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Design and Analysis of an Integral Shaft Bearing

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ABSTRACT

An 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: Finite Element Analysis, Mesh Generation, ANSYS, 2D, 3D, Water Pump Modeling, FEM

I. 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.

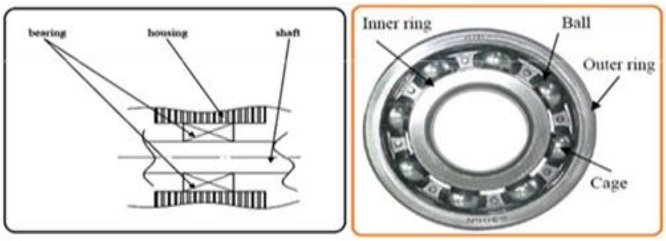

Figure1. Rolling Bearing

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. I n the mechanical system shown in Figure-A, 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.

II. METHODS AND MATERIAL

1. 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.

2. 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.

3. FEA Analysis and Results

A. 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 fine 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 σall = Yield Strength or Ultimate Strength * Factor of Safety, Assume Factor of Safety is 1, Allowable stresses of Materials used in assembly is calculated in following table.

Table.1 Material Allowable Stress

Material	Components	Modulus Of elasticity	Yield Strength	Allowable stress
SAE 52100	Shaft, Balls,rollers, sleeve	210	260	460
Nylon 66	Ball & Roller Cage	36	80	80
Aluminium Alloy	Housing	71	280	310

B. 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.

Table. 2. Integral Shaft Bearing Characteristic

Bearing	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	15.918

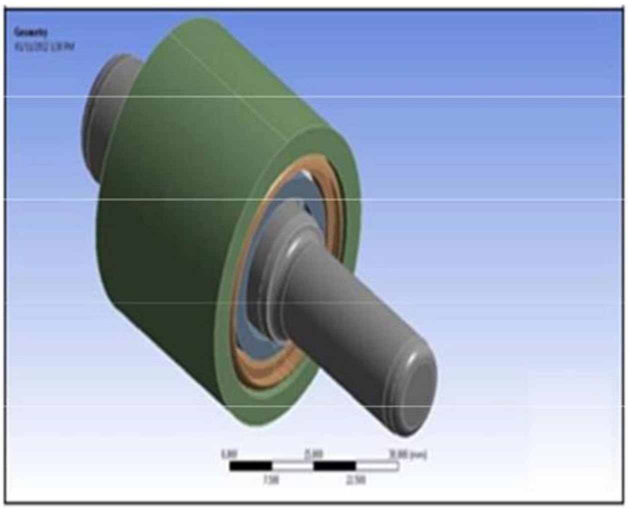

Figure 2. Integral Shaft Bearing

C. **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.

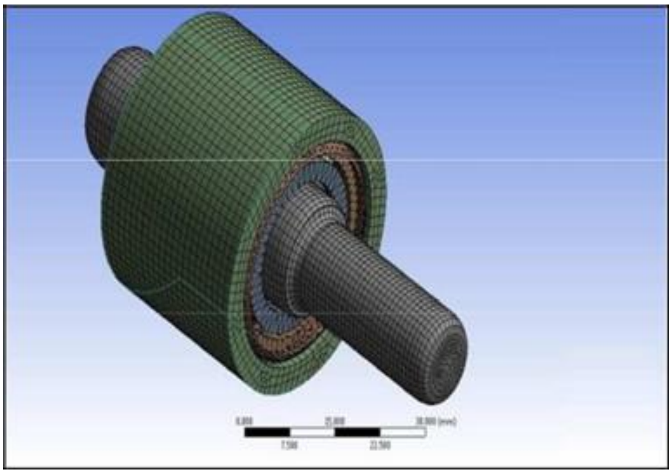

Figure 3. Mesh Generation of Whole Assembly

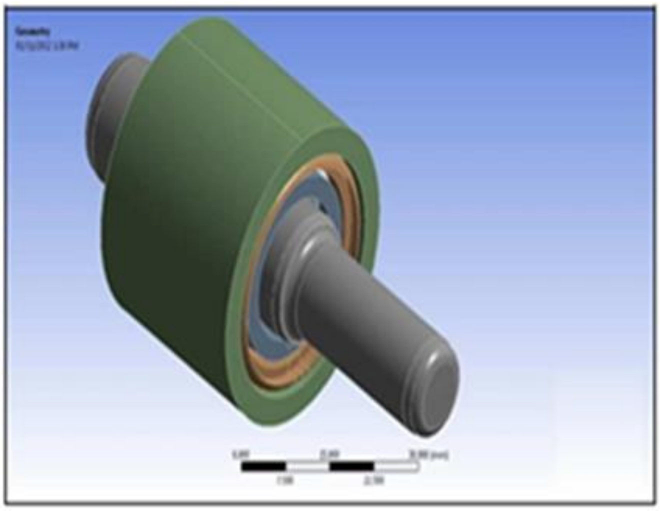

Figure 4. Integral Shaft Bearing

D. **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

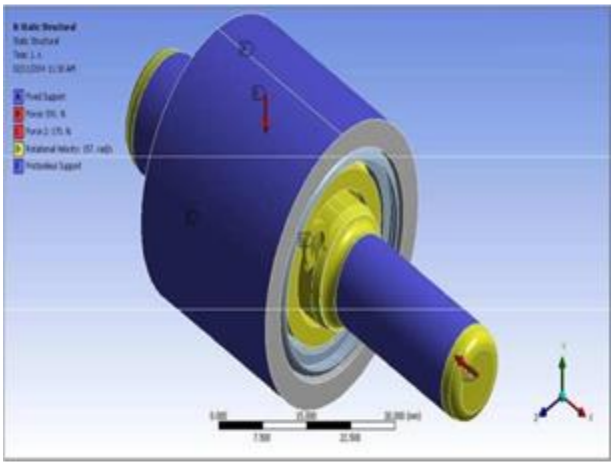

Figure 5. Loading and Boundary Conditions

Table 3. Loading And Boundary Conditions

Speed (pm)		1500
Hub Load (N)		591
% Use		19.7
Water Pump Temperature		-30^{0}C
Housing Diameter	Maximum Inteference	29.2
Sleeve OD		30
Radial Clearence (Roller)	Minimum Clearence	0.02
Radial Clearance (Ball)		0.02

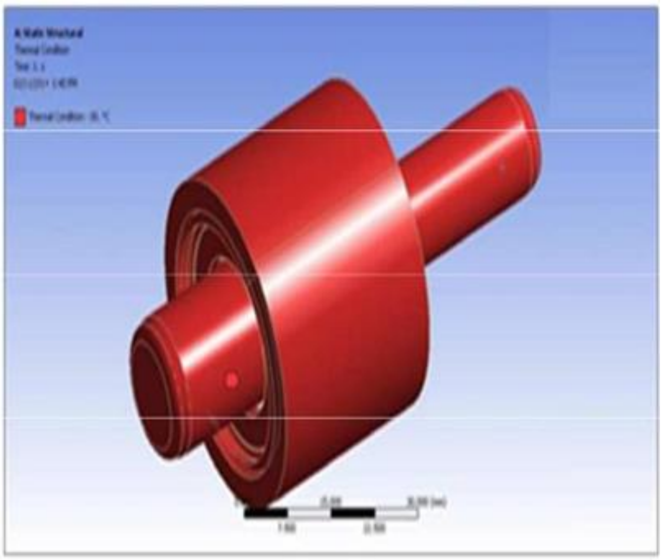

Figure 6. Thermal Conditions

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).

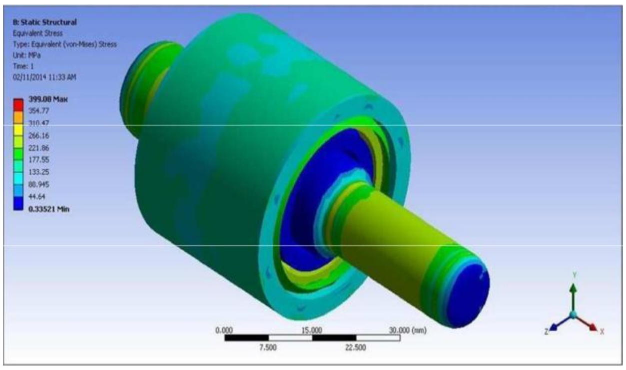

Figure 7. Equivalent Stress Plot

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

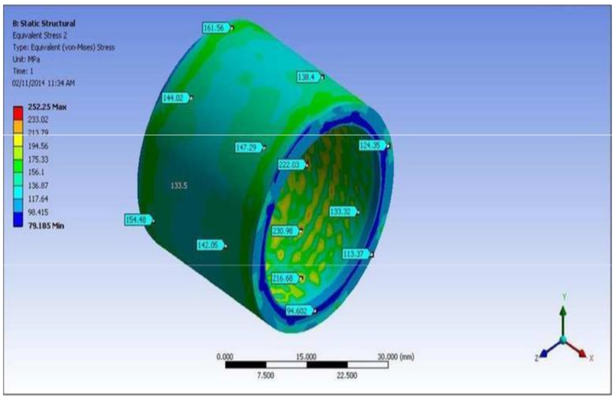

Figure 8. Equivalent Stress for housing

Total Deformations for housing: 0.001194 mm (Max)

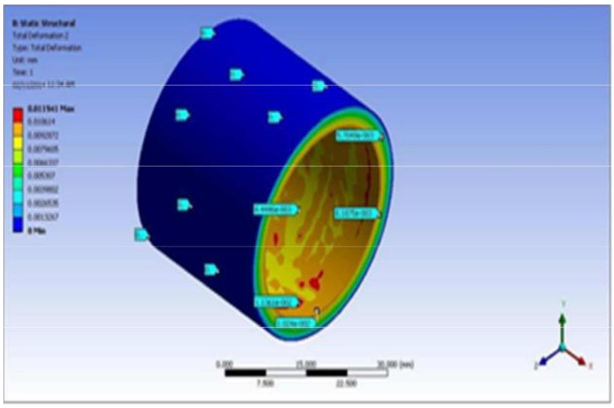

Figure 9. Total Deformation for housing

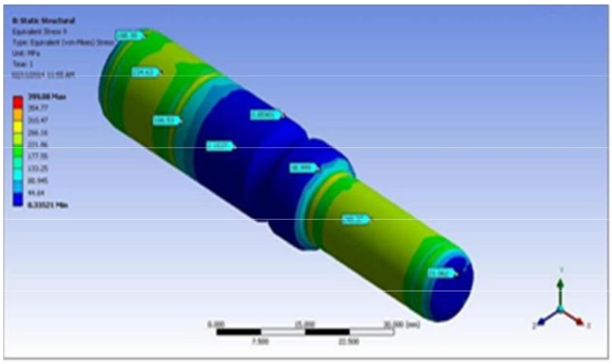

Figure 10. Equivalent Stresses for Shaft

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

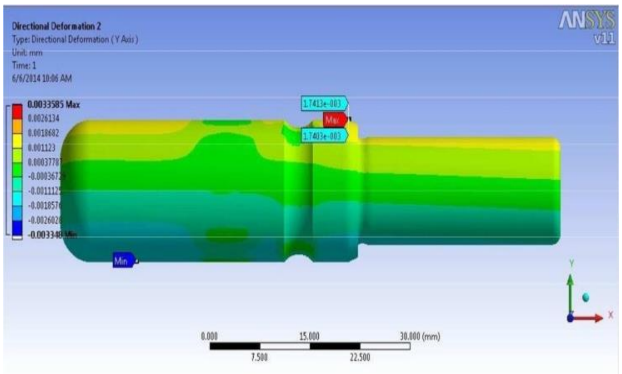

Figure 11. Deformations for Shaft (Y-Direction)

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

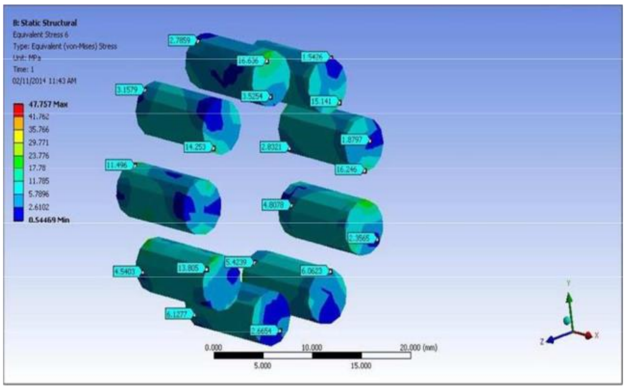

Figure 12. Equivalent Stresses for Rollers

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

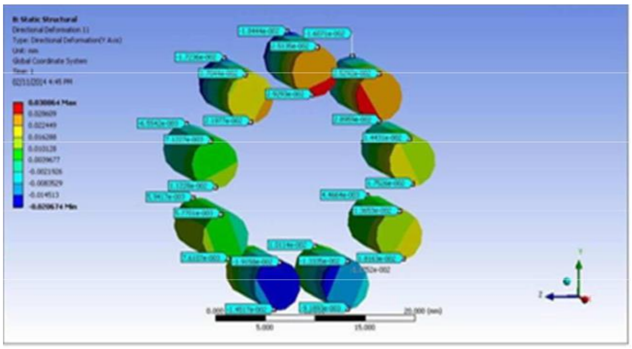

Figure 13. Deformation Plot for Rollers (Y- Direction)

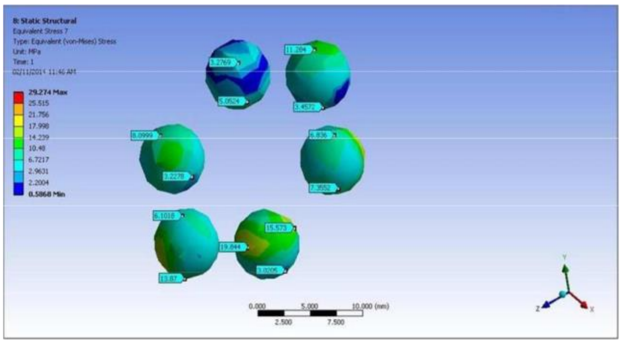

Figure 14. Equivalent Stresses for Ball

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

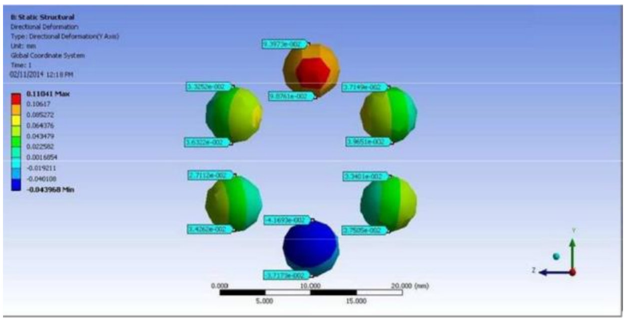

Figure 15. Directional Deformations for Balls (Y-axis)

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

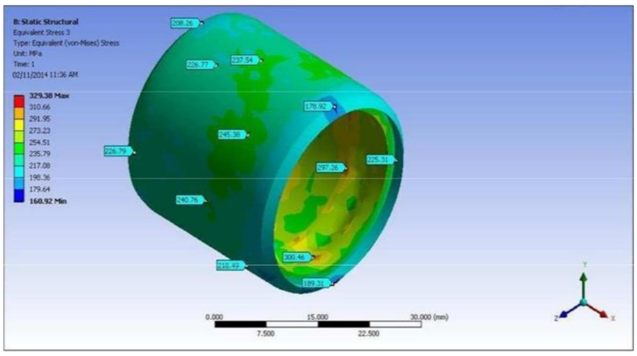

Figure 16. Equivalent Stresses for Sleeve

III. RESULTS AND DISCUSSION

1. Analytical Calculations

A. **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

Lc = life from the table (manufacturers catalog)

Cd = dynamic rating from manufacturer
Pd = design load

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

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

B. **Equivalent Combined Radial load**

For combined radial and thrust loads

P = equivalent radial load
R = actual radial load Ft = actual thrust load
X = radial factor (usually 0.56) V = 1.0 for Pinner=VraceXR +rotatingYFt
= 1.2 for outer race rotating

Given Data:

Dynamic Rating from Manufacturer: 12800 N Radial Load Fr = 591 N

Axial Load Fa = 170 N

Radial Factor = 0.56

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

P = 737.152 N

Bearing Life in Hours L10h

$$L10 = \left(\frac{C}{P}\right)^3$$

L10= 5235.50 million rev

2. 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.

A. 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.

Table 4 Dimensions Comparison After Deformation At Ball Side

Components	Diameter	Deformation	Diameter after deformation
Shaft	14.212	0.003086	14.21508
Balls	6.34	0.00939	6.34939
Sleeve	26.924	0.00315	26.9271

Clearance between ball and sleeve after deformation: 26.92715- [14.215086+ (6.34939*2)] Clearance = 0.013284 mm < 0.02 mm

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

Table 5 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	4.76608
Sleeve	25.455	0.00315	25.4516

Clearance between roller and sleeve after deformation: 25.45165- [15.9067+

(4.766086*2)] Clearance = 0.012778 mm < 0.02 mm

Table 6. 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	4.76608
Sleeve	25.455	0.00315	25.4516

Table 7. 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	4.76608
Sleeve	25.455	0.00315	25.4516

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

Clearance = 0.012778 mm < 0.02 mm

IV. CONCLUSION

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.

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