ANALYSIS OF LATHE SPINDLE USING ANSYS

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ABSTRACT

The machine tool spindle provides the relative motion between the cutting tool and the workpiece which is necessary to perform a material removal operation. In turning, it is the physical link between the machine tool structure and the workpiece, while in processes like milling, drilling or grinding, it links the structure and the cutting tool. This work deals with design and analysis of Lathe Spindle in which the material used for the spindle is alloy steel. The spindle is supported by two bearings separated by different spans. Bearings consist of balls with certain stiffness, which acts as a cushioning effect for the spindle and hence can be considered as a spring in the software (Ansys) for analysis.

Keywords: Spindle, Bearings, FEM

1. INTRODUCTION

The basic structure of a machine tool consists of base and column arrangement which serves as a balancing support for the entire machine. Here, depending on the machining process, the tool is fixed in the tool post and the work piece is held on the chuck of a typical lathe structure. The relative motion is achieved by movements parallel to the three spatial axes. This is achieved by means of linear guide ways and bearings, axial movements along the screws, rack and pinion arrangements etc. The machine is built of heavy steel and iron parts. The base of the machine is rigid and usually is of cast iron

2. STATEMENT OF THE PROBLEM

The present machine tool structures consist of spindle system which play a vital role in the quality of the final product and enhances the overall productivity and efficiency of the machine tool itself.The spindle of a machine, which is rotated by the spindle motor holds the cutting tool, which cuts the work piece, so that it influences the accuracy directly. For instance in machining centers the spindle system could alone account for about 30 to 40 % of stiffness at cutting point between tool and work piece.

Applying the FEM method for modeling a system comprised of an inner and outer race of the bearing and the rolling element between them is considered for the work. This method is being developed for computing the stiffness of the roller bearings and ball bearing. Correct modeling of bearing is one of important conditions for obtaining correct results of simulation; hence it will reduce the number of laboratory tests. It is expected that the results obtained with use of the model developed will allow the determination of stiffness of the bearing.

In the discussed calculation method, it was assumed that the bearing stiffness depends on the stiffness of the outer race, inner race, each rolling element being under load. This stiffness is nonlinear functions of load imposed on the rolling element, thereby dependent on its position in relation to the direction of the force. Such approach allows taking into consideration resulting from the changeable bearing stiffness caused by both, changes in the position of rolling elements in relation to the direction of the force and damage and wear and tear of the interacting bearing elements

In the present work the static analysis and dynamic analysis is carried out for spindle supported on two front bearings and a rear bearing. For the analysis of structure the minimum deflection region near loading point is considered. Bearing stiffness value will be calculated by an iteration procedure, life of bearings is calculated using numerical relations and this paper also describes the modifications required to correct the bearing span so as to get more stability for spindle.

The following are the project specifications for the spindle structure

Angular contact ball bearing (α=15)

Cutting conditions: speed (n) =1200 rpm

Power (p) =2 kW

Cutting diameter (φ) =25.4mm

Tangential cutting force (F_r) =2000N

Axial cutting force (F_a) = 700N

Load is at 110 mm away from spindle nose.

3. OBJECTIVE OF THE WORK

The objective of this work is

1) To carry out the static and dynamic analysis of spindle structure by optimizing the bearing span.
2) Predicting life of bearings.
3) Carrying out the Von mises stress & harmonic analysis.
4) Developing the Mode Shapes to check the vibration characteristics.

4. ELEMENTS USED

4.1 BEAM188 Element Description

BEAM188 is suitable for analyzing slender to moderately stubby/thick beam structures. This element is based on Timoshenko beam theory. Shear deformation effects are included.

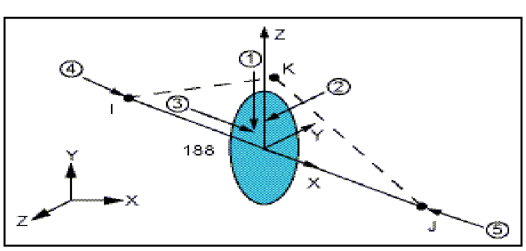


Fig.1 Beam188 geometry

4.2 COMBIN 14 Spring Damper:-

COMBIN-14 has longitudinal or torsional capability in one, two, or three dimensional applications. The longitudinal spring-damper option is a uniaxial tension-compression element with up to three degrees of freedom at each node: translations in the nodal x, y, and z directions. No bending or torsion is considered. The torsional spring-damper option is a purely rotational element with three degrees of freedom at each node: rotations about the nodal x, y, and z axes. No bending or axial loads are considered.

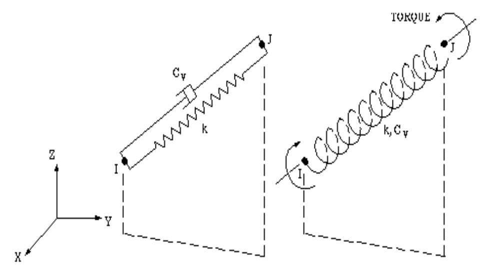


Fig.2 COMBIN14 Spring-Damper

4.2 Specifications

1. **Spindle specifications- steel**
   - Young's modulus – 2.1e5 N/mm^2
   - Poisson's ratio - 0.3
   - Density – 7.8 e-9 kg/mm^3
   - Cutting Diameter – 25.4 mm
   - Speed=1200 rpm
   - Front bearings – Ø65 × Ø 120 mm
   - Rear bearing – Ø 65 × Ø 120 mm

2. **Boundary conditions**

   Load at 110 mm away from spindle nose in radial direction = 2000N

   Load at 110 mm away from spindle nose in Axial direction = 700N

3. **Bearing stiffness**

   Front bearing

   K_y = C13 = - C19 = C64 = 84986
   K_z = C24 = -C30 = C69 = 84986
   K_x = C1=-C7=C58= 2511.5

   Rear bearing

   K_y = C13 = -C19 = C64 = 69732
   K_z = C24 = -C30 = C69 = 69732
   K_x = C1=-C7=C58= 2509.25

ISSN 2229-5518

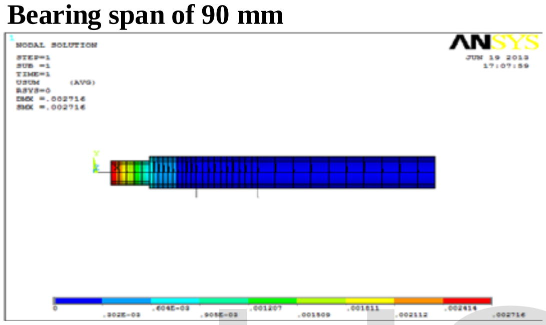
5. RESULTS AND DISCUSIONS

5.1 Results of Analysis

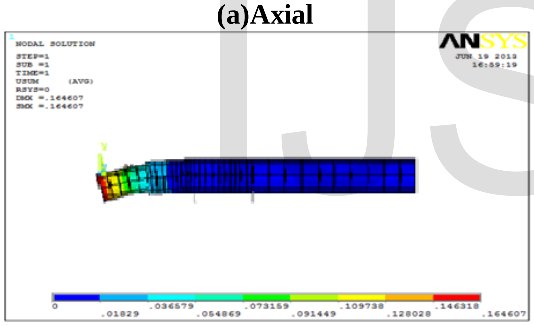
The Finite Element Model with the boundary conditions is submitted in Ansys for Static Analysis. As for output request FEM model results of spindle are obtained which include mainly deflection. For checking the deflections of spindle in axial and radial direction go for sub grid solution under query results mode.

5.1.1 Deflection of spindle for different bearing span

1] Bearing span of 90 mm



(a)Axial



(b)Radial

Fig 3 Showing deflection for 90mm in (a) Axial and (b) Radial direction

Table 1 Stiffness and deflection values for bearing span of 90 mm

Bearing span 90 mm	Front bearing stiffness N/mm	Rear bearing stiffness N/mm	Radial deflection mm	Axial deflection mm
First iteration	153320	152389	0.15012	0.00313
Second iteration	200100	199639	0.16143	0.00293
Third iteration	200243	198834	0.16460	0.00271

Iterative procedure:-

a] First iteration values for front bearing

1] Rolling element load

$$Q = \frac{5F_r}{iZ\cos\alpha}$$

$$Q = \frac{5\times 203.87}{1\times 17\cos 15}$$

Q=62.08 Kgf

Radial deformation of angular contact ball bearings

$$\delta_r = \frac{0.002}{\cos\alpha}\sqrt[3]{\frac{Q^2}{D_w}}$$

$$\delta_r = \frac{0.002}{\cos 15}\sqrt[3]{\frac{62.08^2}{15.4}}$$

δ_r=0.0130446 mm

Rolling element load

$$Q = \frac{5F_a}{iZ\sin\alpha}$$

$$Q = \frac{5\times 71.35}{1\times 17\sin 15}$$

Q=81.081 Kgf

Axial deformation of angular contact ball bearings

$$\delta_a = \frac{0.002}{\sin\alpha}\sqrt[3]{\frac{Q^2}{D_w}}$$

$$\delta_a = \frac{0.002}{\sin 15}\sqrt[3]{\frac{81.081^2}{15.4}}$$

δ_a =0.0581812 mm

Radial and axial stiffness

$$K_r = \frac{P}{\delta_r}$$

$$K_r = \frac{2000}{0.0130446}$$

K_r=153320.14 N/mm

$$K_a = \frac{P}{\delta_a}$$

$$K_a = \frac{700}{0.0581812}$$

K_a=12,031.3778 N/mm

b] Second iteration

3] Rolling element load in radial direction

$$Q = \frac{5F_r}{iZ\cos\alpha}$$

$$Q = \frac{5 \times 421.24}{1 \times 17\cos 15}$$

Q=128.26 Kgf

Radial deformation of angular contact ball bearings

$$\delta_r = \frac{0.002}{\cos\alpha}\sqrt[3]{\frac{Q^2}{D_w}}$$

$$\delta_r = \frac{0.002}{\cos 15}\sqrt[3]{\frac{128.26^2}{15.4}}$$

δ_r=0.0211604 mm

4] Rolling element load in axial direction

$$Q = \frac{5F_a}{iZ\sin\alpha}$$

$$Q = \frac{5 \times 35.68}{1 \times 17\sin 15}$$

Q=40.55

Axial deformation of angular contact ball bearings

$$\delta_a = \frac{0.002}{\sin\alpha}\sqrt[3]{\frac{Q^2}{D_w}}$$

$$\delta_a = \frac{0.002}{\sin 15}\sqrt[3]{\frac{40.55^2}{15.4}}$$

δ_a =0.079824 mm

5] Radial and axial stiffness

$$K_r = \frac{P}{\delta_r}$$

$$K_r = \frac{4212.4}{0.0211004}$$

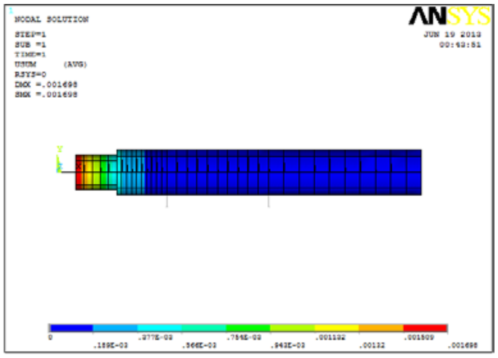
K_r=200590.47 N/mm

$$K_a = \frac{P}{\delta_a}$$

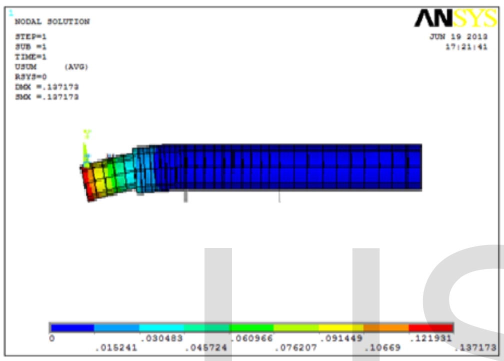
$$K_a = \frac{350.07}{0.079824}$$

$K_a = 4385.52$ N/mm

2] Bearing span of 140 mm



(a)Axial



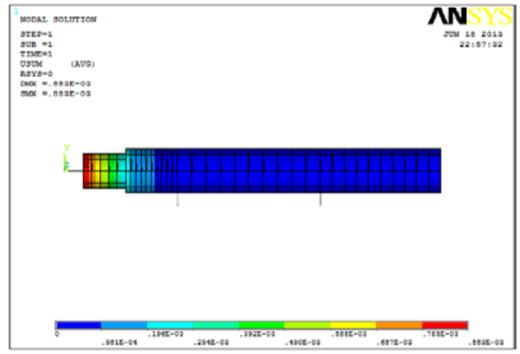
(b)Radial

Fig 4 Showing deflection for 140 mm in(a) Axial and (b) Radial direction

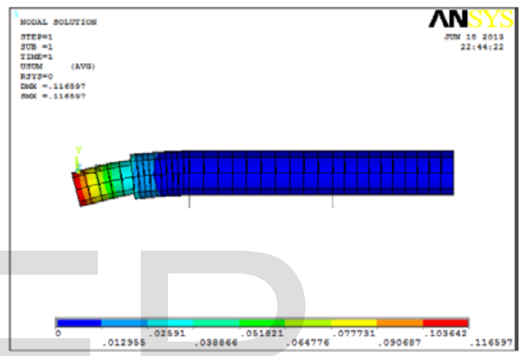
Table.2 Stiffness and deflection values for bearing span of 140 mm

Bearing span 140 mm	Front bearing stiffnes s N/mm	Rear bearing stiffnes s N/mm	Radial deflecti on mm	Axial deflecti on Mm
First iteration	153320	151994	0.13015	0.00136
Second iteration	183241	181699	0.13513	0.00150
Third Iteratio n	186352	181710	0.13717	0.00169

3] Bearing span of 190 mm



(a)Axial



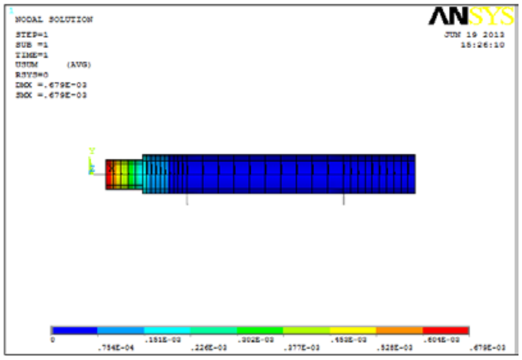
(b) Radial

Fig 5 Showing deflection for 190 mm in (a) Axial and (b) Radial direction

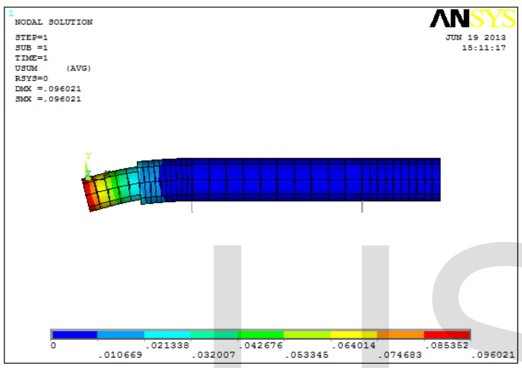
Table.3 Stiffness and deflection values for bearing span of 190 mm

Bearing span 190 mm	Front bearing stiffnes s N/mm	Rear bearing stiffnes s N/mm	Radial deflecti on mm	Axial deflecti on mm
First iteration	153320	150068	0.10021	0.00065
Second iteration	173321	164969	0.11102	0.00071
Third iteration	175232	162920	0.11659	0.00088

4] Bearing span of 240 mm



(a)Axial



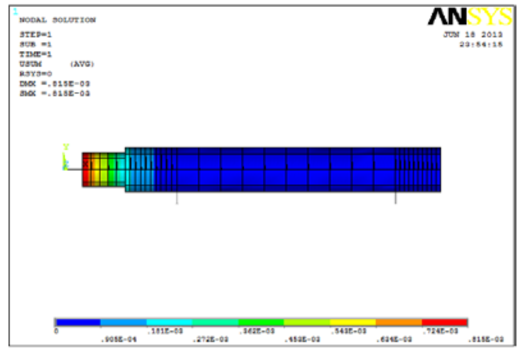
(b)Radial

Fig 6 Showing deflection for 240 mm in (a) Axial and (b) Radial direction

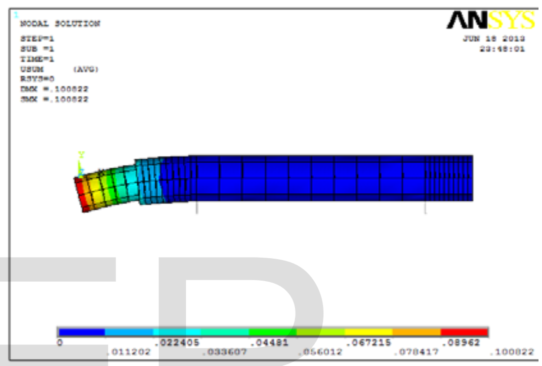
Table 4. Stiffness and deflection values for bearing span of 240 mm

Bearing span 240 mm	Front bearing stiffness N/mm	Rear bearing stiffness N/mm	Radial deflection mm	Axial deflection mm
First iteration	153320	147977	0.07521	0.00031
Second iteration	164321	148759	0.09152	0.00052
Third iteratio	166207	147865	0.09602	0.00067

5] Bearing span of 290 mm



(a)Axial



(b)Radial

Fig 7 Showing deflection for 290 mm in (a) Axial and (b) Radial direction

Table.5 Stiffness and deflection values for bearing span of 290 mm

Bearing span 290 mm	Front bearing stiffness N/mm	Rear bearing stiffness N/mm	Radial deflection mm	Axial deflection mm
First iteration	153320	145858	0.09605	0.00061
Second iteration	161239	141338	0.09921	0.00070
Third iteratio	163962	140994	0.10082	0.00081

Table 6 Deflection and spring values for different bearing span

Sl. No.	Bearing span Mm	Stiffness at 1st Bearing N/mm	Stiffness at 2nd Bearing N/mm	Radial deflection mm	Axial deflection mm
1	90	200243	198834	0.16460	0.00271
2	140	186352	181710	0.13717	0.00169
3	190	175232	162920	0.11659	0.00088
4	240	166207	147865	0.09602	0.00067
5	290	163962	140994	0.10082	0.00081

By looking in to Table 6 it can be decided that the optimum bearing span having length 240 mm has got minimum deflection.

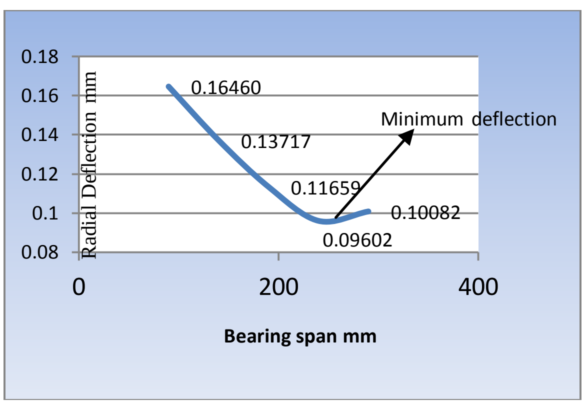


Fig 8 Graph of bearing span v/s Radial deflection

5.2 Modal Analysis

Table 7 Details of modal analysis

MODAL ANALYSIS				
SET	FREQUENCY	LOAD STEP	SUB STEP	CUMULATIVE
1	30.11	1	1	1
2	30.11	1	2	2
3	34.42	1	3	3
4	34.42	1	4	4
5	38.85	1	5	5
6	38.85	1	6	6

Modal analysis is used to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. It can also be a starting point for another, more detailed, dynamic analysis, such as transient dynamic analysis, a harmonic response analysis, or a spectrum analysis.

The spindle model is now subjected to modal analysis. The Table 6.9 shows the frequencies at which the model resonates. Machine frequency when maximum rpm condition is taken is 20 Hz. The mode shapes obtained must be away from this frequency range so as to make that resonance does not occur.

ISSN 2229-5518

5.3 Mode Shapes

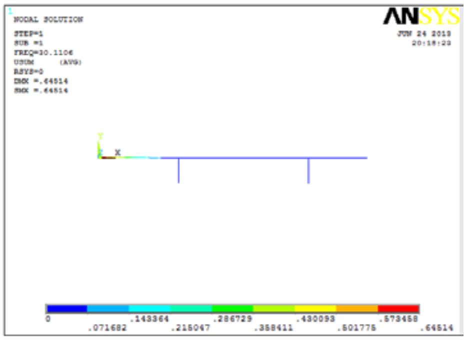


Fig 9 Mode shape 1

Fig 9 shows the mode shape 1 at freq 30.11 Hz. It can see from the Fig that the mode shape is only in vertical direction at loading point

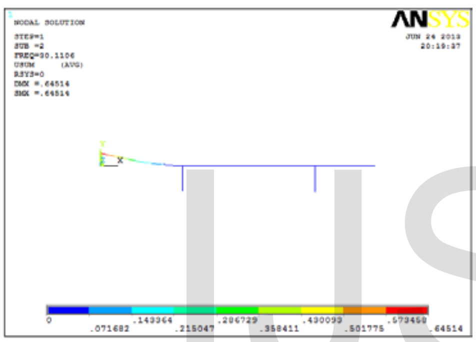


Fig 10 Mode shape 2

Fig 10 shows the mode shape 2 at freq 30.11 Hz It can be seen that there is bending mode in vertical direction at loading point

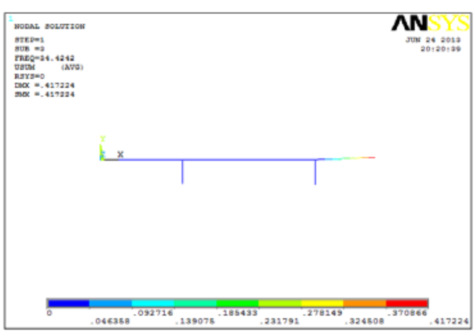


Fig 11 Mode shape 3

Fig 11 shows the mode shape 3 at freq 34.42 Hz. It can be seen that there is bending mode in vertical direction on other direction of the loading point.

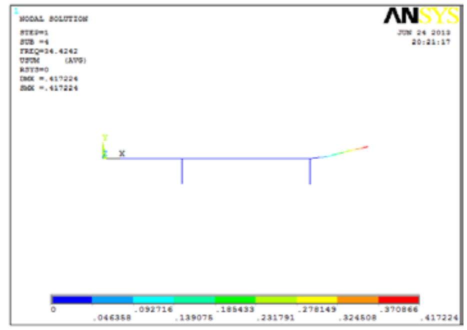


Fig 12 Mode shape 4

Fig 12 shows the mode shape 4 at freq 34.42 Hz. It can be seen that there is bending mode in vertical direction on other direction of the loading point.

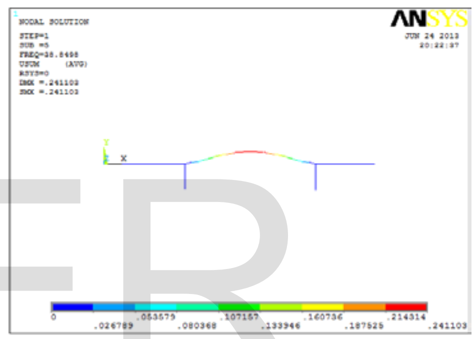


Fig 13 Mode shape 5

Fig 13 shows the mode shape 5 at freq 38.85 Hz. Here in this mode bending mode is in horizontal direction at the center of the structure.

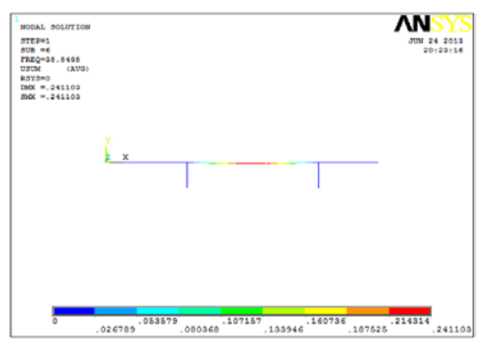


Fig 14 Mode shape 6

Fig 14 shows the mode shape 6 at freq 38.85 Hz. Here in this mode bending mode is in horizontal direction at the center of the structure.

5.4 Von mises Stress Analysis

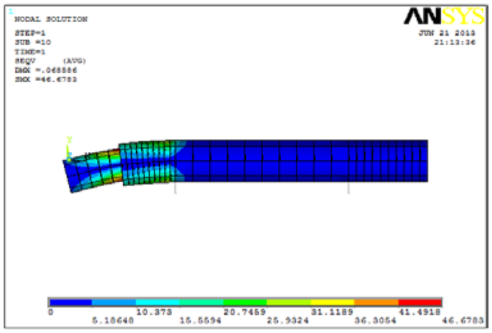


Fig 15 von mises stress for 240 mm span

According to the von Mises's theory, a ductile solid will yield when the distortion energy density reaches a critical value for that material. Since this should be true for uniaxial stress state also, the critical value of the distortional energy can be estimated from the uniaxial test. At the instance of yielding in a uniaxial tensile test, the state of stress in terms of principal stress is given by: $\sigma 1 = \sigma Y$ (yield stress) and $\sigma 2 = \sigma 3 = 0$
Thus, the energy density is the critical value of the distortional energy density for the material. Then according to von Mises's failure criterion, the material under multi-axial loading will yield when the distortional energy is equal to or greater than the critical value for the material. Thus, the distortion energy theory can be stated that material yields when the von Mises stress exceeds the yield stress obtained in a uniaxial tensile test.

For bearing of 240 mm span which is optimized value voin misses stresses are calculated using the ansys software and the results are as shown in the fig 4.14. So allowable stress for steel is much higher than the results found in the software. So material is well within safe condition.

5.5 Harmonic Analysis

Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (*harmonically*) with time. The idea is to calculate the structure's response at several frequencies and obtain a graph of some response quantity (usually displacements) versus frequency. "Peak" responses are then identified on the graph and stresses reviewed at those peak frequencies.

It is required to carry out harmonic analysis to confirm whether the spindle is resonating within the operating range of the spindle. The operating range is the range of spindle speeds which varies up to 1200 RPM (1200/60= 20 Hz).

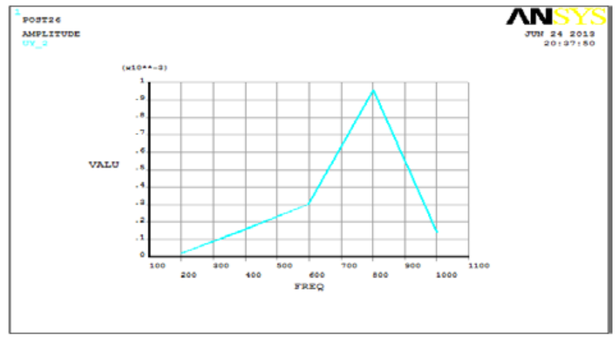


Fig 16 Amplitude V/s Frequency

The Fig4.14 shows that the maximum amplitude is 0.0095 mm. Maximum amplitude is of our concern and it is not in the range of working frequency. So thus it has negligible effect.

6. CONCLUSION

Its application has been carried out for practical All Geared lathe with 2kW spindle and spindle speed of 1200 rpm. The application showed the utility of the above procedure for modeling spindle systems and as an application optimum bearing span of bearings has been calculated, natural frequencies and mode shapes of the spindle have been calculated.

1] Optimum bearing span of 240 mm is considered for fixing distance between front and rear bearings.
2] The vibration analysis showed that no resonance occurs predicted results are verified with analytical models.
3] Von mises Stress Analysis shows that deflection is on the safer side.

From results and discussions deflection value for the 240 mm bearing span is 0.096021 mm. Radial and axial stiffness values are calculated for front bearings while for rear bearing only radial stiffness is calculated since if rear bearing is constrained there will be thermal deformation which leads to buckling therefore there will negligible axial stiffness.

The simplified approach has been developed for analysis of spindle supported on two bearings is developed an iterative procedure is presented for computing reactions of bearings under load and also for calculating bearings stiffness in general 3 to 4 iterations are sufficient to get convergent results. The procedure makes use of beam188 element for spindle and Combin14 Spring Damper element for springs.

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