Structural Analysis of Foot Step Bearing

To study the various factors that influence life of the bearing in rotating systems.

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Abstract **- To study the various factors when the bearing is used to support a shaft which is placed vertically with axial load that influence the life of the bearing. Years of field experience and laboratory testings, together with advanced calculations, form the base for development of computer programs used to analyze and dimension the rotating system where bearing and shaft plays an important role.**

Keywords – **Bearing Design Parameters, Load Factors, Equations, Structural Analysis Results.**

I. INTRODUCTION

Bearings are one of the key components that determine the service interval for rotating machinery; they can even limit the life of the machine. Therefore, knowledge and understanding of bearings are important to both designers and users. In selecting the bearing to be used, one must consider a variety of factors: Loads, static and dynamic factors, Speed and running pattern, Stiffness, Temperature levels and gradients, heat conduction, Lubrication: viscosity, stiffness, durability, Degree of contamination. Shielding, Mounting, Maintenance, Lifetime and service time. A bearing selection or analysis is futile without rotor dynamic data from the shaft and other rotating parts.

To ensure the long life of a product, many factors have to be taken into account. A poor installation with unfavourable inlet conditions, unfavourable duty points, bad anchoring etc can cause structural disturbances that may be detrimental. While natural frequencies in the structure from poorly designed supports of the pump, pipes, valves etc are often the cause of high vibration levels. These factors describe some of the knowledge and experience including the precautions to be taken into consideration in the design and analysis of shaft and bearings.

II. FAILURE CAUSES

In order to properly dimension the rotating system, the failure causes or modes have to be determined. Most failures are due to unpredictable factors and the cure is to create a robust design that can cope with the unknown. Predictable failures or lifetimes can be calculated although many parameters will vary and statistical considerations have to be taken into account.

2.1 Shaft

The shaft is designed to have an infinite life. This will be the case unless the shaft is overloaded or damaged.

2.1.1 Fatigue crack

The cause of a broken shaft is almost always fatigue. A crack starts at a stress concentration from a keyway or a sharp radius, or, in rare cases, from a material impurity. Flaws in the surface of a shaft, such as scratches, indents or corrosion, may also be the starting point of a fatigue crack. Loads that drive a crack are normally torsion loads from direct online starts or bending loads from the hydraulic end.

2.1.2 Plastic deformation

Plastic deformation can only occur in extreme load cases when debris is squeezed into a radial clearance that results in large deformations.

2.1.3 Defect in shaft

Defects in shafts that have passed the checks in the factory, checks for unbalance, as well as the check at test run are very rare. On site, however, it is important to prevent damage from corrosion, indents etc, by proper handling.

2.2 Bearing

A bearing does not last forever, sooner or later fatigue or wear, or lubricant deterioration will ultimately destroy the bearings ability to function properly. Bearing failure causes in order of likelihood are listed below.

2.2.1 Penetrating fluid or particles

Particles generate high stresses in the bearing components and thereby create premature fatigue failure. Particles also generate wear that shortens life. Particles, especially light alloy particles like zinc, may act as catalysts for the grease and create premature ageing. Consequently, Nilos rings must not be used! If a fluid enters the bearing and is over rolled the fluid acts as a jet and forces the grease out from the raceway. If the fluid is oil, with just a tiny amount of water (0.1%), the bearing´s lifetime is ruined even if it is flooded with that oil. Furthermore, the fluid may also cause corrosion.[1]

2.2.2 High temperature or gradient

High temperatures due to bad cooling or too much heat generated can be seen (if the bearing is not totally ruined) if the grease has darkened and has a feeling of carbon particles dissolved in oil. If synthetic oil is used, it may have polymerized and created a layer on the surface, mainly seen in oil lubrication. A bearing that has been overfilled with grease may also cause a too high temperature. If the temperature is just slightly higher than allowed, the signs are not so obvious; however, the lifetime will be reduced due to lower viscosity and faster degradation of the lubricant. Cages with plastic material will also age faster at increased temperatures. If too much heat is generated, maybe from overload, but with adequate cooling, then the gradient over the bearing can be too high, generating a preload that can destroy the bearing.[2][3]

III. SHAFT CALCULATIONS

3.1 Torsion Frequency:

$$\omega_{kr} = \sqrt{\frac{G \cdot K}{L} \cdot \left[\frac{1}{J_M} + \frac{1}{J_L}\right]}$$

$$G = \frac{E}{2 \cdot (1 + \nu)}$$

L = length
G = torsion modulus
E = modulus of elasticity
ν = Poissons ratio
J_m = inertia motor
J_l = inertia load (impeller)
K = torsion stiffness

3.2 Global Reactions:

$$F_x = \omega^2 \cdot Ub_x$$

$$z_x = \frac{DUb_{xz}}{Ub_x}$$

F_x, F_y = forces in x and y direction
z_x, z_y = position of forces
ω = rotational speed
Ub_x, Ub_y = unbalance
DUb_{xz}, DUb_{yz} = moment unbalance

3.3 Axial Displacement:

$$\delta = \int_L \left[\frac{F \cdot dL}{A \cdot E} + \alpha \cdot dL \cdot \Delta T\right]$$

δ = displacement
F = force
L = length
E = modulus of elasticity
A = cross section area
α = thermal expansion coefficient
ΔT = temperature change

3.4 Shaft Deformation:

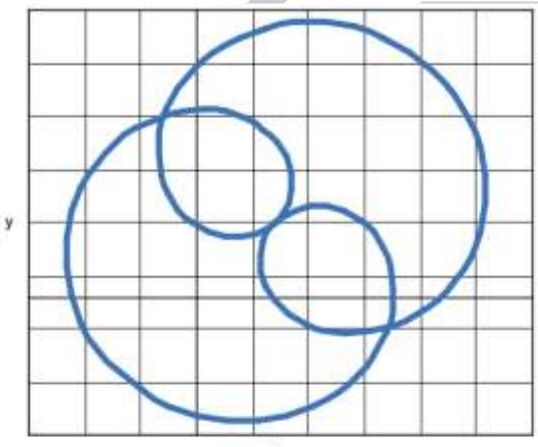

Figure 3.4.1: *The deformation pattern of static & rotational load cases with equal amplitudes & equal phase*[1]

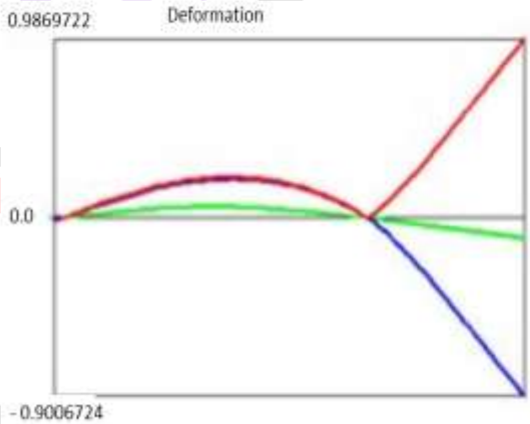

Figure 3.4.2: *The deformation of the static run at one duty point. Red: Magnitude ,Blue & Green : X & Y Data[m]*[4]

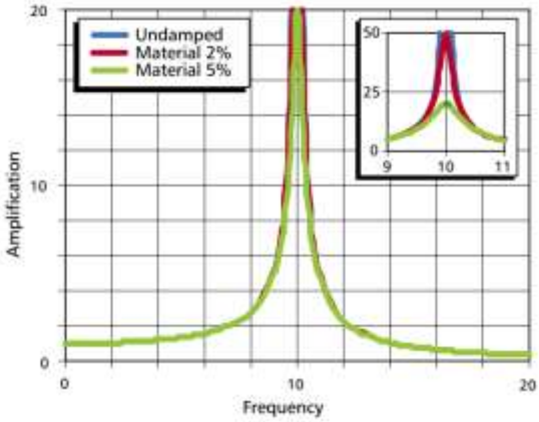

Figure 3.4.3: *The response functions around a single critical frequency for different damping. Max value for 2% damping is 50 and for 5% 20.*[5]

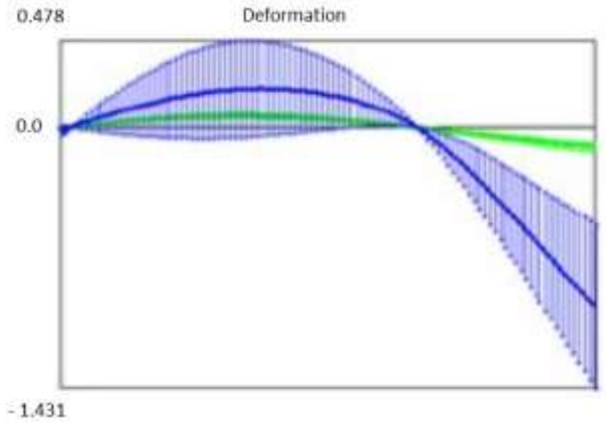

Figure 3.4.4: *The deformation of the static run with the sum of the two dynamic runs superimposed at one duty point. Data from start to end of shaft[m].*[6]

3.5 Angular Motion:

$$\varphi = \frac{M_t \cdot L}{G \cdot K}$$

$$G = \frac{E}{2 \cdot (1 + \nu)}$$

φ = angular motion
K = torsional stiffness
L = length
G = torsion modulus
E = modulus of elasticity
ν = Poissons ratio
M_t = torque

3.6 Max. Tensile Stress:

$$\sigma_T = \sigma_d \cdot K_d + (\sigma_{bstat} - \sigma_{brot} - \sigma_{bbp}) \cdot K_b$$

σ_T = max tensile stress
σ_d = axial stress
$\sigma_{b\,stat}$ = static bending stress
$\sigma_{b\,rot}$ = rotating bending stress
$\sigma_{b\,bp}$ = bending stress from blade pass
K_d = stress concentration, axial
K_b = stress concentration, bending

3.7 Combination of Stresses

$$\sigma_e = \sqrt{\sigma^2 + 3 \cdot \tau^2}$$

σ_e = von Mise stress
σ = tensile stress
τ = shear stress

3.8 Stresses and Fatigue Analysis:

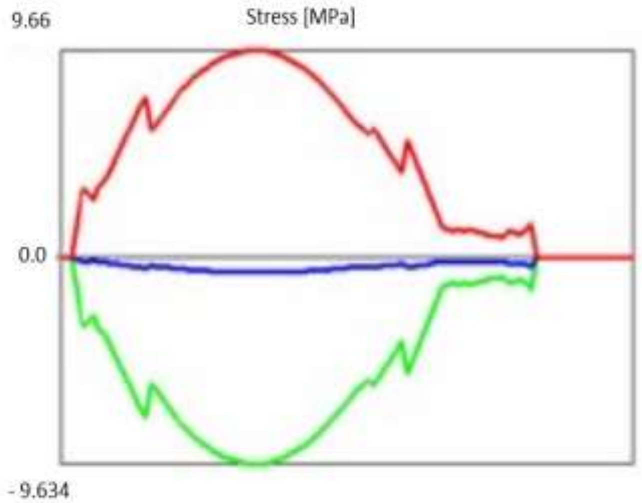

Figure 3.8.1: *The bending stress of the rotating run in one duty point. Red is magnitude and green and blue are x and y-data. Data from start to end of shaft.*[7]

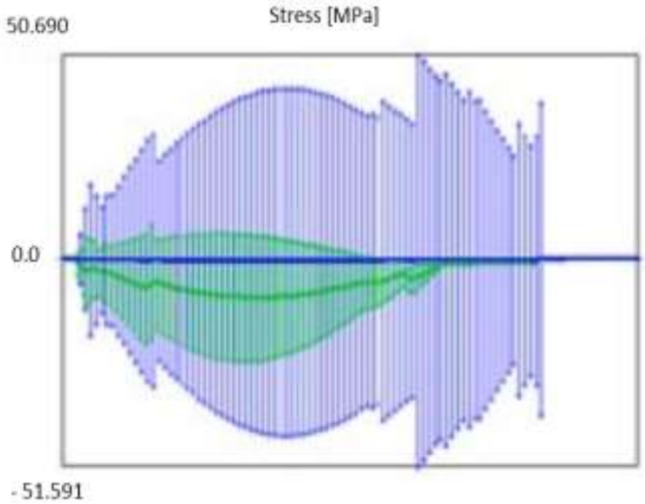

Figure 3.8.2: *The bending stress of the rotating run with the sum of the static and blade pass runs superimposed in one duty point. x and y data only. Data from start to end of shaft.*[7]

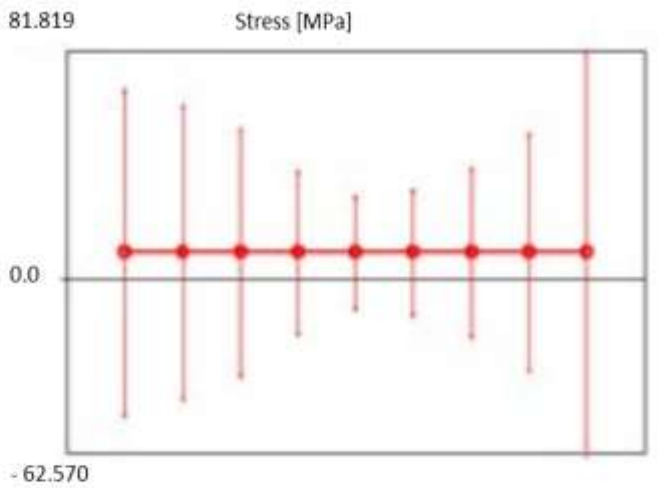

Figure 3.8.3: The bending stress of the rotating run with the sum of the static and blade pass runs superimposed in one position of shaft and in different duty points. Magnitude only.[7]

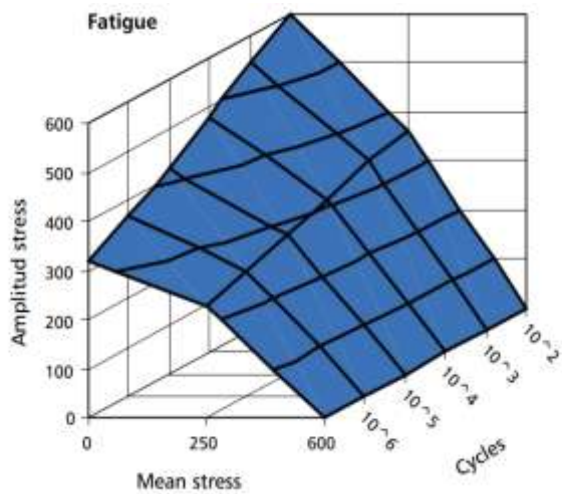

Figure 3.8.4: The combined and linearized Haigh and Wöhler diagram.[7]

IV. STRUCTURAL ANALYSIS IN ANSYS

4.1 Structural Steel Constants :

TABLE 1
Structural Steel > Constants

Density	7850 kg m^-3
Coefficient of Thermal Expansion	1.2e-005 C^-1
Specific Heat	434 J kg^-1 C^-1
Thermal Conductivity	60.5 W m^-1 C^-1
Resistivity	1.7e-007 ohm m
Compressive Yield Strength Pa	2.5e+008
Tensile Yield Strength Pa	2.5e+008
Tensile Ultimate Strength Pa	4.6e+008
Strength Coefficient Pa	9.2e+008
Cyclic Strength Coefficient Pa	1.e+009
Ductility Coefficient	0.213
Young's Modulus Pa	2.e+011
Poisson's Ratio	0.3
Bulk Modulus Pa	1.6667e+011
Shear Modulus Pa	7.6923e+010

TABLE 2
Model (A4) > Mesh

Object Name	*Mesh*
State	Solved
Sizing	
Use Advanced Size Function	Off
Relevance Center	Fine
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	3.2361e-003 m
Inflation Option	Smooth Transition

Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
View Advanced Options	No
Triangle Surface Mesher	Program Controlled
Statistics	
Nodes	58809
Elements	33140

TABLE 3
Model (A4) > Static Structural (A5) > Loads

Object Name	*Fixed Support*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Definition		
Type	Fixed Support	Pressure
Define By		Normal To
Magnitude		1000. Pa (ramped)

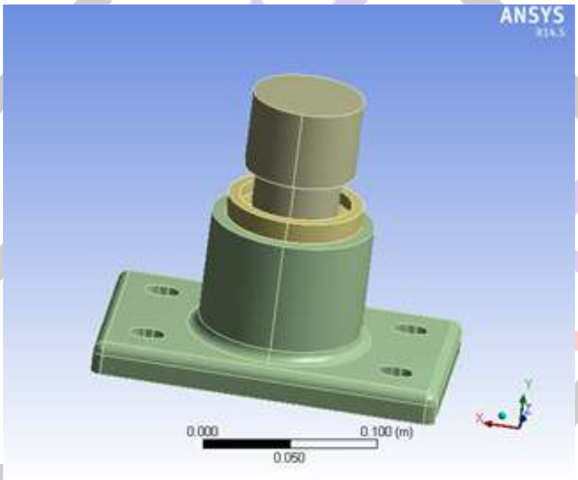

Figure 4.1.1 : *3-D Model*

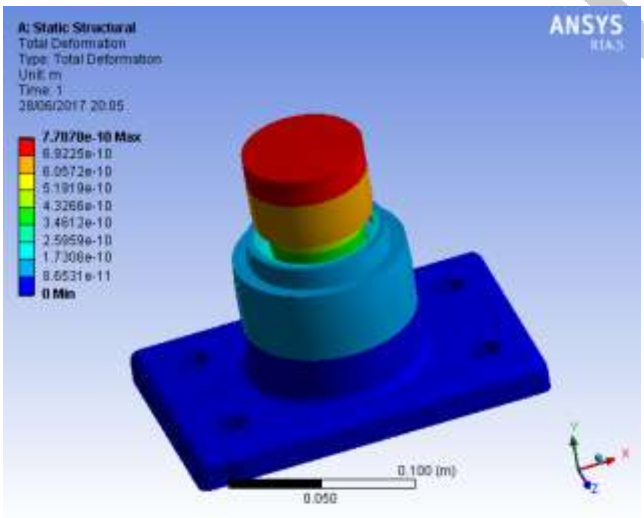

Figure 4.1.2 : *Total Deformation[3D]*

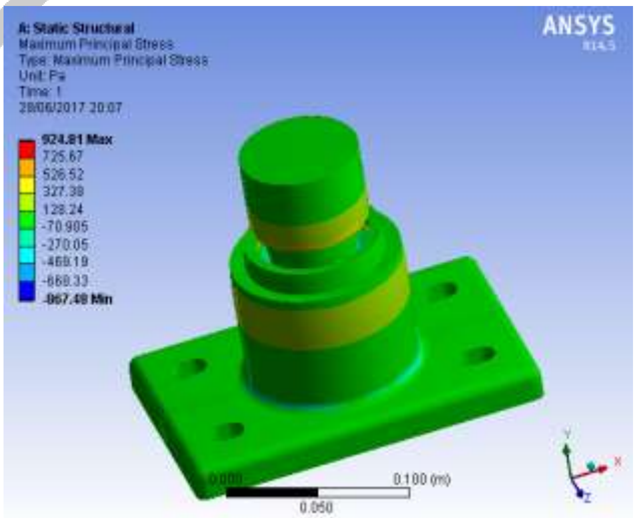

Figure 4.1.3 : *Maximum Principal Stress[3D]*

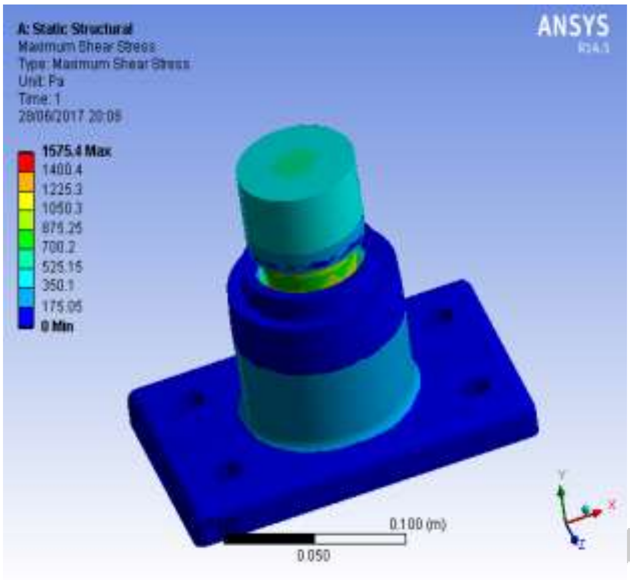

Figure 4.1.4 : *Maximum Shear Stress[3D]*

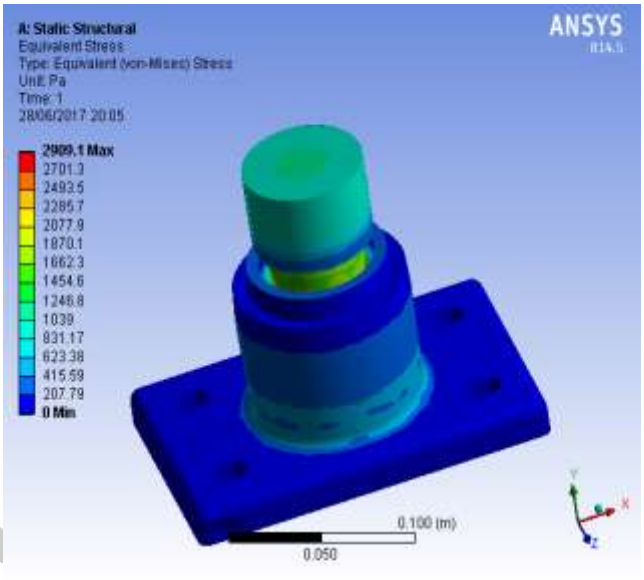

Figure 4.1.5 : *Equivalent Stress[3D]*

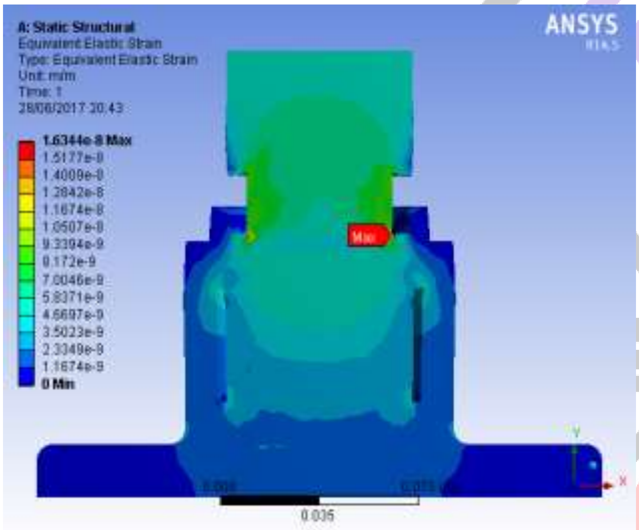

Figure 4.1.6 : *Equivalent Elastic Strain[Section Plane]*

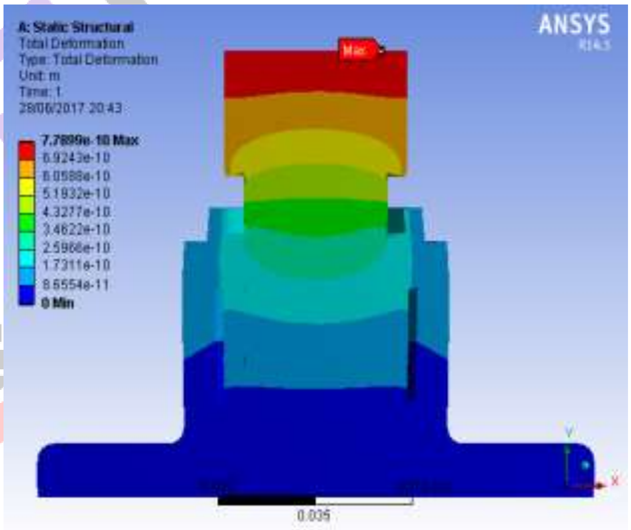

Figure 4.1.7 : *Total Deformation [Section Plane]*

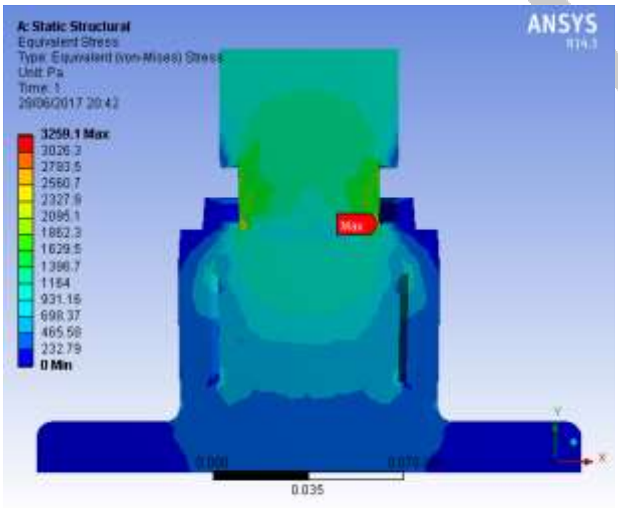

Figure 4.1.8 : *Equivalent Stress [Section Plane]*

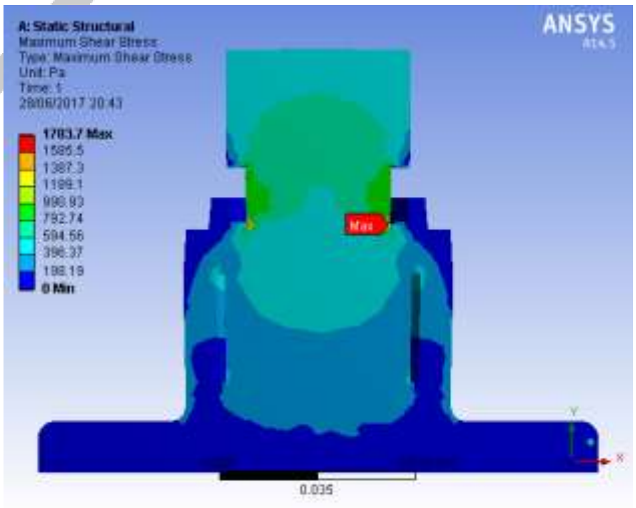

Figure 4.1.9 : *Maximum Shear Stress[Section Plane]*

IV. CONCLUSION

Looking at the results and all parameters related to the given material which can be used for the bearing one can conclude that the bearing is safe under given load conditions as the values of Equivalent Elastic Strain, Total Deformation, Equivalent Stress and Maximum Shear Stress are seen towards the respective parameters related to the material which we have considered.

Area which is under high stresses is easily recognized and the future scope could be the working on that area by considering the various factors like CFD analysis and thermal analysis.

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[6]Ioannides E., Bergling G, Gabelli A. *The SKF formula for rolling bearing.* SKF magazine Evolution no 1, p25-28 (2001)
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