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Static and Dynamic Analysis of Lathe Spindle using ANSYS

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Abstract

There are various parameters over which the performance of the spindle depends. This paper deals with the different parameters of the spindle to find the optimized model from the existing one. Based on the FEM, the static and dynamic analysis has been performed. In the static analysis it considers parameters such as material of the spindle, bearing span, Morse taper. The dynamic analysis has been performed corresponding to the optimized static model and modal shapes are obtained. With the help of Campbell diagram, critical velocity has been found out. The harmonic response is performed to check the design and with the required speed transient analysis is performed. All the analysis is performed in ANSYS Workbench 18'.

Keywords: spindle; bearing; FEM; static analysis; dynamic analysis; Campbell diagram; harmonic response; transient analysis

INTRODUCTION

Spindle tool is an important mechanical constituent which provides relative motion between work piece and cutting tool during material removal operation. The dynamic analysis of spindle bearing system plays a crucial role as it directly affects the machining, productivity as well as the quality of the product. The size of the spindle shaft and diameter used for the bearings are of various types.

C.W. Lin et al. [1] in the paper stated that the FEM analysis approach is the most popular approach for the dynamic analysis of the spindle bearing system because of its capacity to solve the complex model and boundary conditions along with less time calculation. Rishikesh B. Kamthe et al. [2] stated that they have used the Macaulay method for the calculation of deflection of the spindle. They calculated the reaction force and deflection at required points. Anand M. Sharan et al. [3] presented the effect of bearing span and stiffness on the spindle by changing the distance and stiffness. Dumitru D. Nicoara [4] has optimized the bearing locations in their paper where the deflection of the assembly is minimum . Hareesha [5] has proposed the range for the bearing span over which it should be mounted for efficient working. The range for the bearing span should be in between 70 mm to 150 mm.

Yunsong Li et al. [6] has presented in his paper that bearing provides the rotor support in radial, axial and angular stiffness to the rotor. Deping Liu et al. [7] presented that spindle stiffness is highly related with the load capacity and the vibration resistance.

Yuzhong Cao et al. [8] proposed the parameters such as spindle shaft dimensions, housing and bearing locations for designing a spindle.

Osamu Maeda et al. [9] has presented the effect of mesh while doing the operation. The more the density of mesh is, the more is calculation accuracy. It is necessary to go through trial and error method to get the best mesh density.

Santosh Arali et al. [10] has showed that spindle performance affects the machinability, immobility and the particularity, also the obtained stress analysis is less than the given yield strength so the deformation of the spindle is less and neglected.

Satish Kumbar et al. [11] has performed the modal analysis for the spindle bearing system where as Dr. S Shivakumar et al. [12] has done the modal as well as harmonic analysis for the spindle bearing system. Asim Kutlu [13] checked the critical speed of the system by Campbell diagram. It represents a system's response as a function of its oscillation. Jianjun Feng et al. [14] has done the optimization of the bearing system for the grinding machine. The results are validated by comparing with theoretical measurement which is good agreement for the acceptance limits.

Under this paper the objective of the paper is to carry out static analysis of the spindle by optimizing different parameters like material, bearing span and morse taper. The dynamic analysis has been performed corresponding to the optimized static model and modal shapes are obtained. With the help of Campbell diagram, critical velocity has been found out. The harmonic analysis is performed to check the design and with the required speed transient analysis is performed at 6000 rpm. In this paper all the analysis is done by ANSYS work bench'18.

EQUATION OF MOTIONS

The model for analysing a spindle bearing system of lathe machine is depicted in the Figure 1. The spindle is designed to operate on the speed of 6000 rpm. Based on the FEM the work has been done. The general equation of motion is shown below:

Optimum bearing span

There is an effect of bearing span in the deformation of the spindle. The optimum bearing span plays a crucial role for the efficient working of the system. The equation for the optimum bearing span is expressed below: -

$$L_0 = [6EI_L \left(\frac{1}{SA} + \frac{1}{SB} \right) + \left(\frac{6EIL}{a \times SA} \right) Q]^{1/3}$$

Where (L_0) = Bearing span in mm

(Q) = Iterative determination of L_0=4a

(E) = Young's modulus (N/mm^2)

(a) = Length of overhanging portion from the support of bearing.

(S) = P/δ = Overall stiffness, kgf/mm

(S_A) = Radial stiffness near the load point, kgf/mm

(S_B) = Radial stiffness away from the load point, kgf/mm

(I_L) = Moment of Inertia of spindle between bearings (mm^4)

Boundary conditions

For modal analysis it is required to establish the dynamic equation of the spindle bearing system. The dynamic equation of spindle bearing system is shown below: -

i) Equation of forced vibration

$$m\ddot{x} + c\dot{x} + kx = F(x),$$

where $F(x)= F_o\cos\omega t$

Where (m) is the mass of the system (kg), (c) is damping coefficient (Nsec/mm), (k) is the stiffness (N/mm) , F(x) is the excitation vector , F_o is the constant excitation force and (x) is the displacement vector (mm).

From modal analysis the natural frequency of the spindle is found by the material properties, structure, and the damping will have very little effect on the natural frequency of spindle bearing system, therefore F(x) = 0,

ii) Equation of free vibration

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

Assuming the spindle bearing system in the simple harmonic vibration,

iii) $x(t) = x_o\sin(\omega t + \phi)$

Where: x_0 – Amplitude , ω – Angular frequency and ϕ – Phase angle. Substituting (iii) in (ii):

iv) $(k - \omega^2 M)\phi = 0$

The Formula (iv) is used to calculate the modes of spindle bearing system where ϕ_i (i = 1,2,3, …., n).

FINITE ELEMENT MODEL OF SPINDLE

The model for the spindle with bearing mounted on them is shown in Figure 1.

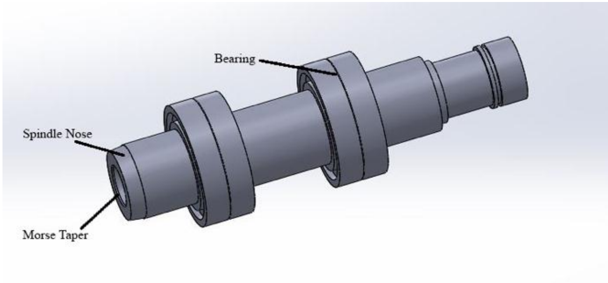

Figure 1: Spindle structure

Radial vibration plays a crucial role on dynamic performance of the spindle. Using solid works the model has been established and the model is imported to ANSYS work bench in which the static, dynamic, harmonic and transient analysis has been performed. The spindle nose taper of the spindle is taken 7.5 degree. The spindle structure comprises of two sets of angular contact ball bearing at the both ends. This type of bearing helps in reducing the axial run out of the spindle and improves the axial stiffness.

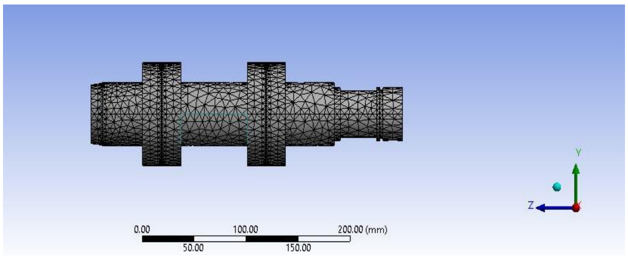

Figure 2: Spindle bearing finite element mode

SPECIFICATION

1. The properties of the materials which are used is listed in the below table:

Table 1: Mechanical and geometrical properties

Material Properties	20MnCr5	Ck45 Steel	Alloy 4140
Young's Modulus (GPa)	200	206	200
Poisson's Ratio	0.28	0.29	0.27
Density (g/cm^3)	7.85	7.87	7.85
Ultimate strength (N/mm^2)	530	515	655
Yield strength (MPa)	385	485	415

2. Bearing stiffness and dimensions [15]

   1) Front Bearing: - 159 N/µm, 55*90*18, SKF7011/ACD

   2) Rear Bearing: - 67N/µm, 55*90*18, SKF7011/C

3. Boundary conditions
   1) Load acting on the spindle nose in radial direction is 2000N.
   2) Distance of cutting point to spindle tip is 110 mm.
   3) Moment acting on the spindle tip is 2.2 *10^5 N mm

RESULTS AND DISCUSSION

Static Analysis

During operation of spindle it experiences different types of forces and varied moments. The applications of these load results in high deformation and stress. This affects the spindle and causes spindle failure. Hence, the design of the spindle should be analysed statistically as well as dynamically with verification.

Static analysis depends on the type of material it is made of, type of bearings, number of bearings, bearing span, Morse taper, spindle nose and so on. Here the spindle is mounted with two sets of bearings at both ends. The bearing used is angular contact ball bearing which constraint the shaft in all directions.

The load acts in both directions i.e. axial and lateral. In most cases the lateral load is more than the axial load and the axial stiffness is more than the bending stiffness. Hence it is important to do lateral loading of static analysis. Therefore we have considered the directional deformation in the Y-axis for doing the analysis.

Effect of material on spindle deformation

The analysis for different types of material with a bearing span of 97 mm, MT 4 and given boundary conditions is presented below: -

1) 20MnCr5

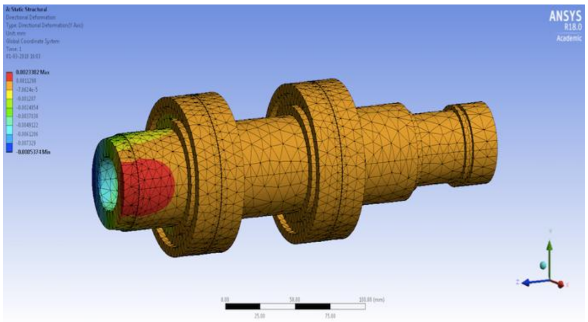

Figure 3: 20MnCr5 deformation

2) Ck45 steel

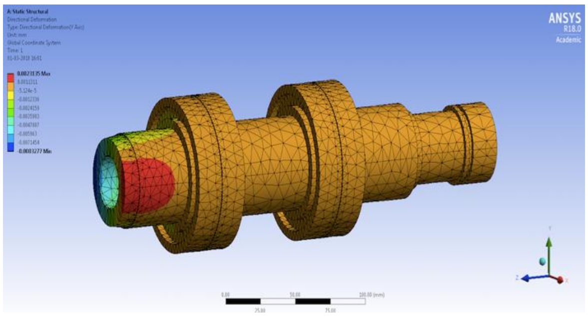

Figure 4: Ck45 Steel deformation

3) Alloy 4140 steel

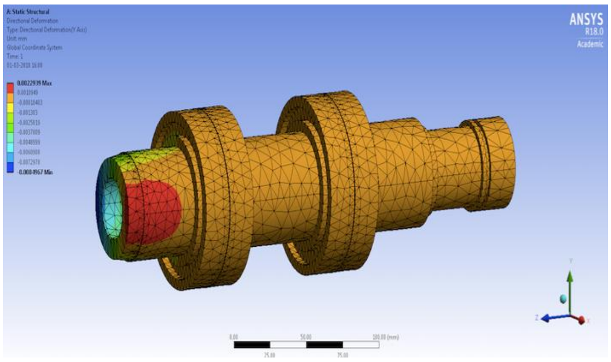

Figure 5: Alloy 4140 deformation

From the figures, the deformation for the 20MnCr5, Ck45 and Alloy 4140 steel are 2.3382 μm, 2.3135 μm and 2.2939 μm respectively. Hence it is clear that the best material is Alloy 4140 steel.

Effect of bearing span on spindle deformation

The analysis of bearing span with material as Alloy 4140 steel, MT 4 and given boundary conditions is presented below: -

1) Bearing span of 93mm

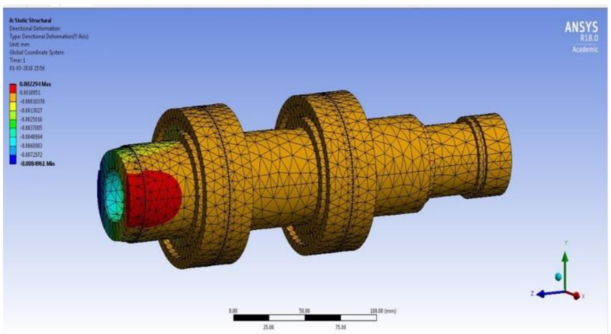

Figure 6: 93mm bearing span deformation

2) Bearing span of 97mm

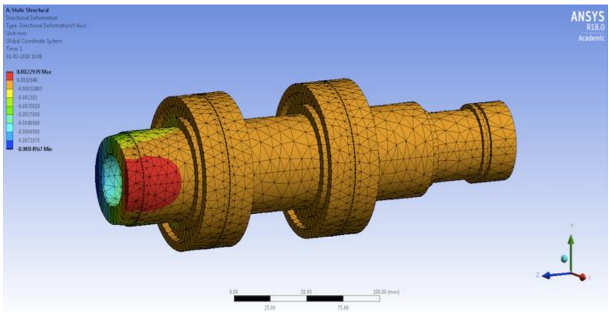

Figure 7: 97 mm bearing span deformation

3) Bearing span of 101mm

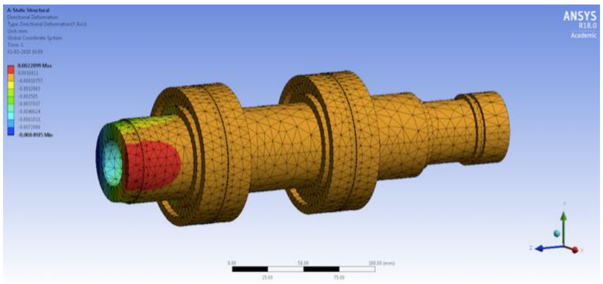

Figure 8: 101 mm bearing span deformation

4) Bearing span of 105mm

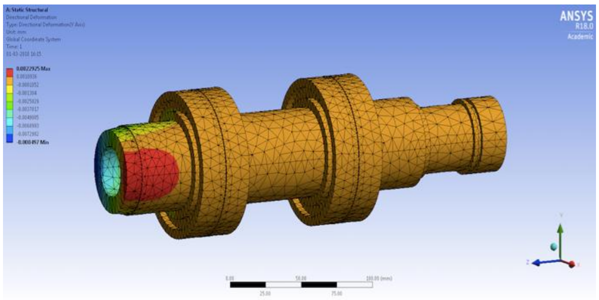

Figure 9: 105 mm bearing span deformation

From the figure, the deformation for the bearing span of 93mm, 97mm, 101mm and 105mm are 2.294µm, 2.2939µm, 2.2899µm and 2.2925µm respectively. Hence the optimum bearing span for the spindle structure is 101mm. The figure-10 shows the variation of deformation with different bearing span.

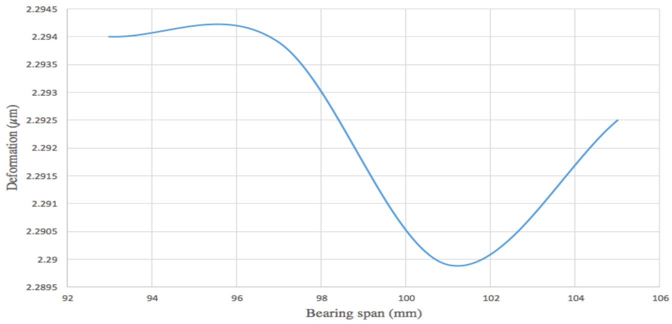

Figure 10: Bearing Span(mm) vs Deformation(µm)

The Table-2 shows the combined effect of materials and bearing spans with the deformation corresponding to them: -

Table 2: Comparison of bearing span, material and deformation

Bearing span (mm)	Spindle Material	Deformation (µm)
93	• 20MnCr5 • Ck45 • Alloy 4140 Steel	• 2.3383 • 2.3136 • 2.294
97	• 20MnCr5 • Ck45 • Alloy 4140 Steel	• 2.3382 • 2.3135 • 2.2939
101	• 20MnCr5 • Ck45 • Alloy 4140 Steel	• 2.3341 • 2.3095 • 2.2899
105	• 20MnCr5 • Ck45 • Alloy 4140 Steel	• 2.3368 • 2.3121 • 2.2925

The result for the different spindle material and different bearing span has been mentioned in the Table 2. It is evident that for all the three materials the deformation is minimum at the bearing span of 101 mm. It is also that if the bearing span is less or more then the reaction forces and the moment produced by the bearing is more which results in the more deformation in the spindle structure.

Effect of Morse taper on spindle deformation

After performing the bearing span and material selection for the spindle, now the best structure is found out by varying the Morse taper. The deformation value for the different Morse taper is shown in the Table-3 : -

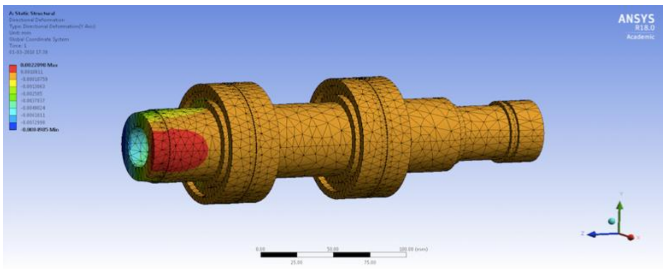

Figure 11: Morse taper deformation of MT6

Table 3: Comparison of Morse taper

Morse Taper	Deformation(μm)
0	2.2938
1	2.2899
2	2.2908
3	2.2899
4	2.2899
5	2.2899
6	2.2898
7	2.2898

Although the deformation in both MT 6 and MT 7 is same, the total deformation of MT 6 is 8.4955μm and the total deformation of MT7 is 8.4985μm. Therefore, the optimized Morse taper is MT6.

Hence, we statically get the most optimized structure i.e. Alloy 4140 steel with a bearing span of 101 and MT 6 taper with spindle nose of 7.5 degree.

The deformation in MT6 taper is the least with deformation of 2.2898μm.

After that we have to verify that whether our design is safe or not by doing the dynamic analysis from which we can know the natural frequencies of the system and from the harmonic analysis we can check that whether the system is safe or not at that frequencies.

Dynamic Analysis

Modal Analysis

The modal analysis of the spindle is performed in ANSYS Workbench 18'. The first six mode shapes are shown below and the frequencies corresponding to the all 6 modes are mentioned in the Table-4: -

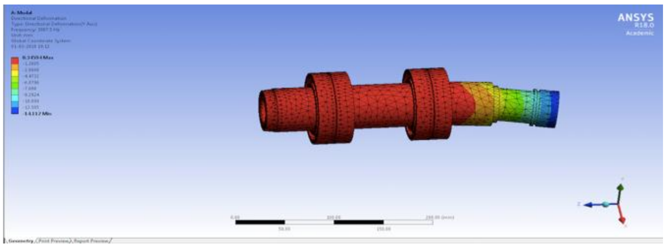

Figure 12: Mode shape 1

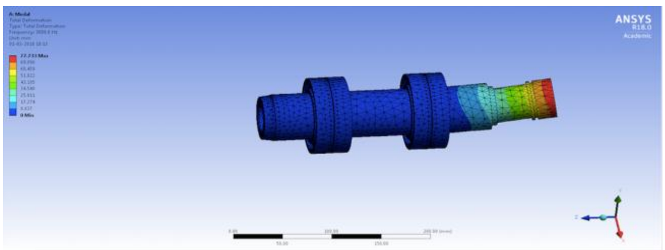

Figure 13: Mode shape 2

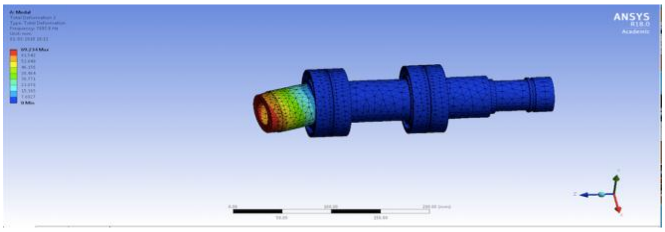

Figure 14: Mode shape 3

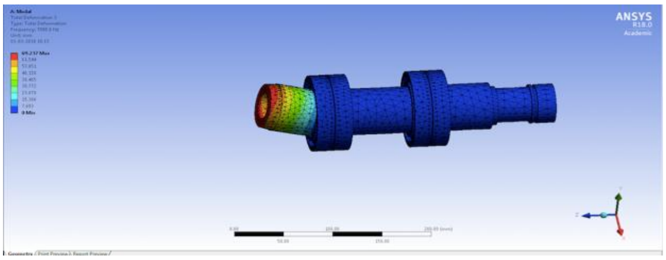

Figure 15: Mode shape 4

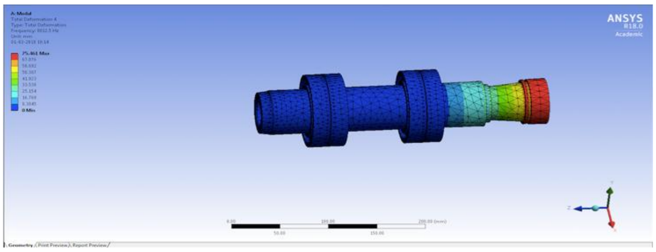

Figure 16: Mode shape 5

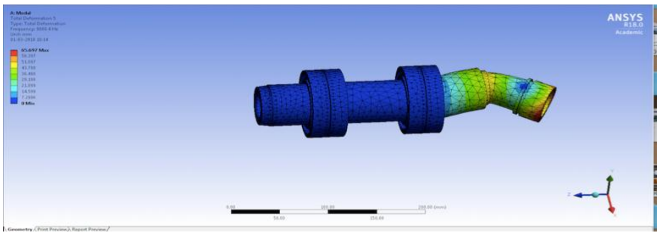

Figure 17: Mode shape 6

Table 4: Modal frequencies (Hz)

Mode	Frequency[Hz]
1	3087.5
2	3088.6
3	7897.9
4	7898.9
5	8012.5
6	9080.4

Therefore, from modal analysis it is clear that the critical frequencies are well beyond the working speed ranges. Guo Bai, Zheng and Pan (2013) [16] states that the maximum rotating speed cannot exceed 75% of its critical speed. For first two modes the working speed are 185280 rpm, which is much more than the working range of spindle, 0-6000 rpm. It is expected for the lathe spindles because of their high rigidity and low working speed ranges. Thus, it proves that the design of spindle can avoid the region within the speed range.

Critical Speed and Campbell Diagram

The working speed range of spindle is (0-6000) rpm and from modal analysis, the critical velocity is much more than it. The analysis is conducted between 0 to 2,00,000 rpm by varying multiple load steps to find the critical speeds with the help of Campbell diagram, it is also possible to obtain the natural frequency with in the speed range.

In Figure 17 and Table 5, the critical speeds of the spindle are given. The natural frequencies are always above 3000 Hz for the spindle speed range of (0-6000) rpm.

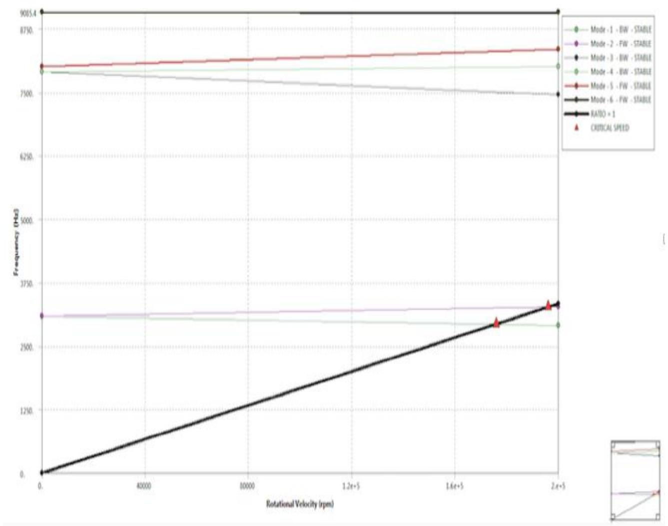

Figure 17: Frequency(Hz) vs Rotational velocity(rpm)

Table 5: Critical speed and frequency

Mode	Whirl Direction	Mode Stability	Critical Speed
1	BW	STABLE	1.7603e+005 rpm
2	FW	STABLE	1.9611e+005 rpm

For mode 1 the critical speed is 1.7603e+005 rpm

For mode 2 the critical speed is 1.9611e+005 rpm

Harmonic Response

It is important to verify our design at the critical frequencies by taking the frequency range of critical speed. At resonance the system reaches the maximum amplitude and results in vibrational disturbance in the system. Hence, the resonance should be stopped for proper functioning. The analysis is done on ANSYS Workbench 18' considering 2500 Hz to 9100 Hz with an incremental of 10Hz.

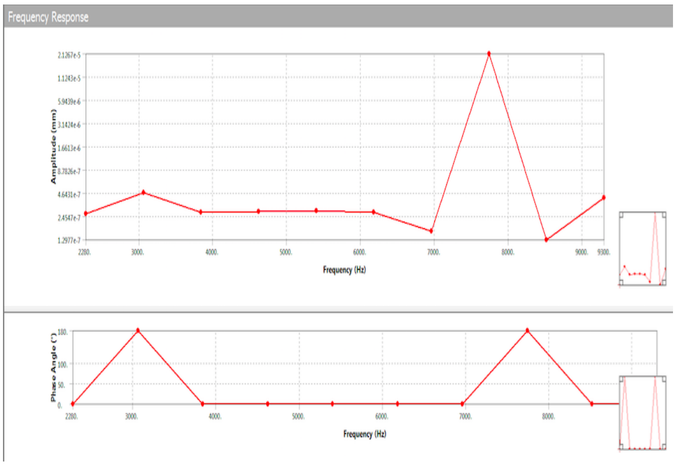

Figure 18: Frequency response

Figure 18 depicts the amplitude response with different frequencies. Form the figure it is clear that the spindle does not reach high deformation at the critical frequencies 2912.9 Hz and 3272.1 Hz is around $4.643*10^{-7}$ mm. Also at 7500 Hz the maximum amplitude is $7*10^{-4}$ mm which is very less.

Transient Analysis

The analysis is done in unsteady condition. A technique helps in finding the dynamic response of a system under a time dependent loads. Transient analysis is a function of time. Under this a force of 2000N is applied with moment of $2.2*10^5$ N mm and a rotational velocity of 6000 rpm. ANSYS'18 is used for transient analysis.

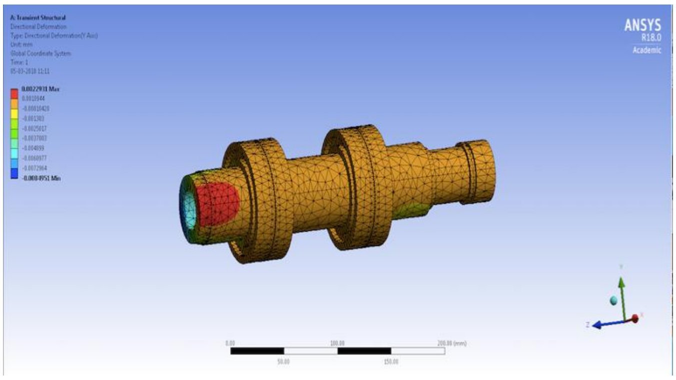

Figure 19: Transient analysis deformation (µm)

Figure 19 shows the deformation of the system in transient analysis and it is clear from the figure that the deformation is 2.2931 µm which is less and good for the working condition.

CONCLUSION

The analysis of spindle is performed on ANSYS Workbench 18'. Spindle structure and bearing plays an important role in failure of spindle bearing system. Here, we proposed the optimization of spindle considering different parameters. The result of static and dynamic analysis are in safe range.

The spindle consists of two sets of angular contact bearings with grease lubrication. The optimized material is Alloy 4140 steel with bearing span of 101mm, MT 6, and spindle nose taper of 7.5 degree. The deflection of the system is 2.2899 μm. According to the modal analysis the frequencies are 3087.5 Hz, 3088.6 Hz, 7897.9 Hz, 7898.9 Hz, 8012.5Hz and 9080.4 Hz and from the Campbell diagram the critical velocities and frequency found are 185274 rpm, 185286 rpm and 2912.9 Hz, 3272.1 Hz respectively. Also from harmonic analysis the critical frequencies are 2912.9 Hz and 3272.1 Hz. The transient analysis is performed at 6000 rpm and the deformation is 2.2931 μm.

The ANSYS work bench is used by the designers as tools, which increases the product quality, reduces cost and consumes less time in design and development.

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