FINITE ELEMENTS METHOD MODELLING OF ROLLING BEARINGS

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Abstract: *This study presents to determine the contact stress in rolling bearings by using analytical and numerical method. Analytical solution is obtained by using Hertizan contact theory. Obtained analytical solution by this theory require comparison with the numerical calculations to obtain more accurate results for contact problems. Because of that the same problems are also examined by using finite element method. The geometry of the model being studied gives different type of contact configurations such as a point or line of contact. In cylindrical roller bearing the contact form is line contact and for the ball bearing the contact characteristic is point contact. High stress occurs on both of these two contact areas. Contact stress causes elastic or plastic deformation and the contact area will change depending on the magnitude of the contact stress. Therefore, it is really important to calculate more accurate stress at the contact area.*

KEYWORDS*: HERTZ CONTACT, NON-LINEAR CONTACT, ROLLER BEARING, FINITE ELEMENT METHOD, ANSYS.*

1. Introduction

Bearings, having a crucial role in the machine design, are the components to provide smooth rotation by reducing friction between the rotating machine parts.

The general structure of bearing are shown in Fig 1, consists of inner and outer rings and the roller elements which vary depending on the application. The cage prevents the roller elements rubbing against each other.

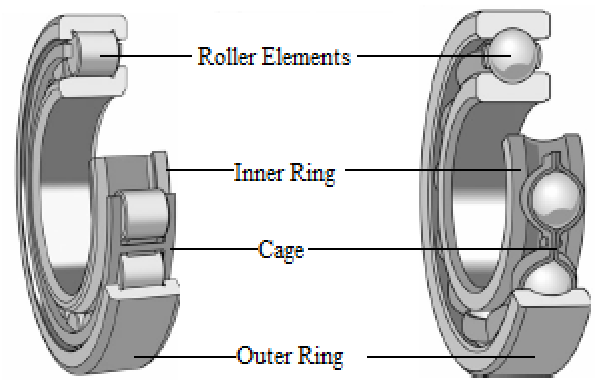


Fig. 1 *Roller bearing structure. a)Cylindrical roller bearing b)Ball bearing*

There are two main functions of roller bearings: First is to reduce frictional forces between moving parts by giving a surface to roll on instead of sliding. The secondary function of bearings is to transmit the force. The roller bearings are classified according to the direction in which the load is applied. Generally bearings carry two kinds of force: radial and axial and some bearings can also carry both radial and axial load at the same time.

In this study a ball bearing and a cylindrical roller bearing are examined.

2. Theoretical Aspects

The force and motion are transferred between machine parts by means of contact phenomena. A contact problem occurs when at least two bodies not mechanically joined touch each other without becoming rigidly attached.

When there are non-conforming surfaces come into mate the contact form can be point or line. The localized stress value on this area is very high. Since the stress is the force over the area, any load on a point contact would be infinite stress as mathematical view. The original analysis of elastic contact stresses was published in1881 by H. Hertz. In his honour, the stresses at the mating surfaces of curvature bodies in compression are called *Hertz contact stresses*.

Hertzian contact stress is one of the major reasons of surface failures especially for bearings, cams, gear teeth, locomotive wheels, valve tappets and pin joints linkages [1].

Hertz calculations are limited to case of contact of elastic bodies with simple quadratic shapes and when he was developing his theory Hertz made some important assumptions which are written below [1]:

i. Contacting bodies are isotropic, homogeneous and elastic.
ii. The contact areas are essentially flat and small relative to the radii of curvature of the undeflected bodies in the vincinity of the interface.
iii. The strains are small and within the elastic limits.
iv. The contacting bodies are perfectly smooth; therefore, friction forces between mated parts need not to be taken into account.

Boussinesq developed Hertz's theory by using theory of elasticity. Boussinesq improved the equations for the state of stress within an isotropic, homogeneous and linearly elastic space, under a concentrated load, which will act perpendicular to the surface. He studied the deformation of semi-infinite solid due to pressure exerted on a small area of its plane surface [2].

Ludenberg in 1939 developed a general theory of elastic contact between two semi-in finite bodies, in which the effect on the stress of the presence of tangential load is taken into consideration.

Midlin investigated the distribution of the tangential forced across the area of contact when one elastic body slides over another.

Lure presented a general three-dimensional punch contact problems

The stress examined in two contacting bodies at a rolling interface is highly dependent on the geometry of the mate surfaces as well as on the loading material properties.

In this study ball and cylindrical roller bearings are examined. In a ball roller bearing, a ball passes over the raceway surface, the theoretical contact form is assumed as a point in this application. A cylindrical roller bearing, cylinders passes over the raceway, the theoretical contact shape is assumed as line contact for this application.

In order to examine contact stresses for cylindrical and ball bearing, traditional Hertzian contact theory is used. According to the theory sphere to sphere contact presents ball bearing and cylinder to cylinder contact presents cylindrical roller bearing

Hertz contact theory equations for point and line contact types, used in analytical solutions, are shown below:

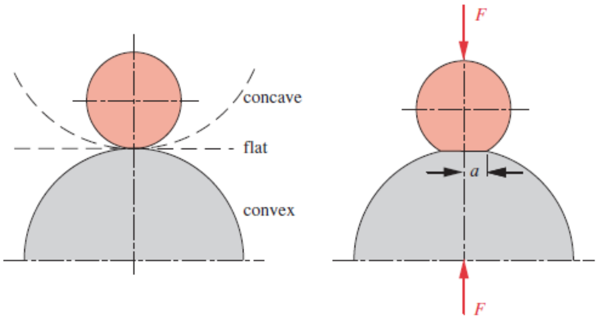


Fig. 2 *Two spheres or cylinders contact zone [3]*

2.1. Point contact:

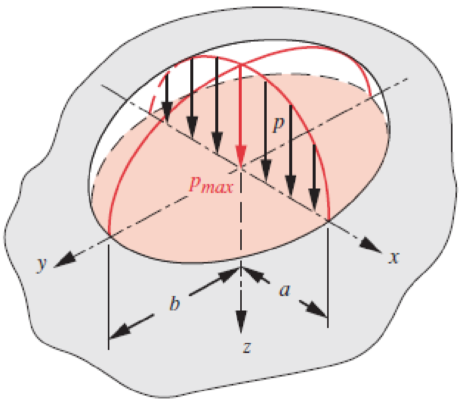


Fig.3 *Spherical contact, pressure distribution.(a=b) [3].*

Two spheres in contact are shown in Fig2. The theoretical contact form is assumed as hemisphere with circular contact shape is shown in Fig3.

Total applied load F on the contact patch is written as follows [4]:

$$F = \frac{2}{3} p a^2 p_{\max} \tag{1}$$

Where a is radius of contact patch. Maximum pressure is written as follows:

$$P_{\max} = \frac{3}{2} \frac{F}{pa^2} \tag{2}$$

Average pressure is written as follows:

$$P_{avg} = \frac{F}{pa^2} \tag{3}$$

Maximum pressure is written depending on average pressure

$$P_{\max} = \frac{3}{2} P_{avg} \tag{4}$$

Material constant can be written as follows:

E_1, E_2 are elasticity modulus and v_1, v_2 are Poisson's ratios of mated materials.

$$m_1 = \frac{1-v_1^2}{E_1} \ , \ m_2 = \frac{1-v_2^2}{E_2} \tag{5}$$

Geometry constant can be written as follows:

$$B = \frac{1}{2}\left(\frac{1}{R_1} + \frac{1}{R_2}\right) \tag{6}$$

The contact-patch radius a is written as follows:

$$a = \frac{p}{4} p_{\max} \frac{m_1 + m_2}{B} \tag{7}$$

$$a = \sqrt[3]{0.375 \frac{m_1 + m_2}{B} F} \tag{8}$$

The pressure distribution within the hemisphere is written as follows:

$$P = P_{\max} = \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{a^2}} \tag{9}$$

Static stress distributions in spherical contact:

$$\sigma_z = -p_{\max} \tag{10}$$

$$\sigma_{x\max} = \sigma_{y\max} = -\left(\frac{1+2v}{2}\right) p_{\max} \tag{11}$$

$$\tau_{xy} = \left(\frac{1-2\upsilon}{3}\right) p_{\max} \tag{12}$$

Von Misses equivalent stress is written as follows:

$$\sigma_{vM} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)} \tag{13}$$

2.2. Line contact:

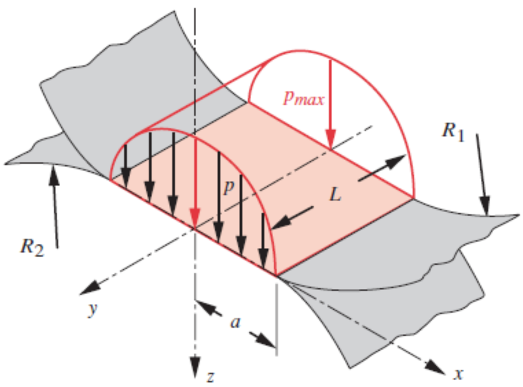


Fig. 4 *Cylindrical contact, pressure distribution.[3].*

When two cylinders roll together, their contact patch will be rectangular as shown in Fig4.

Total applied load F on the contact patch is written as follows [5]:

$$F = \frac{1}{2} p a L p_{\max} \tag{14}$$

where L is the length of contact along the cylinder axis.

$$P_{\max} = \frac{2F}{paL} \tag{15}$$

$$P_{avg} = \frac{F}{2aL} \tag{16}$$

$$P_{\max} @ 1.273 P_{avg} \tag{17}$$

Geometry constant can be written as follows:

$$B = \frac{1}{2}\left(\frac{1}{R_1} + \frac{1}{R_2}\right) \tag{18}$$

Radius of contact patch a is written as follows:

$$a = \sqrt{\frac{2}{p}\frac{m_1 + m_2}{B}\frac{F}{L}} \tag{19}$$

The pressure distribution within the semi-elliptical prism:

$$P = P_{\max} = \sqrt{1 - \frac{x^2}{a^2}} \tag{20}$$

Static stress distributions in cylindrical contact:

$$\sigma_{xn} = -p_{\max}\sqrt{1 - \frac{x^2}{a^2}} \tag{21}$$

$$\sigma_{zn} = -p_{\max}\sqrt{1 - \frac{x^2}{a^2}} \tag{22}$$

$$\tau_{xzn} = 0 \tag{23}$$

Von Misses equivalent stress is written as follows:

$$\sigma_{vM} = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_x \sigma_y + 3\tau_{xz}^2} \tag{23}$$

3. Finite Element Model

All of these theories discuss the contact problem with in the analytical methods and use highly idealised model. Unfortunately these approaches have limited application in engineering.

The more accurate results are examined for rolling bearing such as ball and cylindrical roller bearings contact problem with numerical method .Therefore same problem is also modelled in ANSYS-18.1 workbench environment.

4. Numerical example

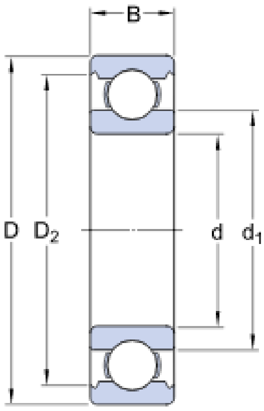
4.1. 6208 Ball Bearing Analytical Solution

Firstly 6208 ball bearing shown in Fig5 is analyzed according to Hertizan contact theory. Applied load is F=5000N.

According to Hertzian theory, the analytical results for 6208 ball bearings are shown in Table 1.

Table.1 *6208 Ball bearing analytical solution results*

$\tau_{\max}$ [MPa]	$P_{\max}$ [MPa]	σ_{vM} [MPa]
2570	5474	2744



d:40	[mm]
D:80	[mm]
B:18	[mm]
d_1:52,6	[mm]
D_2:69,8	[mm]

Fig.5 *6208 Ball bearing dimensions.*

4.2. 6208 Ball Bearing FEA Analysis

Modelling and the analysis of the problem are completed in ANSYS 19-1 Workbench software.

In terms of accuracy and simplicity the problem is modeled for one ball by using cyclic symmetry.

In order to obtain more accurate results, the mesh density over the contact area is increased as shown in Fig6.

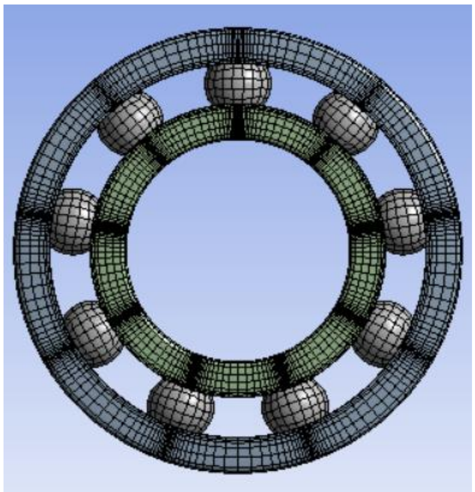


Fig.6 *6208 Ball bearing Solid-FEM model.*

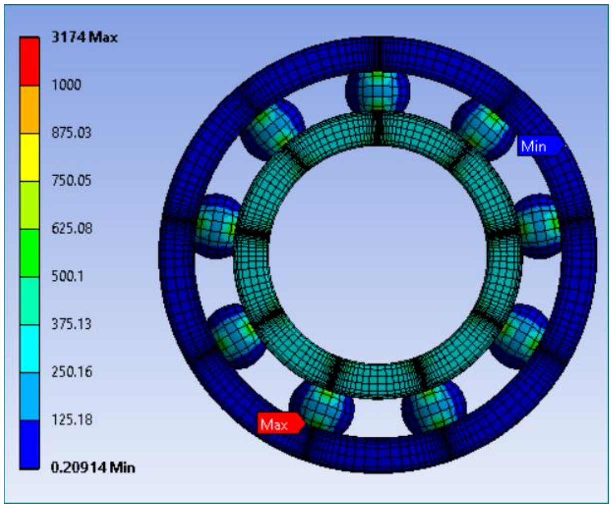
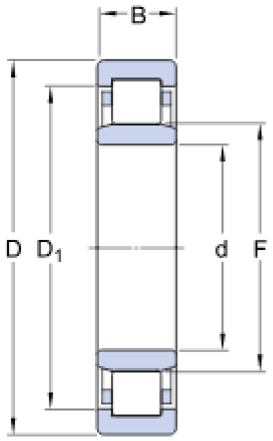


Fig.7 *6208 Ball bearing FEA solution.*

The obtained maximum Von Misses stress is 3174 [MPa].

4.3. NU210 Cylindrical roller bearing analytical Solution

Firstly NU210 cylindrical roller bearing as shown in Fig8 is analyzed according to Hertizan contact theory. Applied load is F=5000N.



d:50	[mm]
D:90	[mm]
B:20	[mm]
F:59,5	[mm]
D_1:77,4	[mm]

Fig.8 *NU210 Cylindrical roller bearing dimensions*

According to Hertzian theory, the analytical results for NU210 cylindrical roller bearings are shown in Table.2

Table.2 NU210 cylindrical bearing analytical solution results.

τ_{max} [MPa]	P_{max} [MPa]	σ_{vM} [MPa]
480	1464	3000

4.4. NU210 Cylindrical roller bearing FEA Solution

In order to obtain more accurate results, the mesh density over the contact area is increased shown in Fig9.

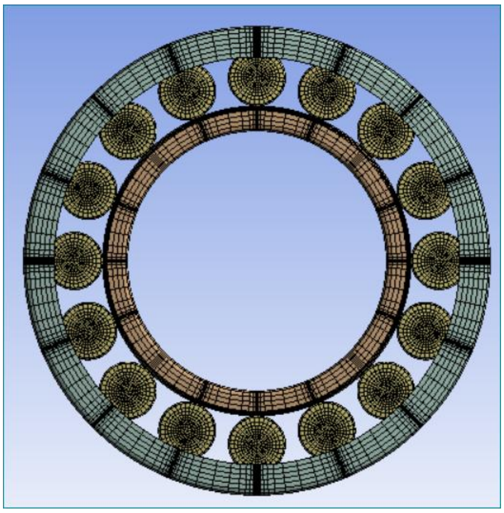


Fig.9 *NU210 Cylindrical roller bearing Solid-FEM model.*

The obtained maximum Von Misses stress is 3781 [MPa].

5. Results and discussion

Comparison of obtained Von Mises stress results for ball and cylindrical roller bearings are presented in Table 3.

Table 3 Comparison of Von Mises stress

Bearing type	Analytical σ_{vM}	FEM σ_{vM}	% Deviation
Ball	2744 [Mpa]	3174 [Mpa]	13,5
Roller	3000 [Mpa]	3781 [Mpa]	20,6

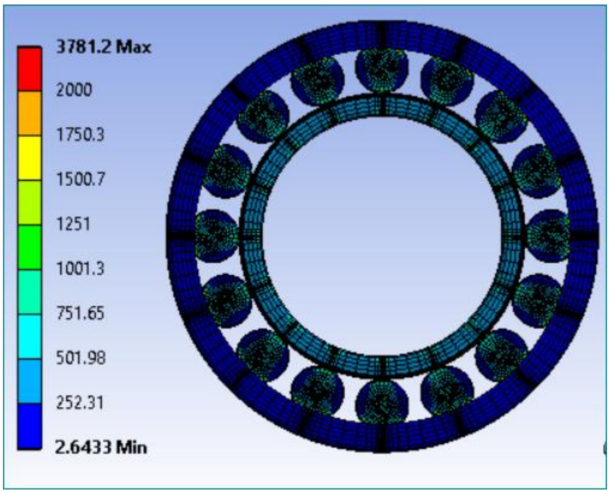


Fig.10 *NU210 cyindrical roller bearing FEA solution.*

All obtained results are acceptable for required Von Mises stress calculation for ball and cylindrical roller bearings.

6. Conclusion

Numerical analysis of rolling bearing contact problem is simulated by using ANSYS workbench environment. Thus the contact stresses are analysed for point and line contact types with 3D model.

For ball bearing applications the contact area is actually an elliptical geometry and the contact form depends on the applied force which means, both of these problems are non-linear and analytical theory has lots of assumptions.

Bearings load capacity and reliability are so important. Therefore the rolling bearings play such a prominent machine elements. Calculation the life of a bearing with considerable accuracy become even more important. Thus making it possible to match the bearing life with the service life of the machine involved.

In this study FEM is used to examine the contact analysis of rolling bearing. Specifically, the normal and tangential forces as well as the rolling friction between mated parts are analysed in order to more accurate results. Numerical analysis of contact problems by computer programs gives more accurate results.

Acknowledgements

This work was supported by Research Fund of the Yildiz Technical Üniversity (BAP) Project Number: FYL-2018-3491.

7. Literature

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