FATIGUE ANALYSIS OF DEEP GROOVE BALL BEARING CAGE TO INCREASE ITS LIFE USING ANSYS

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Abstract*—Analysis of Bearing Cage is the main aim of this research paper, so highly focus on it. And we will discuss deep groove ball bearing cage's fatigue analysis also with various Analytical method & experimenting setups. Here modeling done with solid-works 2016 & Steady Analysis also done with the help of ANSYS, and some Analytical method is investigated. Fatigue life, Stress, Deformation these three are the main factors & as per statics, small percentages of bearing failure is due to subsurface failure. Damaged by surface fatigue lies in the wear of steel cage so in that case replacing cage material to composites which have a higher fatigue resistance & strong adhesive bonding. With the Analytical help of design working hours of bearing life can be calculated & will be validated using ANSYS tool.

***Index Terms*—Pressed Steel Cage, Composites, Deep Groove Ball Bearing, Fatigue life, ANSYS, Stress, Deformation, Bearing Cage.**

I. INTRODUCTION

Since 2600 BC of the Ancient Egyptian's long history Construction of "Black Stones & Pyramids can be move with the help of roller bearing, As bearing provides the relative motion & reducing friction between two surfaces as an example : Shaft & Housing & Also guides & supports as the term bearing means "To Bear" means "To Support Another". Four Major parts in the all types of bearing: 1) Outer race (Outer ring), 2) Inner race (Inner ring), 3) Rolling Element (i.e. Roller, sleeve, needle, ball), 4) Cage/Separator (Retainer). **[1]**

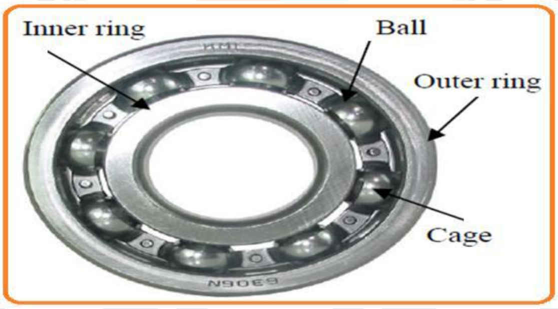


Figure 1. Deep Groove Ball Bearing.

II. THEORY

Finding Reasonable reaction forces in both inner ring & outer ring by using FEM Simulation is one of our goals, Also Our main goal is to increasing fatigue life of the bearing. Fluctuating loads & Subjective or repetitive metals often cause fatigue failure for that reason in this bearing we will replaced pressed steel cages to an epoxy composite cages because of its strong adhesive bonding & higher fatigue resistance.

As Fatigue failure occurred without any plastic deformation as applied stress range is higher, the shorter the life. It also affected many parameters likes of material, surface finishing, over loading, temperature, etc. of the fatigue life.

III. RESEARCH METHODOLOGY

- The basic understanding of the deep groove ball bearing.
- With the help of modified data of the ball bearing & its cage make 2D/3D ax symmetric model.
- FEM analysis carry out using ANSYS afterwards.
- Comparing FEA result with the analytical method.
- Efficiency & for improving life of the bearing improvement determine safe stress level of the given specimen and its performance. **[2]**

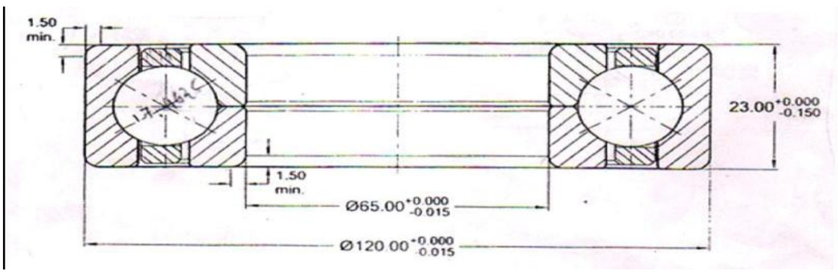


Figure 2. Assembly of Deep Groove Ball Bearing.

As per the figure it's now necessary to find out analytical solution of the given bearing in hours & we just find out life of the bearing at taking two different contact angle. **[1]**

ANALYTICAL DESIGNOFTHEBEARINGIF CONTACT ANGLES $=22.7^{o}$ Contactangles$=22.7^{o}$ BalldiameterD$_w$=13.49375mm RadialloadF$_r$= 5000N AxialloadF$_a$=4000N Bearing Material 440C(SS) Cage Material = Epoxy Carbon Composite RPM=1600 No. ofZ =16 Ballpitchdiameter=92.5mm No. ofrowsi=1	ANALYTICAL DESIGN OFTHE BEARING IF CONTACT ANGLES $=25.7^{o}$ Contactangles$=25.7^{o}$ BalldiameterD$_w$=13.49375mm RadialloadF$_r$=5000N AxialloadF$_a$=4000N Bearing Material 440C(SS) Cage Material = 440C (SS) RPM=2880 No. ofZ =15 Ballpitchdiameter=92.5mm No. ofrowsi=1
So,$D_w \cos\alpha / d_{pw}$ Where,D_w=Balldiameter α=contactangle d_{pw}=Ballpitchdiameter $=(13.49375\times\cos 22.7^{o})/92.5$ $=0.134$ Therefore, asperISO Standards F_c=58.15frominterpolation. Where,F_c=factor of dynamic load	So,$D_w \cos\alpha / d_{pw}$ Where,D_w=Balldiameter α=contactangle d_{pw}=Ballpitchdiameter $=(13.49375\times\cos 25.7^{o})/92.5$ $= 0.131$ Therefore, asperISO Standards F_c=58.15frominterpolation. Where,F_c=factor of dynamic load
Now, Dynamic Load Ratings , $C_r = F_c (i \times \cos\alpha)^{0.7} \times Z^{2/3} \times (D_w)^{1.8}$ **[1]** Where, C_r = Dynamic equivalent radial load $= 58.15(1 \times \cos 22.7)^{0.7} \times (16)^{2/3} \times (13.49375)^{1.8}$ $C_r = 37757.90$ N	**Now, Dynamic Load Ratings ,** $C_r = F_c (i \times \cos\alpha)^{0.7} \times Z^{2/3} \times (D_w)^{1.8}$ **[1]** Where, C_r = Dynamic equivalent radial load $= 58.15(1 \times \cos 25.7)^{0.7} \times (15)^{2/3} \times (13.49375)^{1.8}$ $C_r = 35564.27$ N
Dynamic Equivalent radial load $P_r = X F_r + Y F_a$ **[3]** Where, X = Radial load factor = 1 Y = Thrust load factor = 0 F_r = Radial load F_a = Axial load $P_r = 1 \times 5000 + 0$	**Dynamic Equivalent radial load** $P_r = X F_r + Y F_a$ **[3]** Where, X = Radial load factor = 1 Y = Thrust load factor = 0 F_r = Radial load F_a = Axial load $P_r = 1 \times 5000 + 0$
Static equivalent radial load $P_{or} = X.F_r + Y.F_a$ **[3]** Where, X = 0.35 Y = 0.57 (As per Machinery Handbook table :- 26) From ISO Standards , $P_{or} = 0.35 \times 5000 + 0.57 \times 4000$	**Static equivalent radial load** $P_{or} = X.F_r + Y.F_a$ **[3]** Where, X = 0.35 Y = 0.57 (As per Machinery Handbook table :- 26) From ISO Standards $P_{or} = 0.35 \times 5000 + 0.57 \times 4000$
Basic life ratings $L_0 = (C_r/P_r)^k$ **[1]** Where ,C_r= Basic dynamic load rating P_r = Dynamic radial load K = 3 for ball bearings L_{10} = (37757.90 /5000)^3	**Basic life ratings** $L_0 = (C_r/P_r)^k$ **[1]** Where ,C_r= Basic dynamic load rating P_r = Dynamic radial load K = 3 for ball bearings L_{10} = (35564.27/5000)^3
Life in revolutions $L = L_{10} \times 10^6$ **[1]** $= (C_r/P_r)^k \times 10^6$ $= 430.60 \times 10^6$ Revolutions Where, L = life in revolution	**Life in revolutions** $L = L_{10} \times 10^6$ **[1]** $= (C_r/P_r)^k \times 10^6$ $= 359.85 \times 10^6$ Revolutions Where, L = life in revolution
Life in working hours $L_h = 60 \times N \times L_h$ **[1]** $= L/60 \times N$ $= 430.60 \times 10^6/60 \times 1600$ Where, L_h = Life in hours N = speed of bearing in rpm	**Life in working hours** $L_h = 60 \times N \times L_h$ **[1]** $= L/60 \times N$ $= 359.85 \times 10^6/60 \times 2500$ Where, L_h = Life in hours N = speed of bearing in rpm

Average Pump Working Hour :- 7 hour / day Day :- 20 / Month Month :- 12 / year Total Working Hour :- 7 * 20 * 12	**Average Pump Working** Hour :- 7 hour / day Day :- 20 / Month Month :- 12 / year Total Working Hour :- 7 * 20 * 12
Expected Life in year = Total Hour As per Design / Working Hour = 4485.41 / 1680 = 2.66 Year	**Expected Life in year** = Total Hour As per Design /Working Hour = 2399 / 1680 = 1.42 Year

Table No. 1 Analytical calculation

Parameter	No. Of Balls	Ball diameter	Contact angle	Expected Life in year
Old design	15	13.49375	25.7°	1.42
Developed design	16	13.49375	22.7°	2.66

Table No. 2 Comparison of analytical result

IV. ANSYS RESULT

As we just have make a modeling design on a cad software & according to a different contact angle it will be proceed on ANSYS Software & then compare results of those as we done the analytical calculation**.**

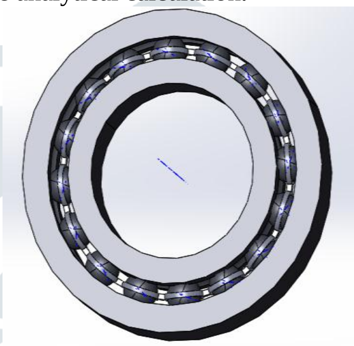
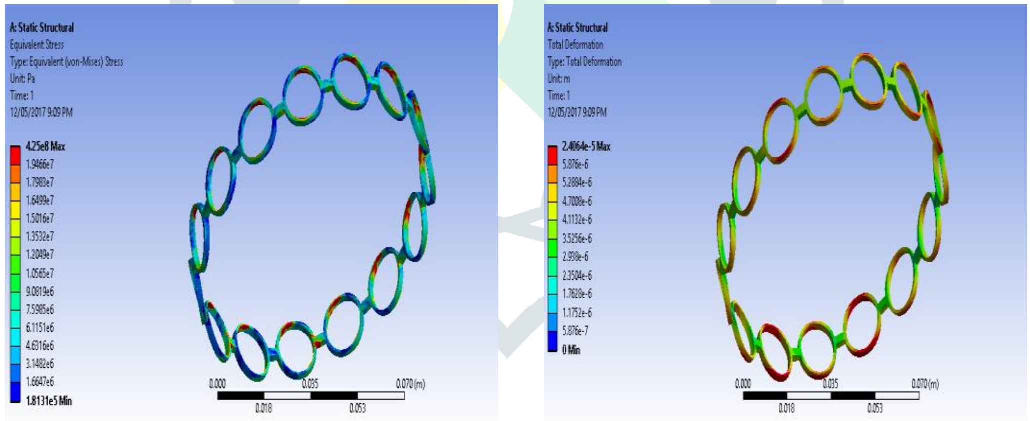


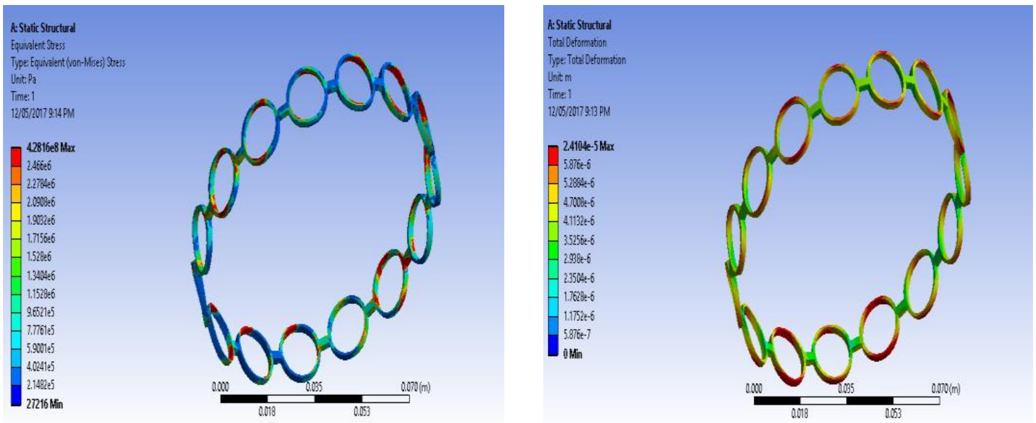
Figure 2. Assembly of Model

V. POSSIBLE SOLUTION

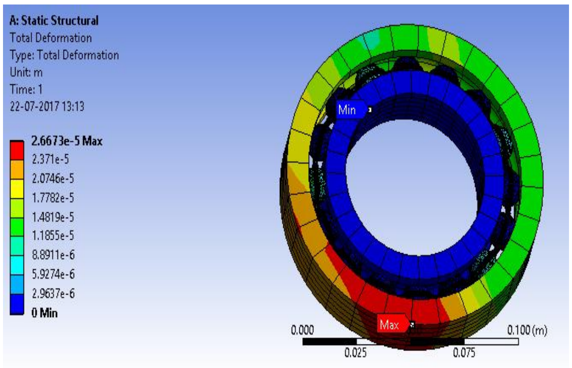
TOTAL STRESS AND DEFORMATION IN STAINLESS STEEL CAGE WITH CONTACT ANGLE 25.7°

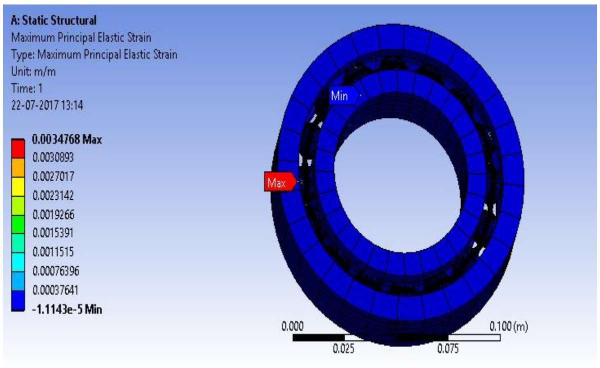


TOTAL STRESS AND DEFORMATION IN EPOXY/CARBON COMPOSITE CAGE WITH CONTACT ANGLE 25.7°



TOTAL STRESS AND DEFORMATION IN STAINLESS STEEL CAGE WITH CONTACT ANGLE 22.7°





VI. RESULTS & CONCLUSION

Sr No	Material of Bearing	Material of Cage	Angle	Stress	Deformation	Expect life in year
1	Stainless steel	Stainless steel	25.7	4.25 Pa max to 1.81 Pa min	0.240 mm max to 0 mm min	1.42
2	Stainless steel	Epoxy/Carbon Composite	25.7	4.28 Pa max to 2.72 Pa min	0.241 mm max to 0 mm min	
3	Stainless steel	Stainless steel	22.7	3.93 Pa max to 3.66 Pa min	0.266 mm max to 0 mm min	
4	Stainless steel	Epoxy/Carbon Composite	22.7	4.80 Pa max to 2.67 Pa min	0.327 mm max to 0 mm min	2.66

	Stainless Steel	**Epoxy/Carbon Composite**
Deformation (mm)	0.266	0.327
Stress (Mpa)	347	428

Table no. 3 Result Analysis

- Since the deformation and stress difference in between the two materials are remaining almost same and within the limits, that leads towards the conclusive evidence that we can replace the epoxy composite material with the structural steel within same area of application.
- As we modified bearing cage design using ANSYS and Analytical method we get a fair result in increased bearing life after changing contact angle from 25.7 to 22.7 with Epoxy/carbon composites as a cage material.

REFERENCES

[1] ISO Standards (IS: 3821, IS 3823:1988, ISO 76:1987).

[2] You Hui-yuan, Zhu Chun-xi, Li Wu-xing, “Contact Analysis on Large Negative Clearance Four-point Contact Ball Bearing”, Procedia Engineering 37 (2012) 174 – 178K.Elissa, “Titleofpaperifknown,” unpublished.