

Finite Element Analysis of Integral Shaft Bearing

Vidyasagar R. Bajaj

M.E (CAD/CAM) Student
Department of Production Engineering
Amurtvahini College of Engineering
Sangamner (India)
Savitribai Phule Pune University
Pune (India)

B. R. Borkar

Associate Professor
Department of Production Engineering
Amurtvahini College of Engineering
Sangamner (India)
Savitribai Phule Pune University
Pune (India)

Abstract: A integral shaft bearing is popular for higher specific load carrying capacity, preventing misalignment defects and eliminating the risk of undesirable distortion of the bearings, rather than conventional one. Integral shaft bearing is used to reduce rotational friction and support radial and axial loads friction in bearings which cause an increase of the temperature and Stresses inside the bearing. If the heat produced cannot be adequately removed from the bearing, the temperature might exceed a certain limit, and as a result the bearing would fail. To analyze the heat flow, temperature distribution and stresses in a bearing system, a typical integral shaft bearing and its environment has been modeled and analyzed using the famous finite element tool ANSYS. In this study we investigate structural and thermal characteristics performance of integral shaft bearing to Analyze temperature distribution and thermal elongation due to friction also its effect on bearing clearances and vice-versa.

Keywords: Integral shaft, Modeling, Meshing, Structural & Thermal Analysis.

1. INTRODUCTION

The term “rolling bearing” includes all forms of roller and ball bearing which permit rotary motion of a shaft. Normally a whole unit of bearing is sold in the market, which includes inner ring, outer ring, rolling element (balls or rollers) and the cage which separates the rolling element from each other.

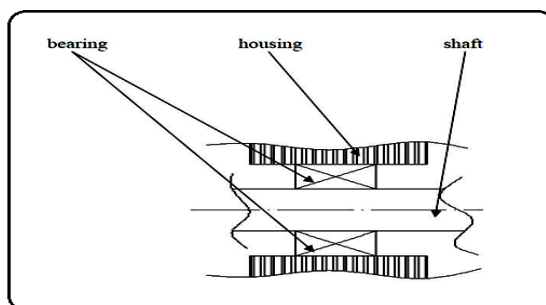


Fig1. Mechanical System

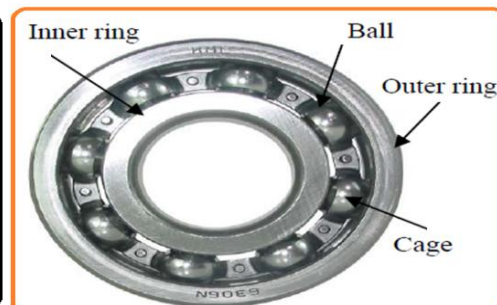


Fig2. Deep Groove Ball Bearing

Rolling bearings are high precision, low cost but commonly used in all kinds of rotary machine. It takes long time for the human being to develop the bearing from the initial idea to the modern rolling bearing. The reason why bearing is used is that first it can transfer moment or force. Secondly and maybe more important is that it can be interchanged easily and conveniently when it's broken. In the mechanical system shown in Figure-1, it is also possible to amount the shaft directly with housing. However, when this mechanism has some problem, the only possibility to recover the function of this system is to replace the housing or the shaft. From the mechanical engineer point of view, both of them are not only very expensive but also time consuming to manufacture a new housing or shaft with the same parameters. However when the bearings are used between them, the situation will be different. Normally there is no relative motion between shaft and inner ring or the outer ring with housing. So it has less possibility for the shaft or housing to be worn out. Usually the bearing first cracks and then the shaft or housing is broken. If the above situation happens it is really easy to figure it out: just buy a new bearing from the market with the same parameter and replace it. That's why bearings are so often used.

2. OBJECTIVE OF RESEARCH

The objective of the Research Project is to:

- To Design of Integral Shaft Bearing for Water Pump
- Modeling & Assembly of bearing components from 2D drawing to 3D model.
- Stress analysis and temperature distribution for Integral Shaft Bearing
- Thermal elongation of components in Integral Shaft Bearing at different temperature & its effect on bearing clearances.

3. NEED OF PROJECT

The integral shaft seals protect the bearing assembly from bearing exposed to coolant and abrasive contaminants. If probable seal failure is not detected and bearing present in pump is not replaced will cause sudden failure and possibly leads to crisis such as snapped bearing shaft. Seal leakage and bearing failure can occurs due to excessive thermal stresses because of high temperature operation of engine, vibration due to high operational speed etc. so determination of bearing failure should be detected at design stage only before manufacturing to reduce extra cost and time to market. Optimization of clearance between rolling element and outer sleeve.

4. FEA ANALYSIS AND RESULTS

4.1. Stress Analysis

Finite element method (FEM) is a numerical method for solving a differential or integral Equation. It has been applied to a number of physical problems, where the governing differential equations are available. The method essentially consists of assuming the piecewise continuous function for the solution and obtaining the parameters of the functions in a manner that reduces the error in the solution. Here we have to find maximum stresses in each component of bearing, and to safe design of components material Maximum Stress should be less than Allowable stress. Design allowable, Allowable stress $\sigma_{all} = \text{Yield Strength or Ultimate Strength} * \text{Factor of Safety}$, Assume Factor of Safety is 1, Allowable stresses of Materials used in assembly is calculated in following table

Table1. Material Allowable Stress

Material	Components	Modulus Of Elasticity (GPa)	Yield strength MPa	Ultimate Strength MPa	Allowable stress MPa
SAE 52100	Shaft, Balls, Rollers, Sleeve	210	260	460	460
Nylon 66	Ball & Roller Cage	36	80	NA	80
Aluminium Alloy	Housing	71	280	310	310

4.2. Geometric Model

Integral Shaft Bearing product details as shown in Table 4.2 as per industrial requirements. Figure 4.2 shows the complete Assembly of Integral Shaft bearing.

Table2. Integral Shaft Bearing Characteristic

Bearing Type	Integral Shaft Bearing
Type of rolling Element	Roller/Ball
No. of Rolling element	15
Components Details for Assembly	Shaft, Ball Cage, Balls, Roller Cage, Rollers, Sleeve
Sleeve Diameter	30 mm
Shaft Diameter	15.918 mm
Ball Diameter	6.35 mm

Part geometry of complete Bearing created in CAD environment of Pro-E 5.0 is successfully imported to ANSYS environment.

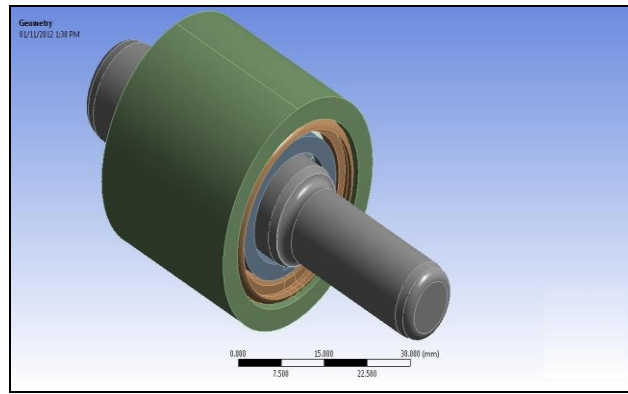


Fig3. Integral Shaft Bearing

4.3. Mesh Generation

In this analysis mesh generation is auto mesh generation with element size is 20. This element size is used for all the body of Integral Shaft Bearing. Hex-dominant method is used for all the parts of Integral Shaft Bearing.

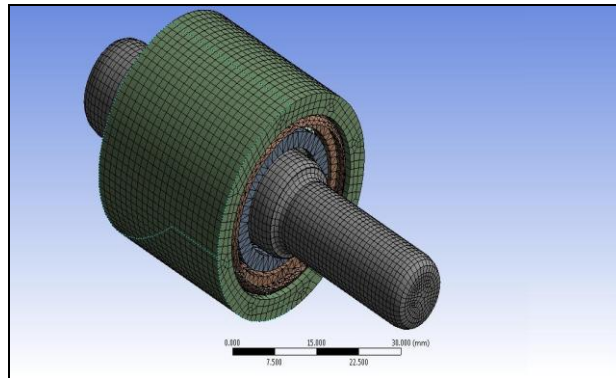


Fig4. Mesh Generation of Whole Assembly

4.4. Loading and Boundary Conditions

Loading and boundary conditions basically consist of two steps first is support and second is applying loads. Following Figure Shows the Supports and Forces

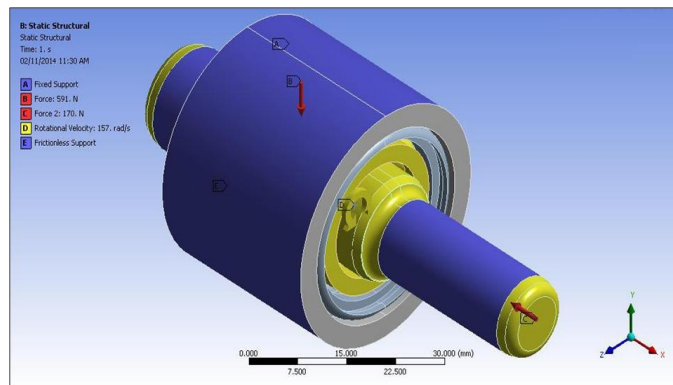


Fig5. Loading and Boundary Conditions

Table3. Loading and boundary conditions

speed (rpm)	Hub load (N)	% Use	Water Pump Temperature	Fit Conditions	Housing diameter	Sleeve OD	Radial clearance (Roller)	Radial clearance (Ball)
1500	591	19.7	-30°C	Max. Interference condition	Max. Interference condition		Minimum clearance	
					29.92	30	0.02	0.02

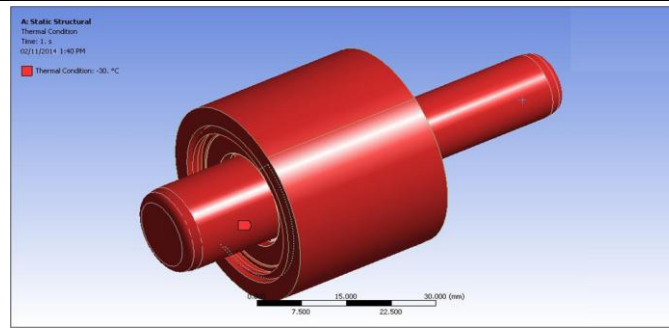


Fig6. Thermal Conditions

4.5. Solutions

After running the solution of above model we get different values of solutions such as, Maximum principle stress, Equivalent Stress, Total deformation and equivalent strain. All the results are described below.

Equivalent Stress: 399.08 Mpa (max)

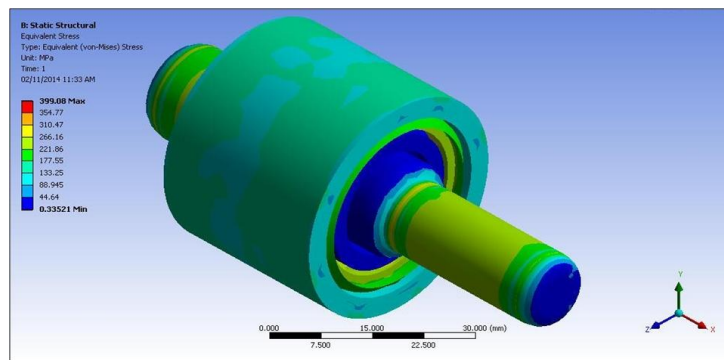


Fig7. Equivalent Stress Plot

Equivalent Stress for housing: Max Stress 252.25 < 310 Mpa (Allowable stress)

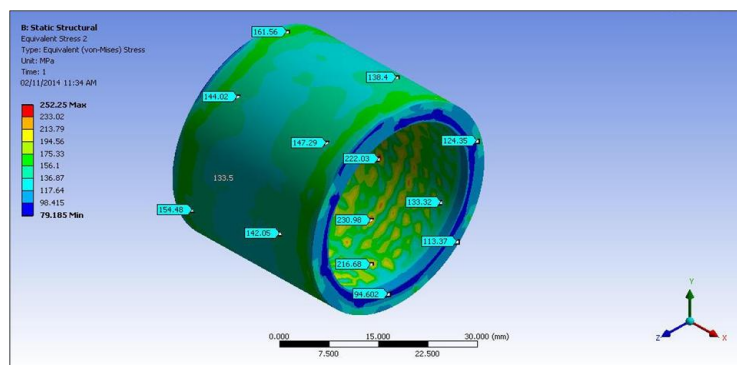


Fig8. Equivalent Stress for housing

Total Deformations for housing: 0.001194 mm (Max)

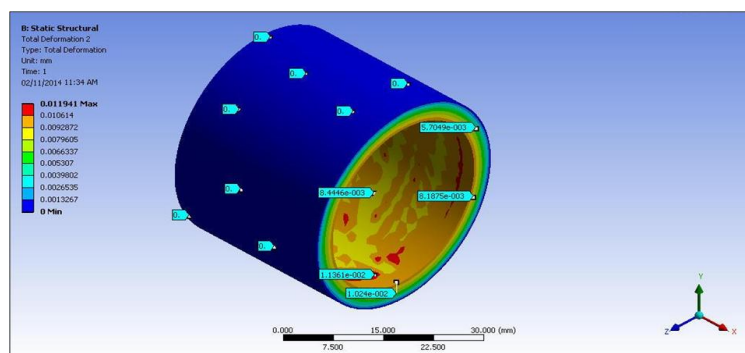


Fig9. Total Deformation for housing

Equivalent Stress for Shaft: Max Stress 399.08 < 460 Mpa (Allowable stress)

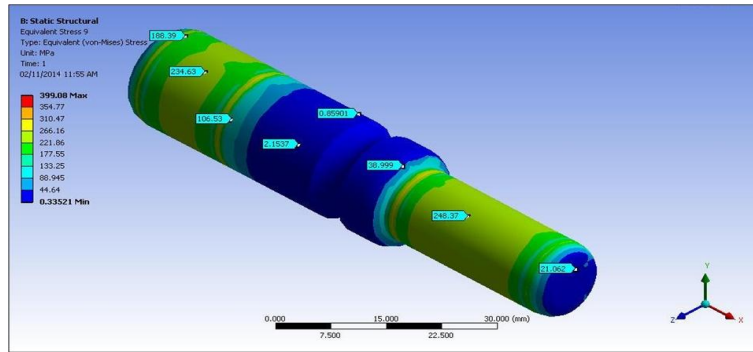


Fig10. Equivalent Stresses for Shaft

Directional Deformations for Shaft (Y-axis): Max. Deformation 0.00335 mm

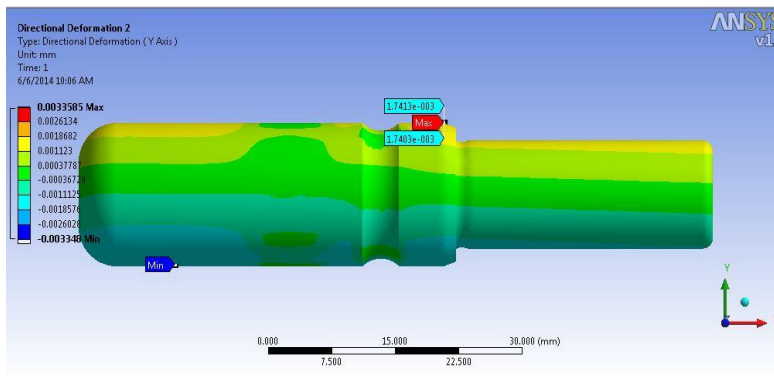


Fig11. Deformations for Shaft (Y-Direction)

Equivalent Stress for Rollers: Max Stress 47.757 < 460 Mpa (Allowable stress)

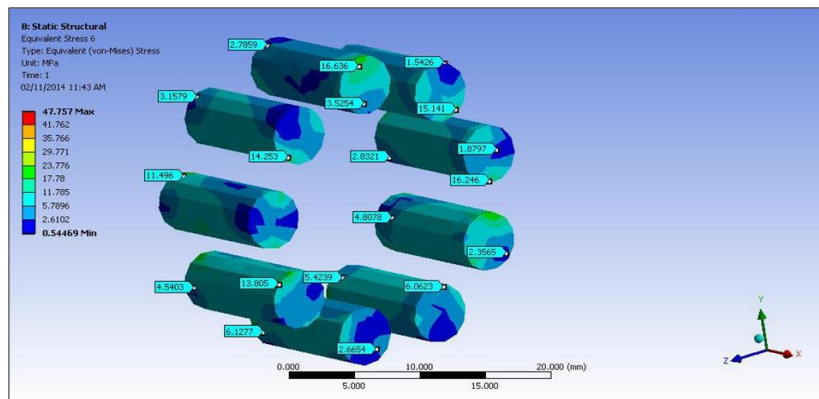


Fig12. Equivalent Stresses for Rollers

Directional Deformations for Rollers (Y-axis): Max. Deformation 0.003086 mm

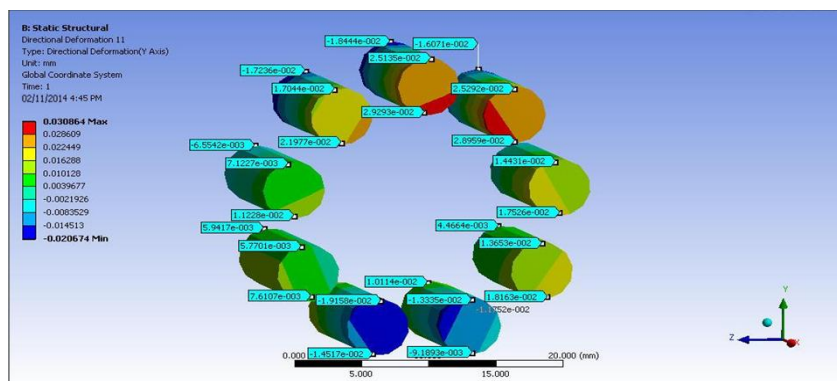


Fig13. Deformation Plot for Rollers (Y- Direction)

Equivalent Stress for Balls: Max Stress 29.274 < 460 Mpa (Allowable stress)

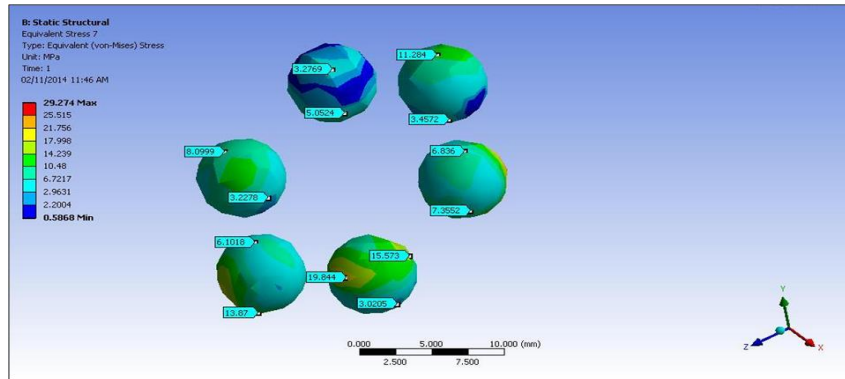


Fig14. Equivalent Stresses for Ball

Directional Deformations for Ball (Y-axis): Max. Deformation 0.00939 mm

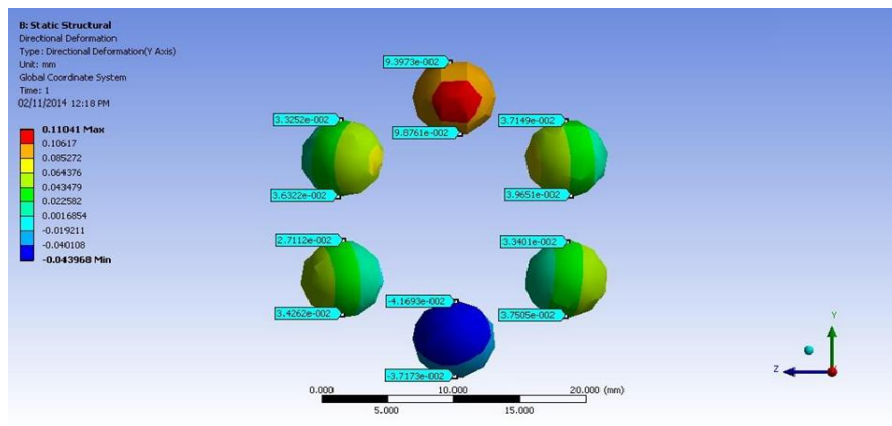


Fig15. Directional Deformations for Balls (Y-axis)

Equivalent Stress for Sleeve: Max Stress 329 < 460 Mpa (Allowable stress)

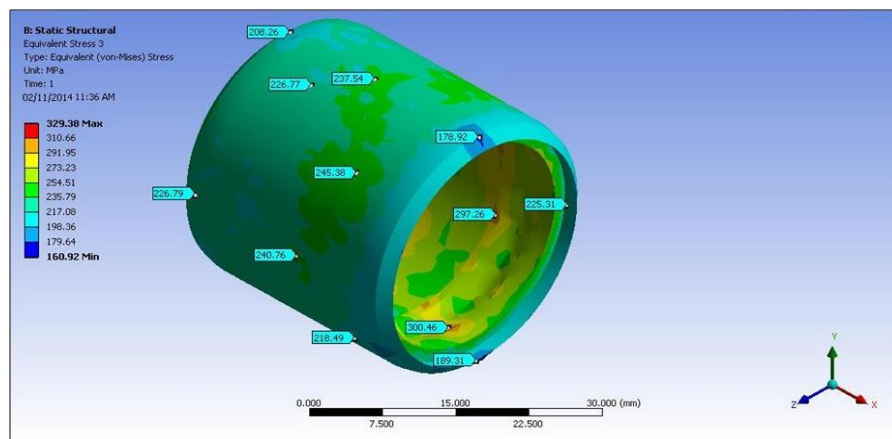


Fig16. Equivalent Stresses for Sleeve

Equivalent Stresses at inner area of sleeve

5. ANALYTICAL CALCULATIONS

5.1. Life of Bearings

The life of the bearing decreases with an increase in the load.

$$\frac{L_d}{L_c} = \left(\frac{C_d}{P_d} \right)^k$$

k = 3 for ball bearings

= 10/3 for roller bearings

L_d = desired life

L_c = life from the table (manufacturers catalog)

C_d = dynamic rating from manufacturer

P_d = design load

The equations can be rewritten as depending upon the variable to be calculated.

$$C_d = P_d \left(\frac{L_d}{L_c} \right)^{1/k} \quad L_d = L_c \left(\frac{C_d}{P_d} \right)^k$$

5.2. Equivalent Combined Radial Load

For combined radial and thrust loads

$$P = V X R + Y F_t$$

P = equivalent radial load

R = actual radial load

F_t = actual thrust load

X = radial factor (usually 0.56)

V = 1.0 for inner race rotating

= 1.2 for outer race rotating

Given Data:

Dynamic Rating from Manufacturer: 12800 N

Radial Load F_r = 591 N

Axial Load F_a = 170 N

Radial Factor = 0.56

There for, equivalent radial load, $P = 0.56 * 1.2 * 591 + 2 * 170$

$$P = 737.152 \text{ N}$$

5.3. Life of Bearing

$$L_{10} = \left(\frac{C}{P} \right)^3 \\ = \left(\frac{12800}{737.152} \right)^3$$

L_{10} = 5235.50 million rev.

5.4. Bearing Life in Hours L_{10h}

$$L_{10h} = L_{10} * 10^6 / 60 n$$

Where n = Integral Shaft Speed = 1500rpm

L_{10h} = 58172.302 hrs.

6. DISCUSSION AND COMPARISON OF RESULTS

Now we have to compare the results of Geometry after deformation. For the comparison purpose we have calculated stresses, deformation, and most important life of Bearing.

6.1. Bearing Clearances Effects

Clearance between rolling element like rollers/ balls and sleeve is most critical area of Integral Shaft bearing. Friction occurs between rolling element and sleeve there is chances of failure of bearing. As per bearing design clearance is 0.02 mm we have to compare this clearance after thermal expansion of bearing components.

6.1.1. At Ball Side clearance (All dimensions are in mm)

Before and after thermal expansion effect on dimensions of Shaft, Ball and Sleeve given in Table 7.1.

Table4. Dimensions Comparison after deformation at Ball Side

Components	Diameter	Deformation	Diameter after deformation
Shaft	14.212	0.003086	14.215086
Balls	6.34	0.00939	6.34939
Sleeve	26.924	0.00315	26.92715

Clearance between ball and sleeve after deformation: $26.92715 - [14.215086 + (6.34939 * 2)]$

Clearance = 0.013284 mm < 0.02 mm

6.1.2. At Roller Side Clearance in Y (All dimensions are in mm)

Before and after thermal expansion effect on dimensions in Y- Direction of Shaft, Rollers and Sleeve given in Table 5.1.

Table5. Dimensions Comparison after deformation at Roller Side

Components	Diameter	Deformation	Diameter after deformation
Shaft	15.905	0.003086	15.9067
Rollers	4.763	0.003086	4.766086
Sleeve	25.455	0.00315	25.45165

Clearance between roller and sleeve after deformation: $25.45165 - [15.9067 + (4.766086 * 2)]$

Clearance = 0.012778 mm < 0.02 mm

7. CONCLUSIONS

It is observed that Equivalent (Von-Mises) Stress Maximum at Shaft which is 189.1 Mpa and allowable stress of Shaft material is 208 MPa, also After Deformation Bearing Clearances are in limit, from this we conclude that design is safe.

Coupled analysis of Thermal and Structural both are equally important to analyze the Stress and Deformation of Integral Shaft Bearing.

By using the Analytical Method Bearing Life is 58172.302 hr, which is also satisfies the design.

FEM analysis is very efficient method for achieving stresses at different loading condition according to Forces & temperature applied to the component from the static analysis. The use of numerical method such as Finite Element Method now a day commonly used to gives detail information about structure or component.

ACKNOWLEDGEMENT

I gratefully acknowledge Department of Production Engineering of Amurtvahini College of Engineering, Sangamner for technical support and providing the research facilities. I would also like to thank to Dr. G. J. Vikhe Principal, Amurtvahini COE, Sangamner and Prof. V. B. Shinde for their help and dedication toward my research and related research, also my friends for their directly & indirectly help, support and excellent co-operation.

REFERENCES

- [1] Pang Chunjun, "Static Analysis of Rolling Bearings Using Finite Element Method", master thesis submitted in University of Stuttgart, May 2009.
- [2] Ring Plus Aqua Ltd, "Shaft Bearing Division", Manufacture Catalog
- [3] Ambepasad.S.Kushwaha, Atul B Wankhade, Dinesh E Mahajan, Darshan K Thakur, " Analysis of the Ball Bearing considering the Thermal (Temperature) and Friction Effects", National Conference on Emerging Trends in Engineering & Technology (VNCET-30 Mar'12).
- [4] Takeo Koyama, "Applying FEM to the Design of Automotive Bearings", Automotive Bearing Technology Department , NSK-1997.

- [5] Prasanna Subbarao Bhamidipati, "FEA Analysis of Novel Design of Cylindrical Roller Bearing", mater thesis submitted in R.C.E, affiliated to Jawaharlal Nehru technological university Hyderabad, India May, 2006
- [6] Brian P. Graney & Ken Starry, "Rolling Element Bearing Analysis", from material Evaluation vol 70, no1, pg 78-85, Jan 2012
- [7] Jafar Takabi & M.M. Khonsari, "Experimental testing and thermal analysis of ball bearings", Department of Mechanical Engineering, 2508 Patrick Taylor Hall, Louisiana State University, Baton Rouge, LA 70803, United States, 19 October 2012.
- [8] Laniado-Jacome Edwin," Numerical Model to Study of Contact Force in A Cylindrical Roller Bearing with Technical Mechanical Event Simulation", Mechanical engineering department, Carlos III University, Madrid, 28911, Spain.
- [9] Jun Sun, Gui Changlin, "Hydrodynamic lubrication analysis of journal bearing considering misalignment caused by shaft deformation", School of Mechanical and Automotive Engineering, Hefei University of Technology, No. 193 Tun Xi Road, Hefei, Anhui 230009, China, Received 12 August 2003; received in revised form 28 February 2004; accepted 7 May 2004.
- [10] Huei, L. (2011). Finite element simulations with ANSYS workbench 13 theory – applications – case studies.
- [11] V.B Bhandari," Design of Machine Elements", Tata-McGraw Hill-2009.