has no noteworthy effect on vibration. The frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners are nearly identical.

ACKNOWLEDGEMENTS

Research is never complete without help of wonderful people behind. Pertaining to this research, I would like to express my sincere gratitude towards my supervisor, Prof. Dr. Sanjeev Maharjan for his continuous effort and the guidance he has given me in accomplishing my dissertation. Constant brainstorming, idea generation on refining the main motive of the research with him has given me great confidence in narrowing down the objective of the study. My due respect to the Department of Mechanical Engineering for the constant coordination and support. I am grateful to all the department professors and lecturers for providing valuable suggestions and kind support throughout the project.

Finally, I am greatly indebted towards my family who have stood beside me as an encouragement as well as in providing favorable environment to complete my research work.

TABLE OF CONTENTS

COPYRIGHT……………………………………………………………………......2

ABSTRACT………………………………….…………………..………......4

ACKNOWLEDGEMENT………………………………….....………...………5

List of Tables 8

List of Figures..9

LIST OF ABBREVIATIONS 10

CHAPTER ONE: INTRODUCTION 11

1.1 Background 11

1.2 Problem Statement 13

1.3 Objectives...14

1.3.1 Main Objective..14

1.3.2 Specific Objectives 14

CHAPTER TWO: LITERATURE REVIEW 15

2.1 Modal Analysis...17

CHAPTER THREE: METHODOLOGY OF RESEARCH..................................21

3.1 Literature Review 21

3.2 Model Development of Spindle 22

3.3 Analysis in ANSYS Workbench 2016 22

3.4 Iteration of Process 3.3..22

3.5 Comparisons of Dynamic Property 22

3.6 Compilation, Discussion and Presentation of Final Report 22

3.7 Autodesk Inventor Professional 2015 23

3.8 Geometric Modelling 23

3.9 ANSYS Model 23

3.10 Defining cases for study 24

3.10.1 Case I: Model of Spindle...24

3.10.2 Case II: Model of spindle with rectangular shaped stiffeners near tool holder....25

3.10.3 Case III: Model of spindle with circular shaped stiffeners near tool holder....26

3.10.4 Case IV: Model of spindle with stiffeners on groove....26

3.10.5 Case V: Model of spindle with rectangular and circular shaped stiffeners near tool holder....27

3.10.6 Case VI: Model of spindle with stiffeners on groove and rectangular shaped stiffeners near tool holder....27

3.10.7 Case VII: Model of spindle with stiffeners on groove and circular shaped stiffeners near tool holder....28

3.10.8 Case VIII: Model of spindle with stiffeners on groove with rectangular and circular stiffeners near tool holder....28

3.10.9 Model of Stiffeners....29

CHAPTER FOUR: RESULTS AND DISCUSSION....30

4.1 Modal analysis of the spindle....30

4.2 Effect of stiffeners on amplitude of modal displacement....31

4.3 Effect of stiffeners on variation of mode frequency of the spindle....34

4.4 Effect of stiffeners on variation of mode shape and deformation behavior of spindle by changing shape of stiffeners. 38

4.5 Effect on other modes....42

Mode 4....42

Mode 5....46

Mode 9....50

CHAPTER FIVE: CONCLUSIONS AND RECOMMENDATIONS....56

REFERENCES....57

LIST OF TABLES

Table 3.1:Parameters used for analysis..24
Table 4.1:First ten modes with their frequency on spindle only model.......................30
Table 4.2:Frequency, deformation and modal amplitude difference on first mode.....31
Table 4.3:Frequency, deformation and modal amplitude difference on second mode 35
Table 4.4:Frequency, deformation and modal amplitude difference on third mode ...39
Table 4.5:Frequency, deformation and modal amplitude difference on fourth mode .42
Table 4.6:Frequency, deformation and modal amplitude difference on fifth mode 47
Table 4.7:Frequency, deformation and modal amplitude difference on ninth mode...51

LIST OF FIGURES

Figure 3:1: Research method used in this study 21
Figure 3:2: Spindle with Label 23
Figure 3:3: Case I: Model of spindle 25
Figure 3:4: Case II : Model of spindle with rectangular shaped stiffeners.................25
Figure 3:5: Case III Model of spindle with circular shaped stiffeners 26
Figure 3:6: Case IV Model of spindle with stiffeners on groove 26
Figure 3:7: Case V Model of spindle with rectangular and circular shaped stiffeners near tool handler 27
Figure 3:8: Case VI Model of spindle with stiffeners on groove and rectangular shaped stiffeners near tool handler..27
Figure 3:9: Case VII Model of spindle with stiffeners on groove and circular shaped stiffeners near tool handler..28
Figure 3:10: Case VIII Model of spindle with stiffeners on groove with rectangular and circular shaped stiffeners near tool handler 28
Figure 3.11: Model of rectangular and circular stiffeners 29
Figure 3.12: Model of stiffeners on groove 29
Figure 4.1(a-h): First Mode for Case I to Case VIII... 30-34
Figure 4.2(a-h): Second Mode for Case I to Case VIII... 35-38
Figure 4.3(a-h): Third Mode for Case I to Case VIII 39-41
Figure 4.4(a-h): Fourth Mode for Case I to Case VIII.. 43-46
Figure 4.5(a-h): Fifth Mode for Case I to Case VIII... 47-50
Figure 4.6(a-h): Ninth Mode for Case I to Case VIII 51-54

LIST OF ABBREVIATIONS

CNC	Computer Numerical Controlled
FEA	Finite Element Analysis
FEM	Finite Element Method
3D	Three Dimension
Hz	Hertz
m	Meter
f_i	i-th natural frequency
M	Mass Matrix
C	Damping Matrix
K	Stiffness Matrix
X	Displacement response vector
F	Exciting force vector
ϕ	Eigen vector or mode shape
ω	Circular natural frequency
λ	Eigen value
I	Identity matrix
A	Square matrix
det	determinant of a matrix
ϕ_i	i-th mode shape
ξ	i-th modal displacement

CHAPTER ONE: INTRODUCTION

1.1 Background

CNC machine is a numerical control machine in which a code is given to the system for the desired result, and the machine then executes the command. Drilling and turning can be accomplished with a CNC milling machine. Such milling machine has been generally used in manufacturing arena these days. CNC machine tool must be better designed and constructed. It must be accurate than the conventional machine tool. It should minimize all the idle time by increasing the productivity and eliminate the tedious and costly time-consuming machining operation.

Vibration taking place on machine tools has been a grave shortcoming for engineers since long. Undesired relative vibrations between the work-piece and the tool downgraded the excellence of the machine surfaces during engraving. A self-excited vibration difficulty happening in significant degrees of material removal, lead to the inevitable flexibility among the cutting tool and workpiece is machine tool chatter. Furthermore, it affects coarser surface quality and dimensional error of the workpiece, along with excessively piercing noise and accelerated tool erosion. In general, one of the main limiting elements to consider while developing a production process is chatter. (Erol Turkes, 2017).

The study of the dynamic properties of structures under vibrational excitation is known as modal analysis. A set of discrete modes of vibration can be used to describe the dynamic behavior of a structure across a specified frequency range. The modal parameters are the parameters that describe each mode's natural frequency or resonance frequency (modal) damping mode forms. Vibration difficulties induced by these resonances or modes can be analyzed and understood by using the modal parameters to describe the structure. The resonances of a structure must be detected and quantified in order to better comprehend any structural vibration problem. Defining the structure's modal parameters is a frequent technique to do this. Spindle deformations, both static and dynamic, play a vital role in machining process tolerance integrity and stability, affecting component quality and productivity. Modal analysis is critical for comprehending and optimizing structures' underlying dynamic behavior, resulting in lighter, stronger, and safer structures that perform better.

Computers and digital control system are responsible for higher accuracy and productivity. The overall accuracy is the combination of accuracy of electric as well as mechanical system. High accuracy and high productivity seem contradictory simultaneously. Because high productivity requires high feed rate, depth of cut, etc. which is responsible for generation of high wear, heat dissipation, thermal deterioration, vibration of machine etc.

The stability of the spindle of a CNC machine is investigated in this thesis, as well as the identification of vibrations that occur. A spindle is made up of a motor, a taper for holding tools, and a shaft that connects all of the individual parts. Spindles rotate on an axis that receives movement commands from the CNC controller. The resulting vibration result is confirmed by doing a dynamic analysis with ANSYS Finite Element Analysis (FEA).

The spindle arrangement is a critical component of the system, and its dynamic qualities have a direct bearing on its performance on the machine's processing precision (Liu, 2010). The analytical and FEA methodologies are currently the most popular spindle dynamic analysis methods (Woodford, 1991). FEA has been used in numerous research, with the findings proving the method's dependability and ability to analyze spindle assembly dynamics (Bathe, 1976). Chatter vibration causes an unsteady removal process and poor surface quality in machining processes.

The stability lobe diagram is a standard method for identifying chatter-free machining settings (Kong Jiali, 2017). The connection at the tool holder-spindle interface must be examined so as to develop stability lobe diagrams (Weiwei Zhang, 2017). The tool holder-spindle contact connection should be evaluated for the distributed-springs connection, however the relative competitiveness over the rigid-connection hypothesis was not stated (Qin-Yuan, 2011). The conclusions of the rigid-connection supposition at the tool holder-spindle interface were closely matched with the deductions from the spring-connection assumption, and both simulations agree with the experiment well. Within a particular frequency range, the rigid-connection approach is able to correctly imitate the mode (Mehdi Namazi, 2007). As a result, the rigid-connection connection is assumed in this work at the interaction between the tool holder and the spindle.

1.2 Problem Statement

With the growth of the sector, the demand for CNC accuracy has skyrocketed. There are a number of elements that affect the quality of a machining's ultimate result. The state of the machine tool itself, the quality of the cutter, and the dynamics of the operation are just a few of them. The smoothness of the surface profile is influenced by the machine tool's structural dynamics, the cutting process's dynamics, and workpiece tool interactions. Modal analysis is a technique for analyzing milling vibrations. Modal testing results were used to determine the inherent frequencies of the spindle and the form of its vibration modes. As a result, the dynamic properties of a CNC spindle at resonance frequencies and vibration shapes are the subject of this project. The spindle was activated by an impulse force, resulting in an excited vibration response.

The stiffness and mass of the CNC spindle have a direct relationship with the frequency of vibration. The natural frequencies that describe each mode are known as modal parameters. Vibration difficulties induced by such resonances or modes can be analyzed and understood by using modal parameters to describe the structure. Vibration is a prevalent problem that affects machining performance as well as spindle life. This demonstrates that there is a vibration issue that affects the surface finish. A structural study must be completed before moving on to spindle optimization or active chatter suppression.

Analysis and calculation of the primary components, as well as the dynamic performance of the spindle, are critical for assuring optimal performance and accuracy design of a CNC milling machine. This can be done by reducing vibration of a spindle. So, it is good idea to determine ways to reduce vibration and compare the approach.

1.3 Objectives

1.3.1 Main Objective

The main objective of this research is to study the vibration behavior of CNC milling spindle by using modal analysis and comparing the frequency resulted by addition of stiffeners.

1.3.2 Specific Objectives

The specific objectives of the thesis are:

1. To study modal analysis of the spindle for frequency and deformation.
2. To analyze effect of stiffeners on variation of mode frequency of the spindle.
3. To analyze mode shape and deformation behavior of spindle by changing shape of stiffeners.

CHAPTER TWO: LITERATURE REVIEW

This research work is about the vibration analysis of spindle in CNC Milling machine, which mainly deals with the analysis of the spindle. Limited number of studies has carried out on the spindle itself and rather researches are based on cutting tools and tool holder assembly. However, similar analysis is commonly carried out in engines of automobiles. Thus, emphasis has been laid on modal analysis of spindle in the case while considering similar findings on similarly related engine parts to carry out the performance evaluation of the spindle.

The study and computation of the primary components, as well as the spindle's dynamic performance, are critical. The researcher used Finite Element Analysis and the Modal Impact Hammer Testing method to conduct experimental modal testing. The mode shape's inherent frequency was established, and a comparative analysis was conducted. The results of this project show that while the mode shape simulation of experimental data is not exactly the same as the finite element analysis, it is generally in accord. (Nik Muhamad Firdaus, 2013).

In each frequency range, the dynamic response of a machine tool structure can be modelled as a series of discrete vibration modes. The vibration difficulties induced by resonances can be explored by modeling the structure with modal parameters (Silva, 1997). It is still hard to anticipate the dynamic characteristics of machine tool components with merely a high degree of precision drafting. However, significant progress has been made in the advancement of experimental approaches for determining machine tool structure mode forms.

By employing stiffness measurements and modal analysis, several researchers have presented static and dynamic analysis of machine tool structures engaged in machining systems (Altintas, 2000), (Erhan Budak, 1994), (M. Polacek, 1963). The notion of modal analysis of machine tool structures is given, which is based on a parametric description of the measured responses (Kim, 1983).

The finite element approach for modal analysis of machine tool constructions has been described in research. The machine tool structure's first ten natural frequencies and mode shapes are determined in their research (Suo Xian Yuan, 2010).

It was shown how to use experimental and analytical modal analysis approaches to evaluate the structural dynamic processes of a vertical machining center. In their research, the experimental modal analysis of several milling machine components is carried out, and the theoretical mode form of the components is determined using FEM. (Anayet Ullah Patwari, 2009) .

A method for analyzing machine tool characteristics using modal analysis and harmonic response, as well as conducting modal experiments to determine modal frequency, damping, and form, was developed (Zhijun Wun, 2010). Meanwhile, to get precise dynamic characteristics, simulation and modal analysis have been performed using FEM.

FEM was used to analyze the instability of the machining process, but the conclusions were not supported by experimental results (Keith Rouch:, 2002). FEM modal analysis is a significant tool for assisting machine tool structure design in terms of dynamic stiffness; additionally, FEM mode shapes offer useful guidance for optimizing a machine tool's dynamic properties (Siamak Pedrammehr, 2012).

The use of FEA was to model and analyze the tool holder and spindle assembly. The rigid-connection is believed to exist at the tool holder-spindle interface. To model the rigid-connection, the author employs the ANSYS finite element analysis software. As a result, the current FEA modeling and analytic approach for studying tool holder-spindle assembly dynamics is possible (Jun Wang, 2012).

Natural frequency was determined by a researcher who dealt with finite element analysis of a car's engine bracket. The engine bracket serves as a structure for supporting the engine. The point to consider is engine bracket vibration and fatigue, which can lead to structural failure if vibration and stress levels are too high (Umesh S. Ghorpade, 2012).

The natural frequency of the engine bracket was optimized using finite element analysis and the usage of several lightweight materials. In comparison to cast iron, the frequency of a bracket made of Magnesium alloy was found to be optimal (Lakshmi Kala, July 2015).

The engine mounts play a crucial role in keeping the vehicle's powertrain components secure (Jasvir Singh Dhillon, September 2014). With this in mind, the current research looks at stiffener modeling, modal analysis, and mass optimization for a CNC milling machine.

2.1 Modal Analysis

Modal analysis provides valuable insights to the dynamic’s characteristics of the structure in the form of natural frequencies, damping factors and mode shapes. It is a linear analysis that doesn't utilize any excitations or loads. Mode frequencies are dependent only on two things: stiffness and mass. Mass increases the frequencies while stiffness decreases vibration. Modal analysis provides engineers with information regarding how the design may respond to different types of dynamics loading and can be used for example to avoid resonant vibrations that can be harmful to the structure. The vibration response of a linear time-invariant dynamic system can be described as a linear combination of a set of simple harmonic motions called natural modes of vibration, according to modal analysis. Mode shapes are the special initial displacements of a system that cause it to vibrate harmonically.

2.1.1. Modal Analysis Theory

The n-th order linear differential equation with constant coefficients can be written as,

$$\boldsymbol{M\ddot{X}} + \boldsymbol{C\dot{X}} + \boldsymbol{KX} = \boldsymbol{F}$$

M stands for mass, C for damping, K for stiffness, X for displacement response vector, and F for exciting force vector.

The differential equation of an undamped free vibration system is used to simplify the problem, is

$$\boldsymbol{M\ddot{X}} + \boldsymbol{KX} = \mathbf{0}$$

To determine the equation of motion for normal modes and natural frequencies, a particular reduced form of the equation of motion is required. In terms of matrix arrangement, when both damping and applied load are absent, the equation of motion becomes,

$$[M]\{\ddot{u}\}+[K]\{u\}=0 \qquad (2\text{-}1)$$

where:

[M] = mass matrix

[K] = stiffness matrix

This equation is for undamped free vibration. Considering a harmonic solution of the type to solve Equation 2-1,

$$\{u\}=\{\Phi\}\sin\omega t \qquad (2\text{-}2)$$

where:

$\{\Phi\}$ stands for the mode shape or also known as eigen value while ω stands for the circular natural frequency

This harmonic has a physical significance and is also fundamental to the numerical solution. The vibrating structure's degrees of freedom move in lockstep, as indicated by the harmonic form of the solution. During motion, the structural structure does not change its basic shape; only the amplitude varies (Aerospace Engineering, 2014).

When the assumed harmonic solution is differentiated and replaced into the equation of motion, the result is as follows:

$$-\omega^2[M]\{\Phi\}\sin\omega t+[K]\{\Phi\}\sin\omega t \qquad (2\text{-}3)$$

which, after being simplified, is

$$([K] - \omega^2 [M])\ \{\Phi\} = 0 \qquad (2\text{-}4)$$

This eigenequation, which serves as the foundation of the eigen value problem, is an array of standardized algebraic equations for the eigenvector's elements. In linear matrix algebra, an eigenvalue problem is a special equation with various applications. An eigenvalue problem in its most basic form is

$$[A-\lambda I]x = 0 \qquad (2\text{-}5)$$

Where A stands for the square matrix, eigenvalues are the eigenvectors, I is the identity matrix, and x is the eigen vector. The physical interpretations of natural frequencies and mode shapes emerge from the representation of stiffness and mass in the eigenequation in structural analysis. As a result, the equation 2-4 shows how to formulate the eigen equation using K, ω, and M with $\omega^2 = \lambda$.

For equation 2-4, there are two alternative solutions:

1. If determinant of $([K]-\omega^2 [M])$ is not equal to zero,

$$\{\Phi\} = 0 \tag{2-6}$$

This is the simplest solution, which provides no useful information from a physical standpoint because it symbolizes the absence of motion. If det $\left([K]-\omega^2[M]\right)=0$, then a non-trivial solution ($\{\phi\} \neq 0$) is obtained for

$$([K]-\omega^2 [M]) \{\Phi\} = 0 \tag{2-7}$$

From the standpoint of structural engineering, the fundamental mathematical eigenvalue problem is reduced to solving the equation of the following form

$$\text{Determinant of } ([K] -\omega^2 [M]) = 0 \tag{2-8}$$

or

$$\text{Determinant of } ([K]-\lambda[M]) = 0 \tag{2-9}$$

where $\lambda = \omega^2$

Only a subset of discrete eigenvalues has a determinant of zero λ_i or ω_i^2**.** Eigenvector $\{\varphi_i\}$ satisfies Eq. 2-7 and agrees with the respective eigenvalue. Consequently, equation 2-7 is written as

$$([K]-\omega_i^2[M]) \{\Phi_i\} =0 \qquad i = 1,2,3,4... \tag{2-10}$$

The structure's free vibration mode is defined by each eigenvalue and eigenvector. The i-th eigenvalue λ_i has the following relationship with the i-th natural frequency.

$$f_i = \frac{\omega_i}{2\pi} \tag{2-11}$$

Where, f_i stands for i-th natural frequency and ω_i is $\sqrt{\lambda_i}$. The total of eigen values with eigen vectors is the same as the total of degrees of freedom with mass or dynamic degrees of freedom.

Natural frequencies and mode shapes have several properties which are relevant in dynamic investigations. Firstly, whenever a linear elastic structure vibrates freely or forcefully, its deflected shape is a linear composite of the entire normal modes at any given instant.

$$\{u\} = \sum (\Phi_i)\xi_i$$

$$\{u\} = \textstyle\sum_i (\Phi_i)\xi_i \tag{2-12}$$

Where {u} stands for physical displacement vector, { ϕ_i }stands for i-th mode shape, and ξ_i stands for i-th modal displacement. Secondly, if [K] and [M] are symmetric and real (as all popular structural finite elements are), the following mathematical properties apply:

$$\{\varphi_i\}^T[M]\{\varphi_j\} = 0 \quad \text{if i } \neq \text{j} \tag{2-13}$$

$\{\varphi_j\}^T[M]\{\varphi_j\} = m_j = j - th\ simplified\ mass$ Equation 2-14, with

$$\{\varphi_i\}^T[K]\{\varphi_j\} = 0 \quad \text{if i } \neq \text{j} \tag{2-15}$$

$$\{\varphi_j\}^T[K]\{\varphi_j\} = k_j = j - th\ simpified\ stiffness = \omega^2 m_j \tag{2-16}$$

Also, from Eq. 2-14 and 2-16 Rayleigh's equation is attained

$$\omega_j^2 = \frac{\{\varphi_j\}^T[K]\{\varphi_j\}}{\{\varphi_j\}^T[M]\{\varphi_j\}} \tag{2-17}$$

The orthogonality property of normal modes is defined by equations. 2-13 and 2-15, which assures that the individual normal mode is different to the rest. Theoretically, orthogonality of modes indicates different mode shape is distinct, plus, same mode shape is not achievable by combining other modes in a linear fashion.

Furthermore, assuming generalized stiffness as well as generalized mass, a fundamental mode of the structure can indeed be depicted. This comes in handy when creating analogous dynamic models and synthesis in component mode.

A structure can either shift like a rigid body or a full or partial way if it is not totally confined in space. There is one natural frequency equal to zero for unique potential element of rigid-body system or deformation. Rigid-body modes are zero-frequency modes. The motion of a structure in a stress-free state is represented by rigid-body motion of entire or portion of the structure. Stress-free rigid-body modes are important for dynamic assessments of unrestrained constructions like satellites, planes. Rigid-body modes may also indicate modeling flaws or an insufficient constraint set. For big buckling and normal mode issues, iteration-based Lanzcos approach is chosen over reduction method to solve the equation (Engineering, 2013).

CHAPTER THREE: METHODOLOGY OF RESEARCH

This research is a case study, which aims to study vibrational behavior of spindle on different stiffeners condition. Analyzing of effects on mode shapes, mode frequency of spindle by variation in shape and position for the stiffeners were considered. Modelling was done then simulations were run to reach to the conclusion of this study.

The following methods have been carried out to meet the objectives of the research:

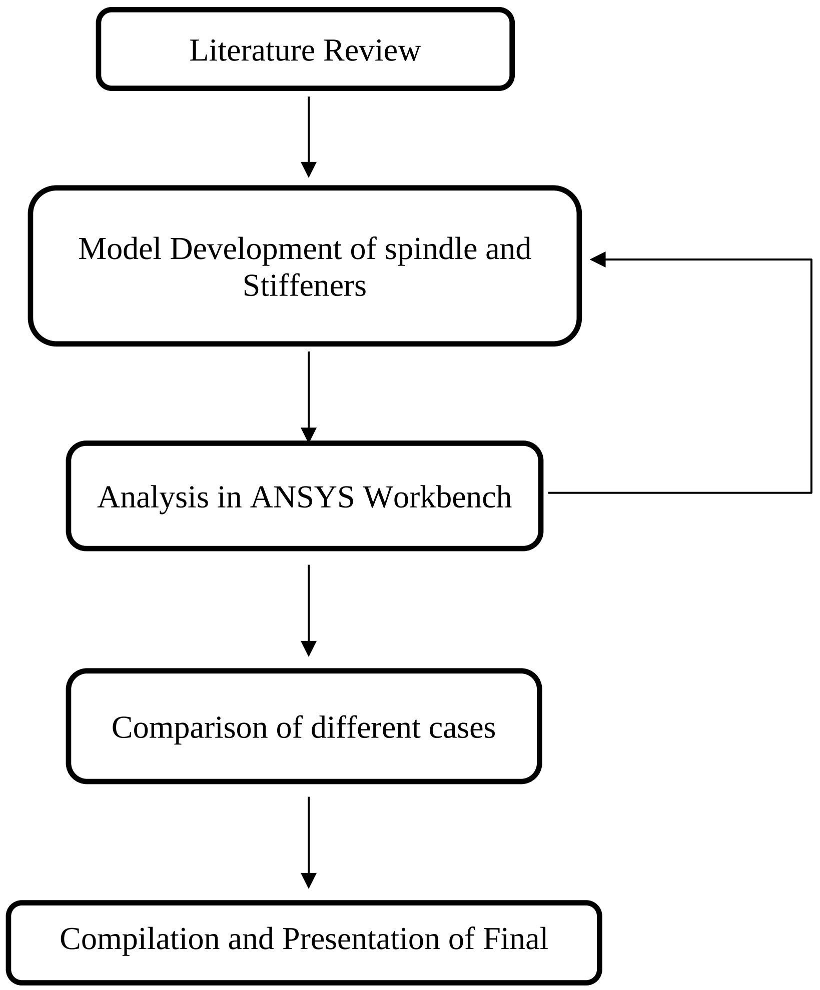

Figure 3.1: Research method used in this study

The above steps have been explained in detail below:

3.1 Literature Review

Reviewing the articles and journal papers provide insights to the degree up to which any works or researches or studies are put forth by the scholars from the past to the present. Considering the importance of material selection for reducing the peak vibration and resultant vibration load due to each of the different available construction materials, various research papers regarding strength, reliability and weight have been studied. Also, the accuracy and acceptance of used methodology for calculating the peak vibration load, researches regarding the accuracy of various calculation method had been gone through.

3.2 Model Development of Spindle

Spindle model was made in Autodesk Inventor 2015 professional. Reference for spindle was Machifit ER11 Air-cooled spindle. Stiffeners were also made. First structural steel was used as material. Then process 3.3 was carried out.

3.3 Analysis in ANSYS Workbench 2016

For modal analysis in ANSYS, initially the modal was brought to modal template. Then the boundary condition was set to fixed to see the free vibration. Ten modes of frequency were set. Then model was solved and frequencies were analyzed.

3.4 Iteration of Process 3.3

Spindle vibration depends upon the properties of stiffener. So, in order to know the variation in vibration, various shapes were selected for stiffener. Common material for spindle used is structural steel while Galvanized Steel was used for stiffener. Then process 3 was iterated with varying features in shape and position of stiffeners.

3.5 Comparisons of Dynamic Property

Mode shape and deformation results were observed.

3.6 Compilation, Discussion and Presentation of Final Report

This was the last phase, which includes compilation of all information, comparison and presenting them in a formal report. From the results obtained, discussions and recommendations were made for reasoning then filed in the report.

3.7 Autodesk Inventor Professional 2015

Autodesk Inventor is a computer-aided design software program for 3D mechanical design, simulation, visualization, and documentation. In this thesis, Autodesk Inventor Professional 2015 was utilized for modeling..

3.8 Geometric Modelling

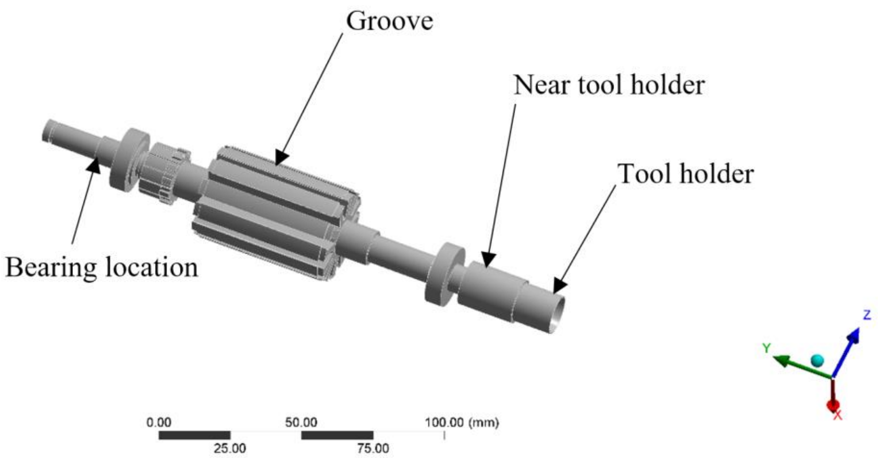

Figure 3.2: Spindle with Label

Inventor model was created for the spindle and stiffeners was added later. Total of eight cases had been presented for comparison. They were named Case I to Case VIII as defined on figure 3.3 to 3.10 and henceforth be mentioned in case numbers. Figure 3.2 shows the location of bearing, groove and tool holder.

3.9 ANSYS Model

The modal analysis was performed using the ANSYS Workbench 16.0 Finite Element software. When it comes to optimizing the shape of the spindle, modal analysis comes in handy. The advantages of utilizing ANSYS included the ability to correctly visualize and model mode shapes. As a result, the deformations in the spindle could be precisely located. ANSYS Workbench also provides a graphical representation of the number of modes vs. frequency. Block Lanczos technique, Power Dynamics approach, Subspace procedure, Reduction technique, Asymmetric approach, and Damping procedure are some of the modal methods available in ANSYS. This research uses the Block Lanczos technique. This is one among the most poular efficient approaches for handling big matrix eigenvalue problems, and is characterized by a recursive vector matrix vector matrix formula. Lanczos approach is an orthogonal matrix multiplication method that can be used to obtain a hypothesis for the modal matrix of a discrete model of an outstanding structure. The low-order modal space is created, as a result of which, it can successfully simulate the structure's discrete model.

The vibration of the structure is expressed as a linear superposition of lower order modes having a smaller impact on the structure than higher order modes. It analyzes the dynamic behavior of structure, that normally require five to ten orders, and the dynamic features of low order modes of structure play a critical function. The first ten order natural frequency and deformation type of the spindle were computed in this research.

Table 3.1: Parameters used for the spindle

Material	
Assignment	Structural Steel
Bounding Box	
Length X	3.3851 e-002m
Length Y	0.19545m
Length Z	3.3851 e-002m
Properties	
Volume	4.8293 e-005 m^3
Mass	0.3791 kg

3.10 Defining cases for study

Total of eight cases were presented for study. Fixed support was applied at bearing location. Stiffeners were applied on different location (on groove and near tool holder) to see effect on mode shape results and deformation. Eight cases were defined below in brief. We referred to the Figure 3.2 for placement of the stiffeners.

3.10.1 Case I: Model of Spindle

This was the model of spindle. The spindle had bearing support where fixed support was applied. The spindle had groove to hold motor. Furthest away from the support lied the tool holder. No any stiffeners were applied. Analysis run on this model was referred to as Case I.

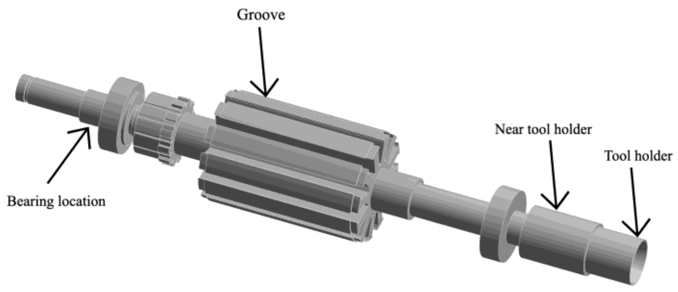

Figure 3.3: Case I: Model of Spindle

3.10.2 Case II: Model of spindle with rectangular shaped stiffeners near tool holder

This was the model of spindle where rectangular shaped stiffeners were applied near tool holder. The spindle had bearing support where fixed support was applied. No other stiffeners were applied elsewhere. This model was referred to as case II.

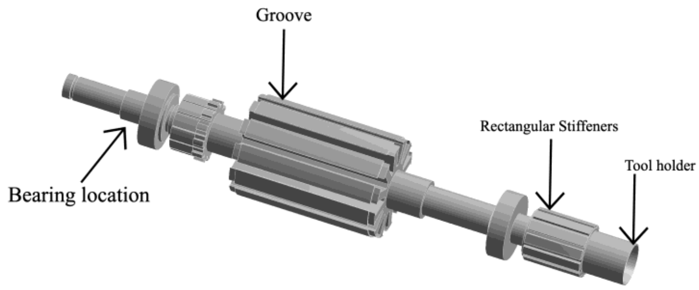

Figure 3.4: Case II: Model of spindle with rectangular shaped stiffeners near tool holder

3.10.3 Case III: Model of spindle with circular shaped stiffeners near tool holder

This was the model of spindle where circular shaped stiffeners were applied near tool holder. The spindle had bearing support where fixed support was applied. No other stiffeners were applied elsewhere. This model was referred to as case III.

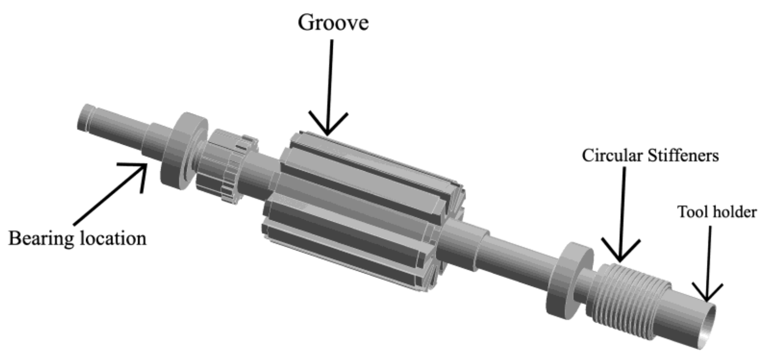

Figure 3.5: Case III: Model of spindle with circular shaped stiffeners near tool holder

3.10.4 Case IV: Model of spindle with stiffeners on groove

This was the model of spindle where stiffeners were applied on groove. The spindle had bearing support where fixed support was applied. No other stiffeners were applied elsewhere. This model was referred to as case IV.

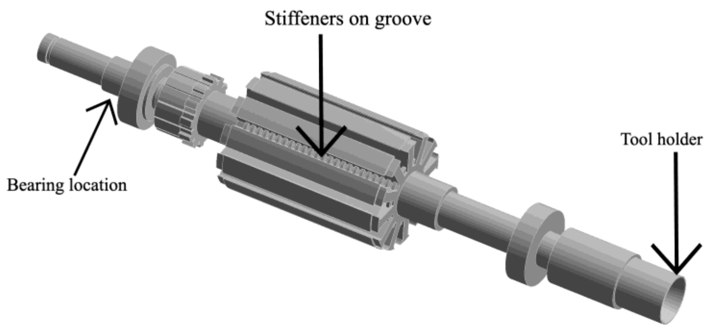

Figure 3.6: Case IV: Model of spindle with stiffeners on groove

3.10.5 Case V: Model of spindle with rectangular and circular shaped stiffeners near tool holder

This was the model of spindle where rectangular and circular shaped stiffeners were applied on groove. The spindle had bearing support where fixed support was applied. No other stiffeners were applied elsewhere. This was the combination of the stiffeners described in case I and case II. This model was referred to as case V.

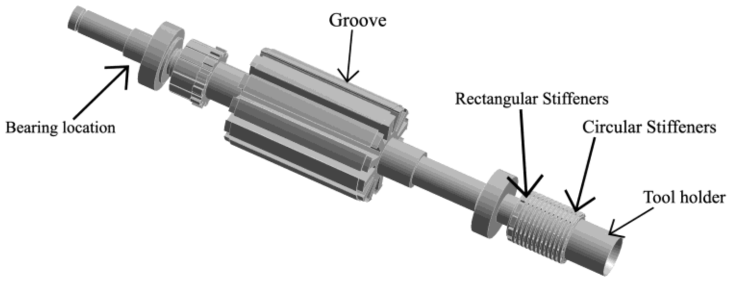

Figure 3.7: Case V: Model of spindle with rectangular and circular shaped stiffeners near tool holder

3.10.6 Case VI: Model of spindle with stiffeners on groove and rectangular shaped stiffeners near tool holder

This was the model of spindle where stiffeners were applied on groove and rectangular shaped stiffeners were applied near tool holder. The spindle had bearing support where fixed support was applied. No other stiffeners were applied. This was the combination of stiffeners referred in case II and case IV. This model was referred to as case VI.

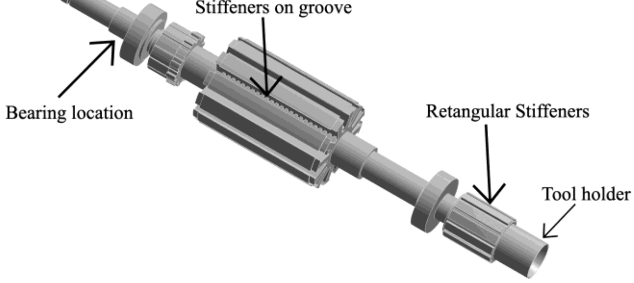

Figure 3.8: Case VI: Model of spindle with stiffeners on groove and rectangular shaped stiffeners near tool holder

3.10.7 Case VII: Model of spindle with stiffeners on groove and circular shaped stiffeners near tool holder

This was the model of spindle where stiffeners were applied on groove and circular shaped stiffeners were applied near tool holder. The spindle had bearing support where fixed support was applied. No other stiffeners were applied. This was the combination of stiffeners referred in case III and case IV. This model was referred to as case VII.

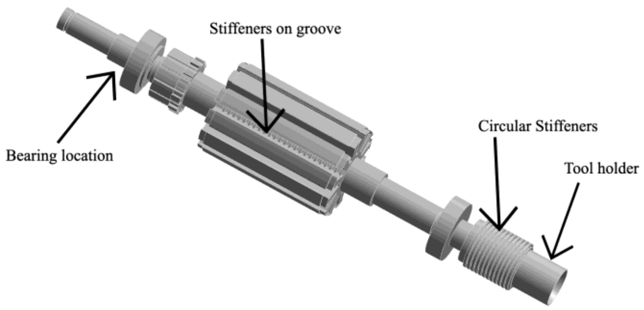

Figure 3.9: Case VII: Model of spindle with stiffeners on groove and circular shaped stiffeners near tool holder

3.10.8 Case VIII: Model of spindle with stiffeners on groove with rectangular and circular stiffeners near tool holder

This was the model of spindle where stiffeners were applied on groove and both rectangular and circular shaped stiffeners were applied near tool holder. This was the combination of stiffeners referred in case II, case III and case IV. This model was referred to as case VIII.

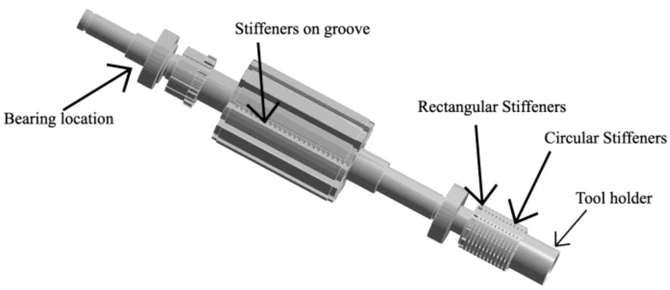

Figure 3.10: Case VIII: Model of spindle with stiffeners on groove with rectangular and circular shaped stiffeners near tool holder

3.10.9 Model of Stiffeners

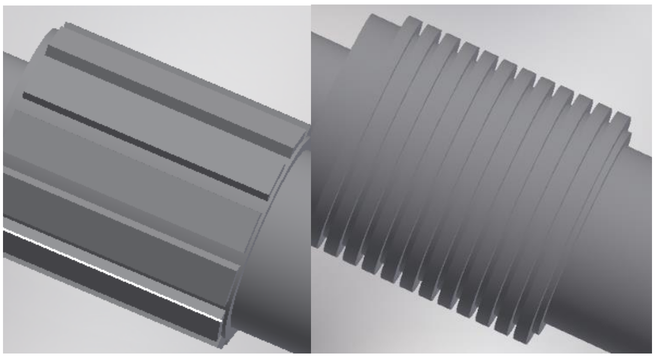

Figure 3.11: Model of rectangular and circular stiffeners

Rectangular and circular shaped stiffeners were modelled. Material used was galvanized steel for stiffeners. 1 mm thickness for stiffeners were preferred.

Dimension of rectangular stiffeners was maintained at 1 mm x 19mm. Twelve rectangular shaped stiffeners were used in each of the Case II, Case V, Case VI and Case VIII.

Dimension of circular stiffeners were maintained at 1 mm thickness with 18 mm diameter. Ten circular stiffeners were used in each of the Case III, Case V, Case VII and Case VIII

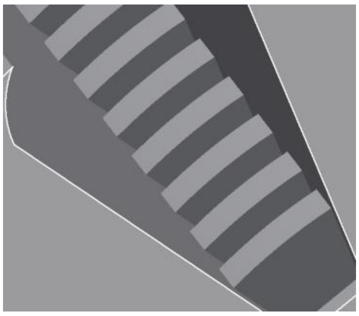

Figure 3.12: Model of stiffeners on groove

Stiffeners on groove were maintained at 10mm x 1 mm. 24 used in each of the Case IV, Case V, Case VI and Case VIII

CHAPTER FOUR: RESULTS AND DISCUSSION

4.1 Modal analysis of the spindle

Modal analysis of the spindle was carried out. Maximum deformation was observed near tool holder for the spindle model in first mode. The spindle initially had maximum modal displacement of 4.04 on the tool holder as shown in figure 4.1(a). The first natural frequency observed was 290.28 Hz.

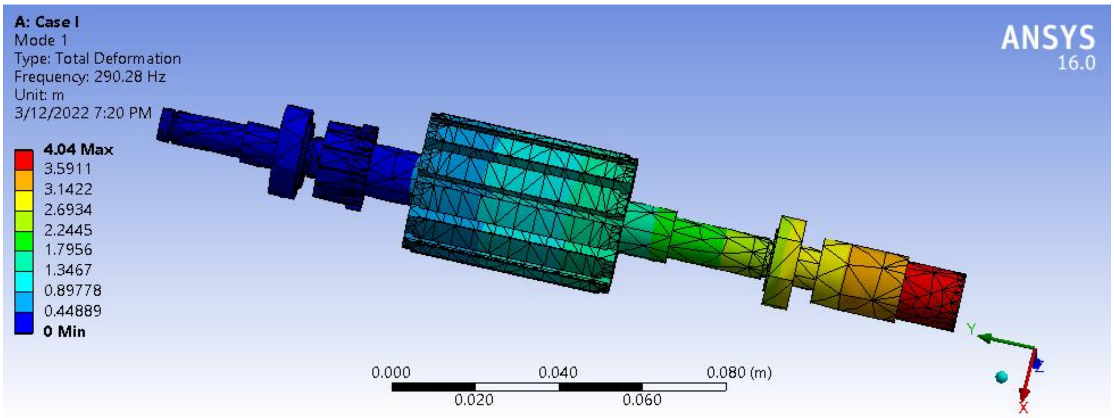

Figure 4.1(a) Case I: First Mode deformation

Table 4.1. First Ten modes with their frequency on spindle only model

Case I	
Mode	Frequency [Hz]
1	290.28
2	290.94
3	1402.6
4	1404.8
5	2016.7
6	4639.1
7	4644
8	4911.1
9	6299.4
10	7622.4

First ten natural frequencies were obtained as shown in Table 4.1. This result was compared with other cases. The input data for material properties were kept same adding only stiffeners. The spindle only model experiences the maximum deformation. The results obtained were used to compare with other cases for further study.

4.2 Effect of stiffeners on amplitude of modal displacement

Modal analysis of spindle with different stiffeners setup cases discussed in 3.10 was carried out. Comparing frequency and deformation for case I to case VIII few observations were made. The first mode frequencies for all eight cases were listed in table 4.2. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. Hence, variation in stiffener was the reason for difference in modal amplitude.

For Mode 1, the spindle initially had maximum modal displacement of 4.04 on the tool holder as shown in figure 4.1(a). The modal displacement was decreased after the application of stiffeners in all rest of the seven cases.

Table 4.2. Frequency, deformation and modal amplitude difference on first mode

Mode 1			
Case	Frequency (Hz)	Deformation (m)	Modal Amplitude difference (m)
I	290.28	4.04	0
II	281.62	3.9451	0.0949
III	281.3	3.9362	0.1038
IV	288.36	4.0202	0.0198
V	277.91	3.9003	0.1397
VI	280.47	3.929	0.111
VII	280.46	3.9202	0.1198
VIII	277.64	3.8867	0.1533

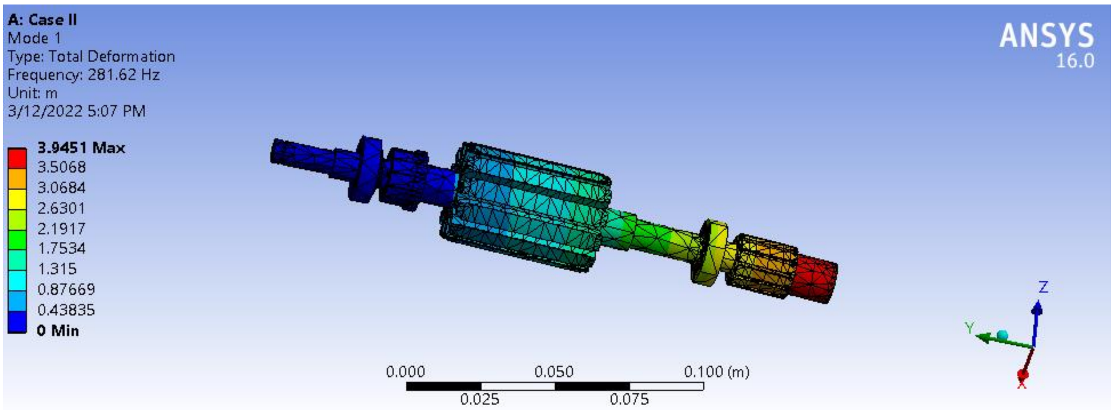

Figure 4.1(b) Case II: First Mode deformation

First mode for case II was at frequency of 281.62 Hz. Deformation was decreased to 3.9451 as compared to 4.04 of case I. Modal amplitude difference was 0.0949.

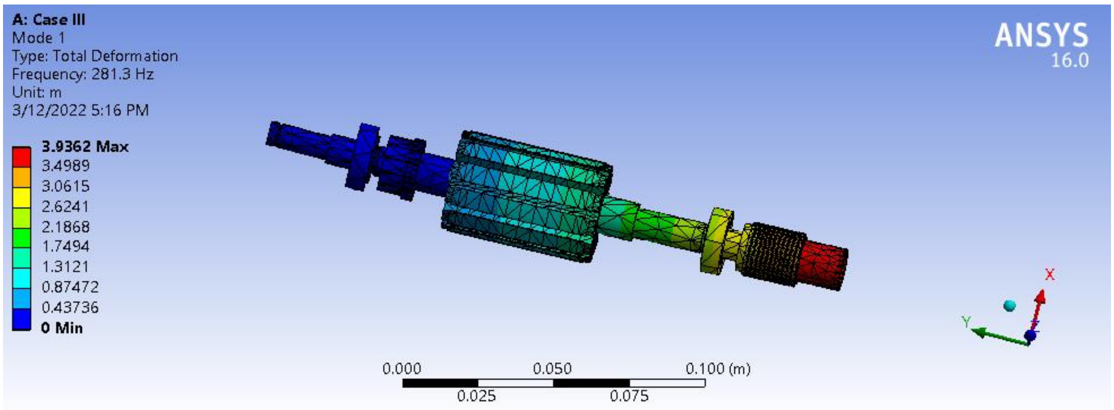

Figure 4.1(c) Case III: First Mode deformation

First mode for case III was at frequency of 281.3 Hz. Deformation was decreased to 3.9362m as compared to 4.04m of case I. Modal amplitude difference was 0.1038m.

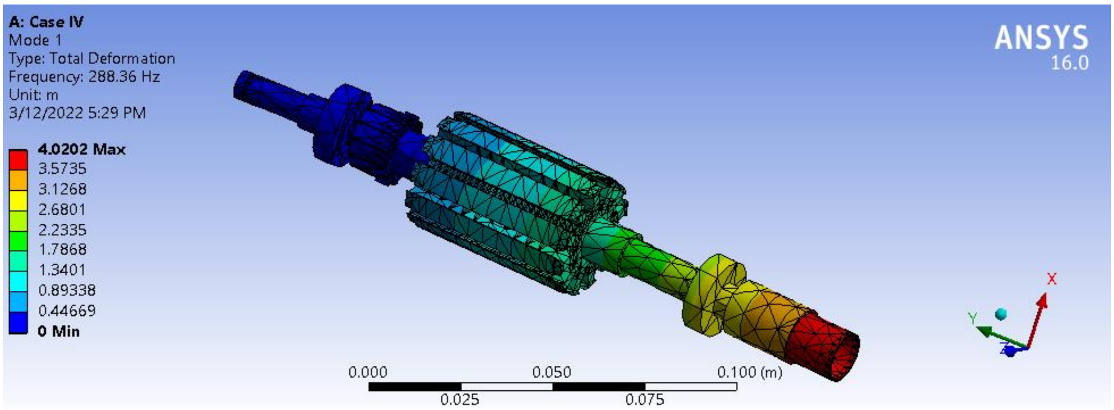

Figure 4.1(d) Case IV: First Mode deformation

First mode for case IV was at frequency of 288.36 Hz. Deformation was decreased to 4.0202m as compared to 4.04m of case I. Modal amplitude difference was 0.0198m.

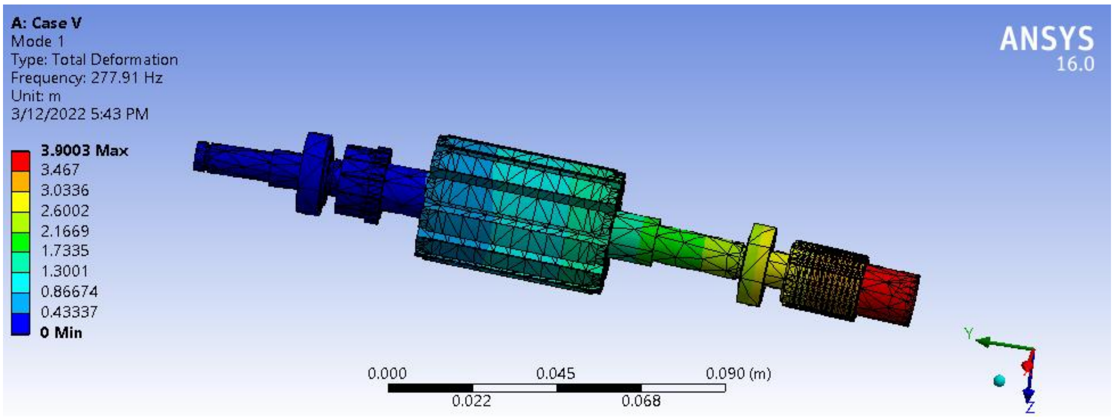

Figure 4.1(e) Case V: First Mode deformation

First mode for case V was at frequency of 277.91 Hz. Deformation was decreased to 3.9003m as compared to 4.04m of case I. Modal amplitude difference was 0.1397m.

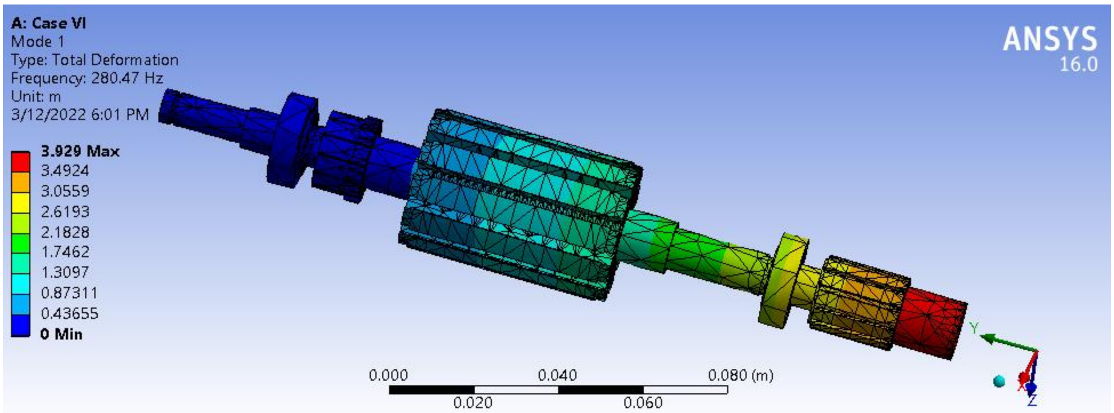

Figure 4.1(f) Case VI: First Mode deformation

First mode for case VI was at frequency of 280.47 Hz. Deformation was decreased to 3.929m as compared to 4.04m of case I. Modal amplitude difference was 0.111m.

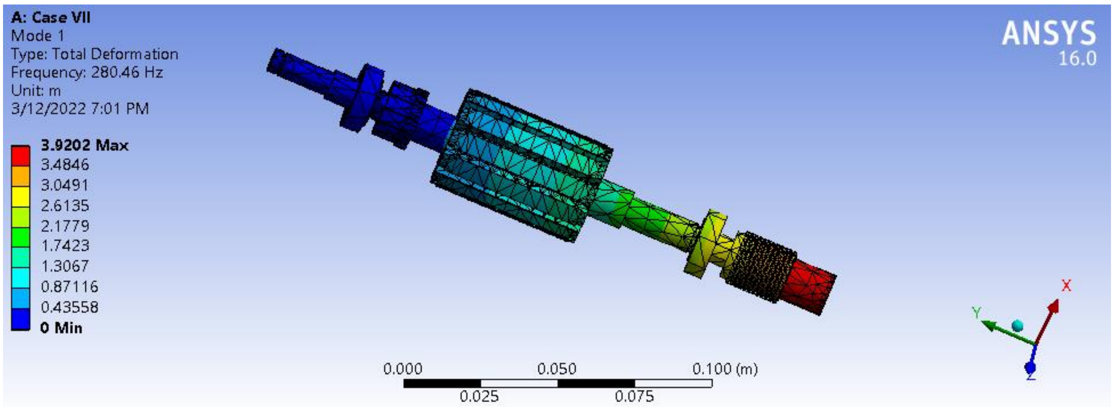

Figure 4.1(g) Case VII: First Mode deformation

First mode for case VII was at frequency of 280.46 Hz. Deformation was decreased to 3.9202m as compared to 4.04m of case I. Modal amplitude difference was 0.1198m.

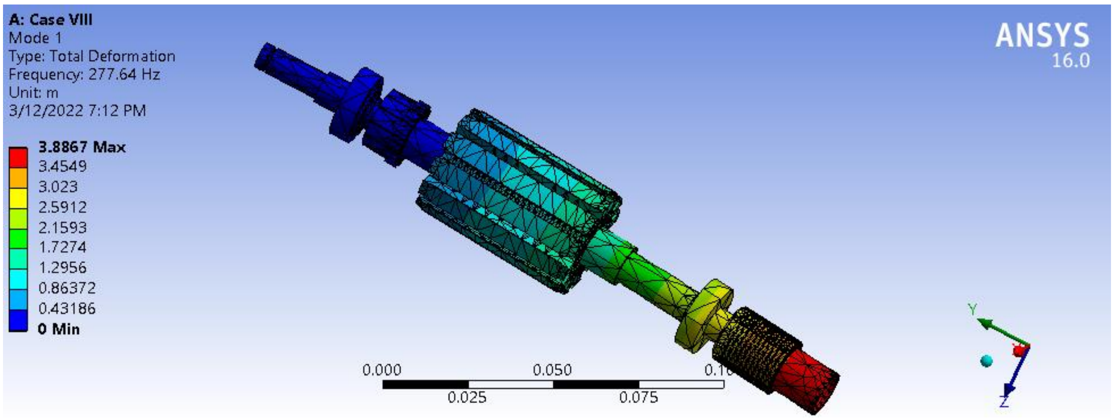

Figure 4.1(h) Case VIII: First Mode deformation

First mode for case VIII was at frequency of 277.64 Hz. Deformation was decreased to 3.9451m as compared to 4.04m of case I. Modal amplitude difference was 0.1533m.

4.3 Effect of stiffeners on variation of mode frequency of the spindle

The second mode frequencies for all eight cases were listed in table 4.3. Comparison of frequency and deformation for case I to case VIII was done to make few observations. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. Hence, variation in stiffener was the reason for difference in mode frequency of the spindle.

The shape of stiffeners did not show noteworthy effect. The frequency 282.11 Hz on case II and 281.86 Hz on case III were nearly identical as seen in Table 4.3. The deformation 3.946 on case II and 3.9373 on case III were nearly identical as well. This meant the frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same can be observed when these two shapes were combined with stiffeners on groove as seen in case VI and case VII.

Table 4.3 Frequency, deformation and modal amplitude difference on second mode

Mode 2			
Case	Frequency (Hz)	Deformation (m)	Modal Amplitude difference (m)
I	290.94	4.0411	0
II	282.11	3.946	0.0951
III	281.86	3.9373	0.1038
IV	288.75	4.0217	0.0194
V	279.11	3.9042	0.1369
VI	281.78	3.9323	0.1088
VII	281.52	3.9233	0.1178
VIII	278.22	3.8884	0.1527

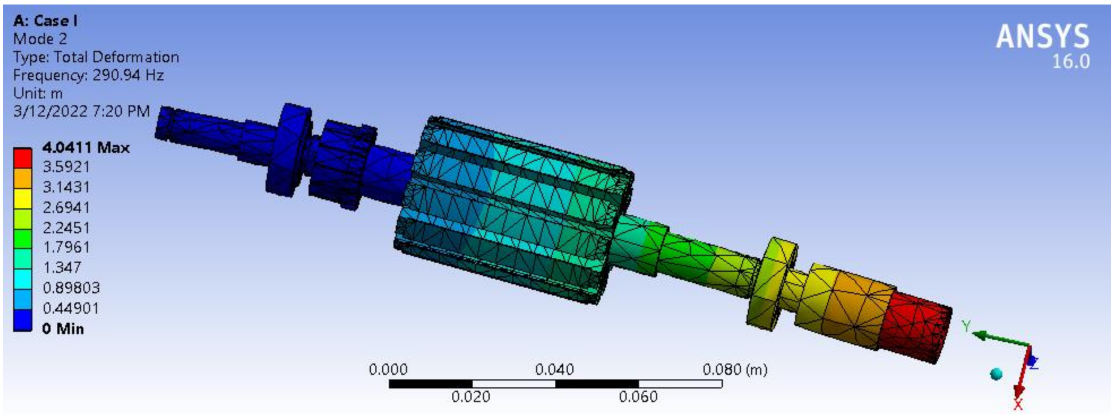

Figure 4.2(a) Case I: Second Mode deformation

Second mode for case I was at frequency of 290.94 Hz.

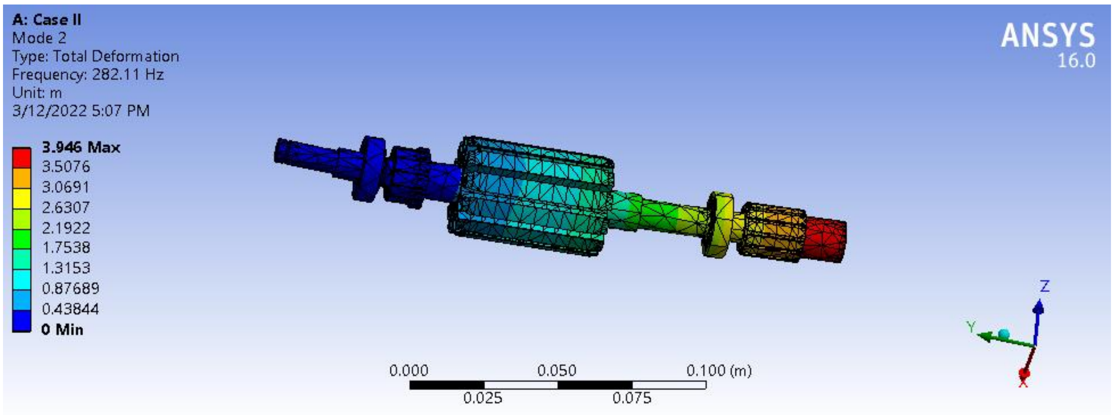

Figure 4.2(b) Case II: Second Mode deformation

Second mode for case II was at frequency of 282.11 Hz as compared to 290.94 Hz of case I.

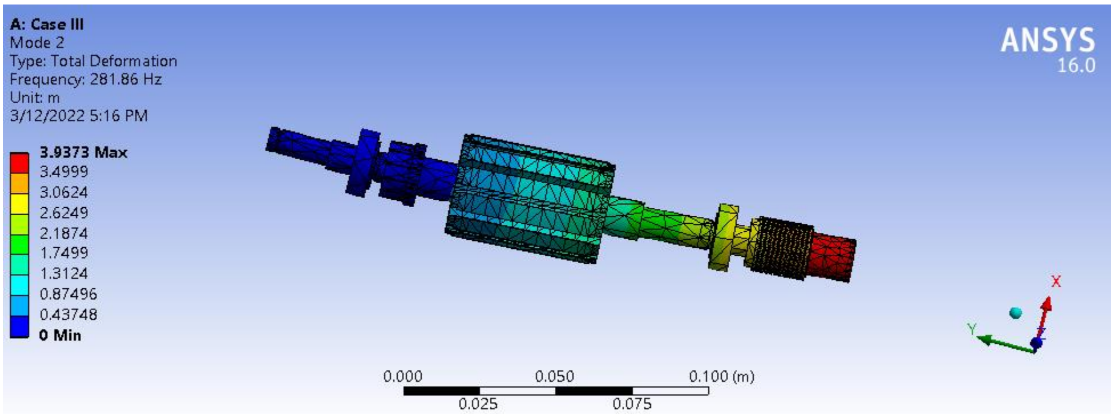

Figure 4.2(c) Case III: Second Mode deformation

Second mode for case III was at frequency of 281.86 Hz as compared to 290.94 Hz of case I. This was nearly identical to case II where frequency was 282.11 Hz.

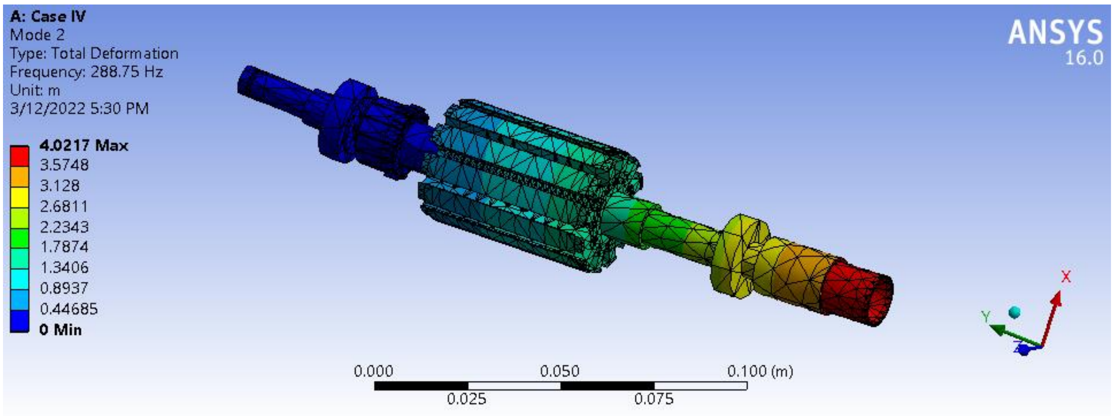

Figure 4.2(d) Case IV: Second Mode deformation

Second mode for case IV was at frequency of 288.75 Hz as compared to 290.94 Hz of case I.

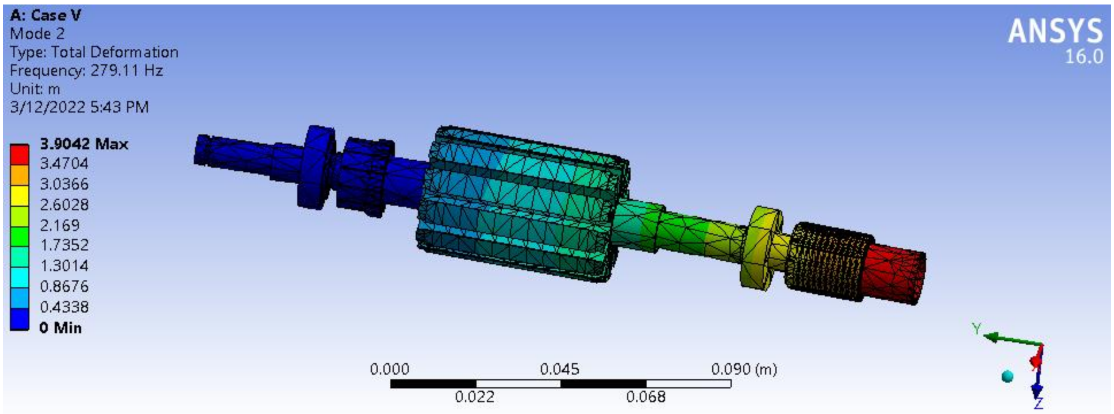

Figure 4.2(e) Case V: Second Mode deformation

Second mode for case V was at frequency of 279.11 Hz as compared to 290.94 Hz of case I.

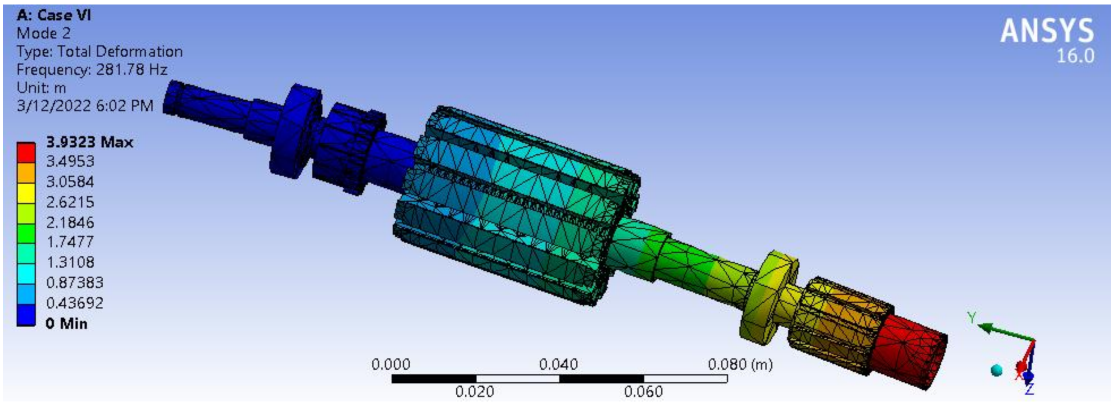

Figure 4.2(f) Case VI: Second Mode deformation

Second mode for case VI was at frequency of 281.78 Hz as compared to 290.94 Hz of case I.

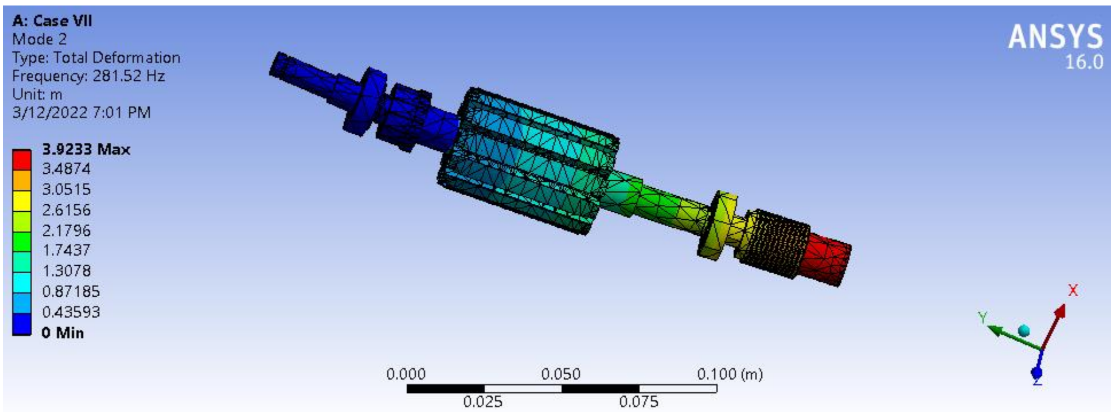

Figure 4.2(g) Case VII: Second Mode deformation

Second mode for case VII was at frequency of 281.52 Hz as compared to 290.94 Hz of case I. This was nearly identical to case VI where frequency was 281.78 Hz.

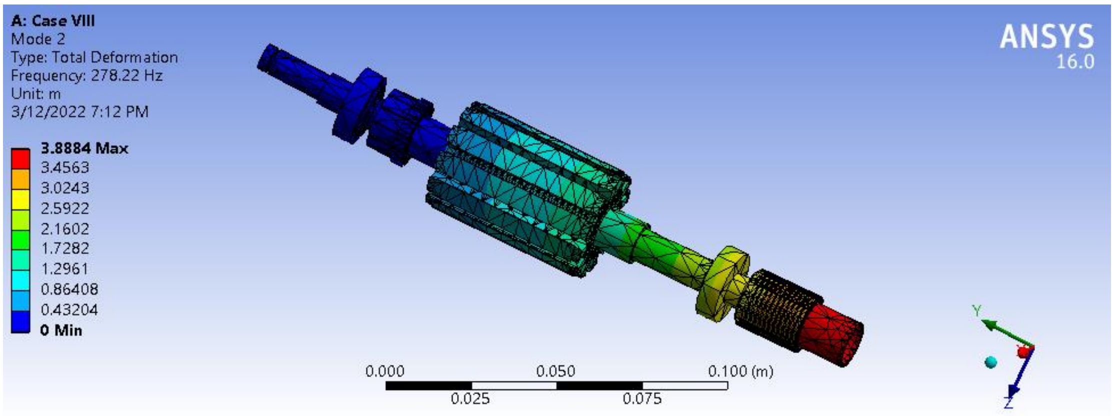

Figure 4.2(h) Case VIII: Second Mode deformation

Second mode for case VIII was at frequency of 278.22 Hz as compared to 290.94 Hz of case I.

We can observe from figure 4.2(b) and figure 4.2(c) that the frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same can be observed when these two shapes were combined with stiffeners on groove from figure 4.2(f) and figure 4.2(g).

4.4 Effect of stiffeners on variation of mode shape and deformation behavior of spindle by changing shape of stiffeners.

Few observations can be made comparing frequency and deformation for case I to case VIII. The third mode frequencies for all eight cases were listed in table 4.3. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. Hence, variation in stiffener was the reason for difference in mode frequency. The deformation shape was found similar in eight cases from figure 4.3(a) to figure 4.3(h). It can be concluded that mode shape will be same irrespective of addition of stiffeners on the spindle.

Table 4.4 Frequency, deformation and modal amplitude difference on third mode

Mode 3			
Case	Frequency (Hz)	Deformation (m)	Modal Amplitude difference (m)
I	1402.6	6.0176	0
II	1373.9	5.7404	0.2772
III	1369.6	5.6904	0.3272
IV	1391.5	6.0083	0.0093
V	1362	5.6035	0.4141
VI	1365.1	5.736	0.2816
VII	1359.5	5.6866	0.331
VIII	1352.8	5.5991	0.4185

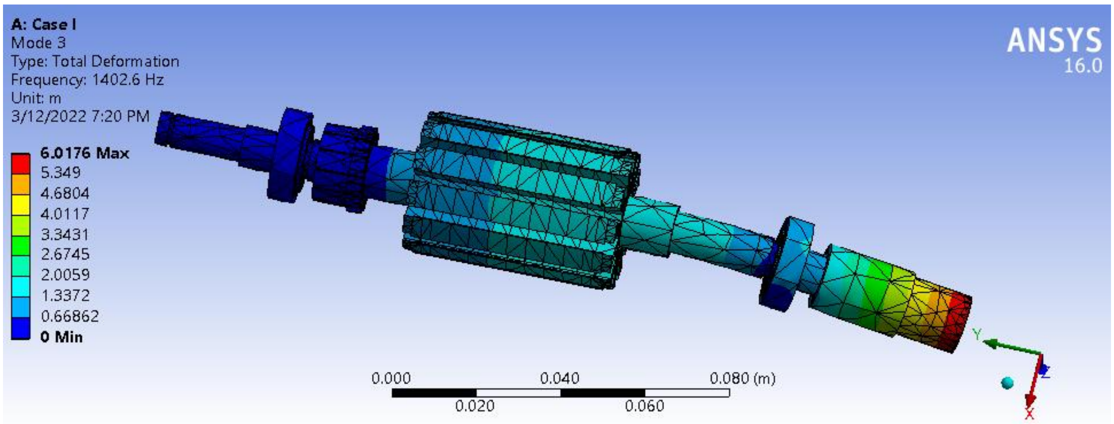

Figure 4.3(a) Case I: Third Mode deformation

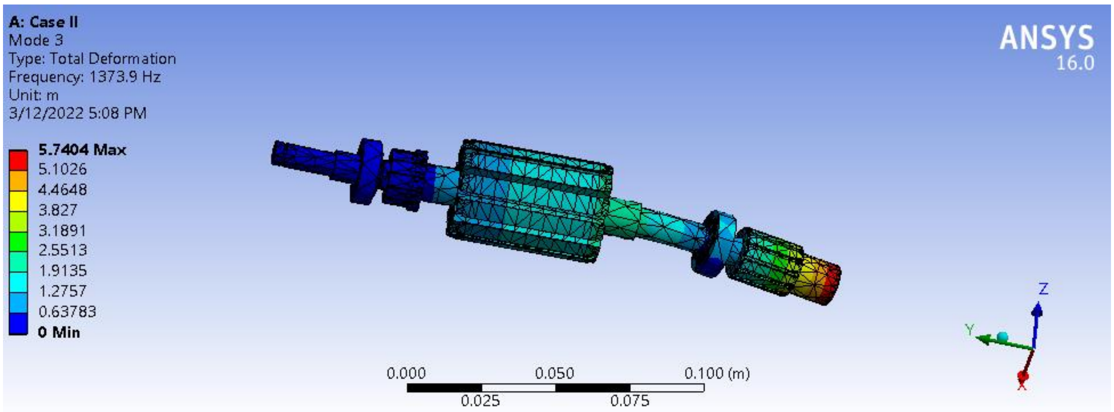

Figure 4.3(b) Case II: Third Mode deformation

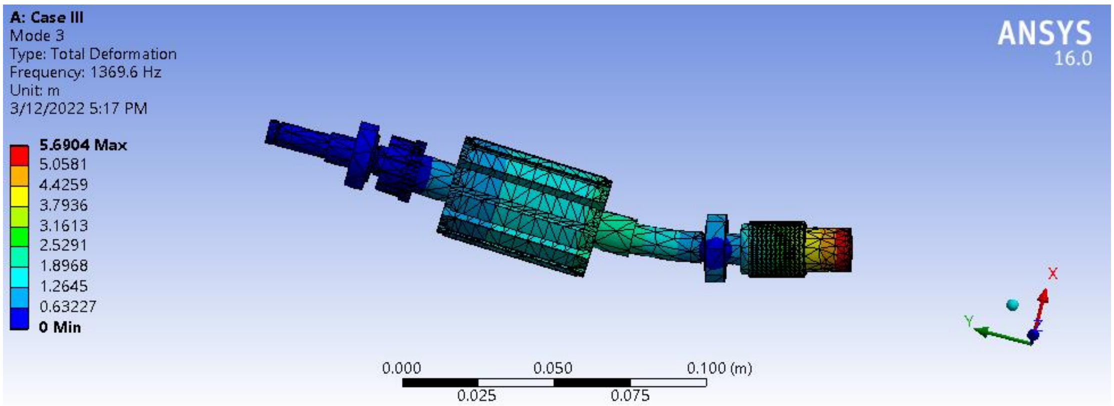

Figure 4.3(c) Case III: Third Mode deformation

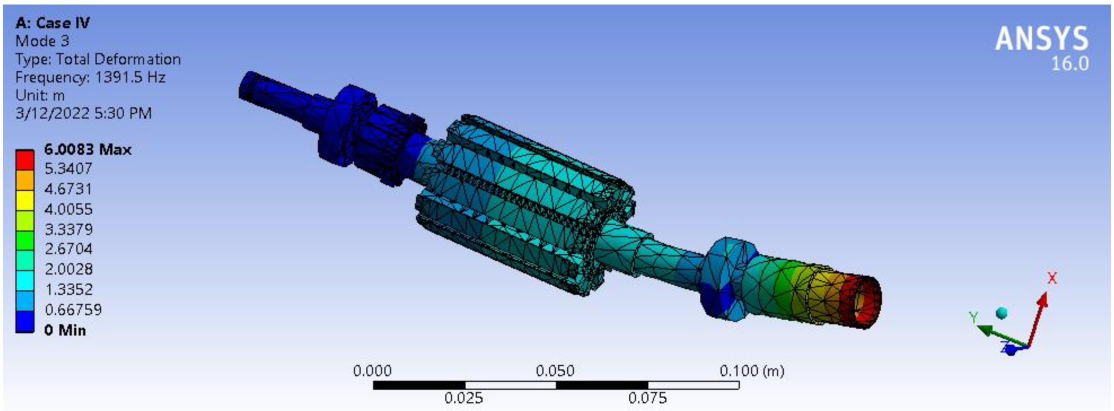

Figure 4.3(d) Case IV: Third Mode deformation

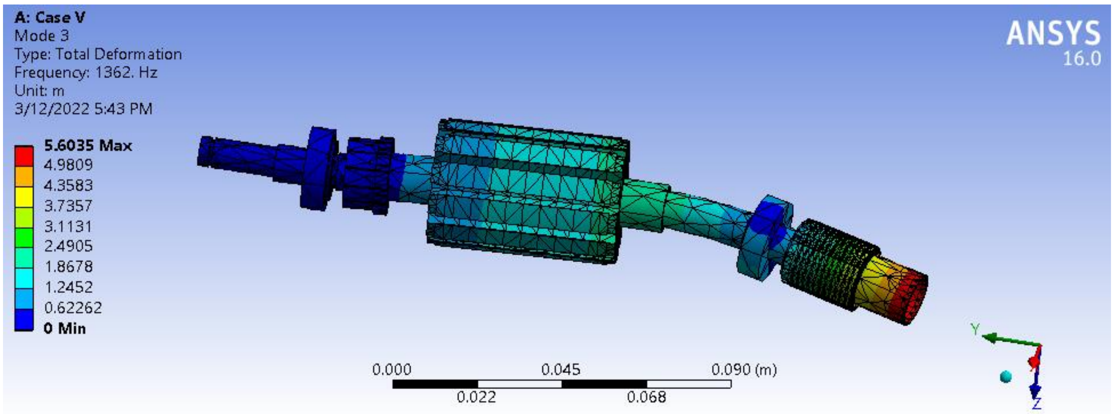

Figure 4.3(e) Case V: Third Mode deformation

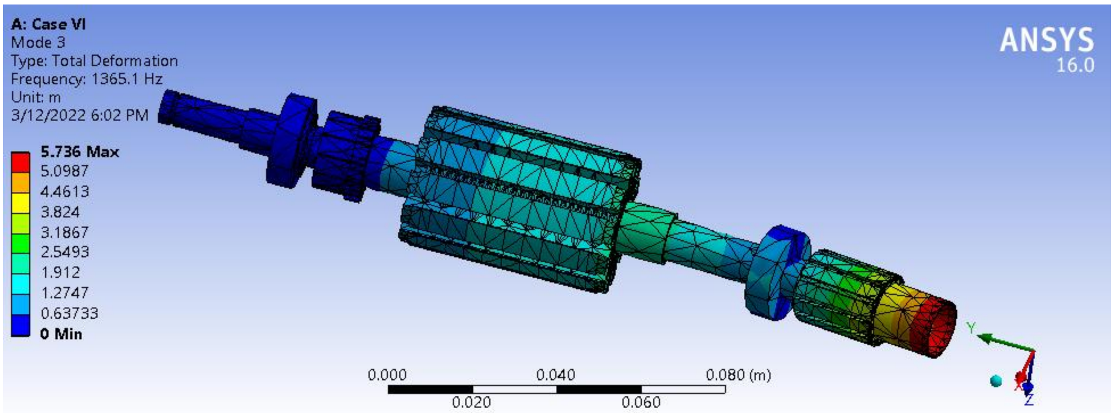

Figure 4.3(f) Case VI: Third Mode deformation

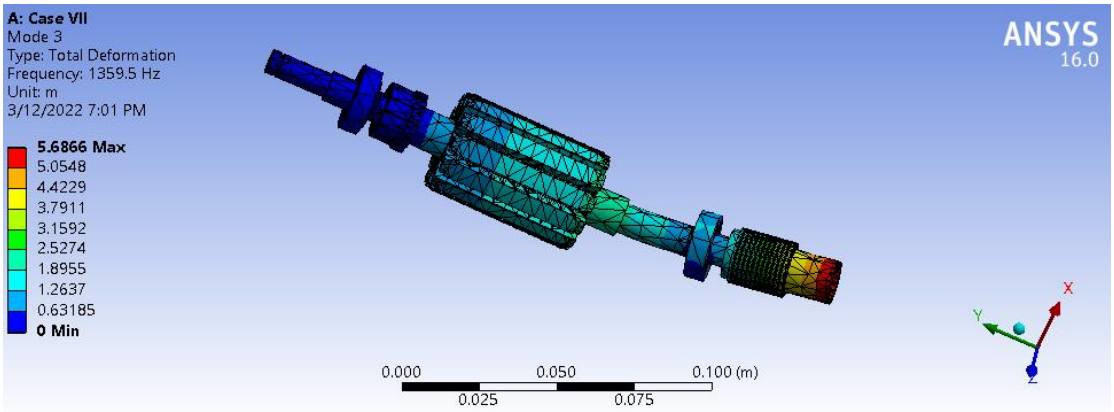

Figure 4.3(g) Case VII: Third Mode deformation

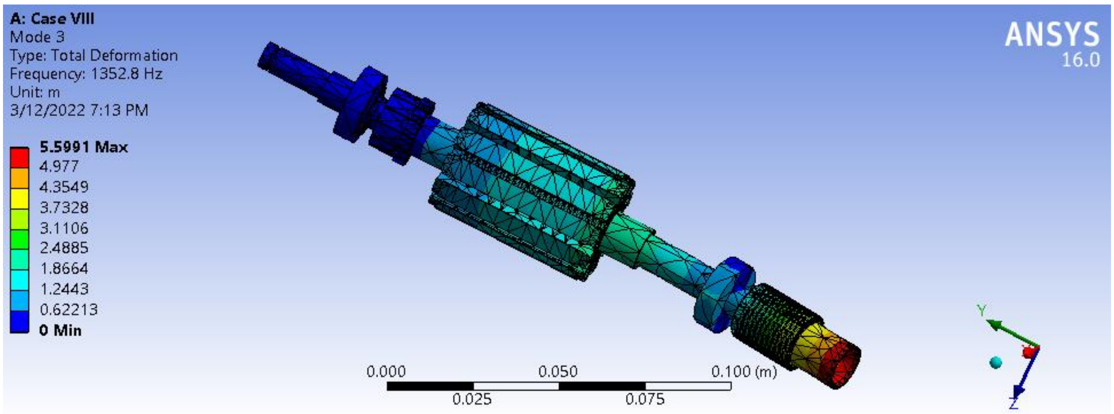

Figure 4.3(h) Case VIII: Third Mode deformation

The rectangular and circular shaped stiffeners put near tool holder in case II and case III respectively showed higher modal amplitude difference than the stiffeners put on groove which is closer to fixed support. Higher value of modal amplitude difference indicates lower deformation.

4.5 Effect on other modes

Mode 4

Tool holder seemed to undergo most deformation according to figure 4.4(a). Comparing frequency and deformation for case I to case VIII few observations were made. The third mode frequencies for all eight cases were listed in table 4.5. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. Hence, variation in stiffener is the reason for difference in mode frequency and modal amplitude difference. The deformation shape was found similar in eight cases. It can be concluded that mode shape will be same irrespective of addition of stiffeners on the spindle.

For fourth mode, the spindle initially had maximum modal displacement of 6.0159 on the tool holder as shown in figure 4.4(a). The modal displacement was decreased after the application of stiffeners in all rest of the seven cases.

Table 4.5 Frequency, deformation and modal amplitude difference on fourth mode

Mode 4			
Case	Frequency (Hz)	Deformation (m)	Modal Amplitude difference (m)
I	1404.8	6.0159	0
II	1375.9	5.7439	0.272
III	1370.6	5.6946	0.3213
IV	1392.5	6.0096	0.0063
V	1363.5	5.6062	0.4097
VI	1365.6	5.7361	0.2798
VII	1360.6	5.6861	0.3298
VIII	1353.3	5.6003	0.4156

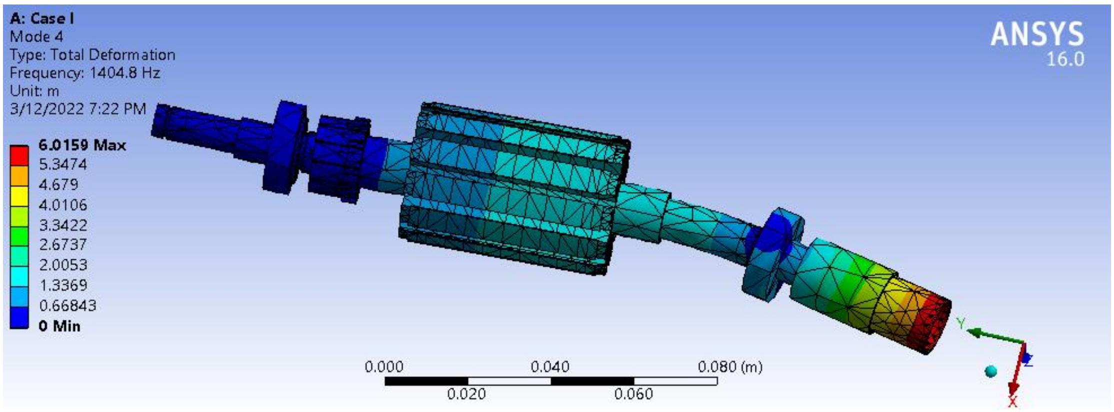

Figure 4.4(a) Case I: Fourth Mode deformation

Fourth mode for case I was at frequency of 1404.8 Hz. Deformation was 6.0159m at tool holder.

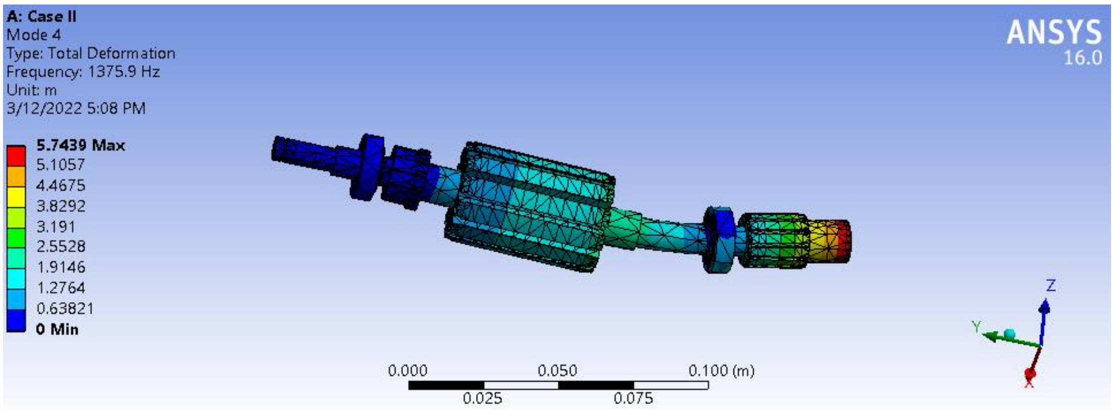

Figure 4.4(b) Case II: Fourth Mode deformation

Fourth mode for case II was at frequency of 1375.9 Hz as compared to 1404.8 Hz of case I. Deformation was decreased to 5.7439m as compared to 6.0159m of case I. Modal amplitude difference was 0.272m.

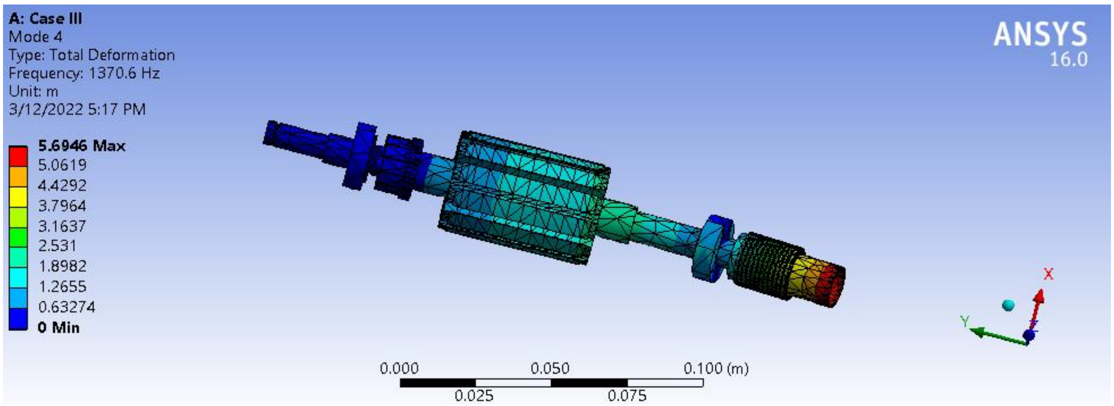

Figure 4.4(c) Case III: Fourth Mode deformation

Fourth mode for case III was at frequency of 1370.6 as compared to 1404.8 Hz of case I. Deformation was decreased to 5.6946m as compared to 6.0159m of case I. Modal amplitude difference was 0.3213m.

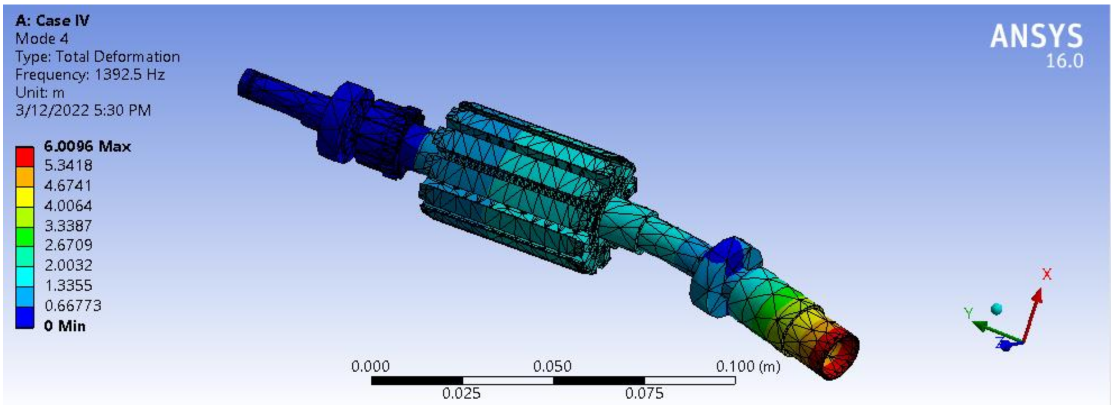

Figure 4.4(d) Case IV: Fourth Mode deformation

Fourth mode for case IV was at frequency of 1392.5 Hz as compared to 1404.8 Hz of case I. Deformation was decreased to 6.0096m as compared to 6.0159m of case I. Modal amplitude difference was 0.0063m.

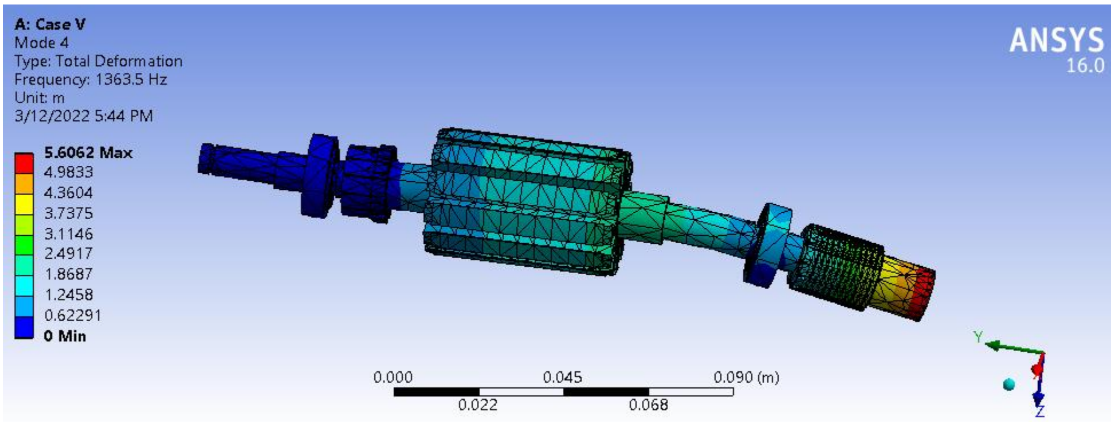

Figure 4.4(e) Case V: Fourth Mode deformation

Fourth mode for case V was at frequency of 1363.5 Hz as compared to 1404.8 Hz of case I. Deformation was decreased to 5.6062m as compared to 6.0159m of case I. Modal amplitude difference was 0.4097m.

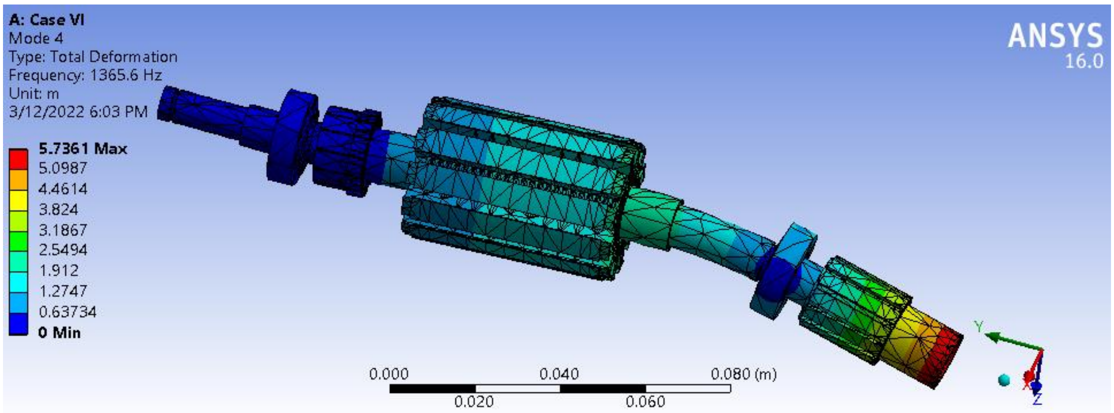

Figure 4.4(f) Case VI: Fourth Mode deformation

Fourth mode for case VI was at frequency of 1365.6 Hz as compared to 1404.8 Hz of case I. Deformation was decreased to 5.7361m as compared to 6.0159m of case I. Modal amplitude difference was 0.2798m.

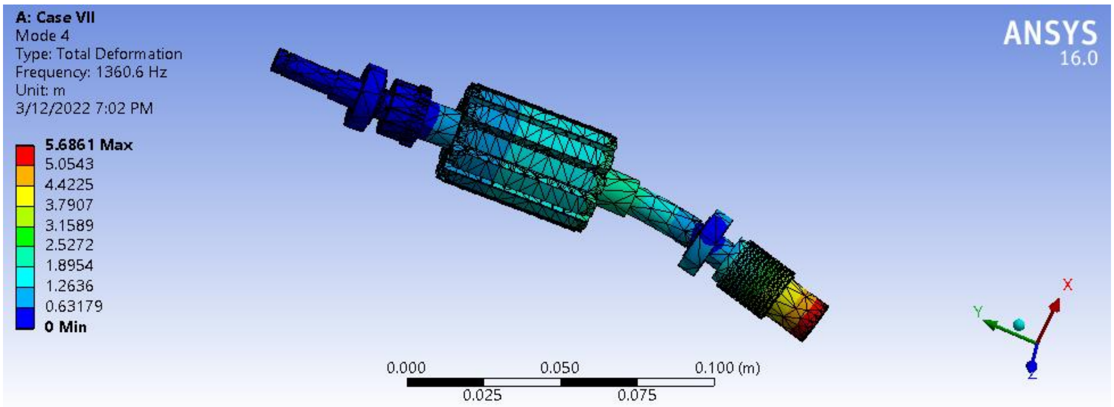

Figure 4.4(g) Case VII: Fourth Mode deformation

Fourth mode for case VII was at frequency of 1360.6 Hz as compared to 1404.8 Hz of case I. Deformation was decreased to 5.6861m as compared to 6.0159m of case I. Modal amplitude difference was 0.3298m.

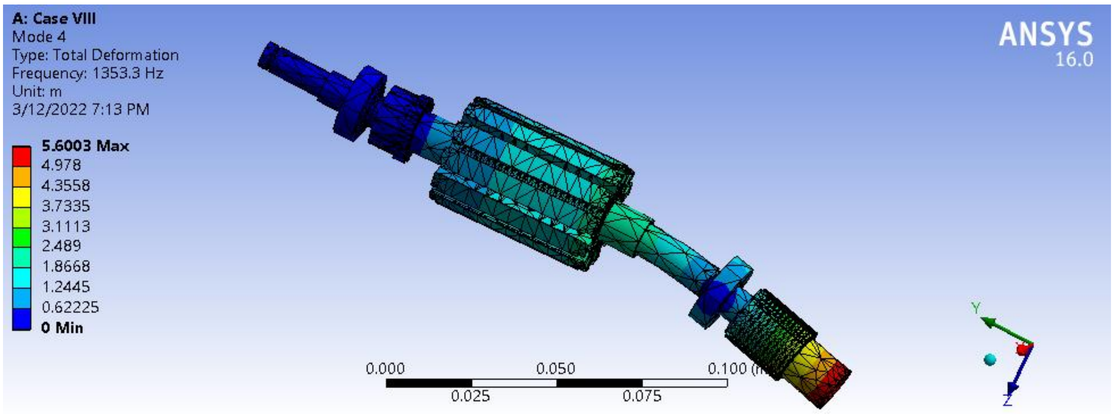

Figure 4.4(h) Case VIII: Fourth Mode deformation

Fourth mode for case VIII was at frequency of 1353.3 Hz as compared to 1404.8 Hz of case I Deformation was decreased to 5.6003m as compared to 6.0159m of case I. Modal amplitude difference was 0.4156m.

We can observe from figure 4.4(b) and figure 4.4(c) that the frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same can be observed when these two shapes were combined with stiffeners on groove from figure 4.4(f) and figure 4.4(g).

Mode 5

Comparing frequency and deformation for case I to case VIII few observations were made. The fifth mode frequencies for all eight cases were listed in table 4.6. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. Hence, variation in stiffener was the reason for difference in mode frequency. The deformation shape was found similar in eight cases. It can be concluded that mode shape will be same irrespective of addition of stiffeners on the spindle.

For fifth mode, the spindle initially had maximum modal displacement of 3.1937 on the groove as shown in figure 4.5(a). The modal displacement was decreased after the application of stiffeners in all rest of the seven cases.

Table 4.6 Frequency, deformation and modal amplitude difference on fifth mode

Mode 5			
Case	Frequency (Hz)	Deformation (m)	Modal Amplitude difference (m)
I	2016.7	3.1937	0
II	1990.2	3.1548	0.0389
III	1992.3	3.1514	0.0423
IV	2004	3.1479	0.0458
V	1981.7	3.1351	0.0586
VI	1985.9	3.1071	0.0866
VII	1986.9	3.1028	0.0909
VIII	1980.3	3.0901	0.1036

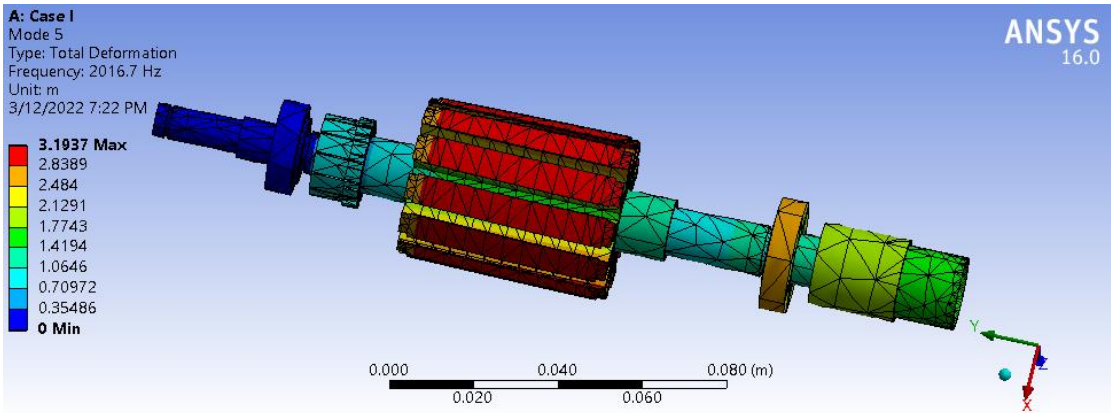

Figure 4.5(a) Case I: Fifth Mode deformation

Fifth mode for case I was at frequency of 2016.7. Maximum deformation was 3.1937m at groove.

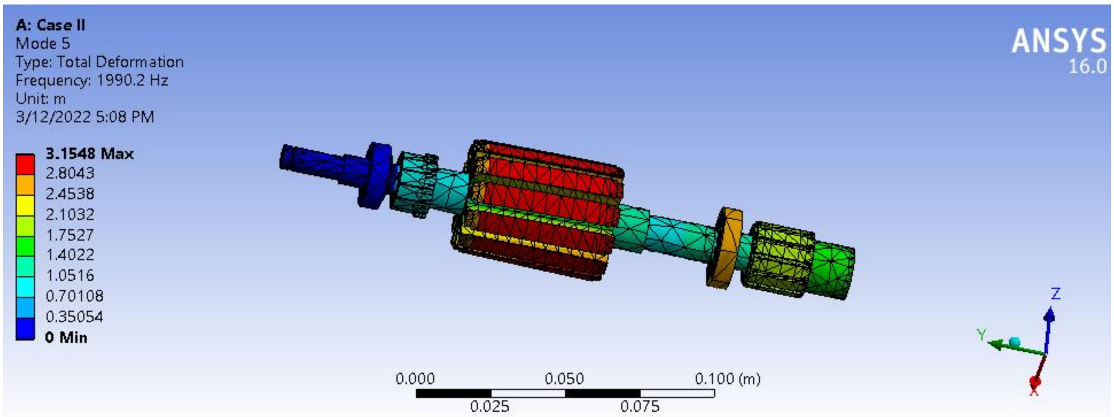

Figure 4.5(b) Case II: Fifth Mode deformation

Fifth mode for case II was at frequency of 1990.2 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1548m as compared to 3.1937m of case I. Modal amplitude difference was 0.0389m.

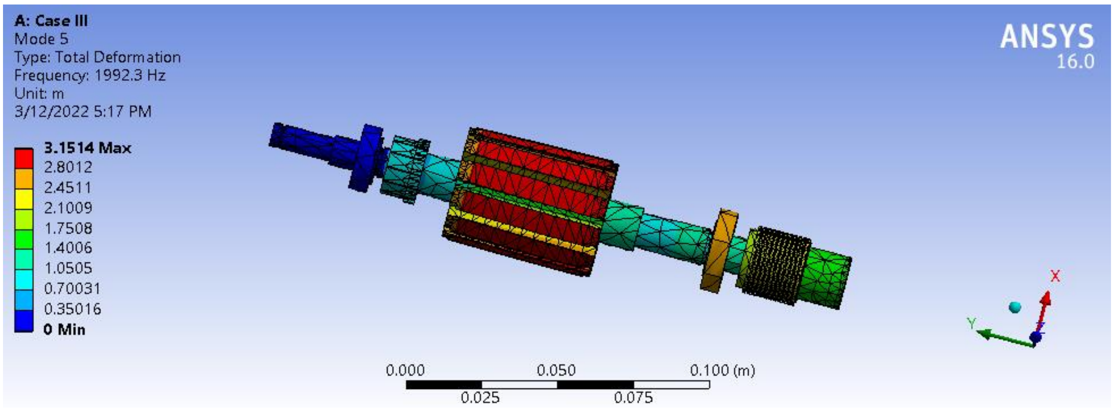

Figure 4.5(c) Case III: Fifth Mode deformation

Fifth mode for case III was at frequency of 1992.3 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1514 as compared to 3.1937m of case I. Modal amplitude difference was 0.0423.

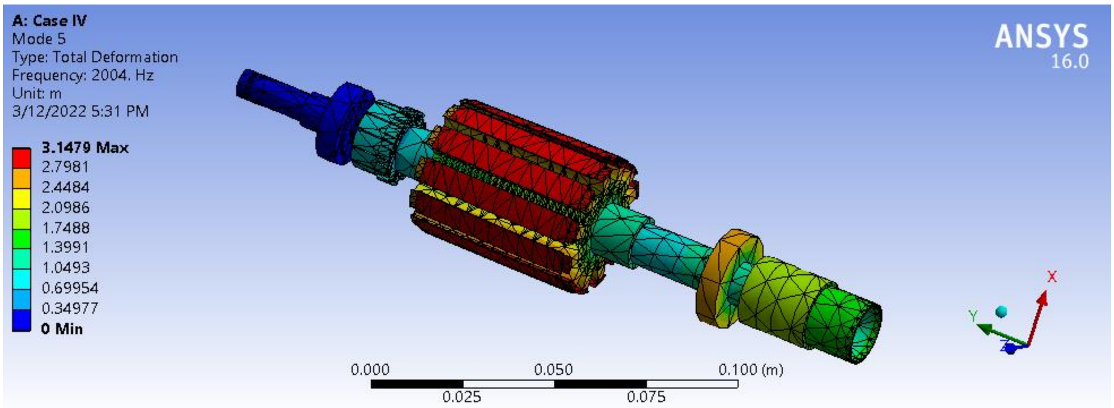

Figure 4.5(d) Case IV: Fifth Mode deformation

Fifth mode for case IV was at frequency of 2004 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1479m as compared to 3.1937m of case I. Modal amplitude difference was 0.0458m.

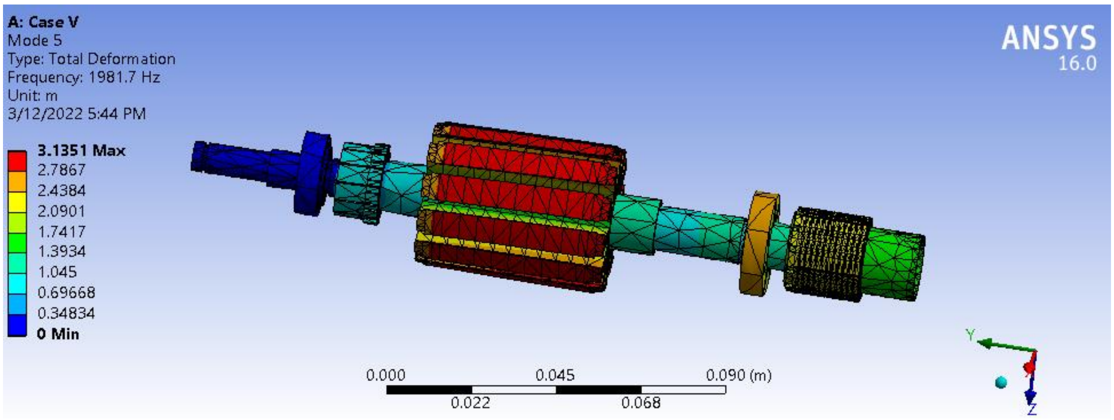

Figure 4.5(e) Case V: Fifth Mode deformation

Fifth mode for case V was at frequency of 1981.7 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1351m as compared to 3.1937m of case I. Modal amplitude difference was 0.0586m.

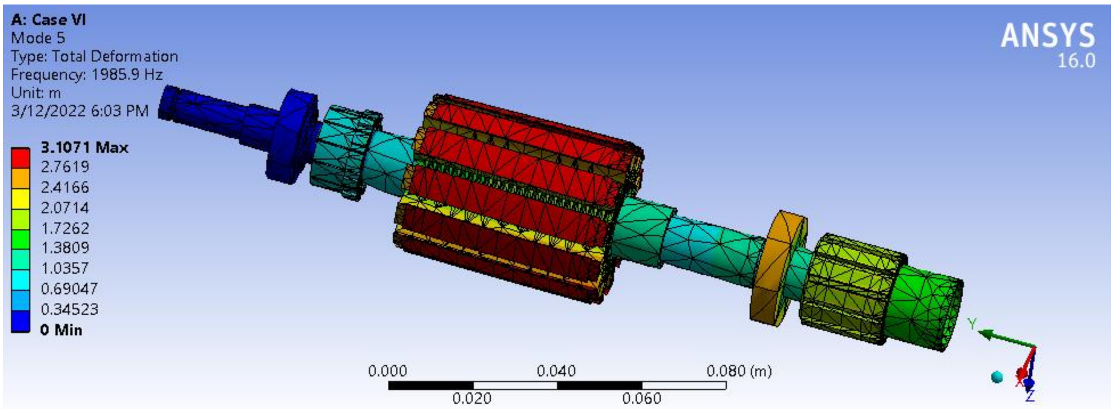

Figure 4.5(f) Case VI: Fifth Mode deformation

Fifth mode for case VI was at frequency of 1985.9 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1071m as compared to 3.1937m of case I. Modal amplitude difference was 0.0866m.

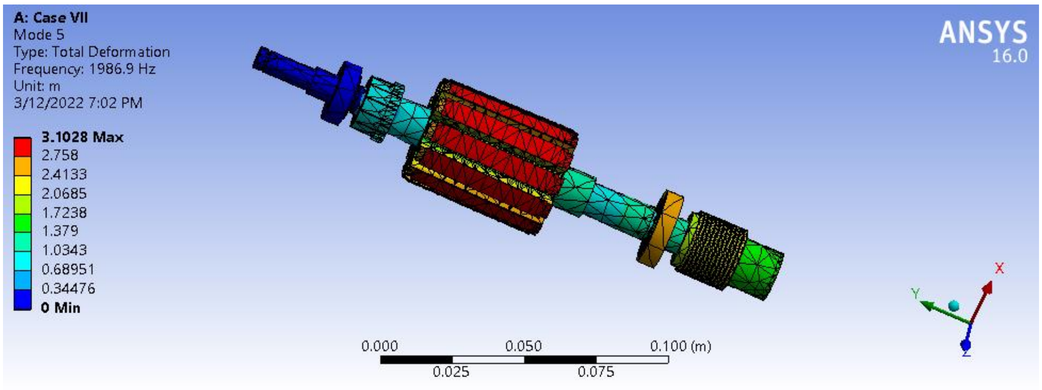

Figure 4.5(g) Case VII: Fifth Mode deformation

Fifth mode for case VII was at frequency of 1986.9 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.1028m as compared to 3.1937m of case I. Modal amplitude difference was 0.0909m.

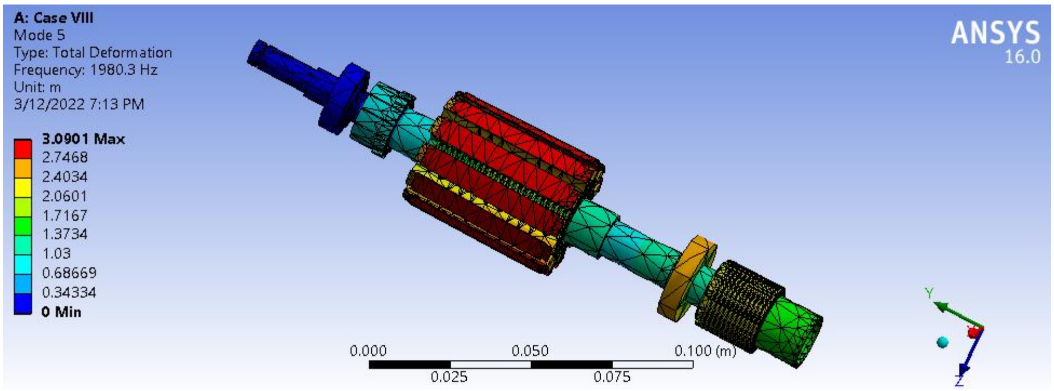

Figure 4.5(h) Case VIII: Fifth Mode deformation

Fifth mode for case VIII was at frequency of 1980.3 Hz as compared to 2016.7 Hz of case I. Deformation was decreased to 3.0901 as compared to 3.1937m of case I. Modal amplitude difference was 0.1036m.

We can observe from figure 4.5(b) and figure 4.5(c) that the frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same can be observed when these two shapes were combined with stiffeners on groove from figure 4.5(f) and figure 4.5(g).

Mode 9

According to figure 4.6(a), for ninth mode, the spindle initially had maximum modal displacement of 2.441 on the tool holder. Comparing frequency and deformation for case I to case VIII few observations were made. The ninth mode frequencies for all eight cases were listed in table 4.6.

For Mode 9, the spindle initially has maximum modal displacement of 2.441 on the tool holder as shown in figure 4.6(a). The modal displacement was decreased after the application of stiffeners in all rest of the seven cases.

The modal displacement was decreased after the application of stiffeners in all rest of the seven cases. Also, combination of all stiffeners at once decreased the deformation the most as observed in case VIII.

Table 4.7 Frequency, deformation and modal amplitude difference on ninth mode

Mode 9			
Case	Frequency (Hz)	Deformation	Modal Amplitude difference
I	6299.4	2.441	0
II	6203.6	2.4343	0.0067
III	6196.3	2.438	0.003
IV	6252.9	2.4093	0.0317
V	6164.2	2.4344	0.0066
VI	6164.9	2.4111	0.0299
VII	6158.2	2.4119	0.0291
VIII	6124.2	2.4056	0.0354

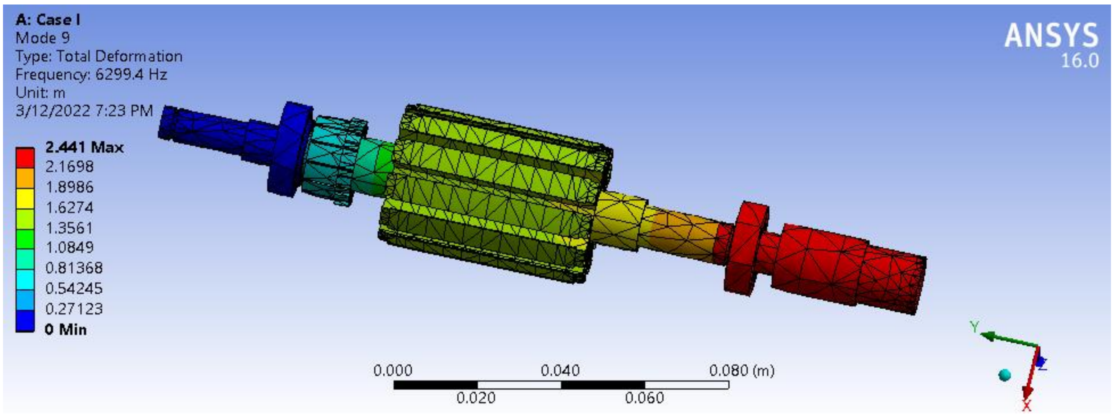

Figure 4.6(a) Case I: Ninth Mode deformation

Ninth mode for case I was at frequency of 6299.4 Hz. Deformation 2.441m.

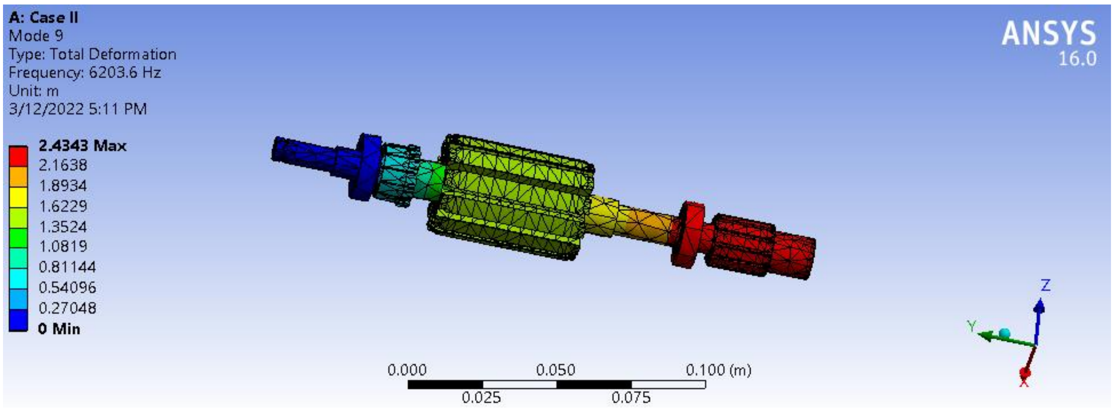

Figure 4.6(b) Case II: Ninth Mode deformation

Ninth mode for case II was at frequency of 6203.6 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 2.4343m as compared to 2.441m of case I. Modal amplitude difference was 0.0067m.

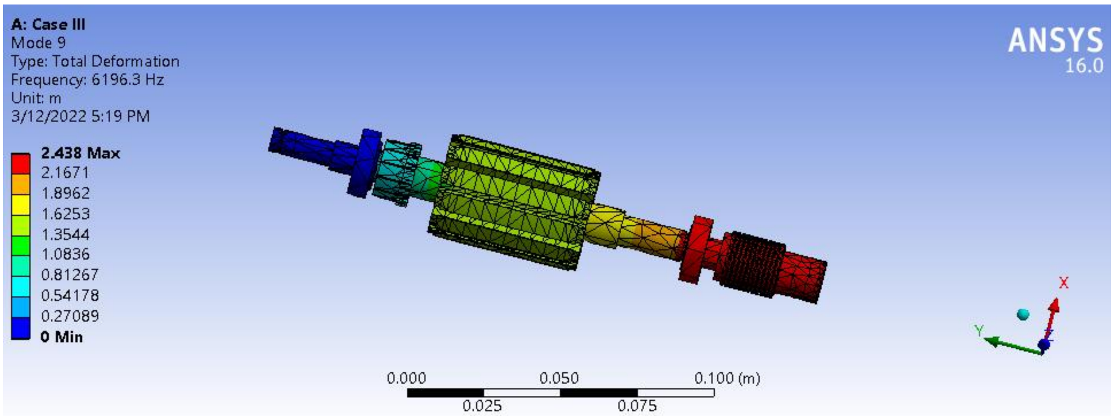

Figure 4.6(c) Case III: Ninth Mode deformation

Ninth mode for case III was at frequency of 6196.3 Hz as compared to 6299.4 Hz of case I Deformation was decreased to 2.438m as compared to 2.441m of case I. Modal amplitude difference was 0.003m.

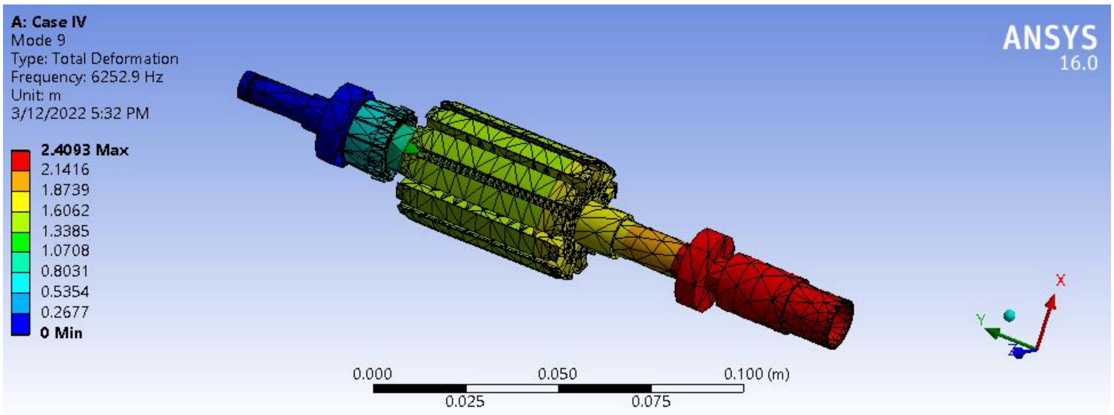

Figure 4.6(d) Case IV: Ninth Mode deformation

Ninth mode for case IV was at frequency of 6252.9 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 2.4093 as compared to 2.441m of case I. Modal amplitude difference was 0.0317m.

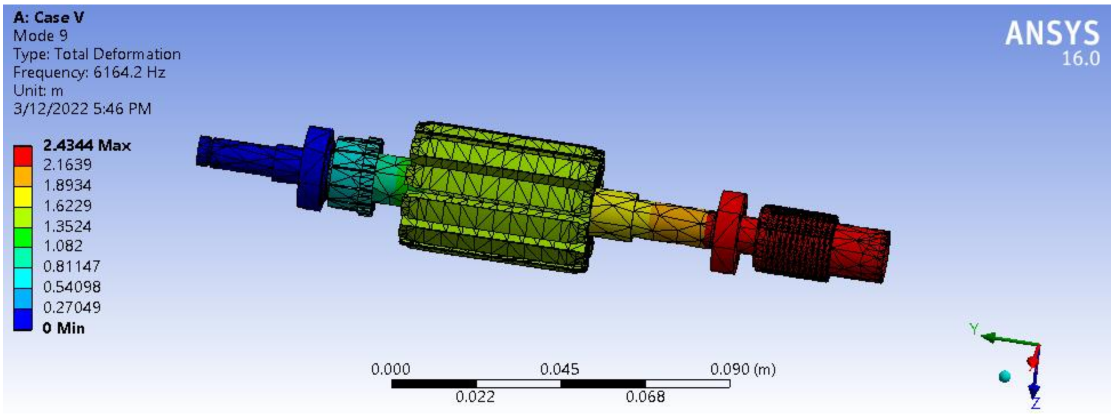

Figure 4.6(e) Case V: Ninth Mode deformation

Ninth mode for case V was at frequency of 6164.2 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 2.4344m as compared to 2.441m of case I. Modal amplitude difference was 0.0066.

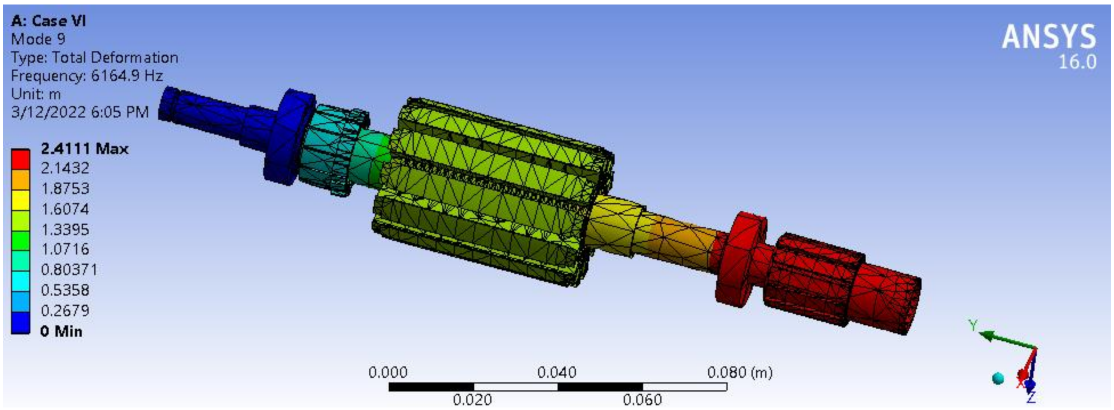

Figure 4.6(f) Case VI: Ninth Mode deformation

Ninth mode for case VI was at frequency of 6164.9 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 2.411m as compared to 2.441m of case I. Modal amplitude difference was 0.0291m.

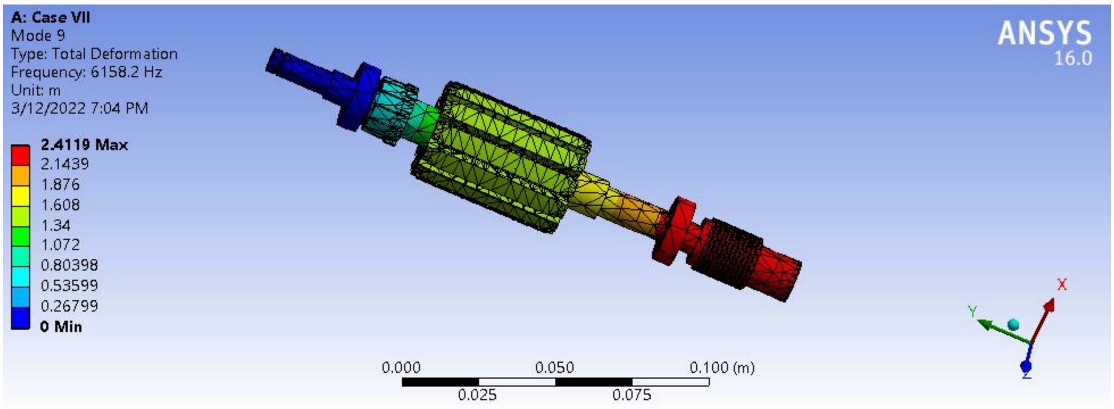

Figure 4.6(g) Case VII: Ninth Mode deformation

Ninth mode for case VII was at frequency of 6158.2 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 2.4119 as compared to 2.441m of case I. Modal amplitude difference was 0.0291m.

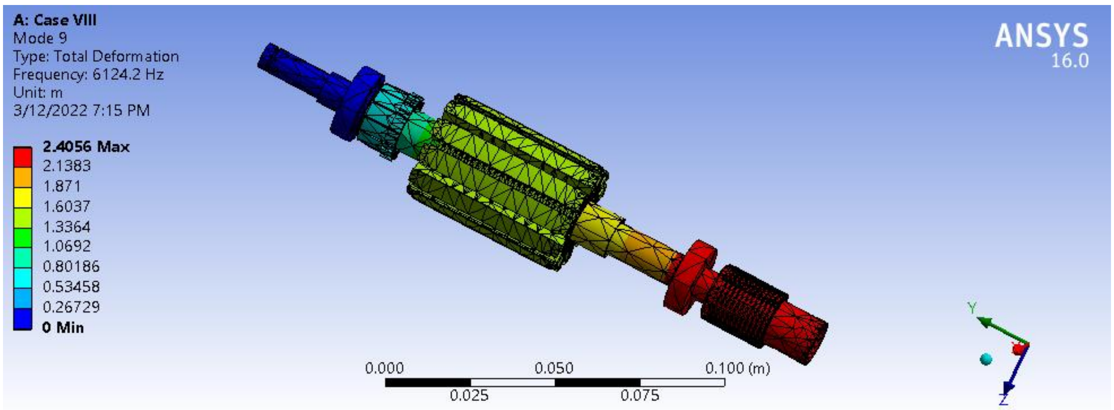

Figure 4.6(h) Case VIII: Ninth Mode deformation

Ninth mode for case VIII was at frequency of 6124.2 Hz as compared to 6299.4 Hz of case I. Deformation was decreased to 4.0202m as compared to 2.441m of case I. Modal amplitude difference was 0.0354m.

We can observe from figure 4.6(b) and figure 4.6(c) that the frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same can be observed when these two shapes were combined with stiffeners on groove from figure 4.6(f) and figure 4.6(g).

Results till mode 10 were observed however results of mode 6, mode 7, mode 8 and mode 10 were omitted in this paper because those results did not add value for analyzing. In other words, similar observations were made on those modes as well. Only results from mode 1, mode 2, mode 3, mode 4, mode 5 and mode 9 were presented for this reason. All the modes showed decrease in modal amplitude displacement, no change in mode shapes and deformation while stiffeners were added.

CHAPTER FIVE: CONCLUSIONS AND RECOMMENDATIONS

The modal analysis of spindle was performed. Taking natural frequency of spindle only as reference, the natural frequency while applying stiffeners were observed. The first natural frequency for spindle only was obtained as 290.28 Hz. It had maximum deformation of 4.04m.

The effect of stiffeners on spindle vibration was also simulated by placing varying shaped stiffeners on different position. Input data of material properties were kept same throughout the modal analysis varying only stiffeners. The deformation shape was found similar in eight cases. It can be concluded that mode shape will be same irrespective of addition of stiffeners on the spindle. The addition of stiffeners had no significant effect on deformation shape of modes. The adding of stiffeners reduced the amplitude of modal displacement to reduce vibration. The initial deformation of 4.04m was reduced to 3.9451m, 3.9362m, 4.0202m, 3.9003m, 3.929m, 3.9202m and 3.8867m respectively for case II to case VIII.

The shape of stiffeners had no noteworthy effect on vibration. The frequency and deformation on model of spindle with rectangular shaped stiffeners and circular shaped stiffeners were nearly identical. Same observations were made when these two shapes were combined with stiffeners on groove.

As modal analysis of spindle had been simulated using ANSYS, the following mentioned work remains as the future work.

1. Determine the damping coefficient of spindle. In this thesis, the natural frequencies were considered. Therefore, the mode frequencies with the damping of spindle structural material need to be analyzed. The damping can be obtained by the linear relation with mass and stiffness of spindle.

2. Set up the stiffeners with varying length and thickness to see its effect on mode shape.

REFERENCES

Aerospace Engineering. (2014, March 22). Retrieved from aerospacengineering.net: https://www.aerospacengineering.net/vibration-analysis-nastran-sol-103

Altintas, Y. (2000). *Manufacturing Automation.* London, UK: Cambridge University Press.

Anayet Ullah Patwari, W. F. (2009). Modal Analysis of the Surface Grinding Machine Structure through FEM and Experimental Test. *Advances in Acoustics and Vibration*.

Bathe, K.-J. (1976). In *Finite Element Procedures in Engineering Analysis.* Prentice.

Engineering, A. (2013, August 1). *aerospacengineering.net*. Retrieved from aerospacengineering.net: https://www.aerospacengineering.net/linear-buckling-analysis/

Erhan Budak, Y. A. (1994). *International Journal of Machine Tools Manufacturing Vol. 34*, 907-918.

Erol Turkes, S. O. (2017). Modelling of dynamic cutting force coefficients and catter stability dependent on shear angle oscillation. *The International Journal of Advanced Manufacturing Technology*.

Jasvir Singh Dhillon, P. R. (September 2014). Design of Engine Mount Bracket for a FSAE Car Using Finite Element Analysis. *ISSN: 2248-9622, Vol. 4, Issue 9 (Version 6)*, 74-81.

Jun Wang, B. W. (2012). Modeling and modal analysis of tool holder-spindle assembly on CNC milling machine using FEA. *Applied Mechanics and Materials Vols. 157-158*, 220-226.

Keith Rouch:, J. R. (2002). Use of Finite Element Structural Models in Analyzing Machine Tool Chatter. *Finite Elements in Analysis and Design, Vol. 38*, 1029-1046.

Kim, K. F.-S. (1983). Modal Analysis of Machine Tool Structures Based on Experimental Data. *Transactions of the ASME Vol. 105*.

Kong Jiali, C. X. (2017). Modal Analysis of CNC Lathe's Spindle Based on Finite Element. *Advances in Engineering Research (AER), Volume 148*.

Lakshmi Kala, V. R. (July 2015). Modeling and Analysis of V6 Engine Mount Bracket. *IJIRSET, Vol. 4, Issue 7*, 5907-5914.

Liu, H. W. (2010). In *Mechanics of materials* (pp. 241-246). Higher Education Pres.

M. Polacek, J. T. (1963). The Stability of Machine Tools against Self Excited Vibrations in Machining. *Proceedings of the ASME International Research in Production*, 465-474.

Mehdi Namazi, Y. A. (2007). Modeling and identification of tool holder-spindle interface dynamics,. *International Journal of Machine Tools & Manufacture 47*, 1333-1341.

Nik Muhamad Firdaus, B. N. (2013). *Modal Analysis on CNC Milling Cutting Tool.* Faculty of Mechanical Engineering UNIVERSITI MALAYSIA PAHANG.

Qin-Yuan, Z. (2011). Modal analysis of 7:24 taper tool system for NC machine based on ANSYS. In *Manufacture Technology and Machine.*

Siamak Pedrammehr, H. F. (2012). Modal Analysis of the Milling Machine Structure through FEM and Experimental Test. *Advanced Materials Research Vols 383-390*, 6717-6721.

Silva, N. M. (1997). *Theoretical and Experimental Modal Analysis.* NY, USA: John Wiley.

Suo Xian Yuan, X. L. (2010). Modal Analysis on the Truss Structures of Machine Tool. *Advanced Materials Research (Volumes 118-120)*, 972-976.

Umesh S. Ghorpade, D. S. (2012). Finite Element Analysis and Natural Frequency Optimization of Engine Bracket. *IJMIE, Vol-2, Iss-3*, 1-6.

Weiwei Zhang, J. K. (2017). The Design of simple slant bed horizontal CNC lathe. Tianjin University of Technology and Education. *Advances in Engineering Research*, 32-39.

Woodford, C. (1991). In E. A. Element, *The finite element mesh* (pp. ,9,43-64.).

Zhijun Wun, C. X. (2010). *International Conference on Digital Manufacturing & Automation (ICDMA) Vol. 1*, (pp. 929-933).