International Journal of Innovative Research in Advanced Engineering (IJIRAE) ISSN: 2349-2163
Volume 1 Issue 6 (July 2014) http://ijirae.com

ANALYSIS OF CNC LATHE SPINDLE FOR MAXIMUM CUTTING FORCE CONDITION AND BEARING LIFE

Santosh Arali[1], V.V.Kulkarni[2]
[1] PG Student, Machine Design, Gogte Institute of Technology Belgaum
[2] Asst.Professor, Department of Mechanical Engineering, Gogte Institute of Technology Belgaum

Abstract*-The present CNC machine structures consist of spindle system which plays a relating to the quality of the final product and the overall productivity and efficiency of the machine tool itself. The spindle of a CNC lathe machine, which is rotated by the main motor, holds the cutting tool, which cuts the work piece, so that the cutting forces are generated which effects the spindle accuracy directly. The forces which are affecting the CNC machine tool spindle are tangential force (F_t), feed force (F_c), radial force (F_r) and will be estimated. Based on maximum cutting force incurred the analysis will be carried out. The main objective is to find the static, fatigue analysis of spindle structure for maximum cutting force condition and predicting life of bearings. From static analysis stress and deformation of the spindle can be found. Stress obtained from the stress analysis is less than the yield strength of the material and deformation of the spindle is very less which can be neglected. Equivalent alternating stress, factor of safety and life of the spindle is found by fatigue analysis and which results are closely matches with the analytical value.

KEY WORDS: CNC, Spindle, Bearings, Cutting Forces, Static and Fatigue analysis, Code Generation.

I. INTRODUCTION

Machine tool spindle is the most important mechanical component in removing metal during machining operations. Spindle is a rotating axis of the machine, which frequently used has a shaft at its heart. The shaft itself is called spindle, it is including three bearing in the front and two bearings in the rear of the spindle. Machine tool spindles lead to unstable chatter vibrations, cutting forces and uneven tensions in the belt and pulleys. This thesis presents static and fatigue analysis by considering cutting forces and tensions in the pulleys. CNC machining is an important technology increasing productivity and reducing production costs. Compared to conventional machine spindles, motorized spindles are equipped with built in motors introduces huge amount of heat into the spindle system as well as additional mass to the spindle shaft. Depending on the machining processes, 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 the movements parallel to the three spatial axes. This can be achieved by the, axial movements are along the screws, rack and pinion arrangements, linear guide ways and bearings etc. The machine is made up of heavy steel material and iron parts. The base of the machine is rigid and usually is of cast iron. The spindle of the machine is hollow and material is $20MnCr_5$.

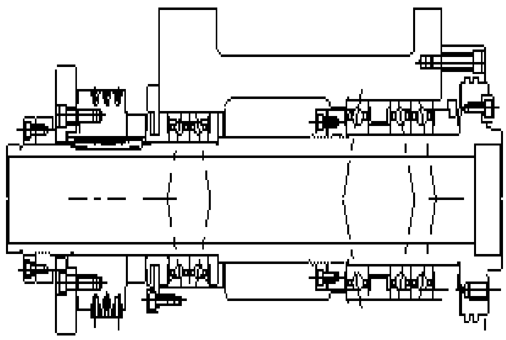


Fig. 1 Bearing arrangement in CNC lathe

II.THEORETICAL ANALYSIS

Theoretical analysis involves the calculations of cutting forces and its effect on the CNC lathe spindle, tensions in the belt, and deformations of the spindle by using Macaulay's Method and equivalent alternating stress, safety of factor by using Modified Goodman method. Following table shows the material properties of concern.

Table 1 Material properties

Physical Properties	Values
Material of the Spindle	$20MnCr_5$
Ultimate Strength (σ_{ult})	682 MPa (N/mm^2)

Yield Strength (S_{yt})	375 MPa (N/mm^2)
Young's Modulus (E)	190×10^3 N/mm^2
Poisson's Ratio (ϑ)	0.27-0.3
Density (ρ)	8030 kg/m^3

A. Total Deformation Using Macaulay's Method of spindle

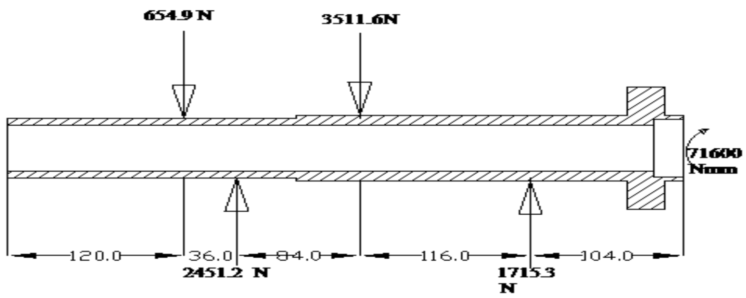


Fig 2 Loading condition of lathe spindle

$$EI\frac{d^2y}{dx^2} = -654.9(x-120)+2451.2(x-156)-3511.6(x-240)+1715.3(x-356) \quad (1)$$

Final equation to find deformation is obtained from the above generalized equation as follows:

$$EI\,y = 723902560-4607760x-654.9\frac{(x-120)3}{6}+2451.2\frac{(x-156)3}{6}-3511.6\frac{(x-240)3}{6}+1751.2\frac{(x-120)3}{6}$$

Y_{max}=0.0066908 mm

B. Fatigue Analysis

Distortion energy theory is used when the factor of safety is to be held in close values and the cause of failure of the component is being investigated. According to the distortion energy theory,

$$\sigma_{eq}=\sqrt{(\sigma_{bmax}{}^2+3(\tau_{max})^2} \quad (2)$$

Where,
σ_{eq}= Equivalent stress (MPa) , σ_{bmax}= Bending Stress(MPa) and τ_{max}= Max shear stress(MPa)

In this section, Von-Mises (equivalent) stress (σ_{eq}), factor of safety (f_s) are calculated. Initially, maximum and minimum bending moments at different points are also calculated as follows with respect to Fig 1.2.Maximumbending moment $(M_b)_{max}$= 198.91 × 10^3 N mm, maximum bending moment $(M)_{bmin}$= 48.02 × 10^3 N mm, maximum torque transmitted by the spindle $(M_t)_{max}$= 71.6 × 10^3 N mm and minimum torque transmitted by the spindle $(M_t)_{min}$= 17.9 × 10^3 N mm

I. "Mean and Amplitude Bending Moments"

Mean bending moment $[M_b]_m$ and amplitude bending moment $[M_b]_a$ are obtained by the following equations:

$$(M_b)_m = ½ ((M_b)_{max} + (M_b)_{min}) \quad (3)$$
$$=123.46 \times 10^3 \text{ N mm}$$

$$(M_b)_a = ½ ((Mb)_{max} - ((Mb)_{max} \quad (4)$$
$$=75.44 \times 10^3 \text{ N mm}$$

II. "Mean and Amplitude Stresses"

Mean Stress $[\sigma_{xm}]$ and Amplitude Stress $[\sigma_{xa}]$ are obtained by the following equations:

$$\sigma_{xm} = \frac{32(M_b)_m}{\pi(d_o^4-d_i^4)}\times d_o \quad (5)$$

Where, $(M_b)_m$ = Mean stress(MPa), d_o = Outer diameter of spindle in mm, d_i = Inner diameter of the spindle in mm.

σ_{xm} = 7.046 N/mm^2

σ_{xa} = $\frac{32(M_b)_a}{\pi(d_o^4-d_i^4)} \times d_o$ (6)

σ_{xa} = 4.28 N/mm^2

III.” Mean and Amplitude Shear Stresses”

Mean shear stress [τ_{xym}] and amplitude shear stress [τ_{xya}] are obtained by the following equations:

τ_{xym} = $\frac{32(M_t)_m}{\pi(d_o^4-d_i^4)} \times d_o$ (7)

τ_{xya} =1.021 N/mm^2

From Von-Mises theorem equation we know that,

σ_{eq} = $\sqrt{(\sigma_{bmax} k_f)^2 + 3(\tau_{max} k_f)^2}$ Where, k_f= Fatigue stress concentration factotr = 3 (8)

= $\sqrt{(7.046 \times 3)^2 + 3(4.08 \times 3)^2}$

σ_m = 30 N /mm^2

Since σ_m= σ_{eq} = 30 N /mm^2, that is equivalent Von-Mises stress is 30 N /mm^2

IV. “Alternating Stress”

Alternating Stress (σ_a) are obtained by the following equation:

σ_a = $\sqrt{(\sigma_{xya} \times k_f)^2 + 3(\tau_{xya} \times k_f)^2}$ (9)

= $\sqrt{(4.28 \times 3)^2 + 3(1.021 \times 3)^2}$

σ_a = 14 N / mm^2

$\tan\theta = \frac{\sigma_m}{\sigma_a}$ and also, $\tan\theta = \frac{S_a}{S_m}$ (10)

Where, S_a= Alternating strength in N/mm^2, S_m= Mean strength in N/mm^2

$\tan\theta$ = 0.4667

θ = 25.01 ≅ 25 degrees

V. “Endurance Strength”

Endurance Strength [S_e] are obtained by the following equation:

$S_e = K_a \times K_b \times K_c \times K_d \times Se' = 0.8 \times 0.75 \times 0.897 \times 0.4347 \times S_e'$ (11)

Where, S_e'= Endurance limit stress of a rotating beam specimen subjected to reversed bending stress(N/mm^2), S_e= Endurance limit stress (N/mm^2), K_a = surface finish factor, For machined or cold rolled process the value of the K_a = 0.8, K_b = size factor, diameter > 50 the value of the K_b = 0.75, K_c=reliable factor, For 90% reliability, the value of the K_c= 0.897, K_d = modifying factor K_d = modifying factor is given by $K_d = \frac{1}{K_t}$ = 0.4347, S_e'=0.5 S_{ut} (for steel) = 0.5 × 682.

S_e' = 341 N/ mm^2

S_e = 79.778 N / mm^2

From modified Goodman method,

$\frac{S_a}{S_e} + \frac{S_m}{S_{yt}}$ =1 where, Where, S_y = Yield tensile strength (N/mm^2), S_e = Modified Endurance limit (N/mm^2), put all above values, (12)

$\frac{0.48076}{79.778} + \frac{S_m}{375}$ =1

S_m =115 N / mm^2

S_a =55.28 N / mm^2

VI. "Factor of safety (f_s)"

Factor of safety (f_s) are obtained by the following equation:

$$f_s = \frac{S_a}{\sigma_a} = \frac{55.28}{14} = 3.9 \tag{13}$$

III FATIGUE AND STATIC ANALYSIS BY FEA

A. Static Analysis of the Spindle

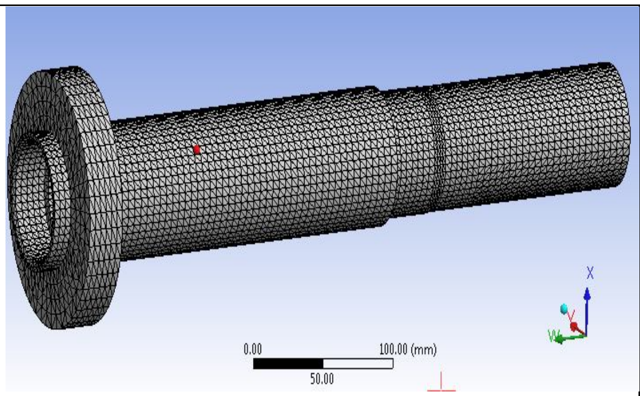


Fig. 3 Meshed spindle component in Ansys workbench

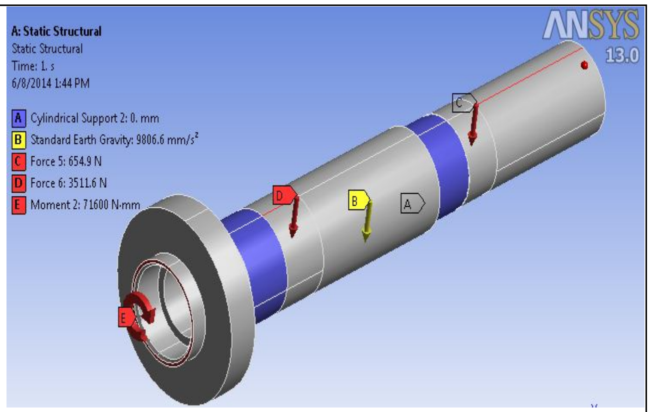


Fig. 4 Boundary conditions of spindle in Ansys workbench

Two loads for the spindle are applied on the F_z negative direction shown in fig. 4. One load is tensions in the belt and another one is maximum cutting force during machining. One torque is applied on end of the spindle in the clockwise direction. The meshed CNC spindle component is shown in above fig. 3.

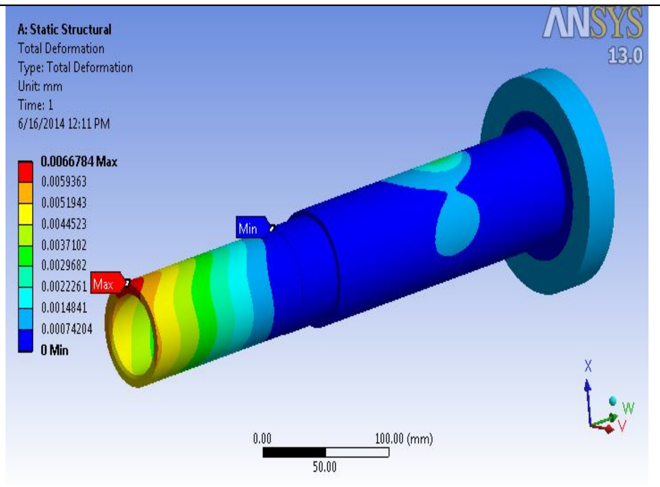


Fig.5 Total deformation of the spindle

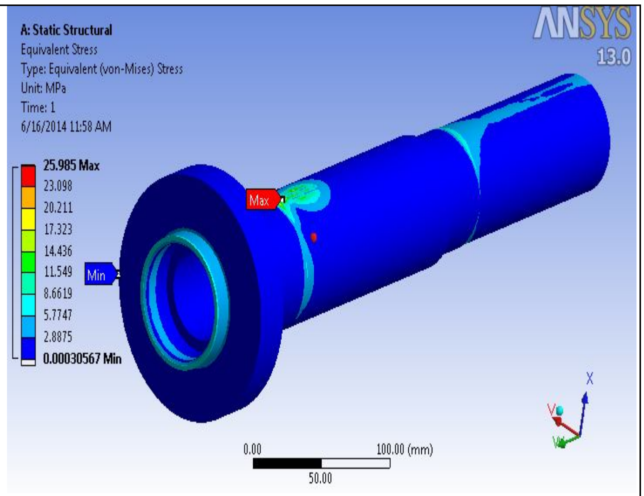


Fig.6 Von-Mises stress on spindle

Table 2 Deformation comparison between theoretical and fem

Approach	Theoretical	FEM	% Error
Maximum deflection of the spindle (in mm)	0.0066908	0.0066784	0.1853

From the above table 2 it is conclude that analytical method value is very nearer to the value of FEM, in terms of error there is only 0.1853% error between theoretical and FEM value. From the ANSYS result shown in, Fig 7 (stress) and it is confirmed that the maximum (Von-mises) stress is 25.985 N/mm^2, which is in the limit of yield strength of 20MnCr$_5$ steel 375 N/mm^2. Hence the design of the spindle is safe. From the ANSYS result shown in, Fig.8 (stress) and it is confirmed that the maximum (Von-mises) stress is 25.985 N/mm^2, which is in the limit of yield strength of $20MnCr_5$ steel 375 N/mm^2. Hence the design of the spindle is safe. It is clear from the fig.8 the fatigue factor of safety calculated from theoretical is 3.9 and by FEA approach 3.3173 again these values are approximately same.

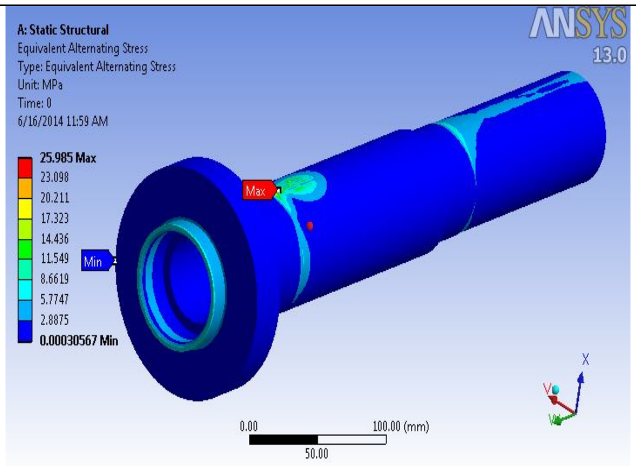


Fig.7 Equivalent alternating stress on spindle

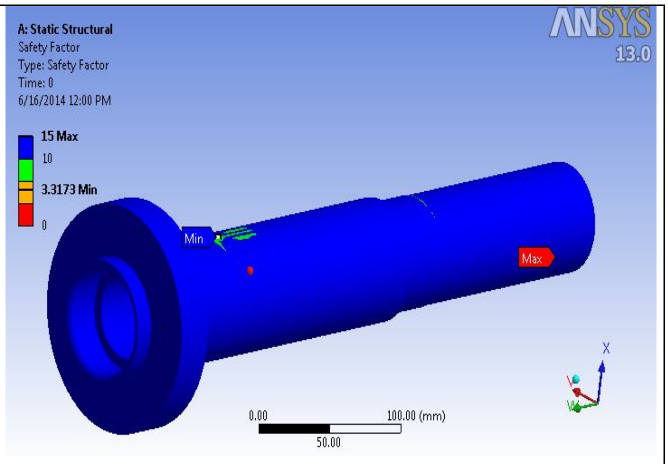


Fig.8 Factor of safety of spindle by FEA

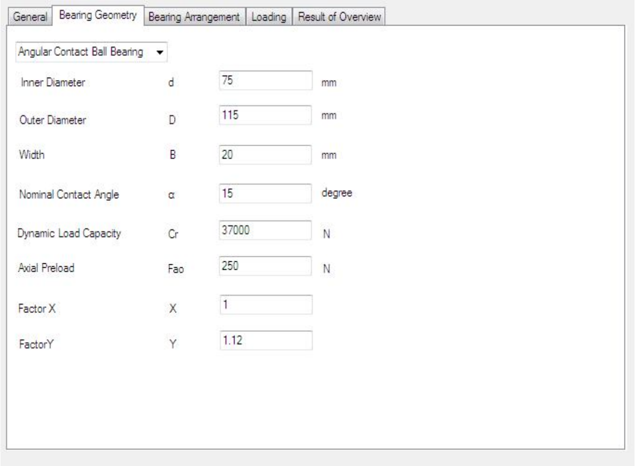
B.Comparisons of theoretical and FEM

Table 3 Fatigue factors comparisons between theoretical and fem

Theoretical Approach	FEM Approach	Percentage Error
$\sigma eq = 30\ N/mm^2$	$\sigma eq = 25.98\ N/mm^2$	13.4%
fs = 3.9	fs = 3.3173	14.94%
Infinite life	Infinite life	

From above Table 3 It is clear that, the fatigue analysis of CNC lathe spindle gives close results by analytical and FEA approach. From optimization point of view the FEA analysis is very useful tool. Also, the value of fatigue factor of safety theoretical and FEM approach are 3.9 and 3.3173. Software used for fatigue analysis CATIA V5, ANSYS13 which gives moderate results.

IV CODE GENERATION TO CALCULATE LIFE OF THE BEARING



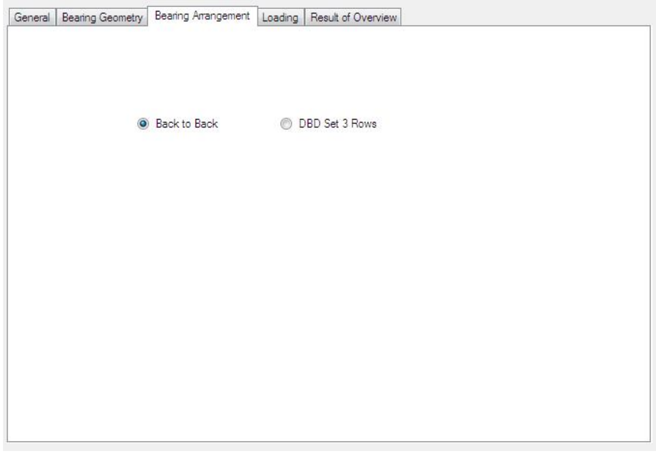


Fig. 9 Layout of bearing configuration in C-programming

Comparisons of theoretical and code generation

Fig 4 Bearing life Comparisons between theoretical and code generation

Bearing type	Theoretical Approach	C-Code Approach	% Error
Front Bearing (Life in Hours)	15,586.37	15,556.92	0.1889
Rear Bearings (Life in Hours)	9198.7	9182.3	0.1782

V CONCLUSION

Usually spindles are mounted in antifriction bearings such spindles are found in several machine structures including lathe, milling machine and all machine tools. The analysis has been carried out for CNC lathe with 7.5kW and spindle speed of 4000 rpm. The total deformation, equivalent stress and factor of safety has been calculated. The fatigue life prediction is performed based on finite element analysis and analytical Method. Using the modified Goodman method, the fatigue life of the CNC lathe spindle has been predicted. This study will help to give information for the manufacturer to improve the fatigue life of the CNC lathe shaft using FEA tools. It can help in reducing the cost, as well as understanding better in which area stresses are more.

1. Static analysis showed that deformation is very small and approximately equal to theoretical values.
2. Von-Mises equivalent stress value by analytical approach 30 N/mm^2 which are nearly same by using FEA approach having difference of 13.4% in both results which is acceptable range.
3. The fatigue factor of safety calculated from theoretical is 3.9 and by FEA approach 3.3173 again these values are approximately same.
4. The life of the front and rear bearings is 9198.7 and 15,586.3 hours.

ACKNOWLEDGEMENT

The author thanks Prof. Vivek V. Kulkarni, Asst. Professor, Department of Mechanical Engineering., KLS Gogte Institute of Technology Belgaum, for his esteemed suggestions, sincere efforts and inspiration given in completing this work. The author is also thankful to one and all to who directly or indirectly encouraged in completing this paper.

REFERENCES

[1] Osamu Maeda, Yuzhong Cao, "*Expert Spindle Design*", Proc of International Journal of Machine Tool and Manufacturer, August 24, 2007

[2] Chi-Wei Lin, Jay F Tu, Jou Kamman "*An integrated thermo mechanical dynamic model to characterize motorized machine tool spindles during very high speed rotation*", International Journal of Machine Tool and Manufacturer, 27 February 2006

[3] Dr. S. Shivakumar, Dr.Anupama N Kallol, Vishwanath Khadakbhavi, "*analysis of lathe spindle using ansys*", International Journal of Scientific & Engineering Research, Vol 4, pp 2229-5518, September-2013

[4] Mr. Sahil, Mr. Jiten Saini *"Fatigue and Modal Analysis of Connecting Rod under Different Loading Conditions"*, Proc of ISSN, pp 2277-9655

[5] Tobias Maier, Michael F. Zaeh, "Modelling of the Thermo mechanical Process Effects on Machine Tool Structures", Proc of CIRP Conference on Process Machine Interactions, 2012

[6] A.Erturk, H.N Ozguven, "Effect Analysis of bearing and interface dynamics on tool point FRF for chatter stability in machine tools using a new analytical model for spindle tool assemblies" International Journal of Machine Tool and Manufacturer, 27 February 2006