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Design, Modelling and Analysis of a Single Raw Four Point Angular Contact Split Ball Bearing to Increase its Life.

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***Abstract* – In this paper design of four points angular contact ball bearing is done which is used in Propeller shaft of an air craft. This type of ball bearing can support radial load and axial load in both directions due to the four points of contact available. The inner ring of a four point angular contact ball bearing is split so because of larger ball quantity, higher carrying capacity is possible. In current work the models of the parts of the bearing are prepared considering the drawings of the bearing. With the help of analytical design life of bearing in working hours can be calculated. With the help of design parameters bearing can be modelled. After modelling, with the help of dynamic analysis of bearing its life in working hours can be calculated.**

***Index terms* - Four point angular contact ball bearings, Split ball bearing, Propeller shaft**

I. INTRODUCTION

An angular contact ball bearing uses axially asymmetric races. Angular contact bearings better support "combined loads". It achieves this by using at least two races to contain the balls and transmit the loads through the balls. In most applications, one race is stationary and the other is attached to the rotating assembly (e.g., a hub or shaft). As one of the bearing races rotates it causes the balls to rotate as well. This kind of bearings can be used in transmissions [6]. In this paper four point angular contact ball bearing is designed and analysed which is used on propeller shaft of aircraft. As a result life of bearing can be calculated which is less because of very high speed[1].

II. LITERATURE REVIEW

It has been studied that all angular-contact ball bearings have similar features regarding geometry, mechanism, and structure [2]. The stiffness of angular-contact ball bearings has a significant influence on the dynamics of a rotating shaft and the precision of the machine system [3] the rolling contact fatigue (RCF) life of rolling elements can be determined by full-scale bearing endurance tests. These tests that are conducted in bearing life test rigs are expensive and time consuming [4]. Bearing designers would like to understand the impact of four variables namely Ball material density, Subsurface residual stress, Gradient in yield strength with depth Raceway surface hardness/yield strength that are thought to affect spall propagation[7]. The normal contact stresses between steel ball and inner and outer ring raceway are calculated under different negative clearance when bearing idling [8]. One important demand on spindle system in modern machine tools is to realize higher rotational speeds in order to increase the machining efficiency [5]. Short spall propagation times of failing main shaft ball bearings of aircraft engines are a serious safety concern for single engine aircraft [7].

III. ANALYTICAL DESIGN OF THE BEARING

A. Input Data

Ball diameter D_w = 13.49375 mm
Radial load F_r = 8000 N
Axial load F_a = 21000 N

Material = 440C
RPM = 25000
No. of balls Z = 15
Ball pitch diameter = 92.5 mm
No. of rows i = 1

B. Contact Angle

$\alpha = \cos^{-1}[1- \{G_r / 2(r_i + r_c - D_w)\}]$[12]
$= \cos^{-1}[1- \{0.15 / 2(7.06 +7.06 - 13.49375\}]$
$\alpha = 28.5^{\circ}$
where,
Gr = radial clearance
r_i = inner groove radius
r_c = outer groove radius
α = contact angle
D_w = Ball diameter
$D_w \times \cos\alpha / d_{pw}$
$= (13.49375 \times \cos 28.5^{\circ}) / 92.5$
$= 0.128$
Where,
D_w = Ball diameter
α = contact angle
d_{pw} = Ball pitch diameter
Therefore, as per ISO standards,
F_c = 58.15 from interpolation.
Where,
F_c = factor for dynamic load rating calculation

C. Dynamic Load Ratings

C_r = Dynamic equivalent radial load
$C_r = F_c (i \times \cos\alpha)^{0.7} \times (Z)^{2/3} \times (D_w)^{1.8}$[11]
$= 58.15(1 \times \cos 28.5)^{0.7} \times (15)^{2/3} \times (13.49375)^{1.8}$
$C_r = 34960.55$ N

D. *Dynamic Equivalent Radial Load*

$P_r = X F_r + y F_a$[1]
Where,
X = radial load factor = 1
Y = Thrust load factor = 0
F_r = Radial load
F_a = Axial load
$P_r = 1 \times 8000 + 0$
$P_r = 8000$N

E. Static Equivalent Radial Load

$P_{or} = X F_r + Y F_a$[1]
Where,
X = 0.5
Y = 0.29
From ISO Standards,
$P_{or} = 0.5 \times 8000 + 0.27 \times 21000$
$P_{or} = 9670$ N

F. Basic Life Ratings

$L_0 = (Cr/P_r)^k$ [11]
Where,
C_r = Basic dynamic load rating
P_r = Dynamic radial load
K = 3 for ball bearings
$L_{10} = 34960.55/8000$
$= 88.457$

G. Life in Revolutions

$L = L_{10} \times 10^6$[11]
$= (C_r/P_r)^k \times 10^6$
$= 83.457 \times 10^6$ Revolutions
Where,
L = life in revolution

H. Life in Working Hours

$L = 60 \times N \times L_h$[11]
$L_h = L/60 \times N$
$= 83.457 \times 106/60 \times 25000$
Where,
L_h = Life in
N = speed of bearing in rpm
L_h = 55.638 hours

IV. ASSEMBLY OF THE BEARING

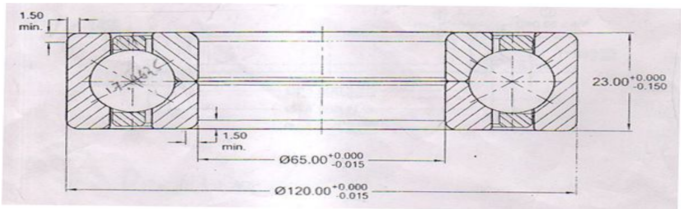


Figure 1 Assembly of the Bearing[10]

V. DEVELOPED DESIGN

Table 1 Developed design

Parameters	No. Of Balls	Ball diameter	Contact angle	Life in working hours
Developed design	15	13.49375	28.5°	55.638

VI. MODELING OF THE BEARING

A. Model and Geometry of Outer Race of the Bearing

Figure 1 Model and Geometry of outer race of the Bearing

B. Model and Geometry of Inner Race of the Bearing

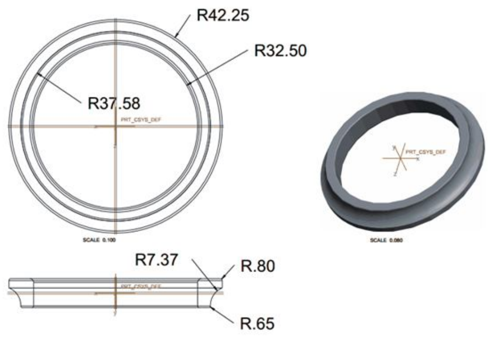


Figure 2 Model and geometry of inner race of the Bearing

C .Model and Geometry of Cage of the Bearing

Figure 3 Model and geometry of cage of the Bearing)

D. Model and geometry of Ball of the Bearing

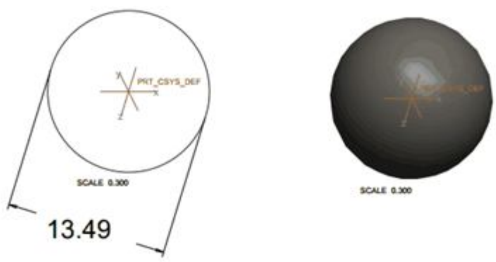


Figure 4 Model and geometry of Ball of the Bearing

E. Assembly Model of the Bearing

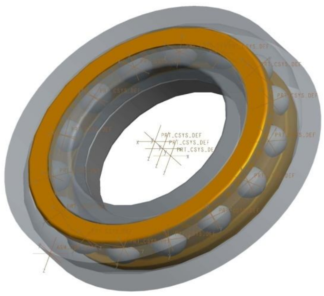


Figure 5 Model and geometry of assembly of the Bearing

VII. DYNAMIC ANALYSIS RESULTS

A. Load Direction for Linear Static Analysis in Static Mode

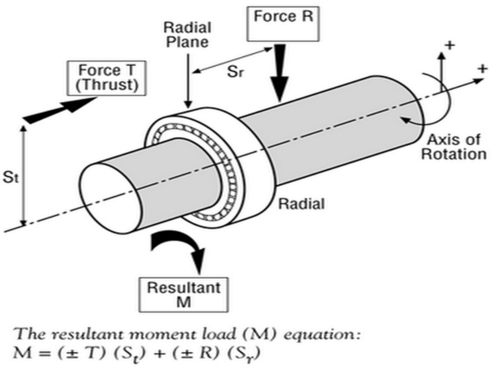


Figure 6 Load directions in static Condition [12]

B. Pre-Processing for Dynamic Analysis of Bearing

For the analysis of bearing boundary condition and force these parameters are required to be applied and then next step is meshing. Meshing is nothing but dividing objet in to small parts for results on that all different parts. In our object meshing is done by tetra method, fixed support applied at cage as a boundary condition, frictionless support applied at inner & outer ring due to lubrication, rotational force applied at a centre (25000 RPM), Axial force applied at outer ring with value of 21000 N.

C. Result Obtained by Dynamic Analysis in Form of Stress

Here, there are results in form of Von-misses Stress produced due to rotation of bearing at 25000 RPM.

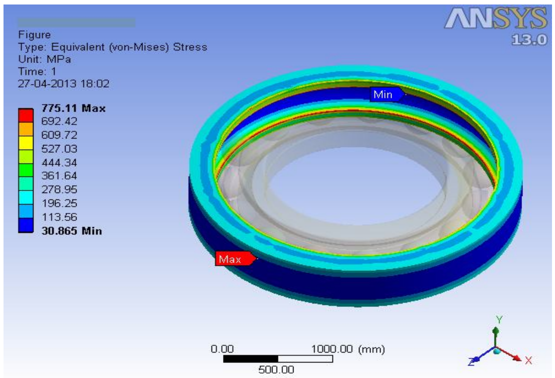


Figure 7 Stresses on Outer Race

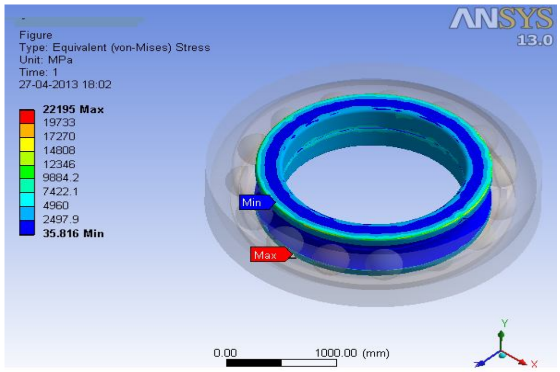


Figure 8 Stresses on Inner Race

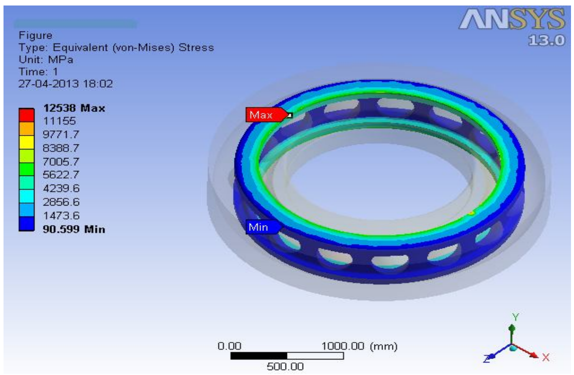
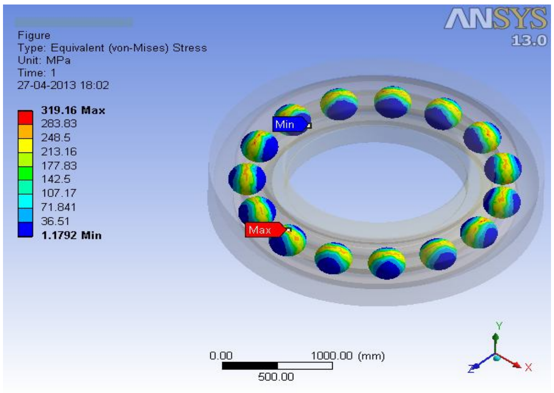


Figure 9 Stresses on Cage



(Figure 10 Stresses on Balls)

D. Results Obtained by Dynamic Analysis in Form of Life

Here, main objective of this research is to increase bearing life in working hours. It is resulted by changing inner grove radius, outer groove radius, contact angle, number of balls, and diameter of balls. After these changes following results are obtained.

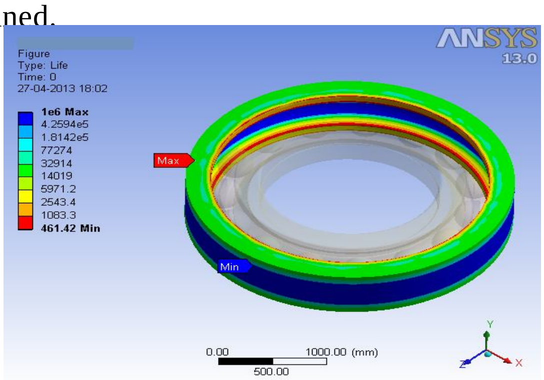


Fig. 11.Life of Outer Race

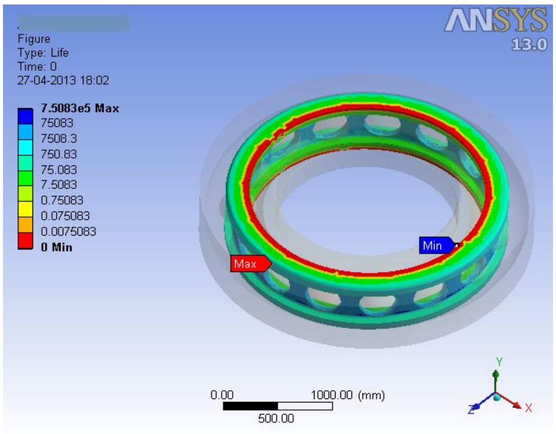


Figure 12 Life of Cage

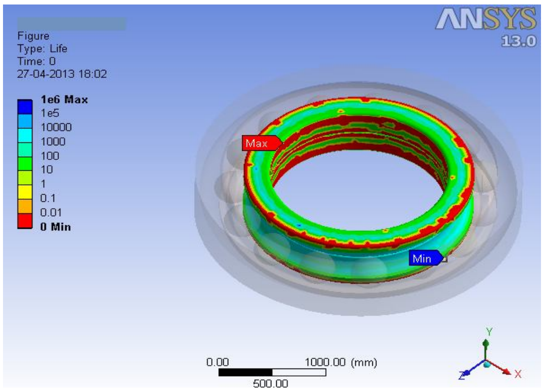


Figure 13 Life of Inner Race

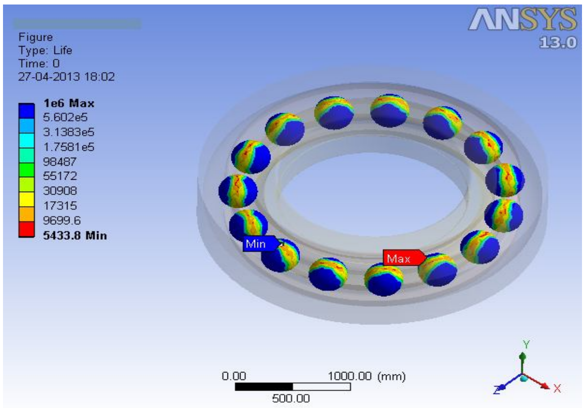


Figure 14 Life of Balls

VIII. RESULTS AND CONCLUSION

Table 2 Comparison of life of the Bearing

Life in working hours by design	55.638 hours
Life in working hours by analysis	70 hours

In analytical design, by increasing numbers of ball, decreasing ball diameter, and changing the contact angle, the life in working hours can be increased. From the results of analytical design and analysis it can be seen that life of bearing is nearly same in both cases. Due to higher stresses, the life in working hours is limited to hours. It can be further increased by changing the parameters, material and lubrication of the Bearing.

IX. REFERENCES

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