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FATIGUE ANALYSIS OF DEEP GROOV BALL BEARING USING ANSYS

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Abstract —*The aim of this research is to focus mainly on Design & Analysis of Bearing and we will discuss about fatigue analysis of deep groove bearing using Ansys & various analytical method and experimental setup. In this study the Modeling of Deep Groove Ball Bearing will be done using Creo parametric 2.0 & the steady analysis of deep groove ball bearing using ANSYS & analytical method is investigated. In this analysis three factors consider Fatigue Life, Stress & Deformation. The main goal is to providing displacement in inner race & to get change in stress and Deformation take place. All the result is based on specific dimension. By providing displacement in inner race and coefficient of friction with degree of rotation in clockwise direction with 500 rpm through which three cases obtained using ANSYS tool. An increasing the displacement value what will be the effect on stress value & Fatigue Life. An increasing the rpm what will be the effect on stress value & Fatigue Life of various parts of bearing. Then by using analytical method Stress, Deformation & fatigue life is to be calculated & will be validated.*
Various researches is help to improving the performance of deep groove ball bearing.

Keywords- *Deep Groove Ball Bearing, Fatigue life, ANSYS, Stress, Deformation.*

I. INTRODUCTION

The use of Bearing has a long history. “since, 2600 BC – The ancient Egyptians use a form of roller Bearing to help move massive bricks or huge block of stone during construction of pyramids”. “A bearing is a machine element which is permit constrains relative motion and reduces friction between two surfaces.” Bearing is a device that supports, guides, and reduces the friction of motion between fixed and moving machine parts. The term “Bearing” is derived from the verb “to bear” it means “to support another”. All type Bearing has a mainly four part • Outer race (Outer ring) • Inner race (Inner ring) • Rolling element (i.e. ball, roller, needle, sleeve) • Cage / separator (retainer) [2]

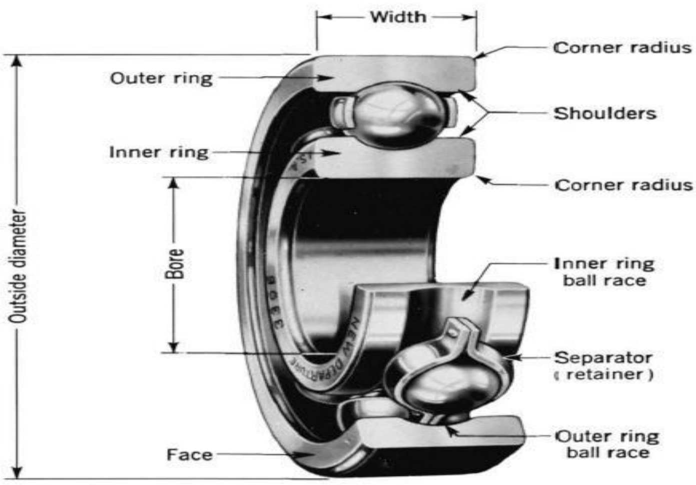


Figure 1. Deep Grove Ball Bearing

II. THEORY

Main Goal of this Study is that increase the fatigue life of Bearing. One of the goals of this study is to find the reasonable reference reaction force from the outer ring to the inner ring by means of FEM simulation.
Fatigue failures occur when metal is subjected to Repetitive and Fluctuating load.

Fatigue failures occur without any plastic deformation. The greater the applied stress rang, the shorter the life. Fatigue life is affected by verity of factors such as Temperature, Surface Finishing, Surface Contact, Overloading, Material etc

III. RESEARCH METHODOLOGY

To understand the basics of deep groove ball bearing.

To make a 2D/3D ax symmetric model with modified design data of ball bearing.

To carry out the FEA (analysis) using ANSYS.

And then compare FEA result with Analytical method.

To improvement on efficiency & life of the ball bearing. and also, Determine the safe stress level of the specimen and it is help to improving performance of ball bearing. **[1]**

IV. DESIGN, DRAWING & ANALYTICAL SOLUTION

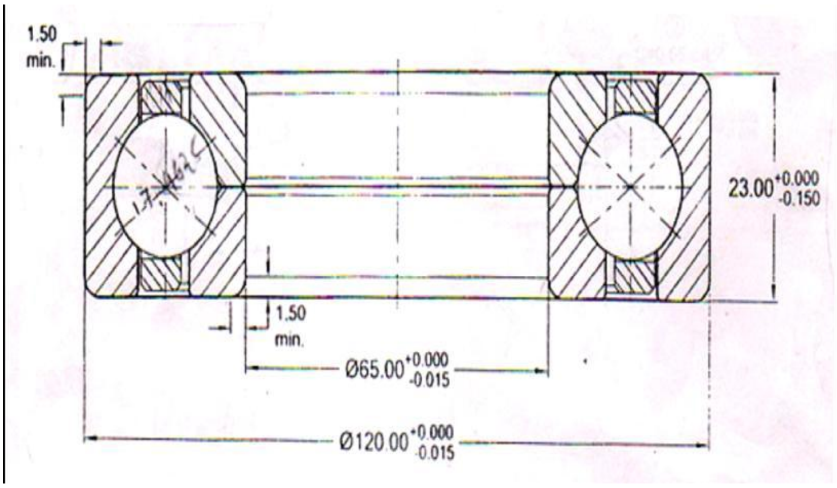


Figure 2. Assembly of Deep Grove Ball Bearing

As per the geometry seen in figure it is necessary to find out analytical solution in terms of life in hours. We just find out life of bearing by taking different contact angle.[2]

ANALYTICAL DESIGN OF THE BEARING IF CONTACT ANGLES = 25.7 ° Contact angles = 25.7 ° Ball dia Dw = 13.49375 mm Radial load Fr = 8000 N Axial load Fa = 21000 N Material 440C (stainless steel) RPM = 25000 No. of Z = 15 Ball pitch diameter = 92.5 mm No. of rows i = 1	ANALYTICAL DESIGN OF THE BEARING IF CONTACT ANGLES = 28.5 ° Contact angles = 28.5 ° Ball dia Dw = 13.49375 mm Radial load Fr = 8000 N Axial load Fa = 21000 N Material 440C (stainless steel) RPM = 25000 No. of Z = 15 Ball pitch diameter = 92.5 mm No. of rows i = 1
So, Dw cosα / dpw Where, Dw = Ball diameter α = contact angle dpw = Ball pitch diameter = (13.49375 × cos 25.7°) / 92.5 = 0.131 Therefore, as per ISO standards, Fc = 58.15	So, Dw cosα / dpw Where, Dw = Ball diameter α = contact angle dpw = Ball pitch diameter = (13.49375 × cos 28.5°) / 92.5 = 0.128 Therefore, as per ISO standards, Fc = 58.15 from interpolation. Where, Fc = factor

from interpolation. Where, Fc = factor	
Now, Dynamic Load Ratings, Cr = Fc (i x cos α)0.7 x 223 x (Dw)1.8 Where, Cr = Dynamic equivalent radial load = 58.15(1 x cos 25.7)0.7 x (15)2/3 x (13.49375)1.8 **Cr=35571.62 N**	**Now, Dynamic Load Ratings,** Cr = Fc (i x cos α)0.7 x 223 x (Dw)1.8 Where, Cr = Dynamic equivalent radial load = 58.15(1 x cos 28.5)0.7 x (15)2/3 x (13.49375)1.8 **Cr=34960.55 N**
Dynamic Equivalent radial load Pr=xFr+yFa Where, X = radial load factor = 1 Y = Thrust load factor = 0 Fr = Radial load Fa = Axial load Pr=1 x 8000 + 0 **Pr=8000N**	**Dynamic Equivalent radial load** Pr=xFr+yFa Where, X = radial load factor = 1 Y = Thrust load factor = 0 Fr = Radial load Fa = Axial load Pr=1 x 8000 + 0 **Pr=8000N**
Static equivalent radial load Por = X.Fr + Y.Fa Where, X = 0.5 Y = 0.29 From ISO Standards, Por = 0.5 x 8000 + 0.27 x 21000 **Por = 9670 N**	**Static equivalent radial load** Por = X.Fr + Y.Fa Where, X = 0.5 Y = 0.29 ISO Standards, Por = 0.5 x 8000 + 0.27 x 21000 **Por = 9670 N**
Basic life ratings L0 = (Cr/Pr)k Where , Cr= Basic dynamic load rating Pr = Dynamic radial load K = 3 for ball bearings L10 = 35571.62/8000 **= 87.9105**	**Basic life ratings** L0 = (Cr/Pr)k Where , Cr= Basic dynamic load rating Pr = Dynamic radial load K = 3 for ball bearings L10 = 34960.55/8000 **= 83.457**
Life in revolutions L = L10 x 106 = (Cr/Pr)k x 106 **= 87.9105 x 106 Revolutions** Where, L = life in revolution	**Life in revolutions** L = L10 x 106 = (Cr/Pr)k x 106 **= 83.457 x 106 Revolutions** Where, L = life in revolution
Life in working hours L= 60 x Nx Lh Lh = L/60 x N = 87.9105 x 106/60 x 25000 Where, Lh = Life in N = speed of bearing in rpm **Lh=58.607 hours**	**Life in working hours** L= 60 x Nx Lh Lh = L/60 x N = 83.457 x 106/60 x 25000 Where, Lh = Life in N = speed of bearing in rpm **Lh=55.638 hours**

Table no. 1 Analytical calculation

Parameters	No. Of Balls	Ball diameter	Contact angle	Life in working hours

Old design	15	13.49375	28.5○	55.638
Developed design	15	13.49375	25.7○	58.607

Table no.2 Comparison of analytical result

IIV. ANSYS RESULT

After analytical calculation of bearing we have just make a design model in cad software and according to different contact angle proceed to anys software & compare the results of those.

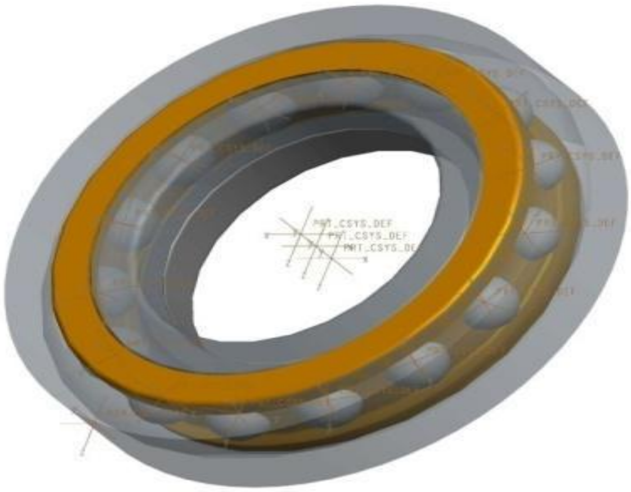
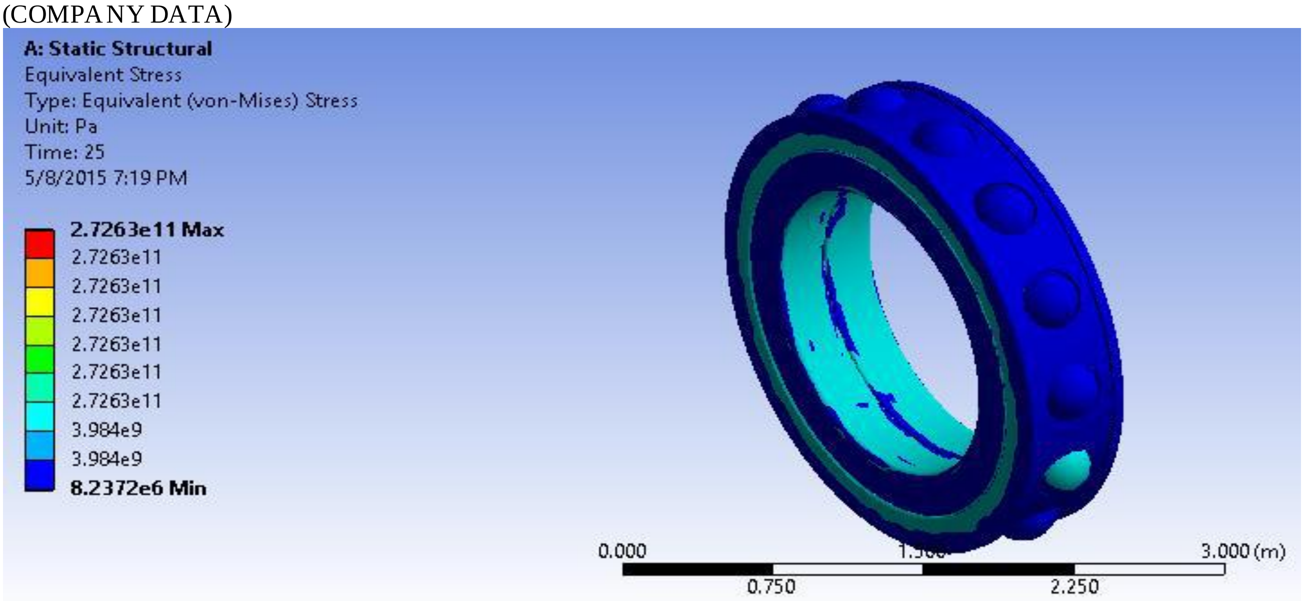
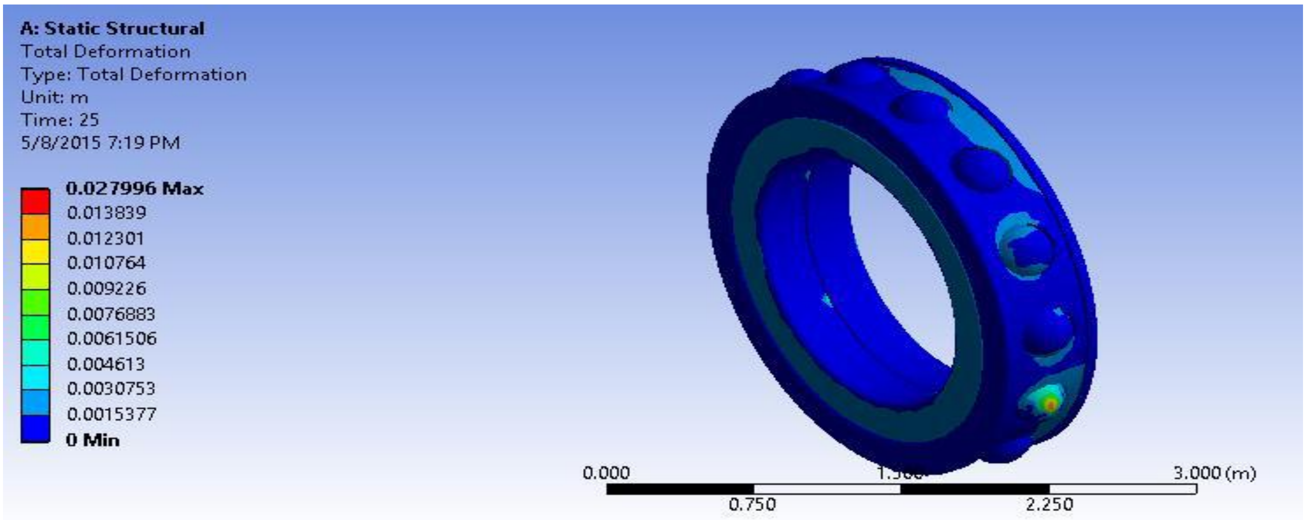


Figure 2. Assembly of model

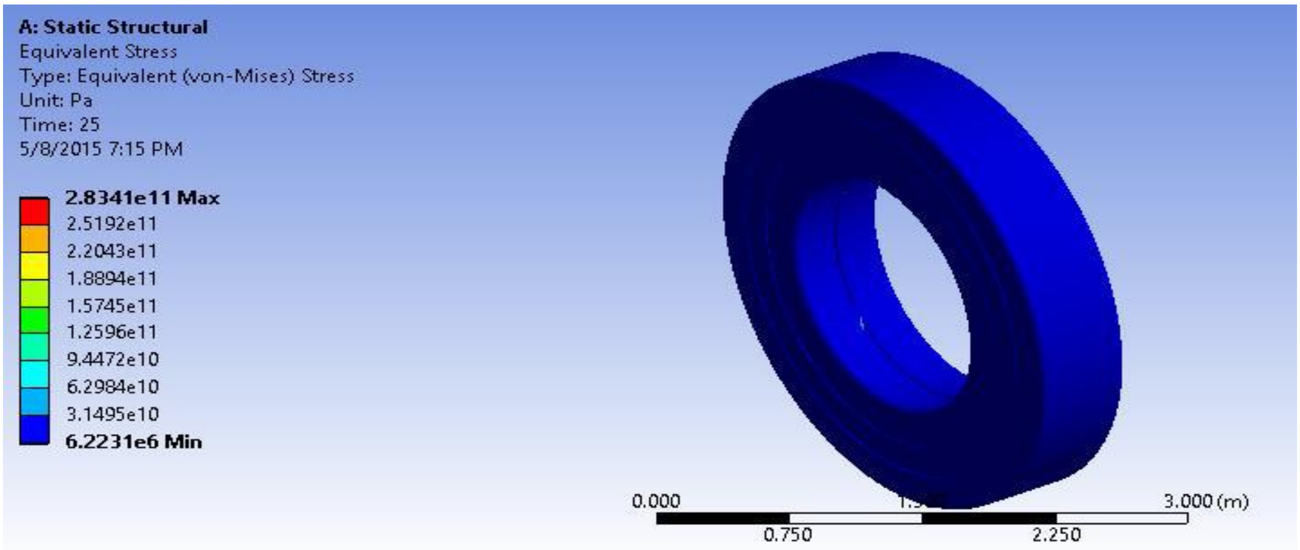
IV. POSSIBLE SOLUTION

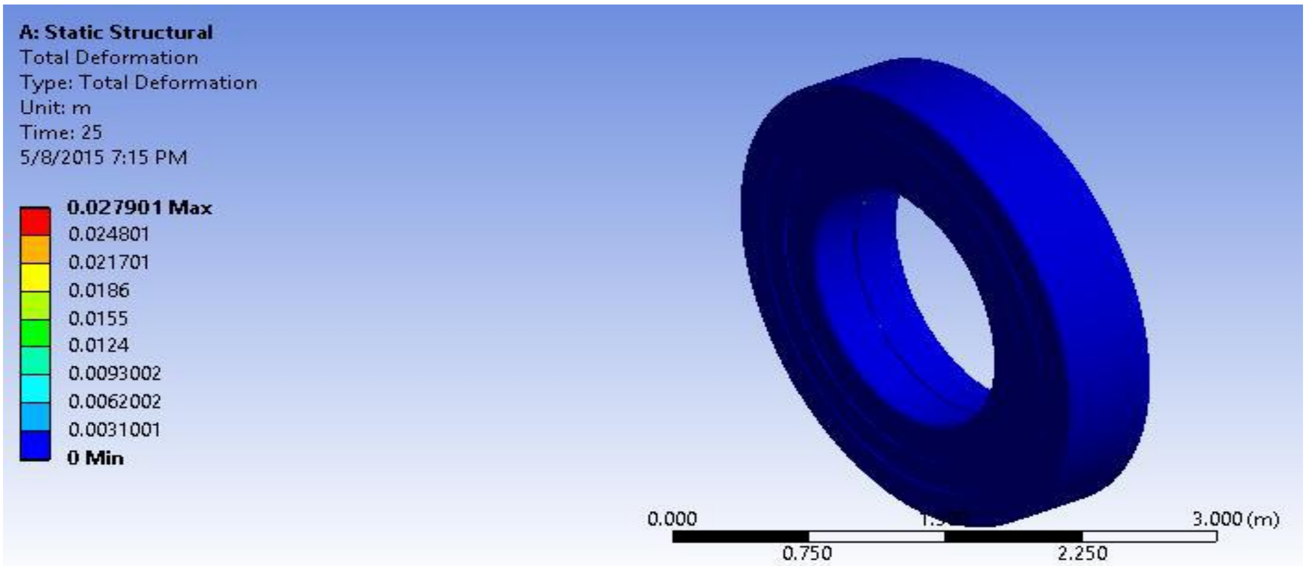
TOTAL STRESS AND TOTAL DEFORMATION IN STAINLESS STEEL WITH CONTACT ANGLE 28.5○ (COMPANY DATA)



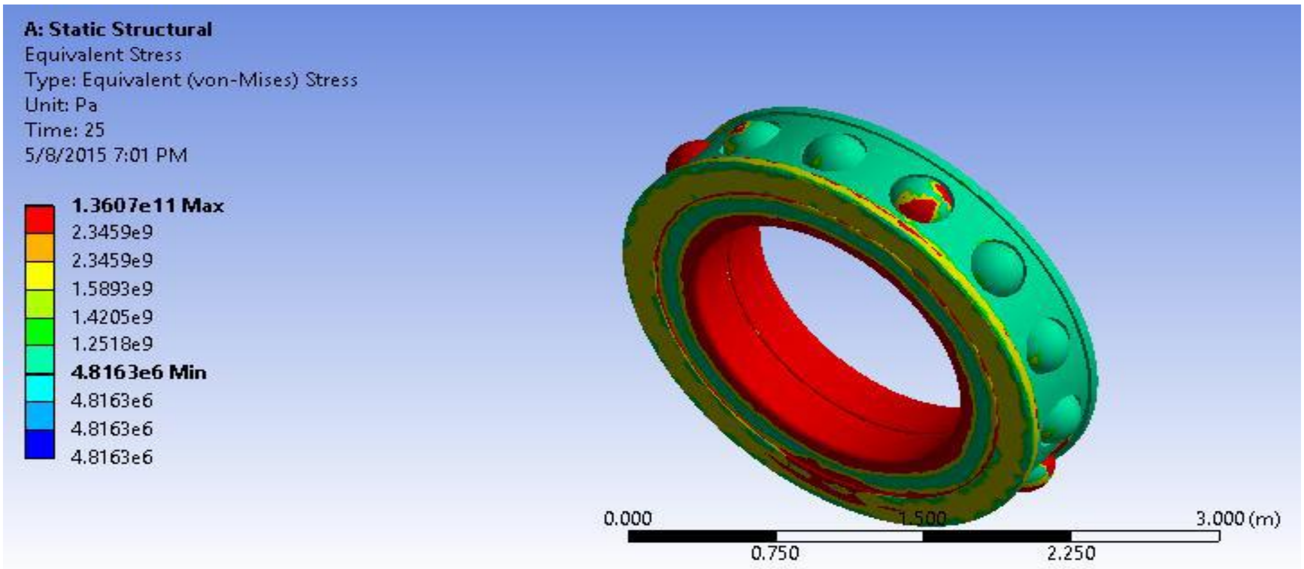


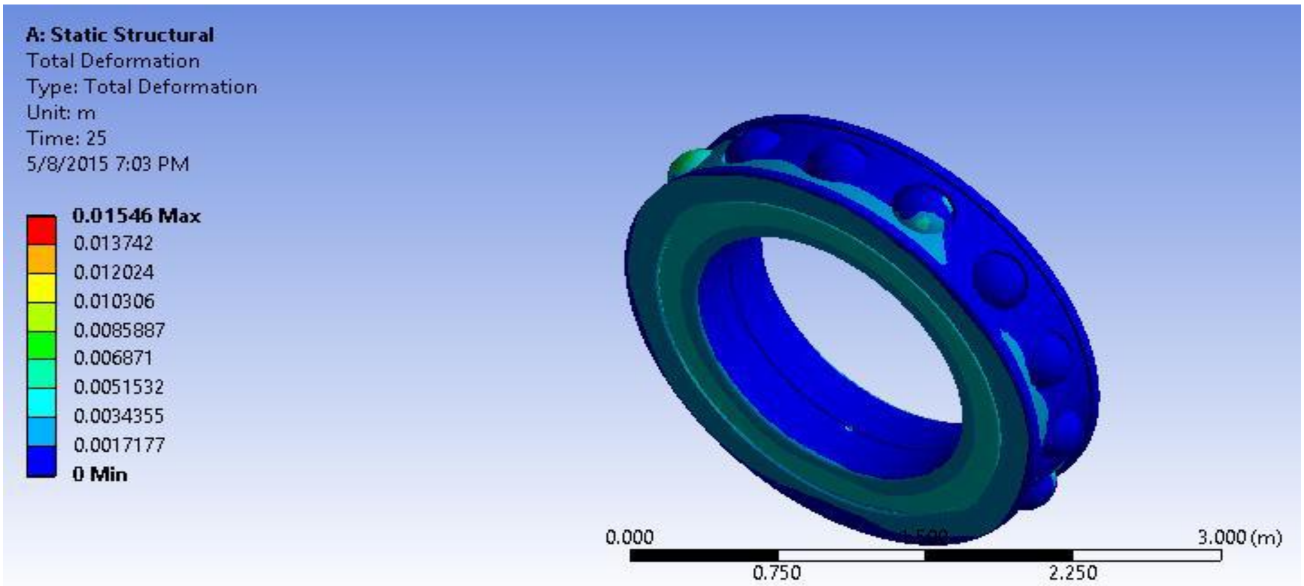
TOTAL STRESS AND TOTAL DEFORMATION IN STRUCTURAL STEEL WITH CONTACT ANGLE 28.5○



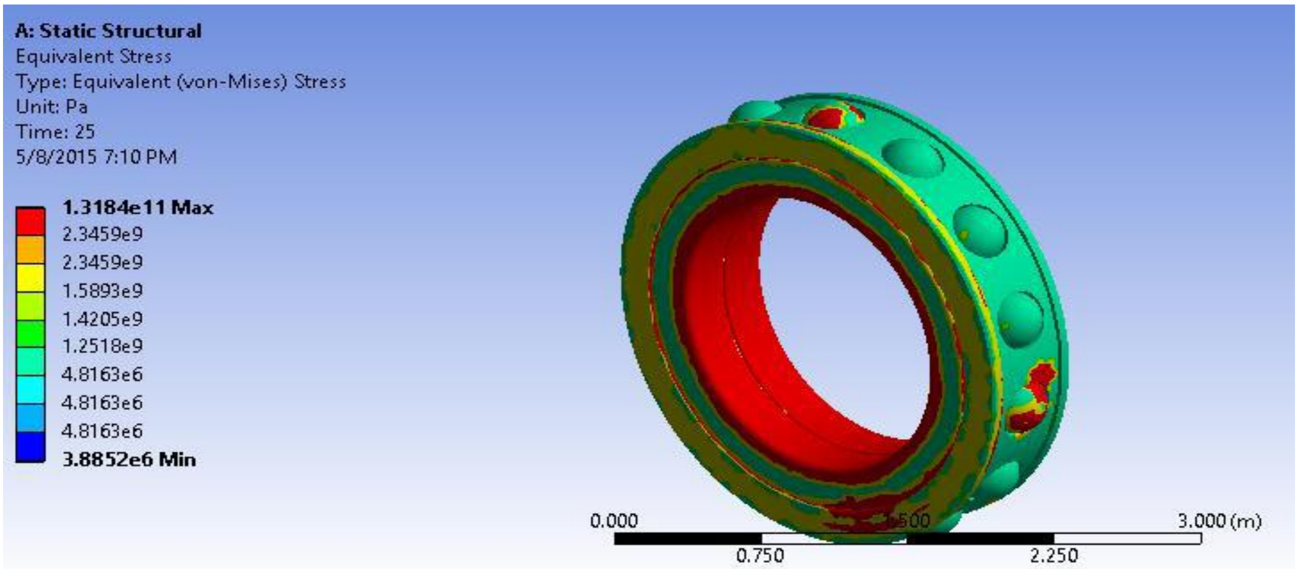


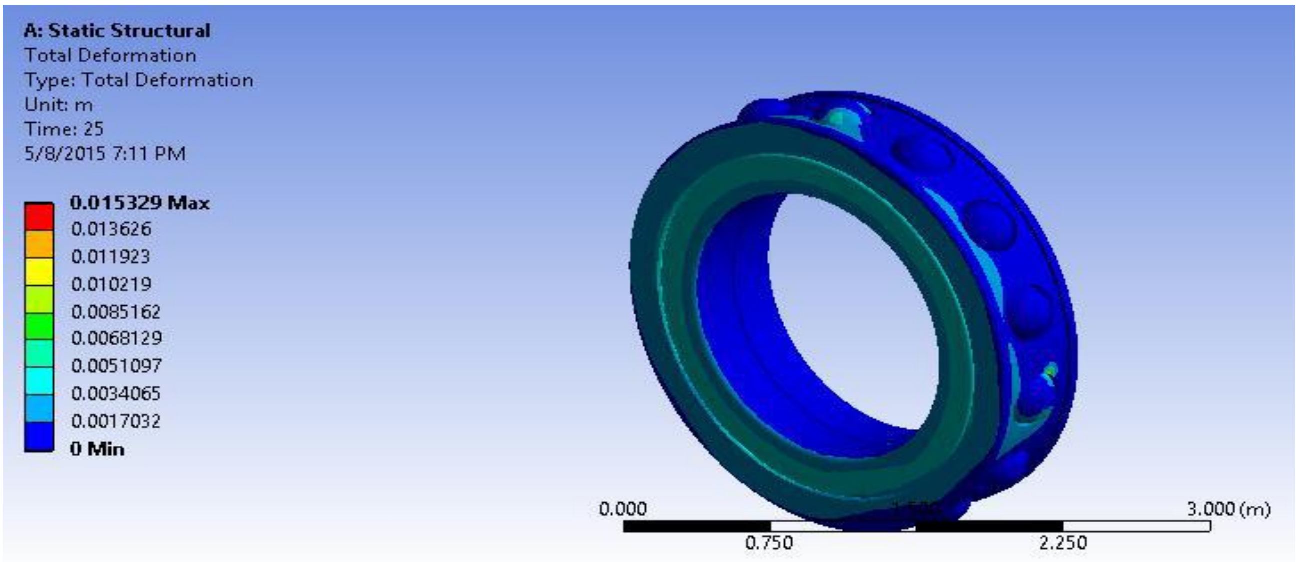
TOTAL STRESS AND TOTAL DEFORMATION IN STRUCTURAL STEEL WITH CONTA CT ANGLE 25.7○





TOTAL STRESS AND TOTAL DEFORMATION IN STAINLESS STEEL WITH CONTACT ANGLE 25.7○





RESULTS & CONCLUSION

Sr No	Material	ANGLE	STRESS	DEFORMATION	LIFE
1	Stainless steel	28.5	2.7253 max to 8.2372 min	0.027996 max to 0 min	55.638
2	Stainless steel	28.5	2.8341 max to 6.2231 min	0.027901 max to 0 min	
3	Stainless steel	25.7	1.3607 max to 4.8163 min	0.01546 max to 0 min	
4	Stainless steel	25.7	1.31484 max to 3.8852 min	0.015329 max to 0 min	58.607

Table no.3 Result analysis

So as per our aim we modify company's bearing design using Ansys and Analytical method and we get fair result with increased life of bearing by changing angle from 28.5to 25.7 with stainless steel material So finally we get fair result of deformation in Angle and Fair life in Analytical method

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