# DESIGN, STATIC, AND MODAL ANALYSIS OF HIGH SPEED MOTORIZED MILLING SPINDLE

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***ABSTRACT** - The increasing demands for higher productivity and lower production costs, high speed machining tools have been widely utilized in the modern production industries. High speed motorized spindle systems are subjected to several effects during high speed rotations that can cause substantial changes in their dynamic and thermal behaviors, leading to chatter, bearing seizure or premature spindle bearing failures. Compared with conventional spindles, motorized spindles are equipped with a built-in-motor, so that power transmission devices such as gears and belts are eliminated. This design also reduces vibrations and achieves high rotational balance, and enables precise control of rotational accelerations and decelerations. The objective of this work is to optimize the parameters influencing the high frequency or high speed milling spindle running at 14000 rpm with power rating 15 kW. The static deflection analysis is carried out to check the spindle stiffness, and the dynamic behavior of spindle is determined by characterizing its properties under different modes of vibration. Modal analysis is carried out using ANSYS work bench software. The modal parameters obtained from the modal analysis can be used to analyze the system behavior. In dynamic analysis every moment spindle shaft changes its behavior so that it is difficult to compare theoretically, but static analysis results can compare with theoretical results.*

***Keyword: Milling, Static, modal, ANSYS, Spindle, Assembly.***

## 1. INTRODUCTION

Modern technology to a great extent relies on the use of high frequency motorized spindle which is a competent technology for significantly ever-increasing productivity and plummeting production costs. On one hand, high precision is essential for the ongoing trend of manufacturing activity which is found especially in electronic industry, automobile industry and machine tool industry. On the other hand, high precision is essential for leading edge research. With the help of CNC technology, machine tools today are not limited to human capabilities and are able to make ultra-precision products down to nanoscales in a much faster manner. The traditional design philosophy of machine tools is multi-functionality and highest precision possible. For example, a shank with spindle together with tailstock can be included onto a standard three axis vertical milling machine to wind up a multifunctional boring-milling-turning machine, which means the machine apparatus is intended to be utilized for different rather than single purposes. However, with the dramatic increase of industry varieties and the growing demand of miniature products, these broadly useful machine instruments are not effective, either in terms of machine time or cost or in manufacturing products with special sizes and precision requirements. In this quick changing corporate world all business are driven by profitability, proficiency and low cost. In assembling businesses there is a developing interest for machine that production completed items at energetic pace with a more elevated amount of consistency. In this respects even machine machines have experienced a transformation and CNC (Computer Numerical Control) machine machines are high sought after on account of their higher level of automation and accuracy. The machine spindle system is one of the most essential parts of a machine tool since its dynamic properties directly influences the cutting ability of the machine tool. The dimensions of the spindle shaft, location, stiffness of the bearings and bearing preload affect the vibration free operation of the spindle. The bearing stiffness is dependent on the preload and is also influenced by the deformation of the spindle shaft with the housing during machining. Angular contact metal ball bearings are most regularly utilized as a part of the CNC machine spindle due to their low-friction properties and ability to withstand external loads in both axial and radial direction. To achieve high speed rotation, motorized spindles have been developed. This type of spindle is equipped with a built in motor as an integrating part of the spindle shaft, eliminating the need for conventional power transmission devices such as gears and belts. This design reduces vibrations, decelerations. However, the high speed rotation and built in motor also introduce a large amounts of heat and rotating mass in to the system, requires precisely regulated cooling and lubrication.

### 1.1 End Milling and Face Milling

The following diagram illustrates face milling and end milling operations. Milling cutting tools are known as multi point cutting tools. In Face milling operation cutters will make the flat surface finish on surface of the object, in this number of inserts are used at the cutter (carbide inserts), the cutting edges are located along its sides. End mills are the tools which have cutting teeth at one end as well as on the sides. These are generally used to refer flat bottom cutters (creating pockets) and rounded cutters. These are made of HSS or cemented carbides.

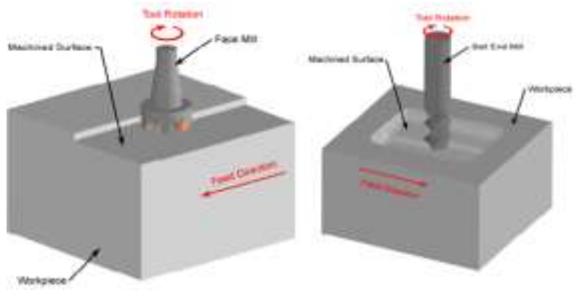


Fig.1.2 Face milling and End milling operations

## 2. OBJECTIVES

This project work focuses on the development of low cost CNC milling spindle. The following are the objectives of this work:
To design the BT-40 milling spindle for a speed of 14000 rpm, by making use of precision bearings and different bearing arrangements. Analytical calculations shall be done for different bearing arrangements by considering the bearing manufacturers data and using suitable equations. Static analysis of spindle assembly for different bearing arrangement will be carried out by using ANSYS Work Bench and the results are compared with the theoretical results. Dynamic Modal analysis of different spindle arrangement would be carried out by using ANSYS Work Bench to determine behaviour of the spindle assembly and to find out natural frequencies and mode shapes. Finally assemble the model using optimum bearing arrangement.

### 2.1 METHODOLOGY

First, develop the parts or subcomponents according to dimensions given by the data using suitable software. Keeping the requirements in mind, assemblies with bearing arrangement will be modelled using SOLID EDGE modelling software. Theoretical deflection of the spindle shaft assemblies will be estimated by considering the appropriate radial cutting force at the nose end of the spindle, material properties of the spindle and radial stiffness of the angular contact ball bearings. Static deflection analysis of the spindle assembly will be carried out in ANSYS by considering the appropriate radial cutting force at the nose end of the spindle assembly, material properties of the spindle and radial stiffness of the angular contact ball bearings. Compare static analysis results with theoretical values, from that we can take optimum bearing arrangement for dynamic analysis. Dynamic Modal analysis of different spindle assemblies will be carried out using ANSYS Work Bench software to obtain the natural frequencies and different mode shape at different speed of the milling spindle and frequency response function. Finally, the optimum design of the spindle would be selected on the results of analysis and basis on the results fabrication will be done.

## 3. COMPONENTS OF THE SPINDLE

**3.1 Parts and materials of the spindle assembly**
Milling spindles usually rotate at high speeds. In integrated motor spindle design, selection of motor becomes the main factor. The motor for the spindle can be selected on the basis of torque and speed requirements, type of cooling arrangement, dimensions of the motor, etc. Angular contact ball bearings are generally preferred for milling spindles. The bearings are preloaded by means of an adjustable locknut. An encoder is required for sensing the spindle speed and spindle position. An integrated motor spindle assembly is shown in the fig. 3.1.

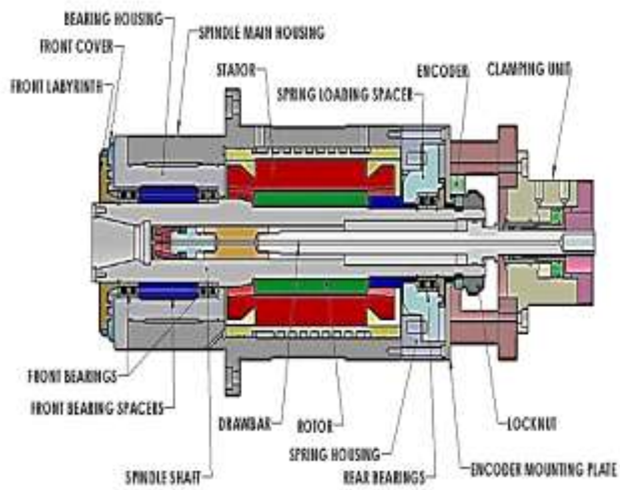


Fig. 3.1 Integrated motor spindle assembly

Material selection is a matter of quality and cost. The properties of the material must be adequate to meet the design requirements and service conditions. The list of the major parts and its materials of the spindle assembly is listed in the table 3.1.

Table 3.1 List of parts of spindle assembly

| Part number | Part name | Material |
|---|---|---|
| 1 | Spindle main housing | SGIron 500/7 |
| 2 | Spindle shaft | En-24 |
| 3 | Bearing housing | En-8 |
| 4 | Front cover | En-8 |
| 5 | Front labyrinth | En-8 |
| 6 | Bearing spacers | En-24 |
| 7 | Spring housing | En-8 |
| 8 | Rear labyrinth | En-8 |
| 9 | Spring loaded spacer | En-8 |
| 10 | Encoder mounting plate | C45 |
| 11 | Drawbar | En-24 |

**3.1.1. Spindle main housing:** As the name indicates it is the outer most cover, or it is casing surrounded to the all parts of the assembly. It is made up of Spheroidal Graphite Iron to obtain better strength and toughness for good pressure tightness and can be welded. Applications of SG: Three jaw and four jaws chucks, gear vices, cutter body and handles etc. through the holes coolant will be enters into the cooling jacket and helps the stator and rotor for cooling.

Fig. 3.2 Spindle Housing

**3.1.2 Spindle shaft:** This is the rotating part of the assembly, to which cutting tool is directly attached. The rotor is integrated part of the shaft and angular contact ball bearings are used to support the spindle shaft. It is made of En-24 Steel to get a high wear resistance and strength, its composition are Steel, 0.4% Carbon and .6% Manganese. Applications En-24: heavy duty axels, shafts, heavy duty gears, and connecting rods, etc.

Fig. 3.3 Spindle Shaft and Bearing Housing

**3.1.3 Bearing housing:** As the name indicates it covers the bearings and holds the bearings. It is made of En-8 medium carbon Steel with hardness range 180-207 HB, to get better wear resistance and good tensile strength. Applications of En8: housing, keys, shafts, and gears, etc.

**3.2 Angular contact bearing features:** Angular contact bearings are most commonly used today in very high speed spindle designs since they provides, High precision, Increases capacity of load carrying, Easily metal cutting spindle can achieve required speed. The contact angles are $0^0$, $15^0$, $25^0$, $60^0$, the lower the contact angle, the larger the radial load carrying capacity, the higher the contact angle the higher the axial loading capacity figure shows the bearing contact angles.

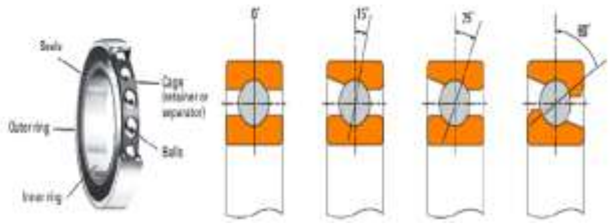

Fig. 3.4 Ball Bearings with Contact Angles

$15^0$- used when loading is primarily radial; for very high speed applications. $25^{o}$ - used when loading is primarily axial. $60^{o}$ - highest axial stiffness; used in ball screw support bearings.

**3.3 Bearing Preload**

In a few applications, the bearings are given an initial burden or load; this implies that the bearings internal clearance is negative before operation. This is called "preload" and is regularly connected to angular contact ball bearings and tapered roller bearings. There are 3 types preloaded occurs they are - **Light preloaded bearing:** These are designed to allow maximum speed and lower stiffness.

**Heavy preload bearing:** This provides high stiffness at low speed, angular contact bearings used.

**Medium preload:** This gives medium stiffness at medium speed.

**3.3.1 Purpose of preload**

Rigidity of the bearing increases, The bearing frequency increases and becomes suitable for high-speed rotation. Shaft run out is suppressed; rotation and position precision is enhanced. Vibration and noise are controlled. Fretting produced by external vibration is prevented. For this project we have selected NSK bearings, with back to back arrangement, Mounting two bearings back-to-back provides a moderately stiff bearing configuration, which can also accommodate tilting moments, when the bearing configuration is back-to-back, than the loading projection lines diverge towards the bearing axis. Axial loads in both directions can be accommodated, but only by one bearing in each direction.

### 3.4 Selection of Bearing Arrangements

For this project we have selected three different bearing arrangements with different bearing stiffness, they are as shown below. Depending on the shaft length and bearing spread distance, two single row tapers or two couple of angular contact ball bearings mounted face-to-face or back-to-back can be used.

**Bearing arrangement 1**

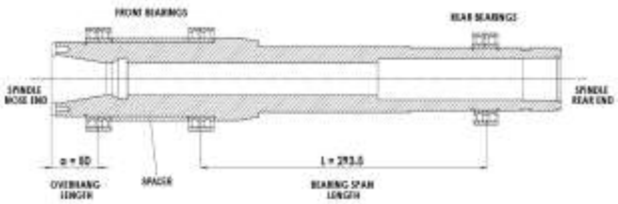


Fig. 3.5 Bearing Arrangement1 with Span Length 293.5 mm.

Fig. 3.5 shows the bearing arrangement 1, having its span length 293.5mm and overhanging length 50mm. In this arrangement there are two sets of bearings are arranged as quadruplet back-to-back (DBTT) at the front separated by a spacer and the rear end of the spindle is provided with one set of bearings as back-to-back (DB) mounting arrangement. With this arrangement the bearings will be able to take loads from both directions axial and radial with high stiffness and high speed.

**Bearing arrangement 2**

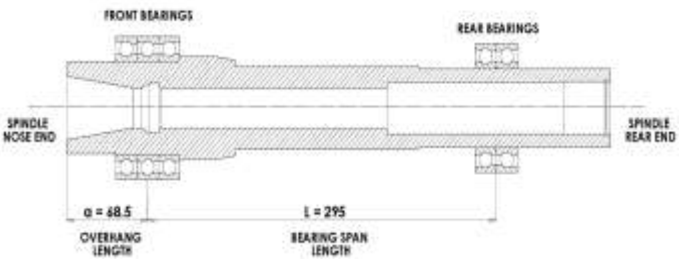


Fig. 3.6 Bearing Arrangement 2 with Span Length 295 mm

Fig. 3.6 shows the bearing arrangement 2, having span length 295 mm and overhanging length 68.5mm. In this arrangement one set of bearings are used at the front and rear end of the spindle with back to back arrange. The rear end of the spindle is supported by one set of bearings arranged in back-to-back (DB) fashion.

**Bearing arrangement 3**

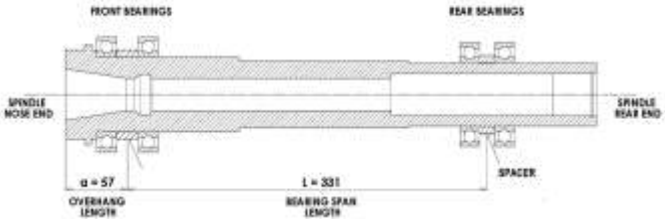


Fig. 3.7 Bearing Arrangement 3 with Span Length 331 mm

Fig. 3.7 shows bearing arrangement 3, with span length 331 mm and having span length 331 mm and overhanging length 57 mm. span length is distance between front and rear bearing, overhanging length is front and nose. In this arrangement one set of bearings is arranged at the front and rear in back-to-back (DB) fashion, on bearing medium preload is applied. The bearings are separated by a spacer. The bearings will be able to carry loads from both the directions. The dimensions of the bearings used for arrangement one, two and three are tabulated in the table 3.10, previous chapter.

## 4. THEORETICAL CALCULATIONS OF CUTTING FORCES AND DEFLECTIONS

### 4.1 THEORETICAL CUTTING FORCES CALCULATIONS

To calculate the theoretical cutting forces acting on the spindle nose for different machining process, CMTI machine tool design data hand book is referred. Forces mainly depend on speed and feed factors. While analyzing cutting tool spindle mainly concentrate on the nose of the spindle assembly.

#### 4.1.1 Face Milling

Diameter of the work piece, **b = 60 mm**
No. of teeth, **Z = 6**, Diameter of the cutter, **D = 80 mm**
Depth of cut, **t = 2 mm**, Cutting speed, **v = 80 m/min** (Table 276, page 655 CMTI hand book), Revolutions per minute, $\mathbf{n} = \frac{1000 \times v}{\pi \times D} = \frac{1000 \times 80}{\pi \times 80} = \mathbf{318.3\ rpm}$, Feed per tooth, $\mathbf{S_z = 0.1\ mm}$, Feed per minute, $\mathbf{S_m} = Sz \times z \times n = 0.1 \times 6 \times 318.3 = \mathbf{191\ mm/min}$, Metal removal rate, $\mathbf{Q} = \frac{b \times t \times Sm}{1000} = \frac{60 \times 2 \times 190.8}{1000} = \mathbf{22.9\ cm^3/min}$
Approach angle, $\mathbf{x = 30^\circ}$, Average chip thickness, $\mathbf{a_s} = \frac{57.3 \times Sz \times (\sin x) \times (\cos \psi 1 - \cos\psi 2)}{\psi s^\circ} = \mathbf{0.045\ mm}$
Work material considered is steel having hardness of 100 HB and average chip thickness of 0.045 mm. Unit power, $\mathbf{U = 0.065\ kW/cm^3/min}$ (Table 269, page 649 CMTI hand book).Considering flank wear = **0.2 mm**. Flank wear Correction factor, $K_h$ **= 1.1** for flank wear 0.2 mm (Table 270, page 650 CMTI hand book). Radial rake angle, $\boldsymbol{\gamma = -5^\circ}$; Radial rake angle Correction factor, $\mathbf{K_\gamma = 1.21}$ (Table 271, page 650 CMTI hand book). Power at

the spindle, **N** = U × $K_h$ × $K_\gamma$ × Q = 0.065 × 1.1 × 1.21 × 22.9 = **1.98 kW.** Tangential cutting force, $\mathbf{P_z} = \frac{6120 \times N}{V}$= 151.47kgf = **1485.9 N**, Radial cutting force, $\mathbf{P_y}$ = 0.35 × $P_z$ = **520 N**, Axial cutting force, $\mathbf{P_x}$ = 0.55 × $P_z$ = **817.3 N** Torque at the spindle, $\mathbf{T_s} = \frac{975 \times N}{n}$= 6.06kgf-m =**59.5 N-m.**

### 4.1.2 End milling

Diameter of the cutter, **D =20 mm**, No. of teeth, **Z =4**
Width of cut, **b =23 mm**, Depth of cut, **t =3 mm**
Cutting speed, **v =80 m/min** (Table 276, page 655 CMTI hand book), Revolutions per minute, $\mathbf{n} = \frac{1000 \times v}{\pi \times D} = \frac{1000 \times 80}{\pi \times 20}$ = **1274 rpm**, Feed per tooth, $\mathbf{S_z}$=**0.1 mm**
Feed per minute, $\mathbf{S_m}$= Sz × z × n = 0.1 × 6 × 1274 = **509.6 mm/min**, Metal removal rate, $\mathbf{Q} = \frac{b \times t \times S_m}{1000} = \frac{23 \times 3 \times 509.6}{1000}$ = **35.162 cm³/min**, Average chip thickness, $\mathbf{a_s} = \frac{114.6 \times S_z \times t}{D \times \psi s^{\circ}} = \frac{114.6 \times 0.1 \times 3}{20 \times 45.8}$= 0.038 **mm.**
Work material considered is steel having hardness of 100 HB. Therefore for hardness value of 100 HB and average chip thickness of 0.045 mm. Unit power, **U** = **0.063 kW/cm³/min (**Table 269, page 649 CMTI hand book). Considering flank wear = **0.2 mm**. Flank wear Correction factor, $\mathbf{K_h}$ **= 1.1** for flank wear 0.2 mm (Table 270, page 650 CMTI hand book). Radial rake angle, **γ = -5°.** Radial rake angle Correction factor, $\mathbf{K_\gamma}$**= 1.21** (Table 271, page 650 CMTI hand book). Power at the spindle, **N** = U × $K_h$ × $K_\gamma$ × Q = 0.063 × 1.1 × 1.21 × 35.162 = **2.95 kW**
Tangential cutting force, $\mathbf{P_z} = \frac{6120 \times N}{V}$= 225.675 kgf = **2213.87 N.** Radial cutting force, $\mathbf{P_y}$ = 0.55 × $P_z$ = **1218 N.** Axial cutting force,$\mathbf{P_x}$= 0.25 × $P_z$ = **553.5 N.** Torque at the spindle, $\mathbf{T_s} = \frac{975 \times N}{n}$= 2.25 kgf.m = **22.15 N**

### 4.2 THEORETICAL CALCULATION OF SPINDLE DEFLECTION

The total deflection of the spindle is due to the elastic deformation of the spindle and the elastic deformation of the bearings and neglecting the effects of housing deformation on the spindle. The t0tal deflection of the bearing system can be calculated by using below given formula. Equation 4.1 indicates deflection formula, for finding nose deflection of the spindle assembly.

$$\boldsymbol{\delta = P\left[\frac{1}{S_A}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L}+\frac{a}{I_a}\right)\right]} \qquad \text{........4.1}$$

- **Bearing Arrangement-1**

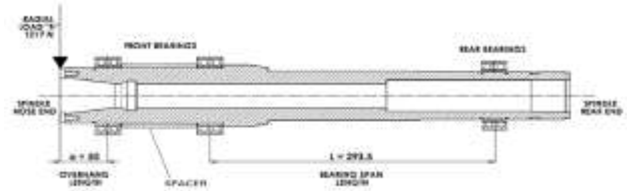


Fig. 4.1 Bearing Arrangement1 with Radial Load 1218 N

Where,
a = Length of overhang in mm = 50 mm, L = Bearing span in mm = 293.5 mm, E =Young's modulus of Spindle material in N/mm² = 210000 N/mm², P = Radial force in N = 1218 N for End Milling, $S_A$ = Stiffness of the front bearing N/mm = 1248000 N/mm, $S_B$ = Stiffness of the rear bearing in N/mm = 540000 N/mm, $I_L$ = Second moment of area of the shaft at the span in mm⁴ = 605113.53 mm⁴, $I_a$ = Second moment of area of the shaft at the overhang in mm⁴ = 780010.53 mm⁴
By substituting all above data in the given 4.1 equation, then the deflection (δ) of the spindle nose is given as below
**For end milling** Radial load **P=1218 N**
**δ = 4.10×10⁻³ mm** or **4.10μm**. For **Face Milling** tangential force, **P = 1486 N** and Nose deflection is **δ = 4.96×10⁻³ mm** or **4.96μm.**
Similarly, calculations for bearing arrangement two and three are as below

- **Bearing arrangement -2**

Bearing arrangement 2 is shown in fig. 4.2.

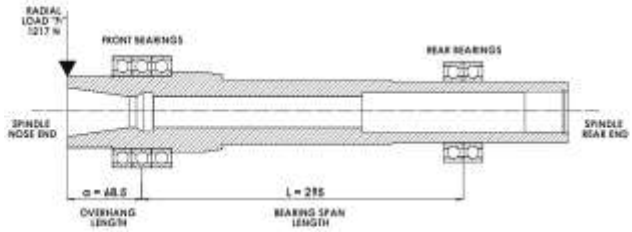


Fig. 4.2.Bearing Arrangement 2 with Radial Load 1218 N

Where, a = 68.5 mm, L = 295 mm, E= 210000 N/mm², P = 1218 N, $S_A$ = 955400 N/mm, $S_B$ = 612000 N/mm, $I_L$ = 600195.7 mm⁴, $I_a$= 774803.56 mm⁴
By substituting the above values in spindle deflection equation, the magnitude of deflection **δ = 7.21 × 10⁻³ mm** or **δ = 7.21 μm** for **end milling,** For **Face Milling, P = 1480 N.** Deflection **δ = 8.90 × 10⁻³ mm** or **δ = 8.90 μm.**

- **Bearing arrangement -3**

Bearing arrangement 3 is shown in fig. 4.3

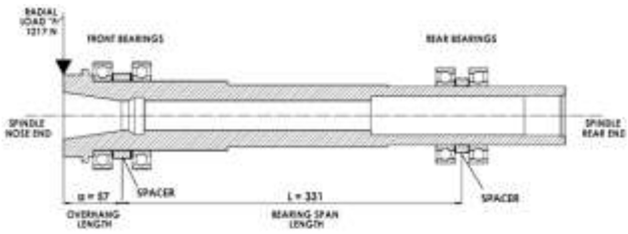


Fig. 4.3 Bearing Arrangement 3 with Radial Load 1218,

Where, a = 57 mm, L = 331 mm, E= 210000 N/mm$^2$, P = 1218 N, $S_A$= 702500 N/mm, $S_B$= 540000 N/mm, $I_L$= 598991.8 mm$^4$ $I_a$= 1183653.15 mm$^4$. By substituting the above values in spindle deflection equation 4.1,the magnitude of deflection, for **end milling, $\delta = 6.12 \times 10^{-3}$ mm** or **$\delta$ = 6.12 µm.** For **Face Milling P = 1480 N,** Deformation **$\delta = 7.50 \times 10^{-3}$ mm** or **$\delta$ = 7.50 µm.**
From the NSK bearing catalog, we have taken stiffness values for different bearing arrangements. The table 4.1 shows front and rear bearings stiffness values for different bearing arrangements.

Table 4.1 Front and Rear Bearing Stiffness values

| Bearing arrangements | Front bearing stiffness (N/mm) | Rear bearing stiffness (N/mm) |
|---|---|---|
| **1** | 1248000 | 540000 |
| **2** | 9554000 | 612000 |
| **3** | 702500 | 540000 |

All the calculated theoretical values are tabulated in table 4.2; the table shows tangential load, radial load, axial load, torque and deflections for different operations and bearing arrangements respectively.

Table 4.2 Theoretical Results summary

| Operation | Tangential Load (N) | Radial Load (N) | Axial Load (N) | Torque (Nm) | Deflection (µm) | | |
|---|---|---|---|---|---|---|---|
| | | | | | 1 | 2 | 3 |
| **Face Milling** | 1486 | 520 | 817 | 59.5 | 4.9 | 8.9 | 7.5 |
| **End milling** | 2214 | 1218 | 553 | 22.2 | 4.1 | 7.2 | 6.2 |

## 5. STATIC ANALYSIS OF SPINDLE ASSEMBLY

**5.1 Introduction** Static deflection analysis determines the impacts of steady state loading conditions on body or structure while neglecting inertia and damping effects, such as those caused by transient or time-varying loads. Static analysis calculates the displacements, stresses, strains, and forces in structures or components caused by loads that do not include inertia and damping effects. An assembly of motorized spindle is consists of a more numbers of different parts and subassemblies, lots of which are complex. The spindle can be modeled as a shaft, supported at each end by bearing sets. This representation can be seen in the fig. 5.1. Instead of bearing we have taken spring element while analyzing, stiffness of the spring is as same as the bearing stiffness.

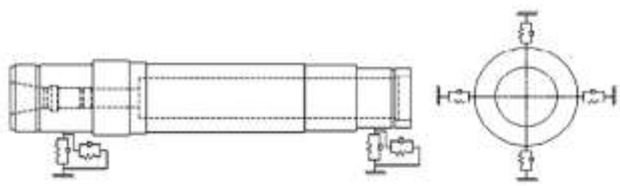

Fig. 5.1 Equivalent Model of Spindle Assembly

### 5.1.1 Bearing arrangement models

The figure show different bearing arrangement 3-d models which is made through Solid Edge /Solid works modeling software, it consists of couple of NSK bearing sets with back to back arrangement as shown and front bearing separated by spacer for arrangement 1&3. With this arrangement the bearings will be able to take loads from both directions, by using high stiffness bearings at the front better rigidity is provided with the proper preloading. We have taken medium bearing preload for analysis purpose.

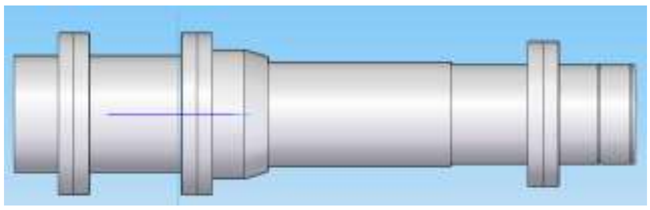

Fig. 5.2 Bearing Arrangement 3-D Model 1

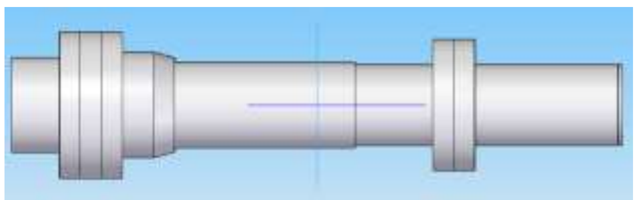

Fig. 5.3 Bearing Arrangement 3-D Model 2

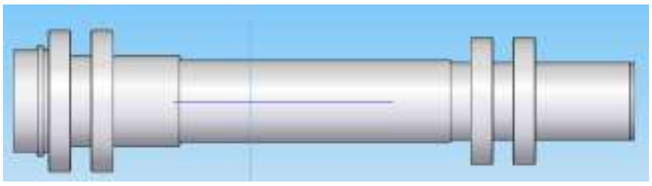

Fig. 5.4 Bearing Arrangement 3-D Model 3

## 5.2 Introduction to finite element analysis

### 5.2.1 Mesh Elements and Constrained model

The dividing the given model into number of finite elements is known as meshing, as shown in diagram. For this four/ten nodes tetrahedron elements are used because it has aerodynamic shape or structure. For high accuracy, elements should be very fine or smaller and smaller. A tetrahedron has four vertices, six edges, and is bounded by four triangular faces; it is a 3-d mesh element. For this project 5 mm element size given to the model. Any analysis system should need boundary conditions for analysis purpose, boundary conditions involves Load, material properties, displacement, gravitational load, thermal or fluid load, etc.

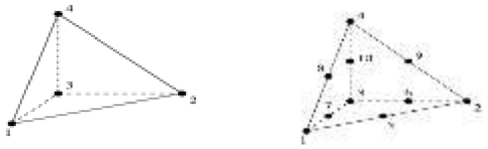

Fig. 5.5 4 node and 10 node tetrahedron

The fig. 5.6 shows meshed and constrained model.

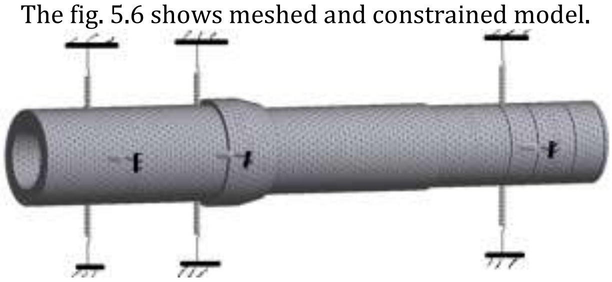

Fig. 5.6 Meshed and constrained 3-D Model

While applying constraints, take a spring element whose stiffness is same as that of bearings used and add body to ground option. In ANSYS APDL give the degrees of freedom as UX, UY, UZ=0 and ROTX, ROTY, ROTZ=1 and for different bearing arrangements real constants are given as spring constant values they are mentioned in table 4.1, for front and rear springs respectively. Along with following material properties also applied to the model to accomplish analysis. After meshed the following mesh statistics are obtained as shown in the table 5.1.

Table 5.1 Material Properties and Mesh Statistics

| Arrangements | 1 | 2 | 3 |
|---|---|---|---|
| Nodes | 122670 | 112387 | 117792 |
| Elements | 80109 | 73025 | 76335 |
| Material properties | Young's modulus (MPa) | Density (kg/mm$^3$) | Poison's ratio |
| | $210\times10^3$ | $7.82\times10^{-6}$ | 0.3 |

### 5.2.2 Loads applied

For this project, calculated load values i.e. radial load is 1218 N for end milling and tangential load 1486 N for face milling applied to different bearing arrangements. The load is applied at nose of the spindle assembly as shown in the fig. 5.7 respectively, instead of bearings, spring elements are used for supporting the spindle while analyzing.

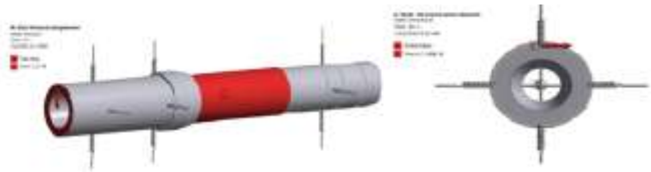

Fig. 5.7 Radial and Tangential Load Applied Model

## 5.3 Results

The static stress and nose deflection analysis results of end milling and face milling operations of the spindle assembly are as shown in the following figures. For this analysis radial and tangential load is applied to the end and face milling respectively.

### 5.3.1 End milling results:

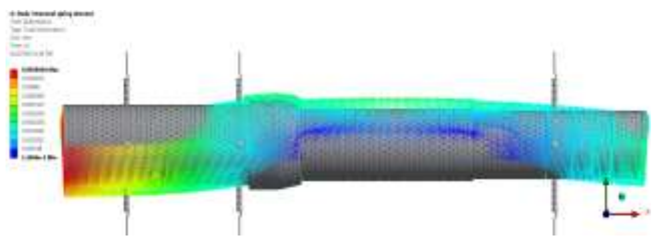

Fig. 5.8 Deflection at the Nose for Bearing Arrangement 1

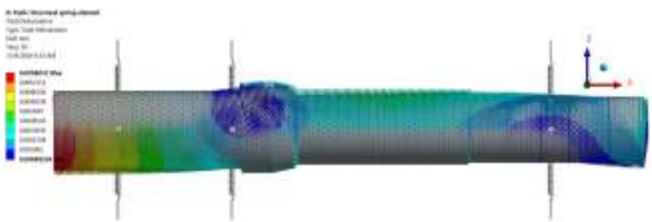

Fig. 5.9 Deflection at the Nose for Bearing Arrangement 2

Fig.5.10 Deflection at Nose for Bearing Arrangement 3

The all above diagrams shows end milling spindle nose deflection analysis results done in the ANSYS work bench 14.5 version of radial load carried 1218 N along the y-direction. The red color indicates maximum deformation and blue color shows least deformations on the spindle. The red color is appearing at the spindle nose i.e. the maximum deformation is available at the nose tip as shown in the figure, here instead of bearing spring is taken for analysis purpose having same stiffness as bearing material. In the above diagram bearing arrangement 1 is more stiffer than the other two bearing arrangement because its deflection is smaller than the other two arrangements. The below table indicates analysis results for end milling operation.

Table 5.2 ANSYS Results for End Milling.

| Bearing Arrangements | End milling deformations (µm) |
|---|---|
| 1 | 5.84 |
| 2 | 8.48 |
| 3 | 7.60 |

**5.3.2 Face milling results:** For this analysis we need to take tangential load as per the analytical calculations **P=1486 N.** and apply at the nose of the spindle assembly.

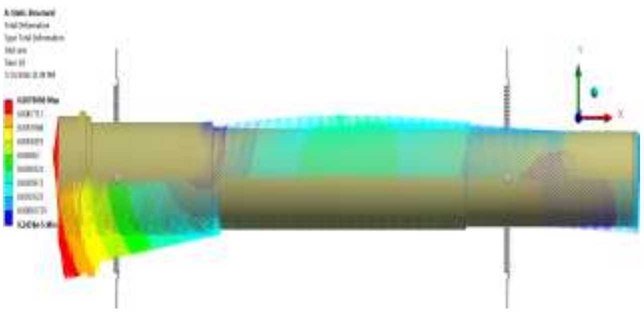

Fig. 5.11 Nose Deflection for Bearing Arrangement 1

Fig. 5.12 Nose Deflection for Bearing Arrangement 2

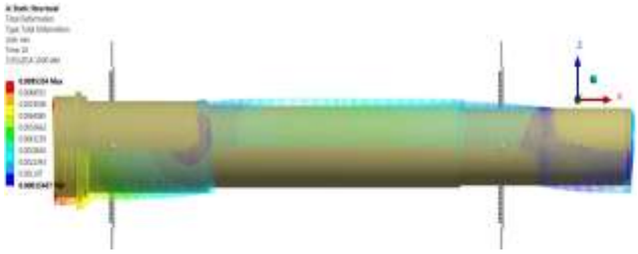

Fig. 5.13 Nose Deflection for Bearing Arrangement3

Similar to end milling, face milling analysis also carried out in ANSYS work bench, here tangential force is applied to the spindle assembly. The spring is used for analysis purpose instead of bearing with same stiffness. Here four springs is attached to the spindle for each bearing component and bearing stiffness is equally divided into four springs. In these bearing arrangements red colour shows maximum deformation of the spindle nose, blue colour shows minimum deformation. The arrangement 1 is having least deformation/deflection so that it high stiffer than the other bearing arrangements. The table indicates analysis results for face milling spindle.

Table 5.3 ANSYS Result of Face Milling.

| Bearing Arrangements | Face milling deformations (µm) |
|---|---|
| 1 | 05.80 |
| 2 | 10.65 |
| 3 | 09.53 |

**5.4 Comparison of Analytical and Numerical Spindle Deflection and spindle stiffness**

The spindle nose deflection/deformation is mainly depends on the stiffness of the bearing used. If the stiffness is high deflection is less and vice versa. Here we have analysed three different bearing arrangements with different stiffness and span length as discussed in the previous pages. The comparison of analytical/theoretical and numerical/ANSYS results for end milling and face milling is as shown in the table. Comparison results are obtained as below.

Table 5.4 Spindle Nose Deflection Comparison Results.

| Operations | Bearing arrangements | Theoretical | ANSYS |
|---|---|---|---|
| End milling | 1 | 4.10 | 5.84 |
| | 2 | 7.21 | 8.48 |
| | 3 | 6.12 | 7.50 |
| Face milling | 1 | 4.96 | 5.80 |
| | 2 | 8.90 | 10.65 |
| | 3 | 7.50 | 9.53 |

Rigidity of the spindle is mainly depends on stiffness of the spindle, Spindle stiffness results for end milling and face milling, based on deflection and load can be calculated as follows. The spindle stiffness can be calculated by using following equation 5.3(a).

$$\text{Spindle stiffness (K)} = \frac{\text{Load applied to spindle}}{\text{Deflection of the spindle}}$$

That is $\mathbf{K = \frac{P}{\delta}}$ **(N/μm)**……… 5.3(a)

Table 5.5 Spindle Stiffness Results

| Operations | Bearing Arrangements | Theoretical | ANSYS |
|---|---|---|---|
| End milling stiffness (N/μm) | 1 | 297.07 | 210.05 |
| | 2 | 168.93 | 143.63 |
| | 3 | 199.01 | 160.26 |
| Face milling stiffness ( N/μm) | 1 | 299.60 | 256.20 |
| | 2 | 165.85 | 139.53 |
| | 3 | 198.13 | 155.92 |

## 6. MODAL ANALYSIS OF SPINDLE ASSEMBLY

### 6.1 Introduction

Modal analysis is the process of determining all the modal parameters, which are then sufficient for formulating a mathematical dynamic model. Most practical noise and vibration issues are related to resonance phenomena, where the operational strengths energize one or more vibration modes. The vibration modes represent the inherent dynamic properties of a free structure (means, there are no forces acting on any structure or component). Modes are associated with structural resonance, resonance is defined as when the external force acting on a body then, external excitation frequency is equal to natural frequency of the system or model is known as resonance. Resonant vibration is caused by collaboration between the inertial and flexible or elastic properties of the materials inside a structure. A typical and valuable method for doing this is to define its modes of vibration. Every mode is characterized by a modal frequency, modal damping, and a mode shapes. Whenever a system is subjected to an external force and then set it to free, it undergoes natural vibrations or free vibrations. The frequency of these free vibrations is called as "natural frequency". At resonant conditions there is a maximum energy transfer between the system and the surrounding. Modes shapes are inherent properties of the material or structure. Modes are mainly depends on material properties such as density, stiffness, damping constants, inertia effect and gyroscopic effect, etc. mode shapes are unique.

### 6.2 Finite Element Model

Finite Element Method is a numerical technique for finding approximate solutions to constrained model. The model which is creates in modeling software for analysis along with given constraints to check the behavior of the object is known as finite element model. The model divided into number of equal parts or finite elements is called meshing (descritization). For this analysis we have taken four/ten nodes tetrahedron mesh elements of element size 5 mm and fixed-fixed constrained as shown in fig. 6.1. Body to ground i.e. fixed-fixed spring elements are used for analysis purpose and no loads are applied. The shape of the tetrahedron mesh elements are shown previous chapter fig. 5.5.

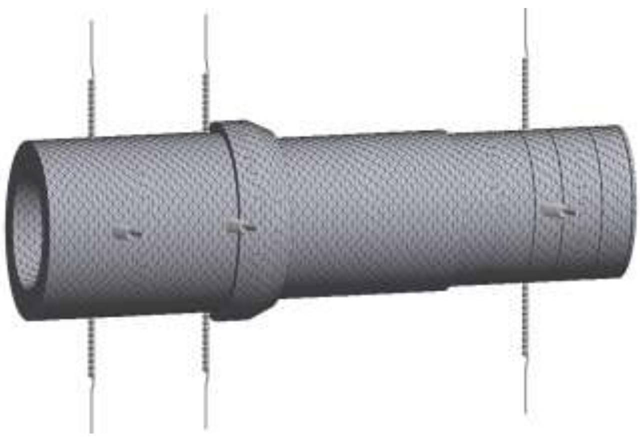

Fig. 6.1 Finite Element Model for Modal Analysis

### 6.3 Modal Analysis Results

- **Bearing arrangement 1**

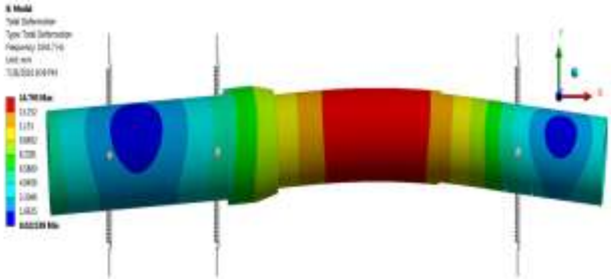

Fig. 6.2 Mode Shape 1 at Natural Frequency 1043.7Hz

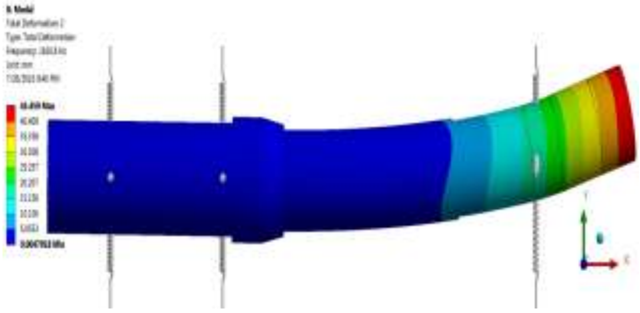

Fig. 6.3 Mode Shape 2 at Natural Frequency 1830.8 Hz

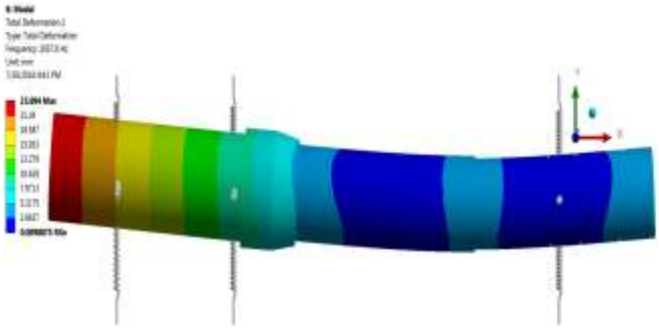

Fig. 6.4 Mode Shape 3 at Natural Frequency 2657.8 Hz

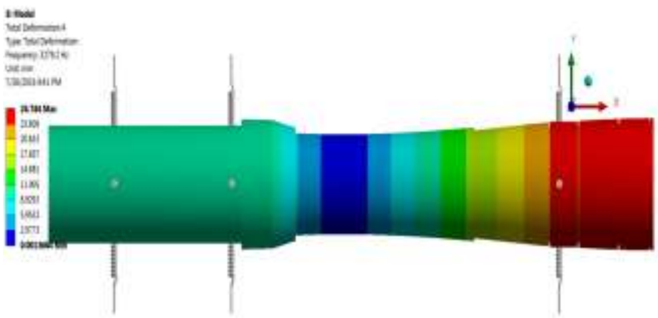

Fig. 6.5 Mode Shape 4 at Natural Frequency 3279.2 Hz

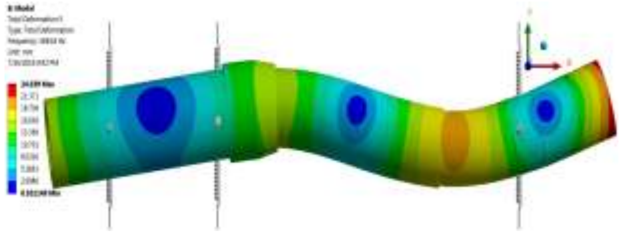

Fig. 6.6 Mode Shape 5 at Natural Frequency 3484.8 Hz

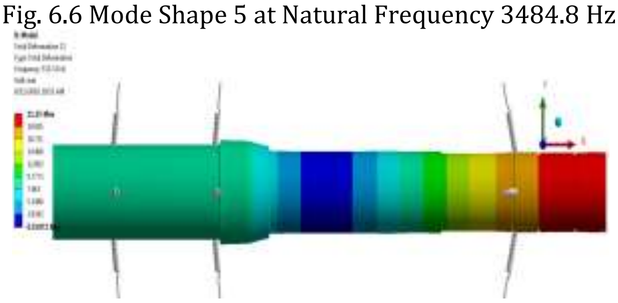

Fig. 6.7 Mode Shape 6 at Natural Frequency 5337.4 Hz

The above diagrams shows different mode shapes at different natural frequencies of spindle shaft for bearing arrangement 1, mode shapes mainly defends on density of the material, boundary conditions, stiffness of the shaft etc. the following table shows natural frequencies and mode shapes of the spindle.

Table 6.1 Bearing Arrangement-1 Mode Shapes and Natural Frequencies

| Number of Modes | Natural Frequency (Hz) | Mode Shapes |
|---|---|---|
| 1 | 1043.7 | Bending |
| 2 | 1830.8 | Bending |
| 3 | 2657.8 | Bending |
| 4 | 3279.2 | Torsion |
| 5 | 3484.8 | Buckling |
| 6 | 5337.4 | Elongation |

- **Bearing arrangement 2**

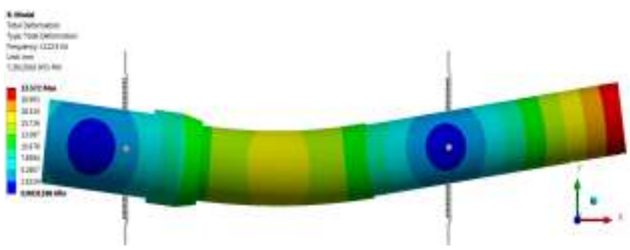

Fig. 6.8 Mode Shape 1 at Natural Frequency 1122.4 Hz

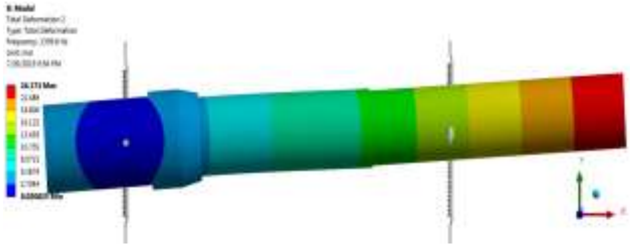

Fig. 6.9 Mode Shape 2 at Natural Frequency 1359.8 Hz

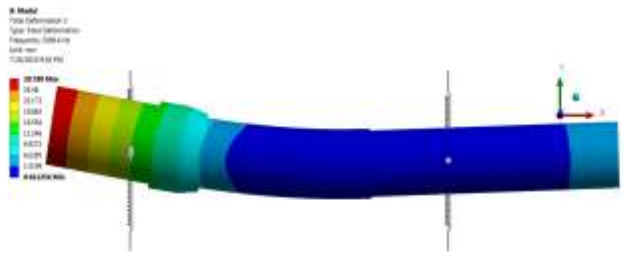

Fig. 6.10 Mode Shape 3 at Natural Frequency 2006.4 Hz

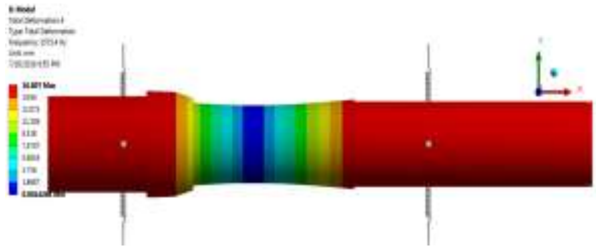

Fig. 6.11Mode Shape 4 at Natural Frequency 3575.4 Hz

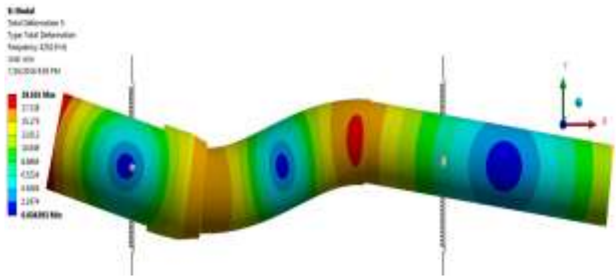

Fig. 6.12 Mode Shape 5 at Natural Frequency 4252.9 Hz

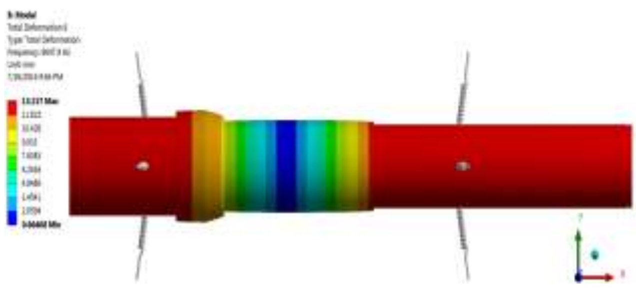

Fig. 6.13 Mode Shape 6 at Natural Frequency 6047.9 Hz

Similarly, bearing arrangement 2 was carried out, and we have got six natural frequencies and its shapes are listed in table 6.2.

Table 6.2 Bearing Arrangement-2 Mode Shapes and Natural Frequencies

| Mode No. | Natural frequency (Hz) | Mode Shapes |
|---|---|---|
| 1 | 1122.4 | Bending |
| 2 | 1359.8 | Bending |
| 3 | 2006.4 | Bending |
| 4 | 3575.4 | Torsion |
| 5 | 4252.9 | Buckling |
| 6 | 6047.9 | Compression |

❖ **Bearing arrangement 3**

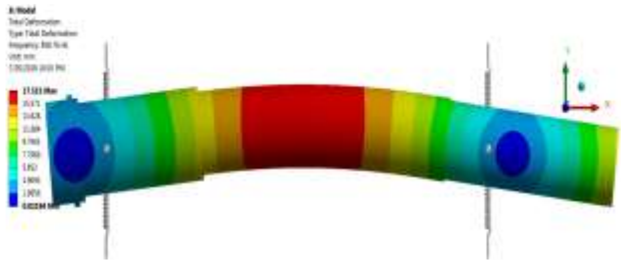

Fig. 6.14 Mode Shape 1 at Natural Frequency 858.76 Hz

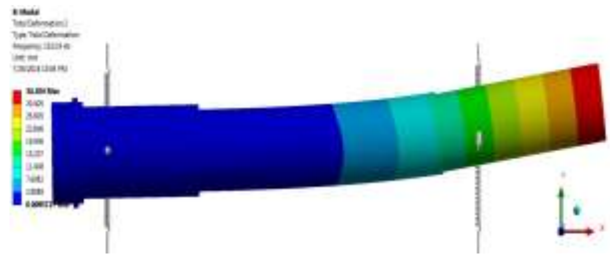

Fig. 6.15 Mode Shape 2 at Natural Frequency 1512.9 Hz

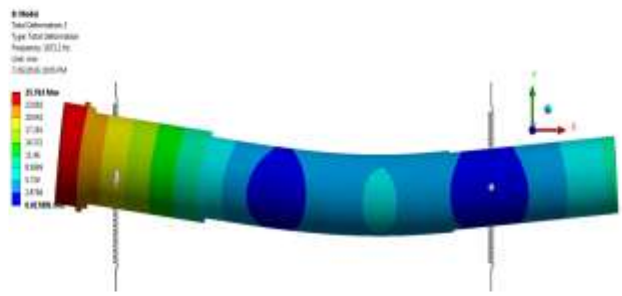

Fig. 6.16 Mode Shape 3 at Natural Frequency 1872.2 Hz

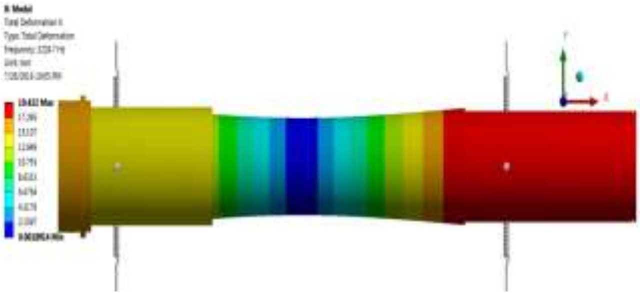

Fig. 6.17 Mode Shape 4 at Natural Frequency 3224.7 Hz

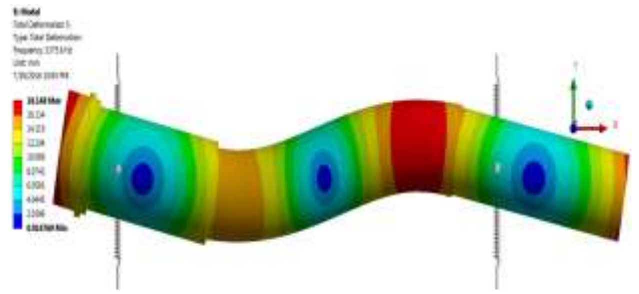

Fig. 6.18 Mode Shape 5 at Natural Frequency 3375.6 Hz

Fig. 6.19 Mode Shape 6 at Natural Frequency 5488.9 Hz

Similarly, for bearing arrangement 3, six numbers of mode shapes and natural frequencies are taken and values are tabulated in the table.

Table 6.3 Bearing Arrangement-3 Mode Shapes and Natural Frequencies

| Mode no. | Natural frequency (Hz) | Mode Shapes |
|---|---|---|
| 1 | 858.76 | Bending |
| 2 | 1512.9 | Bending |
| 3 | 1872.2 | Bending |
| 4 | 3224.2 | Torsion |
| 5 | 3375.6 | Buckling |
| 6 | 5488.9 | Elongation |

From the above modal analysis results, no one frequency is near to the natural frequency of the system so that resonance will not be occurs. The consolidate frequency table 6.4 is given below.

Table 6.4 Results Summary of Natural Frequencies

| Mode No. | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Bearing arrangement-1 (Hz) | 1043 | 1830 | 2657 | 3279 | 3484 | 5337 |
| Bearing arrangement-2 (Hz) | 1122 | 1360 | 2006 | 3575 | 4253 | 6048 |
| Bearing arrangement-3 (Hz) | 858 | 1513 | 1872 | 3224 | 3375 | 5489 |

### 6.4 Final Assembled Milling Spindle

After completion of spindle nose deflection and dynamic modal analysis, we known that bearing arrangement-1 is least deflection and high stiffness spindle, so that CNC milling spindle assembly is completed/prepared using bearing arrangement-1, as shown in the fig.6.20

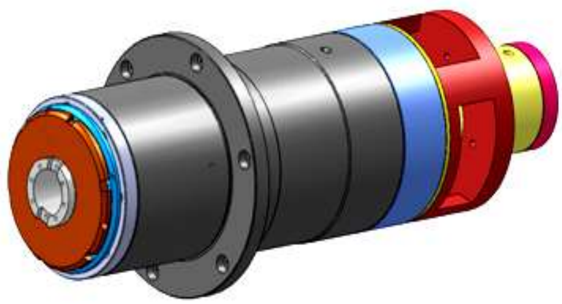

Fig. 6.20 Assembled View of BT – 40 Spindle

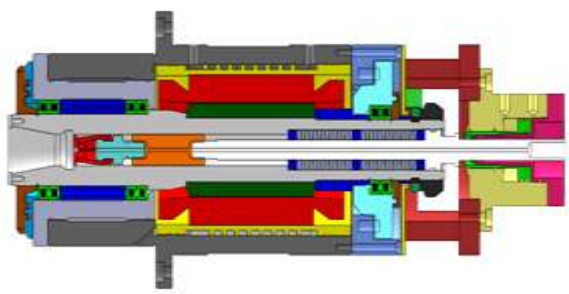

Fig. 6.21 Cross Sectional View of Assembled BT – 40 Spindle

# 7. CONCLUSION

## 7.1 Conclusions:

The below conclusions can be taken from this project work: The BT-40 CNC Milling spindle has been designed to satisfy the required specifications. The design is optimized by proper selection of spindle components and simplifying the design of the spindle parts from the machining and analysis point of view. The spindle deflection is calculated theoretically for three different bearing arrangements with different bearing stiffness and span length. The static stiffness analysis of spindle assembly is carried out using ANSYS work bench 14.5 to find out the spindle nose deflection using spring element. There is a good correlation between the theoretical and ANSYS spindle deflection results. The deflection values obtained for bearing arrangement 1 with NSK bearing configuration is lower compared to that of other bearing arrangements. The bearings with higher stiffness should be located at the front to minimize the deflection, hence bearing arrangement1 is optimized. The deflection and stiffness values obtained for this configuration is given in the following Table 7.1.

Table 7.1 Deflection and Stiffness Values for Optimized Configuration

| Theoretical | | ANSYS | |
|---|---|---|---|
| Deflection (μm) | Stiffness (N/μm) | Deflection (μm) | Stiffness (N/μm) |
| 4.10 | 297.07 | 5.84 | 210.05 |

The modal analysis is carried out using ANSYS work bench-14.5 software to obtain the natural frequencies and the mode shapes for the optimized design, these are the frequency values which should be avoided during operation which will cause resonance.

Table 7.2 Modal Analysis Results for Optimized Configuration

| Mode | 01 | 02 | 03 | 04 | 05 | 06 |
|---|---|---|---|---|---|---|
| Frequency (Hz) | 1040 | 1831 | 2658 | 3279 | 3484 | 5337 |

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