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Break Down Analysis of Bearing

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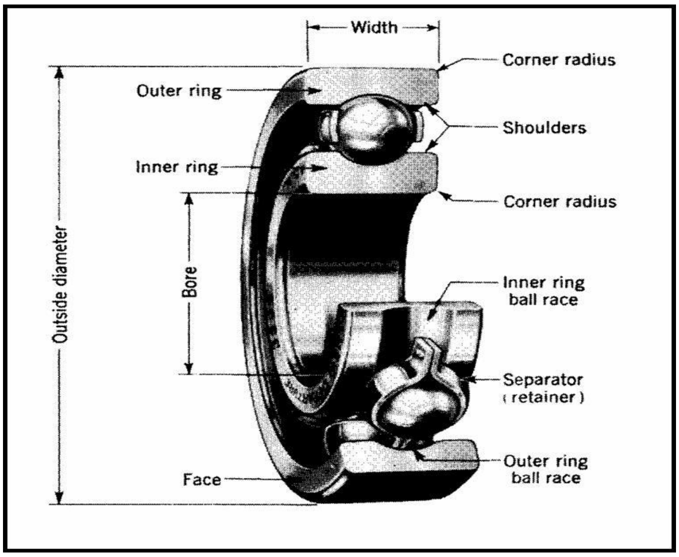
ABSTRACT
The present research work deals is to model new fore wheel bearing of agricultural machine used for ploughing which fail in regular usage. This bearing takes three formulations. They are bearing through Material Change Break Down Analysis The design is done in CATIA and Break Even Analysis is done in ANSYS. And find which one among the Bearing through Design or Bearing through Material Change is solution for the problem aimed for. Break Down Analysis is done in each of the cases to find the maximum forces to be applied on the bearings to get the maximum load that can be bared by the bearings. The objective of our project is to find the solution for the regular failure of bearings of machine through different material. Here we are using three materials i.e. steel, tungsten, zinc
Keywords: design,Finite Element Analysis, Break down Analysis.

I. INTRODUCTION

A bearing is a machine element that constrains relative motion to only the desired motion, and reduces friction between moving parts. The design of the bearing may, for example, provide for free linear movement of the moving part or for free rotation around a fixed axis; or, it may *prevent* a motion by controlling the vectors of normal forces that bear on the moving parts. Many bearings also *facilitate* the desired motion as much as possible, such as by minimizing friction. Bearings are classified broadly according to the type of operation, the motions allowed, or to the directions of the loads (forces) applied to the parts. Now a days the agricultures has been using modern equipment of the great output, in this criteria tractor is one of the main technology used for ploughing .As the soil used for ploughing is much slurry the sudden load impact are possible on front axle of the tractor which lead to the brakeage of bearings fixed to axle and tyres. The bearings used are spherical bearings which have the both parallel and perpendicular movement of rotation of fore wheel bearing of agricultural machine used for ploughing which fail in regular usage. So the bearings are studied at different cases and the modeling of bearings takes three formulations. They are modeling of Bearing through Design, modeling of Bearing through Material Change, Break Down Analysis. The first two formulations are done in CATIA and Break down Analysis is done in ANSYS. And find which one among the modeling of Bearing through
Design or modeling of Bearing through Material Change is solution for the problem aimed for. Break down Analysis is done in each of the cases to find the maximum forces to be applied on the bearings to get the maximum load that can be bared by the bearings.



II. INTRODUCTION TO DESIGN

Design of the bearings is done through data analysis of the bearing specifications and their design catalogues. These ball bearings undergo deforming under different periodic loads while rotation. So as reach the require fatigue nature their loads are calculated in two ways they are analytical method and technical method. Analytical method gives out of the hand calculations by using different scientific formulas and modules with data book of different catalogues. The values are approximate values to satisfy the material properties and technical method gives out the values or visualization of design by using technical software just by updating with the specifications of required design. This project deals with the both analytical method and technical software CATIA.

III. ANALYTICALAPPROACH TO DESIGN

The conditions involved in design of bearings is Basic Static Load Rating of Rolling Contact Bearings The load carried by a non-rotating bearing is called a static load. The basic static load rating is defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which corresponds to a total permanent deformation of the ball (or roller) and race, at the most heavily stressed contact, equal to 0.0001 times the ball (or roller) diameter. In single row angular contact ball bearings, the basic static load relates to the radial component of the load, which causes a purely radial displacement of the bearing rings in relation to each other. According to IS: 3823–1984, the basic static load rating (C0) in Newton's for ball and roller bearings may be obtained as discussed below:

A) For radial ball bearings, the basic static radial load rating (C_0) is given by $C_0 = f_0.i.Z.D^2 \text{ Cos } \alpha$

Where

i = Number of rows of balls in any one bearing,

Z = Number of ball per row,

D = Diameter of balls, in mm,

α = Nominal angle of contact i.e. the nominal angle between the line of action of the ball load and a plane perpendicular to the axis of bearing, and

f_0 = A factor depending upon the type of bearing.

The value of factor (f_0) for bearings made of hardened steel is taken as follows:

f_0 = 3.33, for self-aligning ball bearings

= 12.3, for radial contact and angular contact groove ball bearings.

B) For radial roller bearings, the basic static radial load rating is given by $C_0 = f_0.i.Z.l_e.D \cos \alpha$

Where

i = Number of rows of rollers in the bearing,

Z = Number of rollers per row,

l_e = Effective length of contact between one roller and that ring (or washer)

C) Where the contact is the shortest (in mm). It is equal to the overall length of roller minus roller chamfers or grinding undercuts For thrust ball bearings, the basic static axial load rating is given by $C_0 = f_0.Z.D^2 \sin \alpha$

Where

Z = Number of balls carrying thrust in one direction, and

f_0 = 49, for bearings made of hardened steel.

Considering the values of the bearing

3.1 Calculations

Diameter of balls D=12mm

Number of balls carrying thrust in one direction, Z = 2

Factor depending upon the type of bearing

for bearings made of hardened steel f_0= 49

Nominal angle of contact $\alpha = 35^0$

Therefore

$C_o = f_o Z D^2 \sin \alpha$

$=49x2x12^2x\sin 35$

=8094.91

Static Equivalent Load for Rolling Contact Bearings

The load carried by a non-rotating bearing is called a static load. The basic static load rating is defined as the static radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which corresponds to a total permanent deformation of the ball (or roller) and race, at the most heavily stressed contact, equal to 0.0001 times the ball (or roller) diameter. In single row angular contact ball bearings, the basic static load relates to the radial component of the load, which causes a purely radial displacement of the bearing rings in relation to each other. According to IS: 3823–1984, the basic static load rating (C0) in newtons for ball and roller bearings may be obtained as discussed below:

1. For radial ball bearings, the basic static radial load rating (C_0) is given by

$C_0 = f_0.i.Z.D^2 \cos \alpha$

Where

i = Number of rows of balls in any one bearing,

Z = Number of ball per row,

D = Diameter of balls, in mm,

α = Nominal angle of contact i.e. the nominal angle between the line of action of the ball load and a plane perpendicular to the axis of bearing, and

f_0 = A factor depending upon the type of bearing.

The value of factor (f_0) for bearings made of hardened steel are taken as follows:

f_0 = 3.33, for self-aligning ball bearings

= 12.3, for radial contact and angular contact groove ball bearings.

2. For radial roller bearings, the basic static radial load rating is given by

$C_0 = f_0.i.Z.l_e.D \cos \alpha$

Where

i = Number of rows of rollers in the bearing,

Z = Number of rollers per row,

l_e = Effective length of contact between one roller and that ring (or washer)

Where the contact is the shortest (in mm). It is equal to the overall length of roller minus roller chamfers or grinding undercuts,

3. For thrust ball bearings, the basic static axial load rating is given by

$C_0 = f_0.Z.D^2 \sin \alpha$
Where
Z = Number of balls carrying thrust in one direction, and
f_0 = 49, for bearings made of hardened steel.
Considering the values of the bearing

3.2 Calculations
Diameter of balls D =12mm
Number of balls carrying thrust in one direction, Z = 2
Factor depending upon the type of bearing for bearings made of hardened steel.
f_0= 49,
Nominal angle of contact $\alpha = 35^0$
Therefore

$C_o = f_o Z D^2 \sin \alpha$
$=49x2x12^2x\sin 35$
=8094.91

IV. APPROACH TO CATIA

In Mechanical engineering CATIA enables the creation of 3D parts, from 3D sketches, sheet metal, composites, moulded, forged or tooling parts up to the definition of mechanical assemblies. The software provides advanced technologies for mechanical surfacing & BIW. It provides tools to complete product definition, including functional tolerances as well as kinematics definition. CATIA provides a wide range of applications for tooling design, for both generic tooling and molding.

4.1 DESIGN OF BEARING THROUGH CATIA

CATIA offers a solution to shape design, styling, surfacing workflow and visualization to create, modify, and validate complex innovative shapes from industrial design to Class-A surfacing with the ICEM surfacing technologies. CATIA supports multiple stages of product design whether started from scratch or from 2D sketches. CATIA is able to read and produce STEP format files for reverse engineering and surface reuse. To design a bearing the commands are considered and the required dimension are used to draw the Bearing in the catia soft ware

Design of inner race
Inner diameter of the bearing =50mm
Thickness of the bearing =10mm

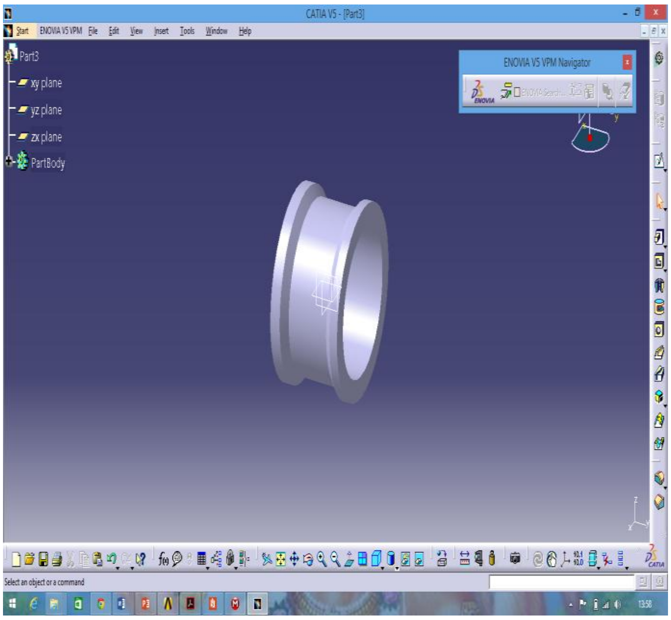
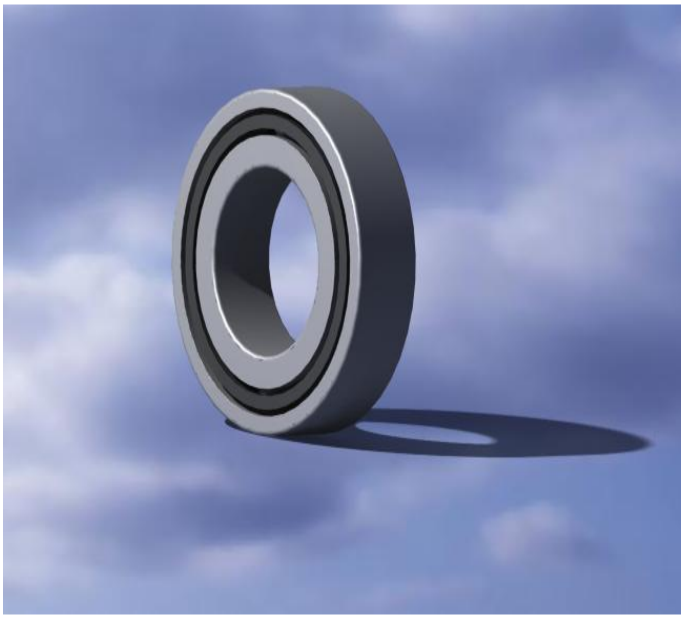


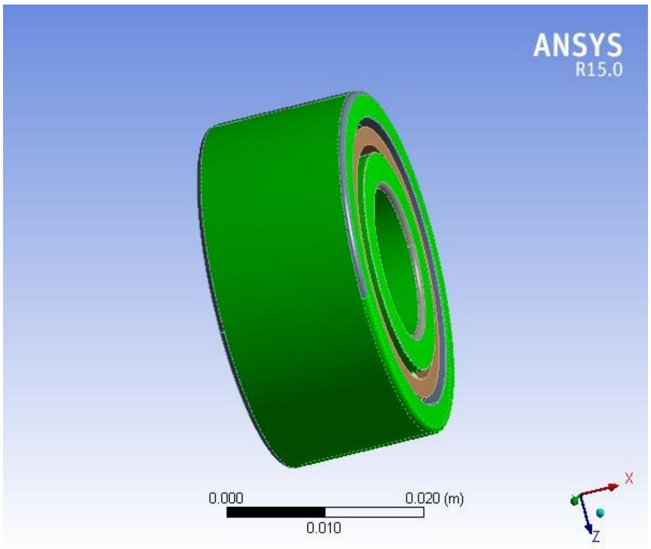
figure: *Inner race*

OVER ALL DESIGN OF THE BEARING
The dimensions under the given specific aspects are
Inner race diameter = 50mm
Thick of inner race =10mm
Outer race diameter =72mm
Thickness of outer race =76mm Crown thickness =4mm



V. FINITE ELEMENT MODEL

The loads acting on the bearings are a perpendicular sudden load they make the bearing break under heavy loads out of fatigue strength.



As the material is required the material is to substitute for the model for the father process here tungsten is used as the material for the balls and steel for the races

Properties

Structural tungsten > Alternating Stress Mean Stress

Table-1

Alternating Stress Pa	Cycles	Mean Stress Pa
3.999e+009	10	0
2.827e+009	20	0
1.896e+009	50	0
1.413e+009	100	0
1.069e+009	200	0
4.41e+008	2000	0
2.62e+008	10000	0
2.14e+008	20000	0
1.38e+008	1.e+005	0
1.14e+008	2.e+005	0
8.62e+007	1.e+006	0

Structural Steel > Strain-Life Parameters

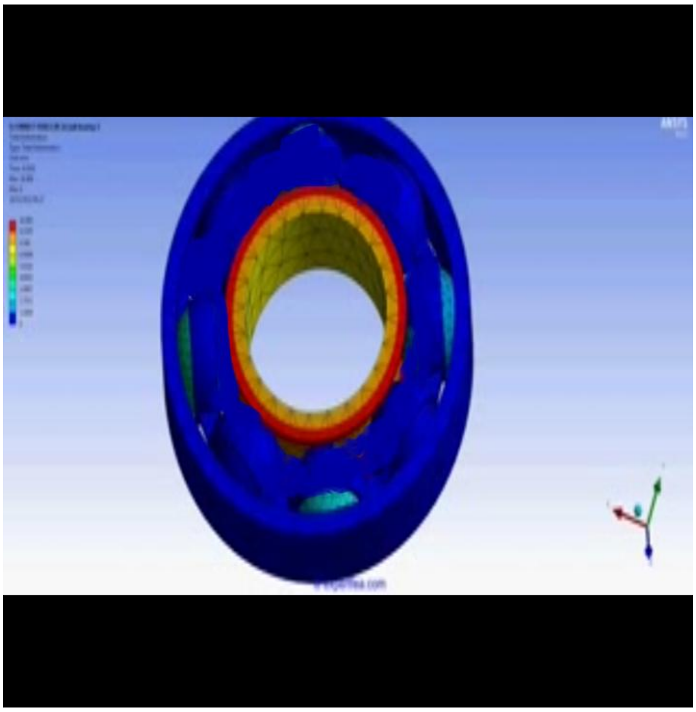
Table-2

Strength Coefficient Pa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient Pa	Cyclic Strain Hardening Exponent
9.2e+008	-0.106	0.213	-0.47	1.e+009	0.2

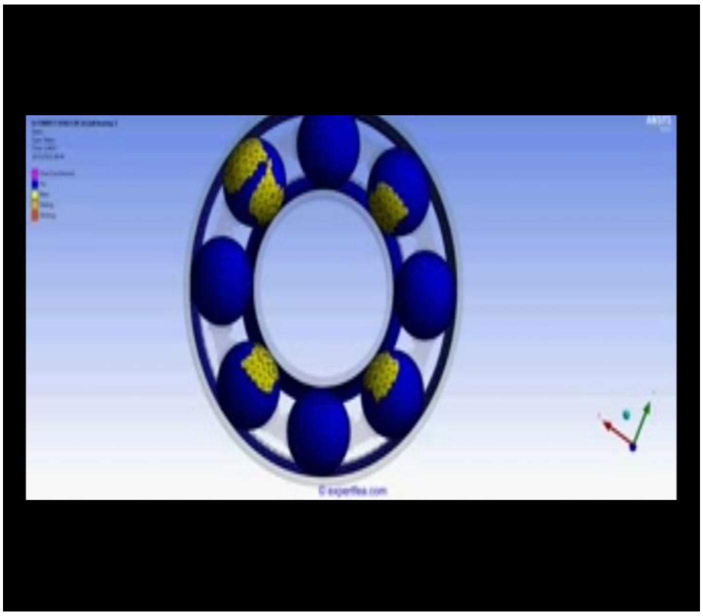
Structural tungsten> Isotropic Elasticity

Table-3

Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
120	2.e+011	0.3	1.6667e+011	7.6923e+010



Failure of inner race



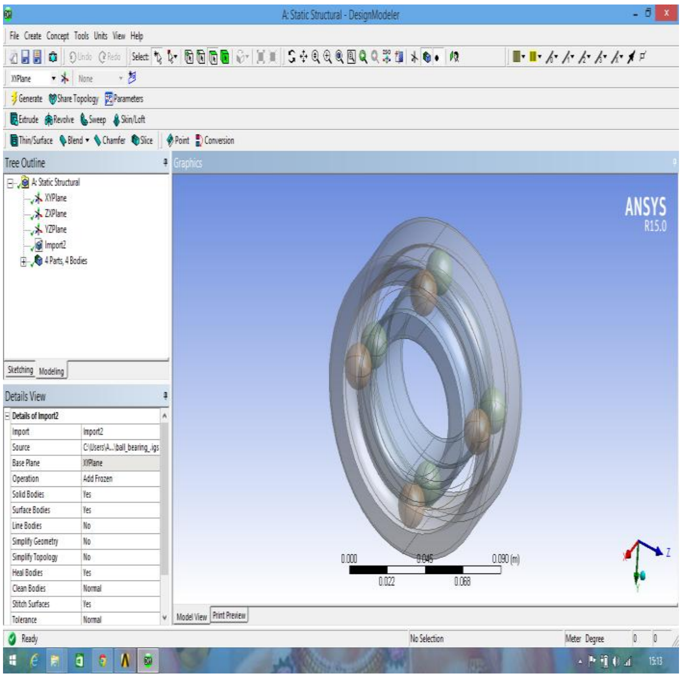
Crushing of balls due to load

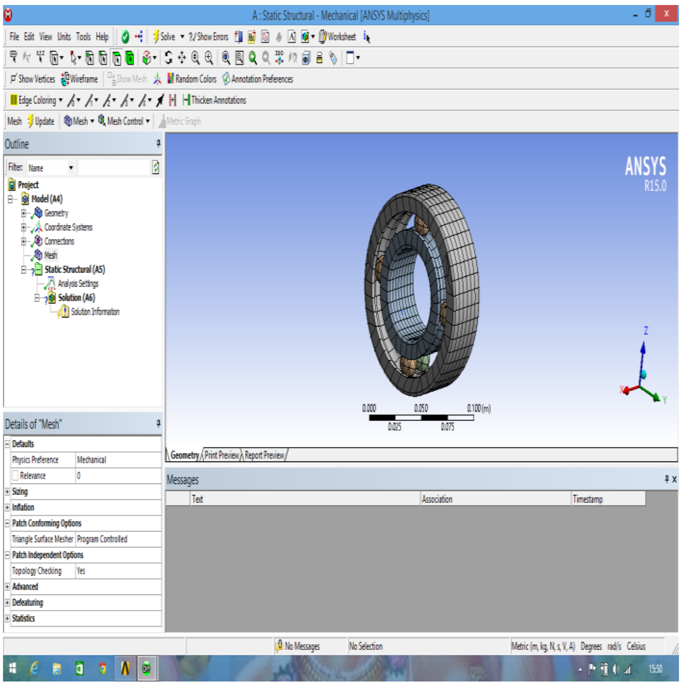
ZINC

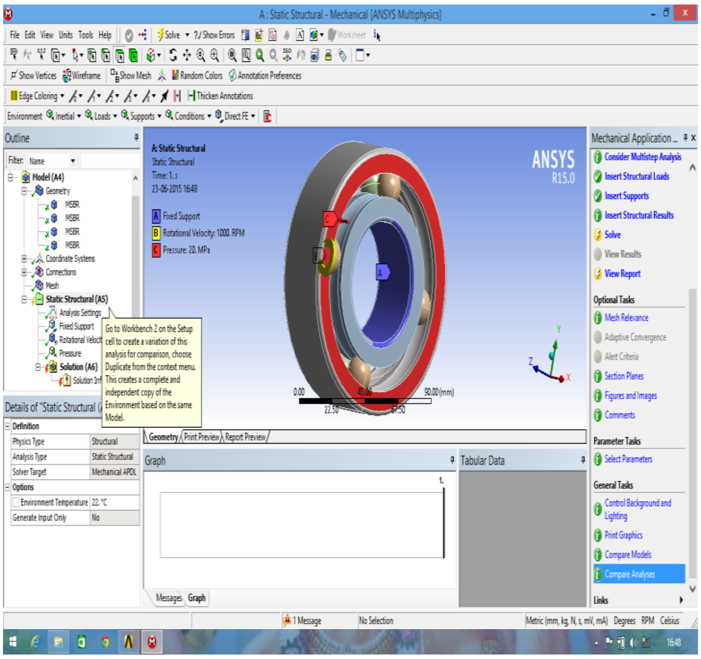
Gruneisen Coefficient	Parameter C1 mm s^-1	Parameter S1	Parameter Quadratic S2 s mm^-1
1.96	3.005e+006	1.581	0

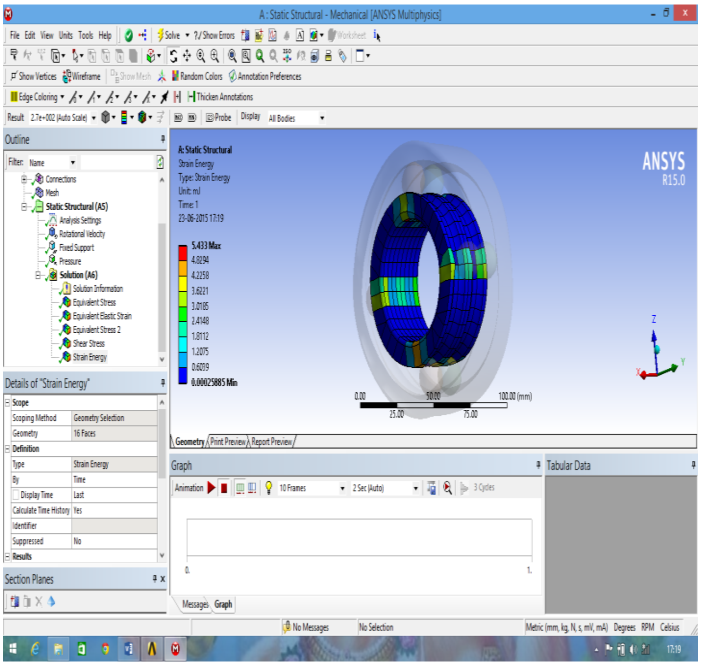
ZINC > Isotropic Elasticity

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
	85	0.27	61.594	33.465





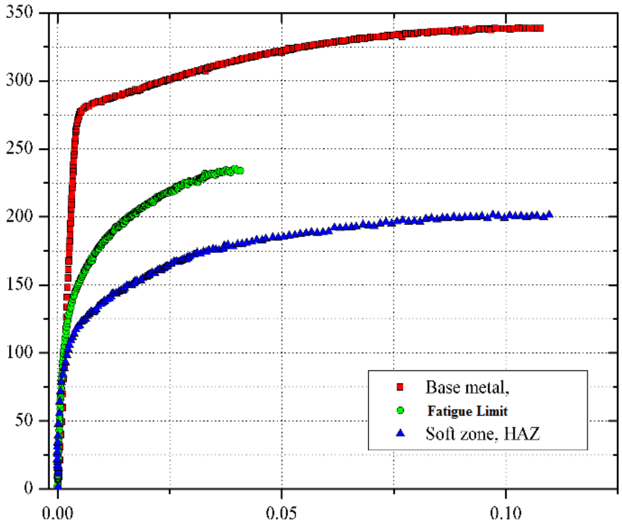




VI. RESULTS

Load component Comparison of the each case

material	Basic capacities in KN			
	Double row angular contact ball bearing		Dynamic load	Static load
	Static (C0)	Dynamic C0)		
Steel material	96.5000	102.000	5982.55	414.56
Tungsten	98.4350	103.4200	6123.7199	674.24
Zinc	86	92.4	7692.35	721.54



Graph 1**fatigue nature**

The above graph shows how the fatigue nature change in different load condition

CONCLUSION

The Project says that how the Load vary with the material and load acting on the bearings under fatigue nature of the material. The fatigue nature of the Zinc alloy is more compared to the steel & Tungsten but comprise of highly qualified design

REFERENCES

[1.] Eschmann et al (1958) stated that when bearings operate under normal conditions of well balanced load and good alignment, fatigue failure begins with small fissures. These fissures are located between the surface of the raceway and the rolling elements, which then gradually propagate to the surface, generating detectable vibrations and increasing noise levels.

[2.] Riddle (1955) observed the fatigue phenomena, known as flaking or spalling and suggested that continued stress causes fragments of the material to break or loose, and produce a localized fatigue.

[3.] Eschmann et al (1958). Eventually, the failure results in rough running of the bearing. This is the initiation of failure in rolling element bearings which reduce the life of the bearing. According to Riddle (1995) external sources include contamination and corrosion.

[4.] Bertele (1990) and Neale (1985) suggested that practically an application of condition monitoring techniques may help in early fault detection. In their study on 23 hydrodynamic lubrication regime occurring in rolling bearing, high contact pressures are developed in the surfaces to deform elastically, giving room for small elliptical contact areas.

[5.] Mitroviü and Tatjana Lazoviü (2002) investigated upon the frictional sliding which follows rolling of balls along the rings raceways. This one causes rolling bearing to wear

[6.] Tomimoto (2003) taking a keen interest in plain bearings, opined that an increase in contaminant concentration could decrease the thickness of the oil film. Larger size particles have greater tendency to cause early fatigue spalling in the contact zone. Besides spalling, contaminant particles can lead to other damage mechanisms, such as scuffing. They originate from the lubricant starvation at the inlet of the contact zone.

[7.] Aktürk (1999) simulated the effect of bearings surface waviness on the vibration by a computer program. 25 The results are obtained in both time domain and frequency domains. Loparo et al (2000) presented a model-based technique for the detection of faults. Jang and Jeong (2003) proposed an analytical method to investigate the stability of a rotating system due to ball bearing waviness

[8.] Ilonen (2005) introduced a generic condition diagnosis tool. The tool successfully detected bearing damage in induction motors using measurements of the stator current or vibration. It is based on discriminative energy functions that reveal discriminative frequency-domain regions where failures can be identified.

[9.] Yhland (1992) proposed a linear model for the vibrations of the shaft bearing system caused by ball bearing geometric imperfections. These imperfections covered are radial and axial waviness of outer and inner rings, ball waviness, and ball diameter oversize.

[10.] Tandon and Nakra (1993) reported visual inspection of the time history of the vibration signals, time wave form indices, probability density function, and probability density moments that are easily analyzed using the time domain analysis. A time wave form index is a single number calculated, based on the raw vibration signal and used for trending and comparisons. The indices include peak value, mean value, rms value, and peak-to-peak amplitude. This method has been applied with only limited success for the detection of the defects.