Int. J. Adv. Res. Sci. Technol. Volume 3, Issue3, 2014, pp.136-151 # ICV 5.14

IJARST

International Journal of Advanced Research in Science and Technology

journal homepage: www.ijarst.com

ISSN 2319 – 1783 (Print)

ISSN 2320 – 1126 (Online)

Modeling and Contact Analysis of Composite (FRP) Material Lamination on Cylindrical Roller Bearing

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A R T I C L E I N F O

Article history:

Received 10Oct. 2014

Accepted 03Nov. 2014

Available online 04Nov. 2014

Keywords:

Cylindrical Roller Bearing,
Contact Analysis,
Fiber Reinforced polymer,
Modeling,
Analysis of Bearing.

A B S T R A C T

Fiber reinforced polymer (FRP) composites are an important class of tribological materials. They possess unique self -lubrication capabilities and low noise which make them suitable for applications like seals, bearings, gears and artificial prosthetic joints. The FRP composite bearings are ideal for high load low speed applications or where normal lubrication is difficult or costly. The friction and wear behavior of FRP composites varies with fiber orientation and sliding direction. For the purpose of fully utilizing the beneficial contact characteristics of FRP composite, it is necessary to obtain an in-depth knowledge of their contact behavior.

In this work, the compliance behavior of FRP composite bearings is studied. The frictional sliding contact between a FRP composite and a rigid parabolic cylinder is analyzed. The influence of sliding direction, fiber and matrix material combinations, volume fraction of the fiber, frictional coefficient and fiber ply orientation on the contact pressure distribution and the contact area for unidirectional FRP composite bearings are evaluated. A finite element model is developed using ANSYS12.0 and the results obtained from the analysis are compared with the analytical results. The influence of sliding direction on the contact pressure distribution for cross FRP composite bearings is studied and compared with unidirectional FRP composite bearings.

Introduction:

A bearing is a device to allow constrained relative motion between two or more parts, typically rotation or linear movement. Bearings may be classified broadly according to the motions they allow and according to their principle of operation as well as by the directions of applied loads they can handle

Roller bearings are used for systems which require exceptionally large load-carrying capacity but cannot feasibly be obtained by a ball bearing. Roller bearings are normally much stiffer structures than the ball bearing with a comparable geometry size and they can provide greater fatigue endurance as well. However they are more expansive than the ball bearings because it is more difficult to manufacture and assemble. Common roller bearings use cylinders of slightly greater length than diameter. Roller bearings typically have higher load capacity than ball bearings, but alower capacity and higher friction under loads perpendicular to the primary supported direction. If the inner and outer races are misaligned, the bearing capacity often drops quickly compared to either a ball bearing or a spherical roller bearing. Roller bearings are the type of rolling-element-bearing dating back to at least 40 BC.

Figure: 1. Cylindrical Roller Bearing

Problem Statement:

The project is concerned with designing the cylindrical roller bearing used in the ic-engine which had a upper bound of stress coming onto the bearing is critical for a steel bearing and using this lead would lead to a catastrophic failure of the engine, testing with the use of laminated FRP material is done by designing in PRO-E and to transfer that assembly into ANSYS and perform, static and contact analysis by using composite material lamination. In spite of the number of investigations devoted to bearing research and analysis there still remains to be developed, a general approach capable of predicting the effects of contact stresses. The aim is to use a numerical approach to develop theoretical models of the behavior of cylindrical roller bearings, predict the effect of bearing stresses and to compare the effects with the theoretical analysis performed in ANSYS

Calculation Procedure for Bearing:

Rating life:

Rating life is defined as the life of a group of apparently identical ball or roller bearings, in number of revolutions or hours, rotating at a given speed, so that 90% of the bearings will complete or exceed before any indication of failure occur. Suppose we consider 100 apparently identical bearings. All the 100 bearings are put onto a shaft rotating at a given speed while it is also acted upon by a load. After some time, one after another, failure of bearings will be observed. When in this process, the tenth bearing fails, then the number of revolutions or hours lapsed is recorded. These figures recorded give the rating life of the bearings or simply L_{10} life (10 % failure). Similarly, L_{50} means, 50 % of the bearings are operational. It is known as median life bearing load.

If two groups of identical bearings are tested under loads P_1 and P_2 for respective lives of L_1 and L_2, then,

$$\frac{L_1}{L_2} = \left(\frac{P_2}{P_1}\right)^a$$

Where,

L: life in millions of revolution or life in hours

a: constant which is 3 for ball bearings and 10/3 forroller bearings

Basic load rating: It is that load which a group of apparently identical bearings can withstand for a rating life of one million revolutions. If say, L is taken as one million then the corresponding load is,

$$C = P\,(L)^{\frac{1}{a}}$$

Where, C is the basic or dynamic load rating

Therefore, for a given load and a given life the value of C represents the load carrying capacity of the bearing for one million revolutions. This value of C, for the purpose of bearing selection, should be lower than that given in the manufacturer's catalogue. Normally the basic or the dynamic load rating as prescribed in the manufacturer's catalogue is a conservative value, therefore the chances of failure of bearing is very less.

Equivalent radial load: The load rating of a bearing is given for radial loads only. Therefore, if a bearing is subjected to both axial and radial load, then an equivalent radial load is estimated as,

or

$$P_e = XVP_r + YP_a$$

Where,

P_e: Equivalent radial load

P_r: Given radial load

P_a: Given axial load

V: Rotation factor (1.0, inner race rotating; 1.2, outer race rotating)

X: A radial factor

Y: An axial factor

The selection Procedure:

Depending on the shaft diameter and magnitude of radial and axial load a suitable type of bearing is to be chosen from the manufacturer's catalogue, either a ball bearing or a roller bearing. The equivalent radial load is to be determined from equation. If it is a tapered bearing then manufacturer's catalogue is to be consulted for the equation given for equivalent radial load. The value of dynamic load rating C is calculated for the given bearing life and equivalent radial load. From the known value of C, a suitable bearing of size that conforms to the shaft is to be chosen. However, some augmentation in the shaft size may be required after a proper bearing is chosen.

Figure: 2. Field marshal I.C. Engine

Table: 1. Specification of Field Marshal I.C Engine

Engine Type		PMV
Power H.P. / kW		36
No.of Cylinder		1
Bore mm		23
Stroke mm		150.6
Speed R.P.M.		265
Specific Fuel Consumption	g/kWh	4800
	g/bhp/h	105
Compression Ratio		17:1
Displacement c.m.3		1432
Fuel		HSD
Lubrication		SAE-30/40
Bare Weight of Engine (Approx.) Kgs.		320
Gross Weight of Engine (Approx.) Kgs.		455

Selection of Bearing:

Calculation:

From the sheet Power = 36kW,

Stroke length = 150.6 mm,

Speed = 4800rpm.

P = 2πNT/60 (by reference 10)

T=36*1000*60/2 πN

T = 71.65 N.m

Now,

Torque = Force * Crank Radius

71.65 = Force * stroke length / 2

=> Force = 951.4 N

Here for a pure cylindrical roller bearing axial force

Fa= 0.

Fr = 951.4 N

Fa / Co = 0 and Fa / Fr = 0. Now for Peq,

$$\therefore P_{eq} = (X * F_r + Y * F_a) * S$$

Here Fa= 0 and X=1, S= 1.

Peq. = Fr = 951.4 N

During a single revolution of the wheel, point A will experience a cycle of stress values varying from zero (when point A lies well outside the contact zone) to a maximum state of stress (when A lies within the contact zone and on the line of action of the 800 lbf force.) We expect point A to "feel" the effects of a semi-elliptical contact pressure distribution as point A moves into and through the contact zone. Thus, we need to calculate the contact stresses for a depth of z = 0.015 inch, which we expect to lie within the contact zone.

Material selection:

The principal fibers in commercial use are various types of glass, carbon and Kevlar. All these fibers can be incorporated into a matrix either in continuous lengths or in discontinuous lengths in the matrix material may be a plastic or rubber polymer, ametal or a ceramic. Comparison of various properties like resistance to bending, elongation and load carrying capacity between composite material and other materials are shown in Table:2

Property	Graphite/ Epoxy	Glass / Epoxy	Steel	Aluminum
Specific gravity	1.6	1.8	7.8	2.6
Young's modulus (GPa)	181.0	138.6	206.8	68.95
Ultimate tensile strength (MPa)	1500	1062	648.1	275.8
Coefficient of Thermal expansion (mm/°C)	0.02	8.6	11.7	23

Modeling of Bearing Using pro/ Engineer:

With the specifications obtained above we model the gear in Pro E by using many relations and parameters in Pro E. When we give the parametric equation of involute it automatically draws the curve. The parametric equation of the involute profile is given as follows.

Cylindrical Roller Bearing Dimensions:

Inner diameter(mm)	65
Outer diameter(mm)	120
Bore diameter(mm)	23
Basic load ratings (N)	106000
Dynamic C	118000
Static CO	15600
Fatigue load pu (N)	
Speed rating lubrication	
Grease (r/min)	4800
Oil (r/min)	5600
Mass (kg)	1.05
Roller diamter(mm)	13
Roller height (mm)	13
Outer ring thickness(mm)	6
Inner ring thickness(mm)	10
Center height(mm)	22
Roller outer distance(mm)	8
inner distance(mm)	4
Bearing number	NU 213

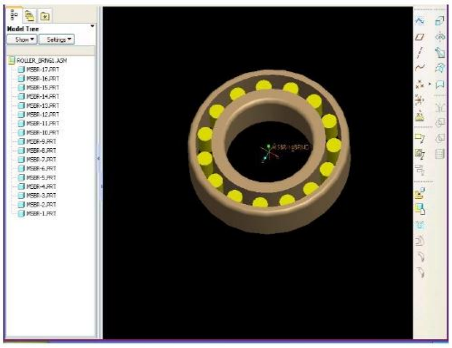

Figure: 3. Modeling of Cylindrical Roller Bearing using pro-e

FE Model of Composite Cylindrical Roller Bearing:

The finite element model of the elastic half- plane is given in Fig. The details of the model and the material properties are given below:

Element type:SOLID46
Real constant: Set 1
No. of layers: 1
Material ID:1
Theta = 0;
Thickness = 0.02 m
Material: E-Glass/Epoxy (Ex=138e9; Ey=Ez=8.96e9;ν = 0.3 ;Gx,Gy,Gz = 7.1e9)
Model : Solid model (Length = Width = Height = 0.02 m)
Mesh : Volume mesh
Size control (Line):
No. of element divisions = 8
No. of nodes = 162
No. of elements (Layered) = 706

FE Model of Cylinder Roller Bearing:

The finite element model of the rigid parabolic cylinder is given in Fig. The details of the model and the material properties are given below:

Element type : SOLID 187
Material: Steel (E = 200 GPa, $\nu = 0.3$)
Material ID =2
Model : Solid model
Radius = 0.008 m,
Length = 0.02m,
Mesh : Volume mesh
Size control (Line):
No. of element divisions = 8
No. of nodes = 1044,
No. of elements = 509

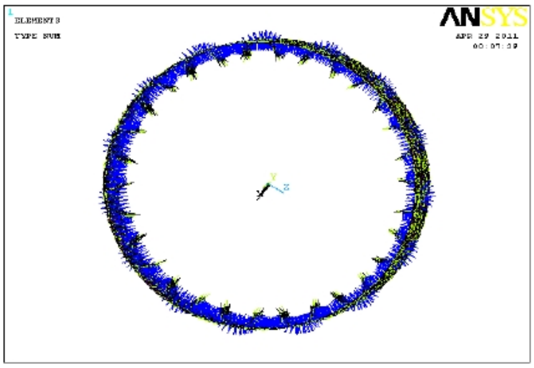

Figure: 4. Contact Element Type Number

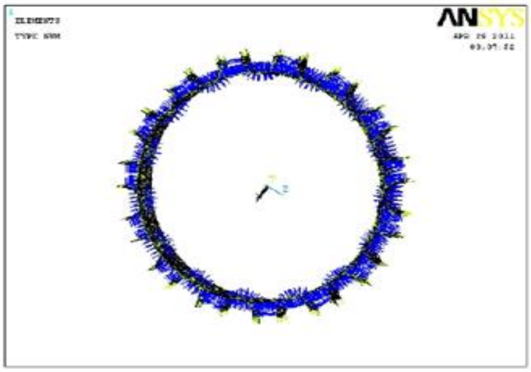

Figure: 5. Contact Element of Bearing

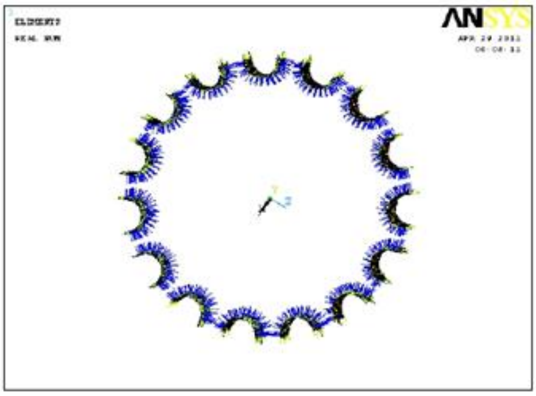

Figure: 6. Contact Element (Real Number)

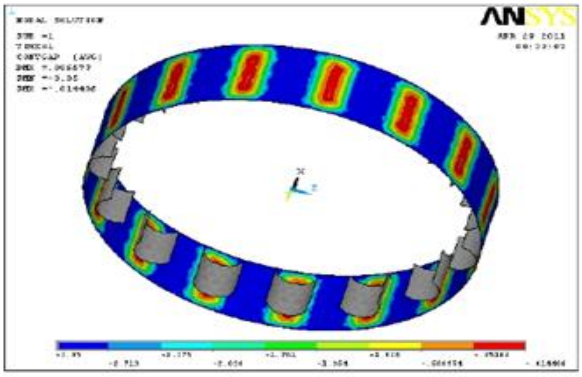

Figure: 7. Contact Element Gap

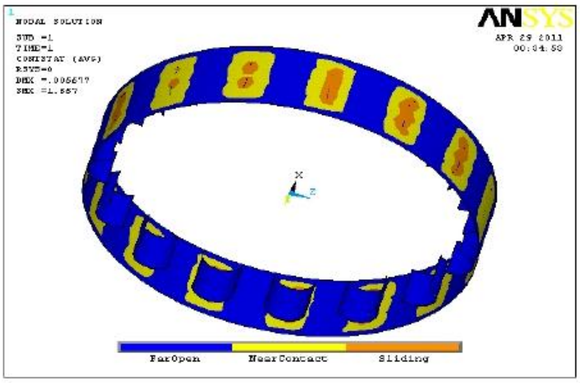

Figure: 8. Contact Status

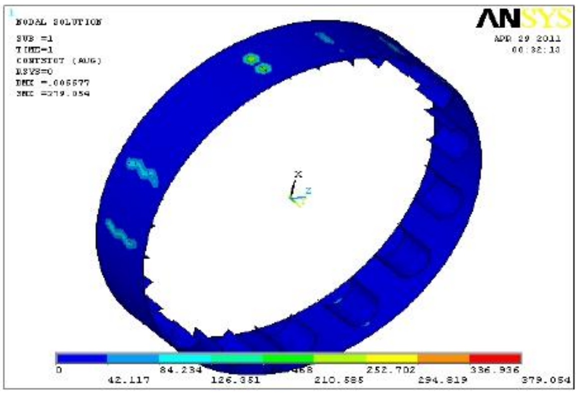

Figure: 9. Bearings at Contact Stresses of Load F=9514N

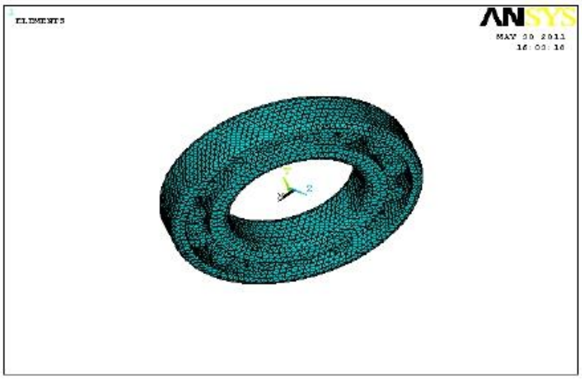

Figure: 10. Meshed Portion of Bearings

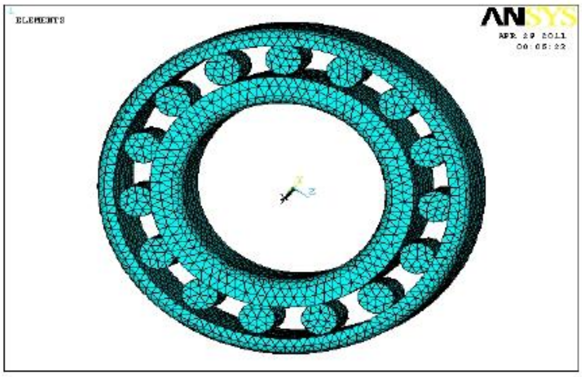

Figure: 11. Fe Modeling of Meshing Of Bearings

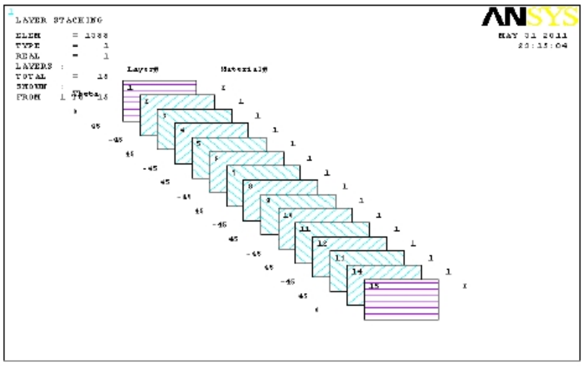

Figure: 12. Composite Layers of Steel and Glass/Epoxy

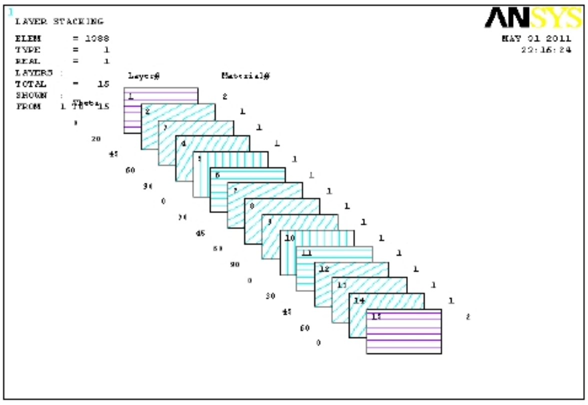

Figure: 13. Composite Layers for Steel and Glass/Epoxy

Composite Analysis at Load F= 7611.2N

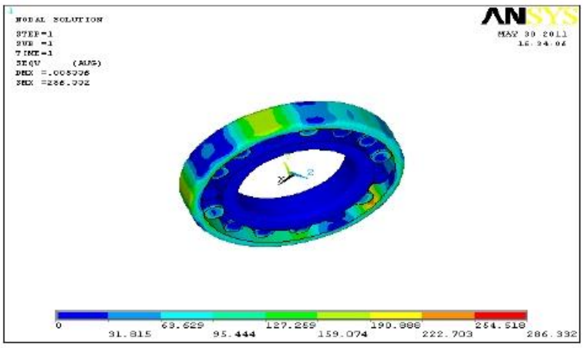

Figure: 14. Fe Modelling Of Contact Pairs

Contact Pair Analysis At Load F=7611.2N

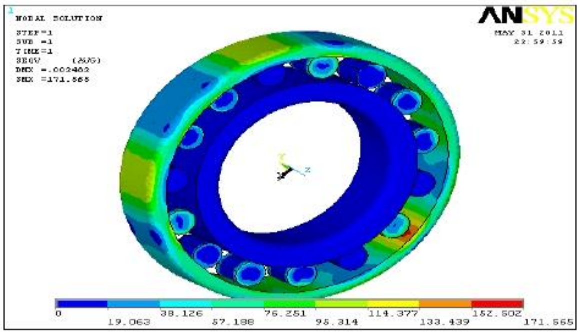

Figure: 15. Fe Modeling Of Composite Materials

Results:

Static Analysis:

After performing the static analysis maximum bending stress is found at the root of the bearing and the results of the static analysis

Table 2: Static Analysis Results for Cylinder

Type of Stress	Maximum Stress(N/mm2)
σ_{xx}	99.755
σ_{yy}	203.178
σ_{zz}	132.434
Von misses stress For Load F=951.4N	224.751

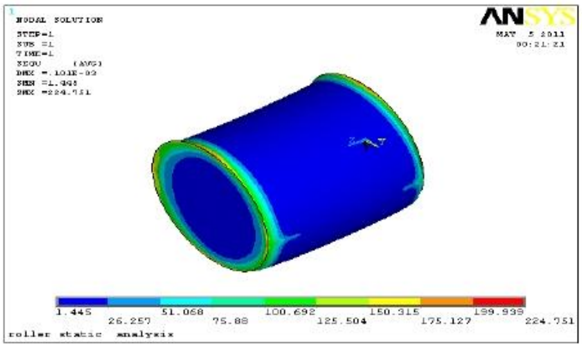

Figure: 16. Static Analysis of Cylinder

Table 3: Static Analysis Results for Outer Ring

Type of Stress	Maximum Stress (N/mm2)
σ_{xx}	105.466
σ_{yy}	249.779
σ_{zz}	101.628
Von misses stress	249.779

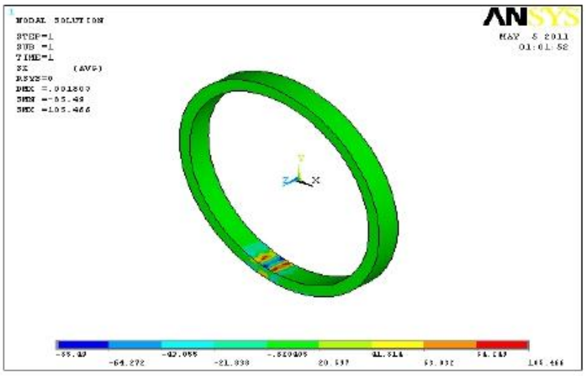

Figure: 17. Static Analysis of Outer Ring

Table: Static Analysis Results for Inner Ring

Type of Stress	Maximum Stress (N/mm2)
σ_{xx}	82.937
σ_{yy}	146.679
σ_{zz}	24.874
Von misses stress	336.476

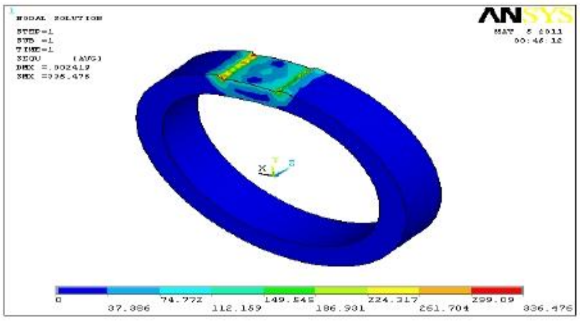

Figure: 18. Static Analysis of Inner Ring

Table: Contact Analysis and Composite AnalysisResults

Load (N)	Numerical Calculation Of Contact Stresses (N/mm2)	Analytical Contact Von Mises Stresses For Steel (N/mm2)	Analytical Composite Von Mises Stresses For Glass/Epoxy (N/mm2)
951.4	35.129	27.045	17.032
1902.8	70.257	61.123	36.673
3805.6	140.516	128.007	81.765
5708.4	212.400	193.539	115.929
7611.2	283.300	286.332	171.565
9514.0	354.000	379.993	247.335

For Load F=7611.2N

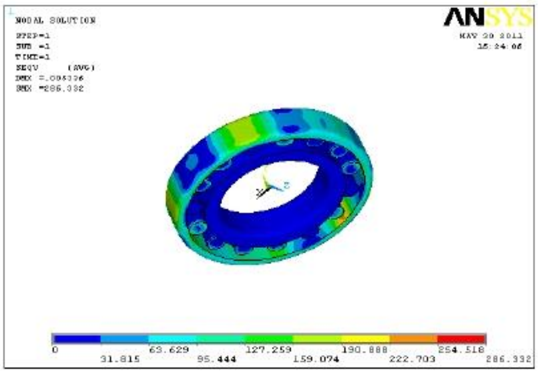

Figure: 19. Von Mises Stress For Contact Pair

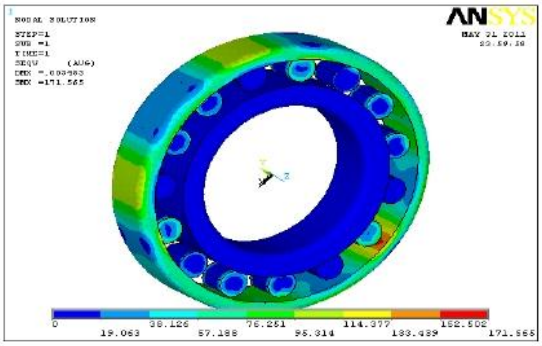

Figure: 20. Von Mises Stress for Composite

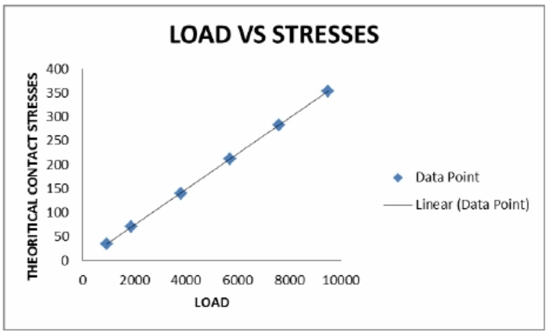

Figure: 21. Load V/S Numerical Contact Stresses

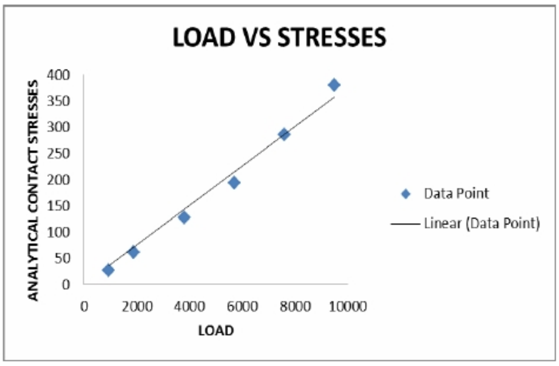

Figure: 22. Load V/S Analytical Contact Stresses

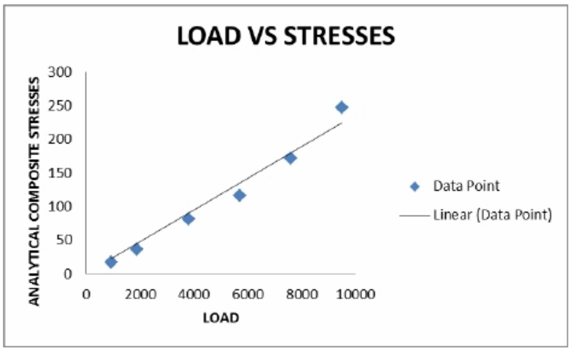

Figure: 23. Load V/S Analytical Composite Stresses

Conclusion:

It was shown that an FEA model could be used to simulate contact between two bodies accurately by verification of contact stresses between two surfaces in contact and comparison with the Hertzian equations.

It is observed from the contact analysis for steel that at F=951.4N, the stress condition is safe, and at F=9514N the stress condition is critical. This is because the yield stress for steel is 250 N/mm^2.

Hence, hybrid composite material (steel and glass/epoxy) is suggested so that the stress levels can be brought within the limits, as the yield stress for glass/epoxy is 485 N/mm^2.

On performing contact analysis on the distribution ofVonmises stress, the theoretical and analytical results when compared aren't varying much. On performing the composite analysis it is observed that

1) Stresses obtained from ANSYS are less when compared to theoretical contact stresses.

2) On reducing the stresses the weight is reduced. As the weight is reduced the power consumption will be less.

3) The performance is within the limits as the yield stress for glass/epoxy is 485N/mm^2.

We suggest the usage of bearings with a lamination of FRP material on the inner side of the outer race of the roller bearing which holds good for a durable operation even in criticalities when operated with whole steel outer race. This usage of FRP also reduces the weight of the bearing which positively affects the power usage of the engine.

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