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PARAMETRIC OPTIMIZATION OF CYLINDRICAL ROLLER BEARING AND COMPARE WITH FEA

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Abstract— Most of the accidents of mechanical systems are majorly caused by bearing failures. Cylindrical roller bearing is one of the critical rotation component in mechanical systems. It can take huge radial loads or two way axial loads and widely used in overloading, steel rolling, metallurgy etc., in this paper, according to the structural properties of cylindrical roller bearing, a mathematical model was established to optimize the design parameters. Being modified the inner ring radius and roller diameters, bearing life was increased through the contact stress reduction. Finite Element Analysis is one of the most powerful approximate method to predict numerical results using ANSYS software. Contact stress for cylindrical rolling components was analytically calculated by Hertizian formula and numerically compared with FEA. From the result comparisons, it was found that, notable amount of contact stress was reduced through the modification of roller diameters and inner ring radius and enhanced the service life of the cylindrical roller bearings.

Keywords— cylindrical roller bearing, mathematical model, contact stress, ANSYS, hertizian, service life.

I. INTRODUCTION

Since being originally introduced, cylindrical roller bearings have been significantly improved, in terms of their performance and working life. This was achieved partially due to the improvement of the properties of the bearing steels, whilst other factors involved design changes, such as the reduction of the stresses of the structures. A major objective has been to decrease the contact stresses at the roller–raceway interfaces, because these are the most heavily stressed areas in a bearing. It has been shown that bearing life is inversely proportional to the stress raised to the ninth power (even higher). For this reason significant efforts have been made to qualify contact stresses in the bearings. The reduction of the contact stresses has been achieved largely by designing the specific surface geometry of the roller– raceway contacts, because the contact-surface geometry has a direct effect on the distribution of contact stresses, and hence, it prescribes the load-carrying capacity of the bearing.

II. LITERATURE SURWAY

Ajinkya Karpe et al [1], have optimized the thicknesses of inner and outer races of a deep groove ball bearing using FEM (ANSYS). 3D model of the roller ball bearing was created in ANSYS workbench. The thicknesses of the inner and outer races of the bearing were to be optimized using parameterization in ANSYS workbench. The basic methodology adopted was the contact stress analysis between the races and the balls. The outer race was kept fixed whereas the inner race was subjected to radial loading and rotation from the shaft. The inner diameter of the bearing and the outer diameter of the bearing were kept fixed according to the application. The bearing ball diameter was also kept fixed. The thicknesses were parameterized by varying the centre to centre distance between the shaft and the bearing balls while maintaining contact between the balls and the grooves of the races. The optimum thicknesses were chosen. Further, material removal from the bearing was also studied.

Prachi Prajapati et al [2], have presented the bearing life optimization of tapered roller bearing. To increase the life of bearings, the dimension of the bearing should be optimum to get the reliability and function of the bearing. In this project the geometry of the bearing component was tried to optimize with the desired life. By optimize the dimension of the Pocket corner radius of bearing component, the total mass of the bearing component. The mass of bearing is reduced by changing the geometry dimension of the bearing component. The optimization has been carried out by keeping the results of the existing bearing constant.

Zhen-huan Ye et al [3], have described the optimization model of fatigue life of cylindrical roller bearings. Aiming to the different geometry parameters and operating parameters of bearings, the effects of assembly interference on bearing fatigue

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life are discussed. The results show that the optimum fatigue life of roller bearings was achieved at negative working clearance with the loads distributed evenly within half ring. The optimum working clearance of roller bearings is not influenced by radial load. But with the increase in pitch diameter or decrease in rolling elements, the optimum assembly interference increases, and accelerating revolution of roller bearings will increase the optimum assembly interference as well.

Prashant M. Jundale et al [4], designed to analyze optimum life and strength of the vehicle differential lock system components (needle roller bearing and differential locking piston) and their effects. It was done by changing dimensions and analyzing the life of needle roller bearing and strength of differential lock piston. Performance parameters of needle roller bearing are analyzed with width and differential lock piston are analyzed with depth. The system is useful to eliminate failures of differential lock system components.

III. PROBLEM STATEMENT

When a bearing roller (cylindrical) is in contact with raceways, excessive pressure peaks occur at the ends of the contact rectangles. This excessive pressure will affect the bearing life.

IV. SELECTION OF BEARING

The bearing was selected according to the required application from P.S.G data book of mechanical design.

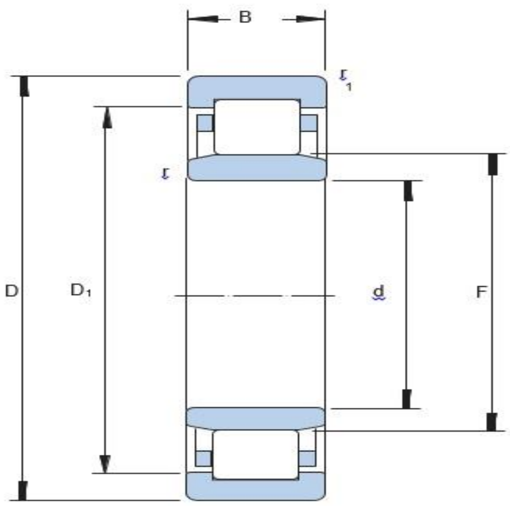
A.BEARING NO: NU2205

Bore diameter	(d) = 25 mm
Outside diameter of bearing	(D) = 52 mm
Thickness	(B) = 18 mm
Fillet radius	(r) = 1.5 mm
Fillet radius	(r_1) = 1 mm
Inner ring outer diameter	(F) = 32 mm
Static capacity	(C_0) = 12200 N
Dynamic capacity	(C) = 16300 N
Max.permissible speed	(n) = 13000 rpm

B.CONTACT STRESS FOR CYLINDRICAL ROLLER BEARING

The contact stress between roller and inner ring of the cylindrical roller bearing (hertizian formula).

$$\sigma = \sqrt{\frac{FE}{2\pi(1-\mu^2)L_{er}}\left[\frac{2}{D_r}+\frac{1}{r_i}\right]}$$



Where

E – Young's modulus of the material (N/mm^2)
F - Radial load per roller (N)
μ - Poisson's ratio
$L_{er} = D_r$ - Effective length of the roller (mm)
D_r - Diameter of the roller (mm)
r_i - Inner ring outer radius of bearing (mm)

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V. PARAMETERIZATION OF ROLLER DIAMETER AND INNER RING RADIUS IN MS-EXCEL

The following table compares the various values obtained for roller diameter and inner ring radius of roller bearing. The material used for the analysis is stainless steel.

Table 4.1 – Parametrization of roller diameter of roller bearing in MS-EXCEL

Load, F (N)	***Young's modulus,E (N/m^2)***	***Effective length,L_{er} (mm)***	***Inner ring radius,r_i (mm)***	***Roller diameter,D_r (mm)***	***Contact stress, σ (MPa)***
400	200000	6.5	16	6.5	892.23
400	200000	9.5	16	9.5	633.81
400	200000	12.5	16	12.5	498.80
400	200000	15.5	16	15.5	415.60
400	200000	18.5	16	18.5	359.03
400	200000	21.5	16	21.5	317.97

Table 4.2 - Parametrization of inner ring radius of roller bearing in MS-EXCEL

Load, F (N)	***Young's modulus,E (N/m^2)***	***Effective length,L_{er} (mm)***	***Roller diameter,D_r (mm)***	***Inner ring radius,r_i (mm)***	***Contact stress,σ (MPa)***
400	200000	6.5	6.5	12	916.99
400	200000	6.5	6.5	16	892.23
400	200000	6.5	6.5	20	877.03
400	200000	6.5	6.5	24	866.76
400	200000	6.5	6.5	28	859.34
400	200000	6.5	6.5	32	853.74

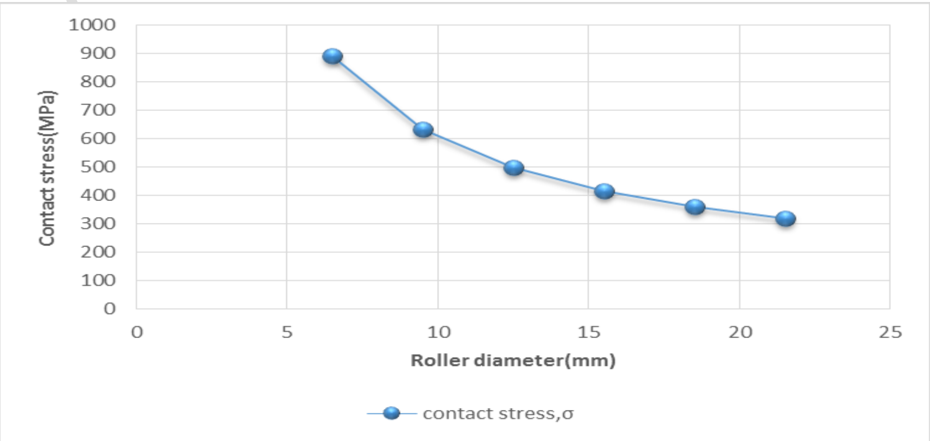

Fig 4.1- Effect of roller diameter on contact stress

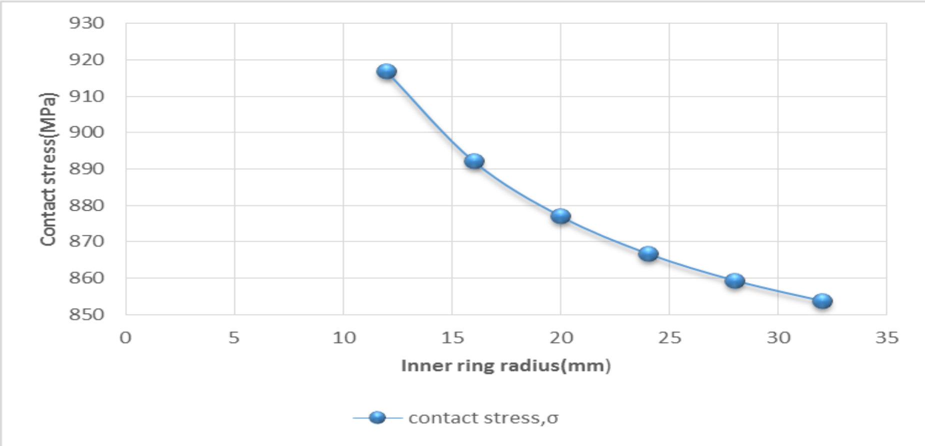

Fig 4.2- Effect of inner ring radius on contact stress

From the above tables 4.1 & 4.2, The contact stress was calculated analytically by varying roller diameter of cylindrical roller bearing from 6.5 mm to 21.5 mm and varying inner ring radius of roller bearing from 12mm to 32 mm with the help of hetizian formula through MS-Excel. The remaining parameters of hertizian contact stress formula are constant.

The above graphs, Shows that the contact stress was considerably reduced by modifying roller diameter and inner ring radius with in the pecified limit of the cylindrical roller bearing.

VI. PARAMETERIZATION OF ROLLER DIAMETER AND INNER RING RADIUS IN ANSYS

A. ANSYS PARAMETERS

- Contact type: Bonded (rollers and ring races)
  Since there is no separation allowed and no sliding allowed between the contact surfaces.
- Contact Body: Inner ring raceway
- Target Body: Bearing rollers
- Bearing Load: 400 N (radial)
- Fixed Support: Outer ring raceway

B. About FE Analysis and ANSYS

- About FEA - The physical problem is idealized as a mathematical model using certain assumptions, which together leads to differential equations governing the Mathematical model. The finite element analysis solves this mathematical model. In brief the basis of finite element analysis is the representation of the body or structure by an assembly of subdivisions called finite elements. These elements are considered as interconnected at the joints, which are called nodes or nodal points. A typical finite element model is comprised of nodes, degrees of freedom, elements material properties, externally applied loads and analysis type. The finite element method is a numerical analysis technique for obtaining approximate solutions to a wide range of engineering problems.

C. Steps in Finite Element Analysis

- *Step 1:*
  First the domain is represented as finite elements. This is called discretization of domain. Mesh generation programs called processors, help in dividing the structure.
- *Step 2:*
  Formulate the properties of each element in stress analysis. It means determining the nodal loads associated with all element deformation stress that is allowed.
- *Step 3:*
  Assemble elements to obtain the finite element model of the structure.
- *Step 4:*

Apply the known loads, nodal forces in stress analysis. In stress analysis the support of the structure has to be specified.

- *Step 5:*
  Solve simultaneous line algebraic equations to determine nodal displacements in the stress analysis.
- *Step 6:*
  Post processors help the user to sort the output and display in the graphical output.

D. CONTACT STRESS ANALYSIS BY MODYFING ROLLER DIAMETER

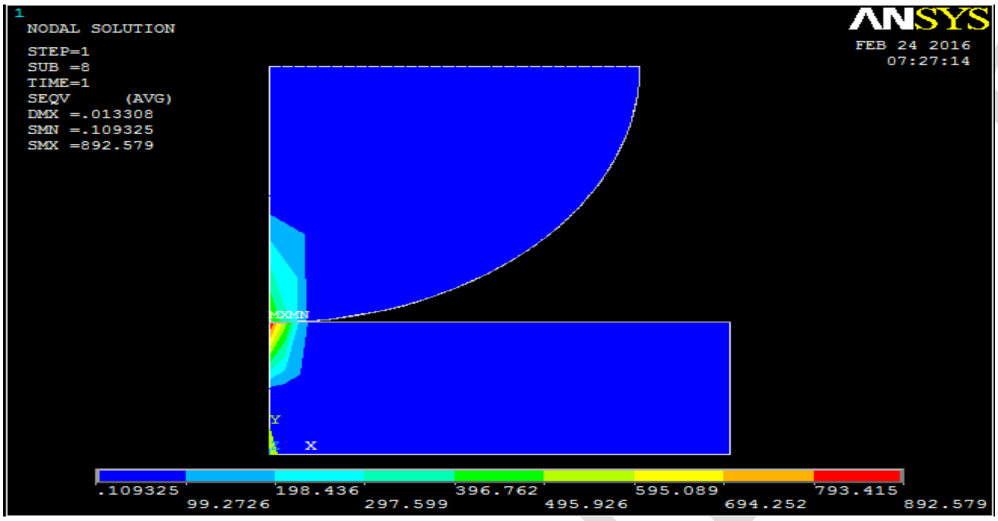

Fig 5.1 Contact stress at roller diameter 6.5 mm

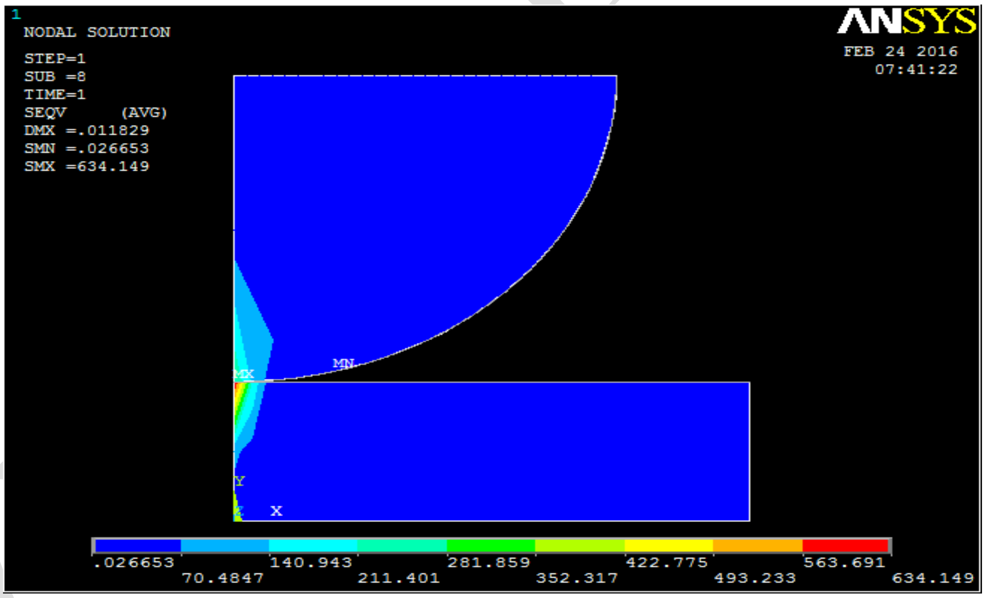

Fig 5.2 – Contact stress at roller diameter 9.5 mm

E. CONTACT STRESS ANALYSIS BY MODIFYING INNER RING RADIUS

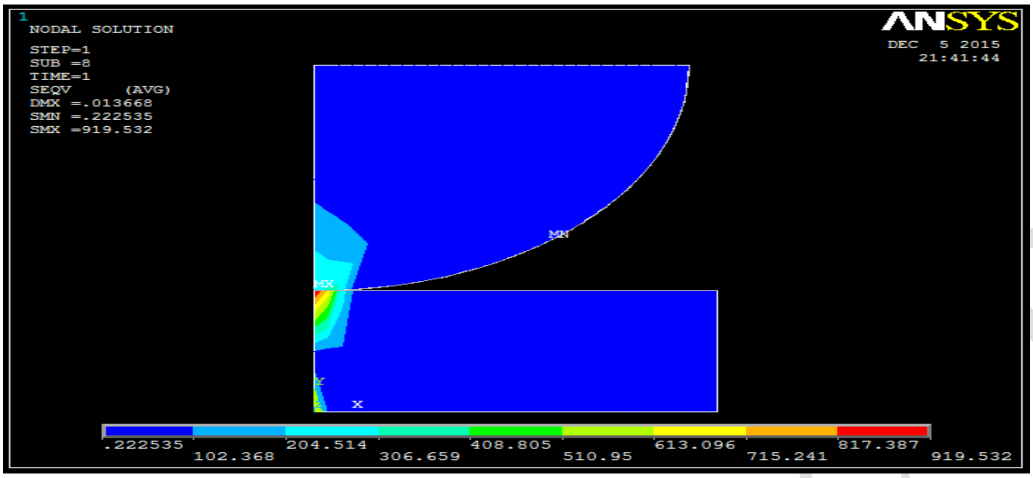

Fig 5.3 – Contact stress at inner ring radius 12 mm

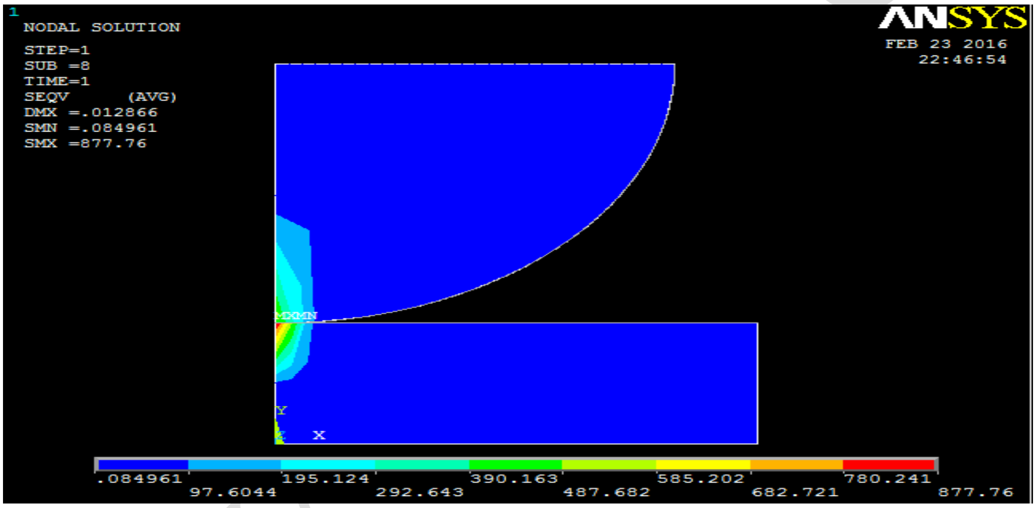

Fig 5.4 – Contact stress at inner ring radius 20 mm

F. About Analysis

The software ANSYS was developed by ANSYS Inc. USA. The ANSYS product family offers the following capabilities linear stress, structural non-linear, dynamic analysis, buckling analysis, buckling, sub structuring, heat transfer, transient, thermal, thermal non-linear, electrostatics, acoustics, electromagnetic and coupled field.

Assumption in

FEA the four primary assumptions, which must be considered in any Finite element based solution, are as below.

(1) Geometry

Geometry must be in its proper context. An FEA solver only understands nodes, and the connectivity of nodes, which are elements. The smaller the element size or the higher the element order, the better the mesh will represent the geometry template it was based on. The inherent assumption when a model is sent is to solve is that the mesh represents the geometry adequately for the study goals.

(2) Material Properties

Specifying a single set of material properties for a part in an FEA study makes the significant assumption that all parts in the production run intended to be represented by the analysis have the same properties. It is also typical to assume that most parts will be isotropic and homogenous.

(3) Mesh

The mesh is our way of communicating the geometry to the FEA solver. The accuracy of the solution depends entirely on the quantity of the mesh. The quality of the mesh is best characterized by the convergence of the problem. The global displacements should converge to a stable value and other results of the interest should converge locally. A less tangible, more subjective measure of the quality of mesh is its appearance and ability to visually convey the geometry it represents.

(4) Boundary Condition

The boundary conditions are those conditions that are placed on the model to represent everything about the system that is not been modeled.

VII. RESULT AND DISCUSSION

After doing an analytical and numerical study of cylindrical roller bearing with the variation of roller diameter and inner ring radius respectively an obtained result are mentioned in following table,

Roller diameter (mm)	Contact stress (N/mm^2) (modified roller diameter)		Inner ring radius (mm)	Contact stress (N/mm^2) (modified inner ring radius)	
	Analytical (MS - Excel)	FEA (ANSYS)		Analytical (MS - Excel)	FEA (ANSYS)
6.5	892.23	892.58	12	916.99	919.53
9.5	633.81	634.15	16	892.23	893.63
12.5	498.80	500.44	20	877.03	877.76
15.5	415.60	417.24	24	866.76	866.70

From the above table, the roller diameter and inner ring radius were used to calculate the contact stresses between bearing roller and inner ring raceway. Contact stresses were analytically calculated by using Hertizian formula and compared with numerical results which was found by ANSYS. The work simulation results in ANSYS and analytical results were nearer. From the results, it was found that while increasing roller diameter and inner ring radius of roller bearing, the contact stresses was reduced.

VIII.CONCLUSION

The goal of present work was to minimize contact stress between roller and raceways by modifying the parameters of the roller bearing. Contact stress is the major cause for bearing failures and it affects the life of bearing. Due to the inverse relation, contact stress reduces the fatigue life of the roller bearing. The roller diameter and inner ring radius of the cylindrical roller bearing were optimized using Hertizian formula through MS-Excel. Analytically calculated contact stress were compared with Finite element analysis (ANSYS) results. The study revealed that, the contact stress was reduced by 64% when we use the optimized values of roller diameter and radius of inner ring and increased service life of the roller bearing.

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