MODIFICATION OF SPHERICAL ROLLER BEARING IN BLENDER RECLAIMER

A Project report submitted in partial fulfillment of the requirements for the award of the degree

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

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ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY AND SCIENCES

(Affiliated to Andhra University, Accredited by NBA & NAAC)

VISAKHAPATNAM

2017-2018

ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY AND SCIENCES

(Affiliated to Andhra University, Accredited by NBA & NAAC with 'A' Grade)

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CERTIFICATE

This is to certify that the Project Report entitled **"MODIFICATION OF SPHERICAL ROLLER BEARING IN BLENDER RECLAIMER"** is a bonafide work carried out by B.ANIL KUMAR (314126520012), Ch. SAGAR PAVAN (314126520032), G.MEGHA SHYAM (314126520059), G.SUNIL KUMAR (314126520066) during the year 2017-2018 under the guidance of **Ms. K. V. RUKMINI M. E, Assistant Professor** in the partial fulfillment of the requirement for the award of Degree of Bachelor of Mechanical Engineering by Andhra University, Visakhapatnam.

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ACKNOWLEDGEMENT

We would like to express our sincere gratitude and thanks to our guide **Ms.K.V RUKMINI** Asst. Professor Dept. of Mechanical Engineering, ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY AND SCIENCES,for her timely guidance and time she has devoted towards us in doing this project.

We would like to express our thanks to **Mr.V.VITAL RAO, Assistant General Manager** Engineering Shops and Foundry Dept., **Visakhapatnam Steel Plant** for his guidance in completing this project. We would like to thank **Mr. K. MALLIKARJUNA RAO, Assistant General Manager** Raw Material Handling Plant Dept., **Visakhapatnam Steel Plant** for his expert tutelage that required for us in completing this project.

Our special thanks to, **Prof.T.V.HANUMANTHA RAO,** Principal of ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY AND SCIENCES and **Dr.B.NAGARAJU,** Head of the Department of Mechanical Engineering for their whole hearted co-operation.

Finally we thank one and all who directly as well as indirectly helped us in the completion of this project.

NOMENCLATURE

SYMBOL	DESCRIPTION	UNITS
m	Mass of ball or roller	mm
D	Ball or roller diameter	mm
d_m	Pitch diameter	mm
C_0	Basic static load	KN
F	Force	N
F_a	Axial force	N
F_r	Radial force	N
J	Mass moment of inertia	kg . mm^2
M	Moment	N . mm
r	Ball Radius	mm
U	Coordinate direction distance	mm
V	Coordinate direction distance	mm
W	Coordinate direction distance	mm
x	Coordinate direction distance	mm
x^1	Coordinate direction distance	mm
y	Coordinate direction distance	mm
y^1	Coordinate direction distance	mm
z	Coordinate direction distance	mm
z^1	Coordinate direction distance	mm
Ψ	The angle between the z axis and z^1 axis	rad
β^1	Angle between projection of the U axis	rad

	on the x^1y^1 plane and the x^1 axis.	
β	Angle between W axis and z^1 axis.	rad
φ	Angle between z axis and radius.	Rad
ρ	Mass density	kg / mm^3
ω_m	Orbital angular velocity of ball	rad / sec
ω_R	Angular velocity of ball about its own axis	rad / sec
P	Applied radial direction load	N
F_x	Force acting in x - direction	N
F_y	Force acting in y - direction	N
F_z	Force acting in z – direction	N
V	Race rotaion factor	-
M_x	Moment in x - direction	N . mm
M_y	Moment in y - direction	N . mm
M_z	Moment in z – direction	N .

ABSTRACT

Blender reclaimer is a machine used to reclaim raw material like iron ore fines, sized iron ore and fluxes. In this blender reclaimer spherical roller bearings are used for carrying relatively heavy radial loads and some axial loads in either direction. They are also extremely resistant to shock loads. These spherical roller bearings are failing mainly due to overloading.

Earlier, 23024CC bearing was used in blender reclaimer. During loading conditions the observed static and dynamic load values are exceeding the permissible load values. Hence to avoid this failure the bearing is replaced by 22224CC bearing. It was found out the applied static and dynamic load values are within the maximum allowable load values of 22224CC bearing.

Using ANSYS Workbench software, static structural analysis is carried out on the considered three bearings. By using the output parameters viz., Equivalent von-Mises stress, Equivalent elastic strain & Strain energy of the three bearings and by comparing their respective parameters with each other, the most suitable bearing is determined.

CHAPTER-1

INTRODUCTION

Visakhapatnam Steel Plant (VSP) annually requires quality raw materials viz. Iron Ore, fluxes (Lime stone, Dolomite); coking and non coking coals etc., to the tune of 12-13 million tonnes for producing 3 million tones of Liquid Steel. To handle such a large volume of incoming raw materials received from different sources and to ensure timely supply of consistent quality of feed materials to different VSP consumers, Raw Materials Handling Plant serves a vital function. This unit is provided with elaborate unloading, blending, stacking and reclaiming facilities viz. Wagon Tipplers, Ground and Trace Hoppers, Stock yards crushing plant, vibrating screens, single/twin boom stickers, wheel on boom and blender reclaimers.

The Blender Reclaimer is a machine used to Reclaim Raw material like Iron Ore fines, Sized Iron Ore and Fluxes. This machine is hydraulically and electrically operated. There are Long Travel, Cross Travel, Two Bucket wheels with 20 no of buckets, which reclaim the above material and send to Sinter Plant, Blast Furnace, Coke Ovens and Steel Melting Shop. The machine can be shifted to one bed to another bed when ever required. The Buckets reclaim Raw Material which passes through Boom Conveyor to Main Conveyor and through no of conveyor paths and finally reaches customer departments. There are 5 reclaimers in raw material handling plant. Out of 5 reclaimers 3 are blender reclaimers called as MUKUND, 1 is stacker reclaimers called as SANDVIK and 1 is scraper reclaimer called as FLSmidth reclaimer. Each blender reclaimer has 1200TPH capacity. These reclaimers are used where large volumes of material must be readily available and/or blending of stockpile materials is critical. A high degree blending is achieved by moving laterally across the stockpile face as the whole machine moves forward.

Fig 1.1: Blender Reclaimer

FEATURES OF RECLAIMER:-

- Construction is robust.
- Working parts are minimum.
- Automatic operation is possible.

ADVANTAGES OF RECLAIMER:-

- Maintenance is low due to less working parts.
- Degree of blending is high.
- Maximum possible homogeneity of material properties.

CHAPTER 2

LITERATURE REVIEW

Among the other mechanical components, researchers pay great attention to the rolling element bearings due to their unquestionable industrial importance. In addition, more faults arising in rotating machines are often linked to bearing faults. The result of many studies show that bearing problems account for over 40% of all machine failures (Schoen et al. 1995). Rolling element bearings generally consist of two rings, an inner and an outer, between which a set of balls or rollers rotate in raceways.

Riddle (1955) stated that improperly installed bearings are often caused by forcing the bearing onto the shaft or in the housing. This produces physical damage in the form of brinelling or false brinelling of the raceways which leads to an early failure. Brinelling is the formation of indentations in the raceways as a result of deformation caused by static overloading.

Riddle (1955) observed the fatigue phenomena, known as flaking or spalling and suggested that continued stress causes fragments of the material to break or loose, and produce a localized fatigue. Once fatigue gets started, the affected area expands rapidly contaminating the lubricant and causing localized overloading over the entire circumference of the raceway, noticed by Eschmann et al (1958). Eventually, the failure results in rough running of the bearing. This is the initiation of failure in rolling element bearings which reduce the life of the bearing.

The presence of water, acids, deteriorated lubrication and even perspiration from careless handling during installations produce bearing corrosion. Eschmann et al (1958) and Riddle (1955) observed the effects of improper lubrication that includes both under and over lubrication. In either case, the rolling elements are not allowed to rotate on the designed oil film causing increased levels of heating. Consequently, the excessive heating causes the grease to break down, which not only reduces its ability to lubricate the bearing elements but also accelerates the failure process.

Rolling element bearings generally consist of two rings, an inner and an outer, between which a set of balls or rollers rotate in raceways. Eschmann et al (1958) stated that when bearings operate under normal conditions of well balanced load and good alignment, fatigue failure begins with small fissures. These fissures are located between the surface of the raceway and the rolling elements, which then gradually propagate to the surface, generating detectable vibrations and increasing noise levels.

Boness (1969a) investigated the factors affecting cage and roller slip in high speed roller bearings. Cage and roller speeds for different radial clearances were measured over a range of radial loads and shaft speeds using several lubricants. These results were analyzed and compared with existing theoretical evidence. To improve the correlation between theory and practice, further theoretical results of cage and roller motion were obtained for bearing conditions relevant to the regime where the fluid prosperities are pressure dependent. The theoretical extent of cage and roller slip shows only a fair agreement with the experimental findings. However, the load below which cage slip will occur was much more accurately forecasted. Boness concluded that increasing the load, or decreasing the shaft speed, results in a reduction in cage and roller slip. Smaller radial clearances also reduce the slip. No cage or roller slip was detected at heavy loads well into the full elastohydrodynamic regime. Evaluation of the operating conditions, in terms of the nondimensional load and speed parameters, indicated that the majority of the experimental results were relevant to the intermediate regime.

Poplawski (1972) investigated a roller bearing model, which includes the effects of full film lubrication at the race contacts, for use in estimating cage slip, roller slip, film thickness, and cage forces for a given geometry and operating condition. The model includes churning loss, cage pilot surface friction, roller pocket friction, cage unbalance as well as drag due to the unloaded rolling elements. The description of the lubricant film thickness, traction, and pressure forces are based upon assumptions introduced by Dowson, which reduce the complex numerical procedure required for a rigorous solution to the isothermal elastohydrodynamic problem to a set of nonlinear equations. Poplawski concluded that cage slippage has become a significant problem in the operation of high speed lightly loaded bearings. Bearing distress caused by slippage appears as a thermal and wear problem that leads to eventual bearing seizure. As the radial load is decreased for a given shaft speed, there exists a region where the cage speed begins to decrease and causes the bearing to go into skid. He pointed out that the present opinion is that 10% slip is tolerable, however, there have been no controlled experiments reported where bearings were run at various slippages and inspected for wear and failure. In addition, the region where slip is initiated or occurs at a lesser load as shaft speed decreased in practice.\

According to Riddle (1995) external sources include contamination and corrosion. Improper lubrication also affects the life of the bearing. The dirt and foreign matter that is commonly present in most of the industrial environment and the abrasive nature of these miniature particles, whose hardness can vary from soft to hard like diamond, cause pitting and sanding action responsible for measurable wear of the balls and raceways.

Aditya et al (2014) tested cylindrical roller bearing in order to investigate its tribological properties. Initially, wear rates are very small and constant over a period of time. This wear is further responsible for fatigue failure of the bearing. Due to increase in wear, lubricant film is not capable to form a hydrodynamic film between roller and inner race. scanning electron microscope of the inner race and rollers have revealed the presence of pits, cracks at the contact surface and wear debris due to surface contact fatigue. An increase in bearing temperature reduces the lubricant film thickness. The lubricant film between the roller and inner race could not be effectively formed due to increase in bearing temperature, which results in direct contact between two metal surfaces and thus wear takes place [20].

Jafar Takabi et al (2015) stated that first type is thermal failure of rolling bearings that can occur at high rotational speeds and large radial loads, and the second type deals with spindle bearings of high–speed machine tools. The results of dynamic simulations for the first type of thermal failure show that the unstable motion of the cage can lead to an ultimate bearing seizure because o f the cage failure due to the large rollers/cage contact forces and high wear rate of the cage. Second type of thermal failure of rolling element bearings, the simulation results reveal that the minimum film thickness at the raceways. Severe surface damage and wear occur at the raceways contact surfaces and eventually the bearing fails. The cause of this failure is the thermal seizure of the spindle bearing due to the rapid rise of the thermally–induced preload inside the bearing assembly with no sign of cage instability [21-22].

R.K. Upadhyay, L.A. Kumaraswamidhas and Md.Sikandar Azam**,** Rolling Contact Fatigue (RCF) occurs due to the result of cyclic stresses developed during operation and mechanism that involve in fretting failure of rolling element bearing. As bearing raceways of non-rotating rolling element bearings exposed to vibration or sliding oscillation false Brinelling occurs. Bearing surface due to false Brinelling tends to damage within a short period, due to cavities created on the bearing raceway. Recommendation towards enhancement of bearing life is also suggested.

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CHAPTER-3

DEFINITION OF THE PROBLEM

A bearing is a machine element which supports another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. A little consideration will show that due to the relative motion between the contact surfaces, a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are in direct contact, there will be rapid wear. In order to reduce frictional resistance and wear and in some cases to carry away the heat generated, a layer of fluid (known as lubricant) may be provided.

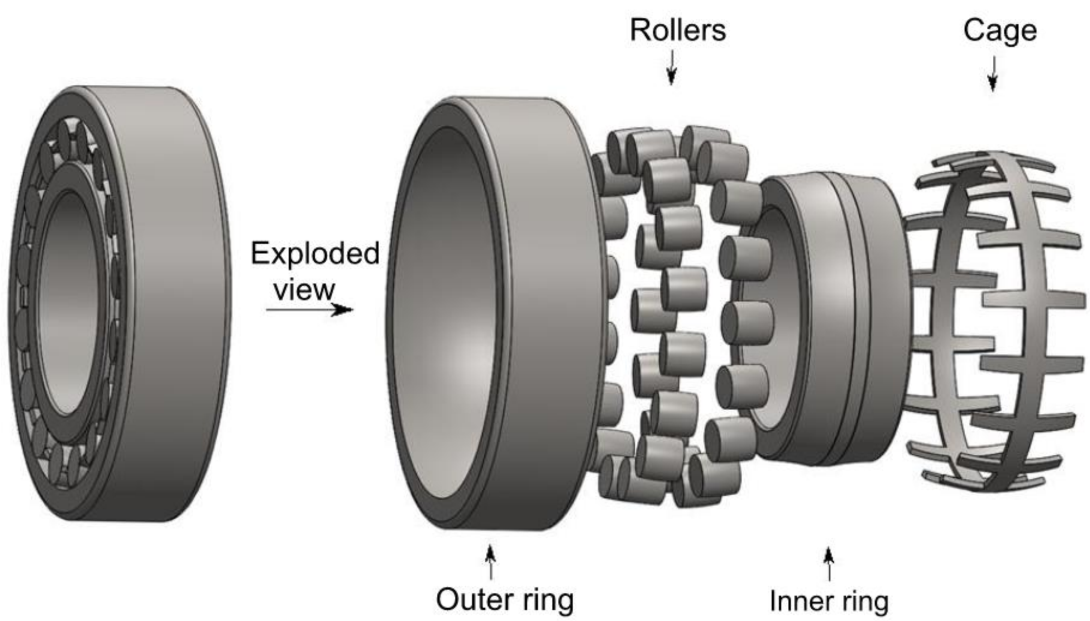
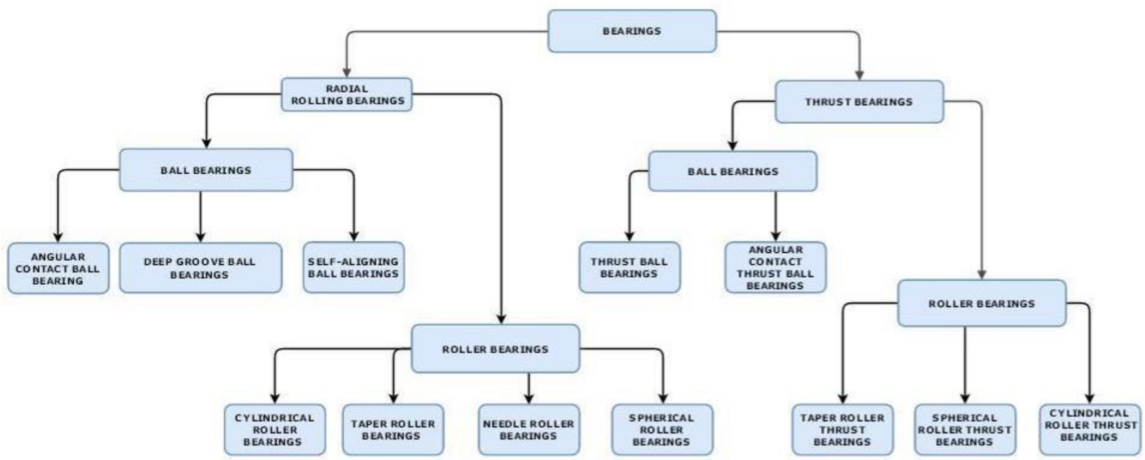


Fig 3.1: Spherical Roller Bearing

3.1 CLASSIFICATION OF BEARINGS:-

Bearings can be split into two groups: Rolling bearings and Sliding bearings. Rolling bearings attempt to eliminate friction and sliding between surfaces in a junction by introducing interfaces such as balls or rollers which rotate or roll in as opposed to sliding. Examples of this type of bearings are axial ball and roller bearings.



3.1.1 ANGULAR CONTACT BALL BEARINGS:-

Angular contact ball bearings have inner and outer ring raceways that are displaced relative to each other in the direction of the bearing axis. This means that these bearings are designed to accommodate combined loads, i.e. simultaneously acting radial and axial loads.

FEATURES OF ANGULAR CONTACT BALL BEARINGS

- The contact angle between the bearing balls and rings is normally 15, 30, or 40 degrees.
- The larger contact angle bearings have greater thrust load capacity.
- Single row bearings can accommodate radial load and axial load in one direction only.

• Double row matched bearings can accommodate radial load and axial load in either direction.

• The DT double tandem arrangement should be used for higher axial loads.

• Angular contact bearings can provide a high degree of accuracy and are capable of running at high speed.

• Cages can be pressed steel, machined copper alloy, molded polyamide, and machined synthetic resin.

MAIN APPLICATIONS:-

• Automotive: Front wheel bearings, transmissions, differential pinion shaft.

• Electrical: High frequency motors.

• Industrial Equipment: Machine tool spindles, pumps, gas turbines, centrifugal separators, and Printing equipment.

3.1.2 DEEP GROVE BALL BEARINGS:-

Deep grove ball bearings are rolling-element bearings with spherical rolling elements. They are often manufactured as single-row, closed and inseparable bearings. Deep groove ball bearings are intended primarily to accommodate radial forces.

FEATURES OF ANGULAR CONTACT BALL BEARINGS

- The inner and outer rings have deep uninterrupted grooves.
- High speed capability.
- Because the rings and balls cannot move out of axial alignment due to the groove, they can also accommodate - in addition to radial forces - high axial forces in two directions.
- Deep groove ball bearings have a comparatively rigid geometry, which means that only a very small angle of incline is possible between the inner and outer rings.

- Sealed and shielded types are filled with an appropriate volume of grease.
- Bearings with a locating snap ring aid axial positioning and eliminate need for housing shoulder.
- Cages can be a pressed steel, machined copper alloy, molded polyamide, synthetic resin, pressed stainless sheet steel.

MAIN APPLICATIONS

- Automotive: Transmissions, electrical devices, truck and trailer equipment
- Off-Highway Vehicles: Construction equipment, agricultural equipment, railroad rolling equipment.
- Electrical Equipment: Standard motors, electric appliances for domestic use.
- Other: Measuring instruments, medical instruments, and miscellaneous industrial equipment.

3.1.3 SELF-ALIGNING BALL BEARINGS:-

Self-aligning ball bearings have two rows of balls, a common sphere raceway in the outer ring and two deep uninterrupted raceway grooves in the inner ring.

FEATURES OF ANGULAR CONTACT BALL BEARINGS

- Self-aligning ball bearings generate less friction than any other type of rolling bearing, which enables them to run cooler even at high speeds.
- Due to low heat generation the bearing temperature is lower, leading to extended bearing life and maintenance intervals.
- Self-aligning ball bearings have low minimum load requirements.
- Very loose conformity between balls and outer ring keeps friction and frictional heat at low levels.
- Self-aligning ball bearings can reduce noise and vibrations levels, for example, in fans.

MAIN APPLICATIONS

Power transmission shafts of wood working and spinning machines and Plummer blocks.

3.1.4 CYLINDRICAL ROLLLER BEARINGS:-

Cylindrical Roller Bearings are bearing in which cylinders are used as the rolling elements as opposed to balls in ball bearings. As such, the rollers have a greater (linear) contact area with the outer ring and are distribute loads across a broader surface.

FEATURES OF ANGULAR CONTACT BALL BEARINGS

- High load carrying capacity.
- High stiffness.
- The surface finish on the contact surfaces of the rollers and raceways supports the formation of a hydrodynamic lubricant film.

MAIN APPLICATIONS

- Cylindrical roller bearings are essential part of any machinery that comes with rotating parts.
- These are widely used in paper-making industry, electric motors, railways, motorcycles, pumps, wind turbines and gear boxes.

3.1.5 TAPERED ROLLER BEARINGS:-

Tapered roller bearings are rolling element bearings that can support axial forces (i.e., they are good thrust bearings) as well as radial forces.

FEATURES OF TAPERED ROLLER BEARINGS

- The inner ring assembly of a tapered roller is referred to as the cone, while the outer ring is called the cup.

- The taper of the rollers and the cone raceways have a common apex on the bearing axis.

- The angle between the cup raceway and the bearing axis is the contact angle.
- Single row bearings can accommodate radial load and axial load in one direction.
- Two single row bearings are used opposing each other to simplify setting of the proper clearance between the two rows.
- Cages material is pressed steel or a pin type.

MAIN APPLICATIONS

- AUTOMOTIVE: Front and rear wheels, transmissions, differential pinion
- OFF-HIGHWAY VEHICLES: Railroad, Construction, and Agricultural equipment
- INDUSTRIAL: Rolling mill equipment, gear boxes, and machine tool spindles.

3.1.6 NEEDLE ROLLER BEARINGS:-

A needle roller bearing is a special type of roller bearing which uses long, thin cylindrical rollers resembling needles.

FEATURES OF NEEDLE ROLLER BEARINGS

- For those applications where minimal cross section height is required.
- The rollers of needle bearings are longer and smaller in diameter than cylindrical rollers.
- Needle roller bearings cannot carry thrust loading but have relatively high radial capacity.
- Cage material is pressed steel.

MAIN APPLICATIONS

- AUTOMOTIVE: Automobile engines, transmissions, pumps, and compressors
- OFF-HIGHWAY VEHICLES: Power shovel, wheel drums
- INDUSTRIAL: Overhead cranes, hoists, and power tools.

3.1.7 SPHERICAL ROLLER BEARINGS:-

A spherical roller bearing is a rolling-element bearing that permits rotation with low friction, and permits angular misalignment. Typically these bearings support a rotating shaft in the bore of the inner ring that may be misaligned in respect to the outer ring.

FEATURES OF SPHERICAL ROLLER BEARINGS

- The bearing is self-aligning and forgiving to errors of up to 1.5° of misalignment between the shaft and housing due to shaft bending.
- Spherical bearings can carry radial and axial load in both directions.
- Suitable for applications where there is heavy radial and impact loading applied.
- The bearing can be manufactured with a cylindrical or taper bore.
- Lubrication grooves and holes can be provided on either the inner or outer rings.
- Cage types can be pressed steel, machined copper alloy.

MAIN APPLICATIONS

- OFF-HIGHWAY: Railroad axle journals
- INDUSTRIAL: Paper manufacturing equipment, gear boxes, continuous casters, rolling mill pinion stands, table rollers, crushers, shaker screens, and printing cylinders.

3.1.8 TAPERED ROLLER THRUST BEARINGS:-

Tapered roller thrust bearings consist of small tapered rollers arranged so that their axes all converge at a point on the axis of the bearing.

FEATURES OF TAPERED THRUST BEARINGS

- This thrust bearing design uses tapered rollers with spherical large ends precisely guided by ribbed shaft and housing washer rings.

- Both the shaft and housing washers have tapered raceways whose apexes along with the rollers, converge on a point on the bearing axis.
- This bearing has a very high thrust load capacity.
- Single direction thrust load designs are normally used, but double direction designs are possible.
- Cage material is normally a machined copper alloy.

MAIN APPLICATIONS

Crane hooks crushers, oil excavator swivels, and rolling mill screw down equipment.

3.1.9 SPHERICAL ROLLER THRUST BEARINGS:-

A spherical roller thrust bearing is a rolling-element bearing of thrust type that permits rotation with low friction, and permits angular misalignment.

FEATURES OF SPHERICAL THRUST BEARINGS

- This bearing uses spherical convex rollers arranged at an angle to the axis of the bearing.
- Due to the rollers and spherical raceways on the washers, some shaft misalignment can be tolerated.
- The large number of rollers, which have an optimum conformity with the washer raceways, enables the bearings to accommodate heavy axial and simultaneously acting radial loads.
- This bearing can support radial load up to 55% of the axial load being carried.
- Spherical thrust bearings are not suitable for high speed operation.
- The cage is a machined copper alloy.

MAIN APPLICATIONS

- Vertical motors, deep well pumps, ship propeller shafts, jib cranes, screw down reducers, and hydroelectric generators.

3.1.10 CYLINDRICAL ROLLER THRUST BEARINGS:-

A cylindricalrollerthrust bearing is a particular type of roller bearing. Like other roller bearings, they permit rotation between parts, but they are designed to support a high axial load while doing this (parallel to the shaft).

FEATURES OF CYLINDRICAL THRUST BEARINGS

- To prevent stress peaks, the roller ends are relieved slightly to modify the line contact between the raceway and rollers.
- Very high axial load capacity and rigidity.
- Accommodates axial load in one direction only.
- Cage is a machined copper alloy material or pressed steel.

MAIN APPLICATIONS

- Oil excavator swivels and steel crane hooks.

3.2 STATIC LOAD CARRYING CAPACITY FOR 23024CC, 22224CC AND 2226CC BEARING USING STRIBECK’S EQUATION

3.2.1 STATIC LOAD CARRYING CAPACITY:

Static load is defined as the load acting on the bearing when the shaft is stationary. It produces permanent deformation in balls and races, which increases with increasing load. The permissible static load, therefore, depends upon the permissible magnitude of permanent deformation. From past experience, it has been found that a total permanent deformation of 0.0001 of the ball or roller diameter occurring at the most heavily stressed ball and race contact, can be tolerated in practice, without any disturbance like noise or vibrations. The static load carrying capacity of a bearing is defined as the static load which corresponds to a total permanent deformation of balls and races, at the most heavily stressed point of contact, equal to 0.0001 of the ball diameter.

STRIBECK'S EQUATION:

Stribeck's equation gives the static load capacity of bearing. It is based on the following assumptions:

(i)The races are rigid and retain their circular shape.

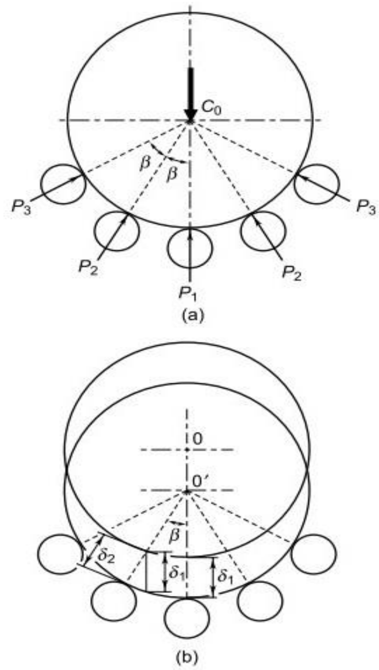
(ii) The balls are equally spaced.

(iii) The balls in the upper half do not support any load.

Figure (a) shows the forces acting on the inner race through the rolling elements, which support the static load C0. It is assumed that there is a single row of balls. Considering the equilibrium of forces in the vertical direction,

$$Co = P_1 + 2P_2 \cos(\beta) + 2P_3 \cos(2\beta) + \ldots \quad (a)$$

As the races are rigid, only balls are deformed. Suppose d_1 is the deformation at the most heavily stressed Ball No.1. Due to this deformation, the inner race is deflected with respect to the outer race through d_1.



As shown in Fig. (b), the centre of the inner ring moves from O to O¢ through the distance δ_1 without changing its shape. Suppose δ_1, δ_2 … are radial deflections at the respective balls.

$$\text{Also, } \delta_2 = \delta_1\cos\beta \text{ (or) } \delta_2/\delta_1 = \cos\beta$$

According to Hertz's equation, the relationship between the load and deflection at each ball is given by,

$$\delta \ \alpha \ (P)^{2/3}$$

Therefore,

$$\delta_1 = C_1 P_1^{2/3} \quad \text{and} \quad \delta_2 = C_1 P_2^{2/3}$$

$$\delta 2/\delta 1 = (P_2/P_1)^{2/3}$$

From Eq. (b) and (c)

$$(P_2/P_1)^{2/3} = \cos\beta$$

Or

$$P_2 = P_1 (\cos\beta)^{3/2}$$

In a similar way,

$$P_3 = P_1 (\cos 2\beta)^{3/2}$$

Substituting these values in Eq. (a),

$$C_0 = P_1 + 2[P_1 (\cos\beta)^{3/2}]\cos\beta + 2[P_1 (\cos 2\beta)^{3/2}]\cos 2\beta + \ldots\ldots.$$

$$= P_1 [1 + 2(\cos\beta)^{5/2} + 2(\cos 2\beta)^{5/2} + \ldots..]$$

Or

$$C_0 = P_1 M$$

Where

$$M = [1 + 2(\cos\beta)^{5/2} + 2(\cos 2\beta)^{5/2} + \ldots\ldots..]$$

If z is the number of balls,

$$\beta = 360/z$$

The values of M for different values of z are tabulated as follows:

Z	8	10	12	15
M	1.84	2.28	2.75	3.47
(z/M)	4.35	4.38	4.36	4.37

It is seen from the above table that (z/M) is practically constant. Stribeck suggested the value for (z/M) as 5.

Or

$$M = (1/5)\ z$$

Substituting this value in Eq. (d),

$$C_0 = (1/5)\ zP_1$$

From experimental evidence, it is found that the force P_1 required to produce a given permanent deformation of the ball is given by,

$$P_1 = kd^2$$

Where *d* is the ball diameter and the factor *k* depends upon the radii of curvature at the point of contact,

and on the modulus of elasticity of materials. From Eqs (f) and (g),

$$C_0 = (kd^2z/5)$$

The above equation is known as Stribeck's equation.

3.3 DYNAMIC LOAD CARRYING CAPACITY FOR 23024CC AND 22224CC BEARING:

3.3.1 DYNAMIC LOAD CARRYING CAPACITY:

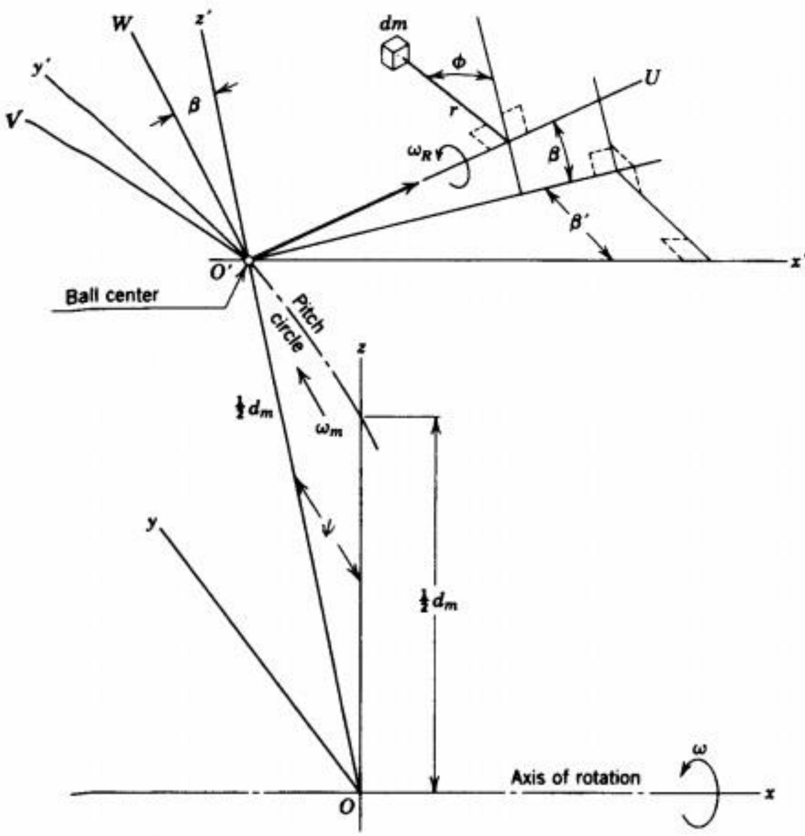
The surfaces of rolling elements and races undergo fatigue failure while the machine is running which imposes limit on the life of bearings. Therefore, dynamic load carrying capacity (*C*) of the bearing must be considered in designing the bearings. The dynamic load carrying capacity is defined as the radial load in radial bearings (or thrust load in thrust bearings) that can be carried for a minimum life of one million revolutions by 90% of the bearings before fatigue crack appears.

DYNAMIC LOADING:-

Body Forces Due to Rolling Element Rotations:-

The development of equations in this section is based on the motions occurring in an angular-contact ball bearing because it is the most general form of rolling bearing. Subsequently, the equations developed can be so restricted as to apply to other ball bearings and also to roller bearings.

Following figure illustrates the instantaneous position of a particle of mass in a ball of an angular-contact ball bearing operating at high rotational speed about an axis x.To simplify the analysis the following coordinate axes systems are introduced:



x, y, z: - A fixed set of Cartesian coordinates with the x axis coincident with the bearing rotational axis.

x^1, y^1, z^1:- A set of Cartesian coordinates with the x^1 axis parallel to the x axis of the fixed set. This set of coordinates has its origin O^1 at the ball center and rotates at orbital speed about the fixed x axis at radius $1/2d_{m.}$

U , V , W :-A set of Cartesian coordinates with origin at ball center O^1 and rotating at orbital speed ω_m. The U axis is collinear with the axis of rotation of the ball about its own centre. The W axis is in the plane of the U axis and z^1 axis; the angle between the W axis and z^1 axis is β.

U, r, φ: A set of polar coordinates rotating with the ball.

In addition to the foregoing coordinate systems, the following symbols are introduced:

β' The angle between the projection of the U axis on the x'y' plane and the x' axis.

Ψ The angle between the z axis and z' axis, that is, the angular position of the ball on the pitch circle.

Consider that an element of mass DM in5the ball has the following instantaneous location in the system of rotating coordinates: U, r, φ,

Since

$$U = U$$

$$V = r \sin \varphi$$

$$W = r \cos \varphi$$

and

$$x' = U \cos \beta \cos \beta' - V \sin \beta' - W \sin \beta \cos \beta'$$

$$y' = U \cos \beta \sin \beta' + V \cos \beta' - W \sin \beta \sin \beta'$$

$$z' = U \sin \beta + W \cos \beta$$

and

$$x = x'$$

$$y = d_m \sin\psi + y' \cos\psi + z' \sin\psi$$

$$z = d_m \cos\psi - y' \sin\psi + z' \cos\psi$$

Therefore, by substitution of equations into and thence into, the following expressions relating the instantaneous position of the element of mass dm to the fixed Cartesian coordinates can be formulated.

$$x = U \cos\beta \cos\beta' - r(\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)$$

$$+ r(\cos\beta \sin\varphi \cos\psi + \cos\beta \cos\varphi \sin\varphi - \sin\beta \sin\beta' \cos\varphi \cos\psi)$$

$$z = d_m \cos\psi + U(-\cos\beta \sin\beta' \sin\psi + \sin\beta \cos\psi) + r(-\cos\beta' \sin\varphi \sin\psi + \cos\beta \cos\varphi \cos\psi + \sin\beta \sin\beta' \cos\varphi \cos\psi)$$

In accordance with Newton's second law of motion, the following relationships can be determined if the rolling element position angle ψ is arbitrarily set equal to 0°;

$$dF_x = xdm$$

$$dF_y = ydm$$

$$dF_z = zdm$$

$$dM_z' = \{-x[U \cos\beta \sin\beta' + r(\cos\beta' \sin\varphi - \sin\beta \sin\beta' \cos\varphi)]$$

$$+ y\ [U \cos\beta \cos\beta' - r(\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)]\}dm$$

$$dM_y' = \{x[U \sin\beta + r \cos\beta \cos\varphi]$$

$$- z[U \cos\beta \cos\beta' - r(\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)]\}dm$$

The net moment about the x axis must be zero for constant speed motion. At each ball location (), (rotational speed of the ball about its own axis U -0) and (orbital speed of the ball about the bearing axis x) are constant; therefore, at ψ=0.

$$x = d^2x/dt^2 = r\omega^2_R(\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)$$

$$y = d^2y/dt^2 = -2\omega_R\omega_m r \cos\beta \sin\varphi$$

$$+\omega^2_m[-U \cos\beta \sin\beta' + r(-\cos\beta' \sin\varphi + \sin\beta \sin\beta' \cos\varphi)]$$

$$+\omega^2_R r (\cos\beta' \cos\varphi + \sin\beta \sin\beta' \sin\varphi)$$

$$z = d^2z/dt^2 = -2\omega_R\omega_m r(\cos\beta' \cos\varphi + \sin\beta \sin\beta' \sin\varphi)$$

$$-\omega^2_m(d_m/2 + U \sin\beta + r \cos\beta \cos\varphi) - \omega^2_R r \cos\beta \cos\varphi.$$

$$F_x' = -\rho \int_{-rR}^{+rR} \int_0^{(rR2-U^2)^{1/2}} \int_0^{2\pi} \ddot{x} r\, dr\, dU\, d\emptyset$$

$$F_y' = -\rho \int_{-rR}^{+rR} \int_0^{(rR2-U^2)^{1/2}} \int_0^{2\pi} \ddot{y} r\, dr\, dU\, d\emptyset$$

$$F_z' = -\rho \int_{-rR}^{+rR} \int_0^{(rR2-U^2)^{1/2}} \int_0^{2\pi} \ddot{z} r\, dr\, dU\, d\emptyset$$

$$M_{z'} = -\rho \int_{-rR}^{+rR} \int_0^{(rR2-U^2)^{\frac{1}{2}}} \int_0^{2\pi} \{-\ddot{x}[U \cos\beta \sin\beta'$$

$$+ r(\cos\beta' \sin\varphi + \sin\beta \sin\beta' \cos\varphi)]$$

$$+ \ddot{y}[U \cos\beta \cos\beta' - r (\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)]\} r\, dr dU\, d\varphi$$

$$M_{y'} = -\rho \int_{-rR}^{+rR} \int_0^{(rR2-U^2)^{\frac{1}{2}}} \int_0^{2\pi} \{x[U \sin\beta + r \cos\beta \cos\varphi - z[U \cos\beta \cos\beta' - r (\sin\beta' \sin\varphi + \sin\beta \cos\beta' \cos\varphi)]\} r\, dr\, du\, d\varphi$$

In the above equations ρ is the mass density of the ball material and r_R is the ball radius. Performing the integrations indicated by equations establishes that the net forces in the x' and y' directions are zero and that

$$F_z' = ½\, m d_m \omega^2_m$$

$$M_y' = J\omega_R\omega_m \sin\beta$$

$$M_z' = -J\omega_R\omega_m \cos\beta \sin\beta' r_R^2$$

In which m is the mass of the ball and J is the moment of the inertia. M and J are defines as follows:

$$m = 1/6\ P\pi D^3$$

$$J\ \ = 1/60\ P\pi D^5$$

Dynamic load of spherical roller bearing is calculated by using the emperical formula

$$F=XVF_r + YF_a$$

PROJECT CALCULATIONS:

BUCKET WHEEL SPECIFICATIONS:-

BUCKET WHEEL	SIZE	DIA 8M
	TYPE	CELL LESS TYPE
	QTY	2
SPEED		4 RPM
MOTOR	MODEL	55KW HAGGLUNDS CA50, LOW SPEED HIGH TORQUE
	QTY	1 (PER BUCKET WHEEL)
HYDRAULIC POWER PACK	MODEL	PEC 403 – 110 KW
	QTY	1
BUCKET	CAPACITY	420 LTRS(INCLUDING RING VOLUME)
	QTY	10 (PER BUCKET WHEEL)
	MATERIAL	HARDOX 400
REPLACEABLE BUCKET LIP	MATERIAL	CR . CARBIDE
LINER PLATES	MATERIAL	MN STEEL
CHAIN	MODEL	64B – 1; DIN 8187/ ISO R606

	CAPACITY	MIN BREAKING LOAD – 600 KN
	MAKE	ROLCON
	LENGTH	46000 MM (APPROX)
SPROCKET(DRIVE AND IDLE)	NO OF TEETH	18 TEETH
	MATERIAL	C 45
	SIZE	P.C.D. 585.1 MM
BUCKET WHEEL SPROCKET	NO OF TEETH	189 TEETH
	MATERIAL	C 45
	SIZE	P.C.D 6112.8 MM

Under static load condition, the weight of the bucket wheel which is 12 tonnes is exerted on the bearing. This bucket wheel is supported by 8 roller bearing. Out of these 8 rollers 4 are placed at the top and 4 at the bottom.

This 12 tonnes of load is exerted only the top 4 rollers. The bucket wheel weight acting on one roller = 12/4

= 3 tonnes

Each bucket wheel consists of 10 buckets and the capacity of each bucket is 420 litres ($0.42m^3$). The density of the iron ore that is carried by bucket is 2.2×10^3 kg/m^3. During running conditions only 4 buckets of the wheel are filled with iron ore.

Therefore, the total capacity of 4 buckets = $0.42m^3 \times 4$

= $1.68m^3$

We know that, Density = Mass/Volume

Mass = Density x Volume

Therefore the total mass of the 4 loaded buckets = $2.2 \times 10^3 \times 1.68$

= 3696 kg

= 3.696 tonnes

When the loaded bucket reaches to the top, the total weight of the bucket is distributed equally between the two rollers, the total load on the one roller

$$= (3.696 \text{ tonnes})/2$$

$$= 1.8 \text{ tonnes}$$

Therefore the total weight acting on one roller $= (3\text{tonnes}) + (1.8\text{tonnes})$

$$= 4.8 \text{ tonnes}$$

$$\text{Sin } 55° = 4.8/x$$

$$x = 5.85 \text{ tonnes} \sim 6 \text{ tonnes}$$

Therefore, the radial force acting on the roller is 6 tonnes.

$$F_r = 6 \text{ tonnes}$$

$$= 6 \times 10^3 \times 9.81$$

$$= 58.86 \text{ KN}$$

The empirical formula for the Basic Static Load is $C_0 = S_0 \times F_r$

Table 3.1: Minimum Safety Factor Values (S_0)

Operating Conditions	Ball Bearing	Roller Bearing
High Rotational accuracy demand	2	3
Normal Rotational accuracy demand (Universal application)	1	1.5
Slight Rotational accuracy deterioration permitted (Low speed, Heavy Loading etc.,)	0.5	1

Let us consider the operating conditions as high rotational accuracy demand for the roller bearing, so the value of S_0 is 3.

Hence the Basic Static Load $C_0 = 3 \text{ x } 58.86$

$= 176.81$ KN

Dynamic load of spherical roller bearing is calculated by using an Emperical formula

$$F = XVF_r + YF_a$$

Where, X = Radial Load Factor = 2.04

Y = Axial Load Factor

V = Race Rotation Factor = 0.54

F_r = Actual Radial Load = 58.86 KN

F_a = Actual Axial Load = 0 KN

Dynamic Load F = 2.04 X 0.54 X 1.3

= 156.282 KN

When a radial or axial load has been mathematically calculated, the actual load on the bearing may be greater than the calculated load because of vibration and shock present during operation of the machine.

The actual load may be calculated using the equation

$$Fr = fw \text{ x } Frc$$

where Fr = Loads applied on bearing (kN) , f_w = Load factor

Frc = Theoretically calculated load (kN)

Table 3.2 : Load Factor Values (f_w)

Operating Conditions	Typical Applications	fw
Smooth operation free from shocks	Electrical motors, Machine tools, air conditioners	1 to 1.2
Normal operation	Air blowers, Compressors, Elevators, Cranes	1.2 to 1.5
Operation accompanied by shocks and vibration	Construction equipment, Crushers, Vibrating screens	1.5 to 3

Therefore, The Basic static load value = 176.81 x 3

= 530.43 KN

The dynamic load value = 156.28 x 3

= 468.84 KN

3.2 FAILURE OF SPHERICAL ROLLER BEARING:

The bearing will fail due to many reasons and some of them are as follows.

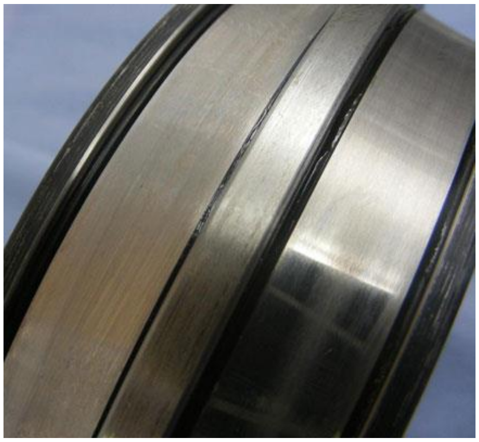
- Flaking and Pitting
- Wear and Fretting
- Rust and Corrosion
- Peeling
- Discoloration
- Smearing
- Creep
- Failure Of Cage

1. FLAKING AND PITTING:-

Flaking is a phenomenon in which the bearing surface turns scaly and peels off due to contact load repeatedly received on the raceway and rolling surface during rotation. Occurrence of flaking indicates that the end of a bearing's service life is near.

Flaking On The Inner Ring Of Spherical Roller Bearing

Pitting is a phenomenon in which small holes of 0.1 mm in depth are generated on the raceway surface by rolling fatigue.



Pitting on inner ring of Spherical Roller Bearing

Flaking and pitting are often found at an early stage. In this case, countermeasures should be taken, after examining the causes.

CAUSES OF FLAKING AND PITTING:-

Flaking and pitting occur early in a bearing's service life under the following conditions:

- During operation, bearing internal clearance becomes narrower than specified.
- Bearing ring is mounted at an inclination by mistake.
- Flaw is created during mounting, or brinelling, nicks, rust, etc. occur on the raceway surface or rolling surface.
- Inaccurate shape of shaft or housing (imperfect circle, depressions on surface.)

COUNTER MEASURES:-

Flaking:

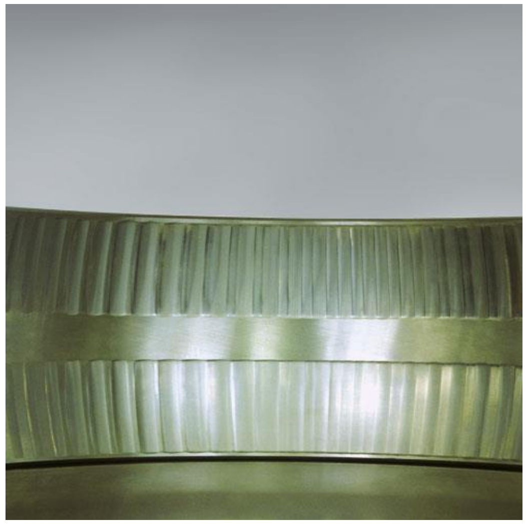
- Use a bearing with heavier rated load.
- Check if abnormal load is being generated.
- Improve lubrication method to ensure better formation of lubricant film, by increasing the viscosity.
- When a failure is discovered at an early stage, the countermeasures described above should be taken, after investigating the causes.

Pitting:

- Increase viscosity of lubricant to ensure better formation of lubricant film. (Care should be taken because foreign matters appear similar to holes caused by brinelling or corrosion.)

2. WEAR AND FRETTING:-

Wear is caused mainly by sliding abrasion on parts including the roller end face and rib, cage pocket surface, cage, and the guide surface of the bearing ring. Wear due to contamination by foreign matter and corrosion occurs not only to the sliding surface but also to the rolling surface.



Outer Ring Of A Spherical Roller Bearing

Fretting is a phenomena which occurs when slight sliding is repeatedly caused on the contact surface. On the fitting surface, fretting corrosion occurs, generating rust like powder.

Fretting Marks On Outside Diameter Of Spherical Roller Bearing

If bearings receive a vibration load when they stop or operate, slight sliding occurs in the section between the rolling element and bearing ring due to elastic distortion. False brinelling, a flaw similar to brinelling, is generated by this condition.

CAUSES OF WEAR AND FRETTING:-

Wear:

- Improper lubricant or shortage of lubricant.
- Contamination by foreign matter(s).

Fretting:

- Vibration load.
- Slight vibration on fitting surface caused by load.

COUNTER MEASURES:-

Wear:

- Review and improvement of lubricant and lubrication method.
- Filtering of oil.
- Improvement of sealing.

Fretting:

- Investigation and countermeasures for the source of vibration.

3.5.1 RUST AND CORROSION:-

Rust and corrosion are pits on the surface of rings and rolling elements and may occur at the rolling element pitch on the rings or over the entire bearing surfaces.

Rollers Of A Spherical Roller Bearing

CAUSES OF RUST AND CORROSION:-

- Entry of corrosive gas or water.
- Improper lubricant.
- Formation of water droplets due to condensation of moisture.
- High temperature and high humidity while stationary.
- Poor rust-preventive treatment during transporting.
- Improper storage conditions.
- Improper handling.

COUNTERMEASURES:-

- Sealing mechanism should be improved.
- Anti-rust treatment during periods of non-running conditions should be given.
- Storage, handling and lubrication methods should be improved.

3.5.2 PEELING:-

Dull or cloudy spots appear on surface along with light wear. From such dull spots, tiny cracks are generated downward to a depth of 5-10µm. Small particles fall off and minor flaking occurs widely.

Outer Ring Of A Spherical Roller Bearing

CAUSES OF PEELING:-

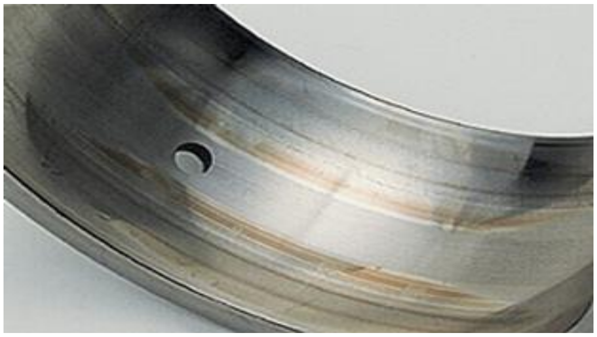
- Unsuitable lubricant.
- Entry of debris into lubrication.
- Rough surface due to poor lubrication.
- Surface roughness of mating rolling parts.

COUNTERMEASURES:-

- Select a proper lubricant.
- Improve the sealing mechanism.
- Improve the surface finish of the rolling mating parts.

3.5.3 DISCOLORATION:-

Discoloration occurs due to reactions with lubricant at higher temperature. This occurs in cages, rolling elements and raceway rings.



OUTER RING OF A SPHERICAL ROLLER BEARING

CAUSES OF DISCOLORATION:-

- Poor lubrication.
- Oil stain due to a reaction with lubricant.
- High temperature.

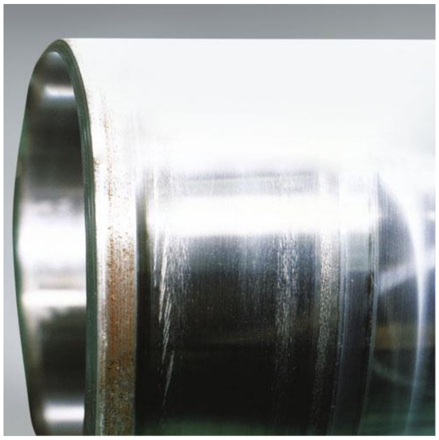
- A brown discoloration of the rolling or sliding surface is caused by adhesion of acidic powders generated by abrasion during operation.
- In general, these powders adhere uniformly to the bearing circumference.

COUNTERMEASURES:-

- Improvement of heat dissipation from bearings.
- Improvement of lubrication.

3.5.4 SMEARING:-

Smearing is a surface damage which occurs from a collection of small seizures between bearing components caused by oil film rupture.



Inner ring of a spherical roller bearing

Outer ring of a spherical roller bearing

CAUSES OF SMEARING:-

- Smearing occurs if the oil film disappears as rolling elements stop rotating due to inappropriate use or improper lubrication, and then starts to slide on the raceway surface.
- In ball bearings smearing is caused by sliding or spinning of balls and in roller bearings, smearing tends to occur when the roller enters into on from the load zone.

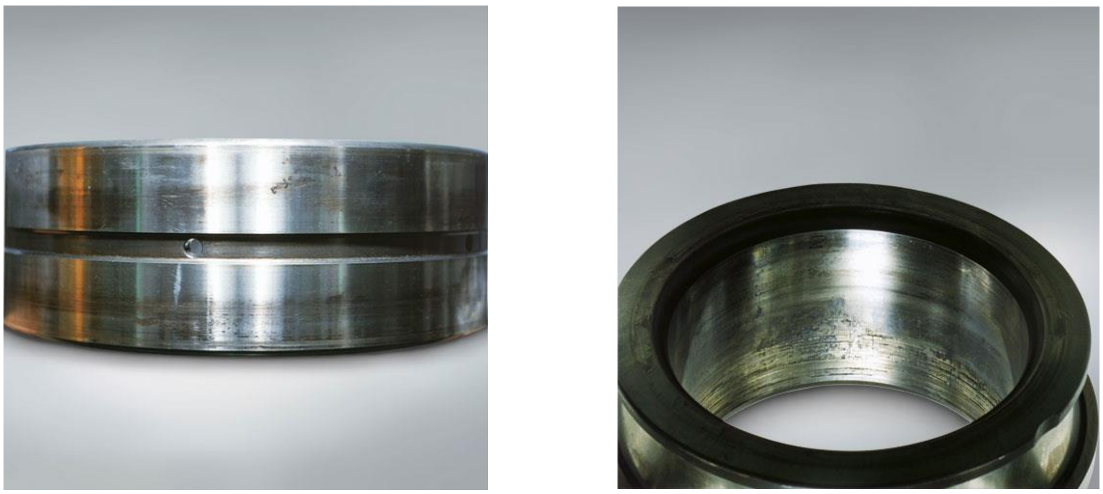
COUNTERMEASURES:-

- Provision for extreme-pressure lubricant.
- Improving the bearing clearance.

- Improving the sealing mechanism.
- Improve the pre-load.

3.5.5 CREEP:-

Creep is the phenomenon in bearings where relative slippage occurs between fitting surfaces and thereby creates a clearance between surfaces. It creates a shiny appearance. The following figure shows the inner and outer ring of spherical roller bearing.



Inner And Outer Ring Of A Spherical Roller Bearing

CAUSES OF CREEP:-

- Creep occurs when interference is too small in relation to the heat or load generated during operation.
- Insufficient sleeve tightening.

COUNTERMEASURES

- Interference should be checked and rotation should be prevented.
- Adhesive should be applied to the fitting surface.
- Lubricant should be applied to the fitting surface.

3.5.6 FAILURE OF CAGE:-

Failure of cage will takes place when all above failures will takes place.

CHAPTER 4

MODELLING AND ANALYSIS

4.1 INTRODUCTION TO SOLIDWORKS:

SolidWorks is a 3D solid modeling package published by dessault systems which allows users to develop full solid models in a simulated environment for both design and analysis in which we can sketch ideas and experiment with different designs to create 3D models. It is used to build parts, assemblies and drawings that take advantage of the familiar microsoft windows graphical user interface. it has capabilities to help you create and edit models efficiently. it includes familiar windows functions such as dragging and resizing windows. It enables the designers to create a mathematically correct solid model of an object that can be stored in a database. When the mathematical model of a part or assembly is associated with the properties of the materials used, we get a solid model that can be used to simulate and predict the behavior of the part or model with finite element and other simulation software. The same soild model can be used to manufacture the object and also contains the information necessary to inspect and assemble the product. The marketing organization can produce sales brochures and videos that introduce the product to potential customers. Solidworks and similar CAD programs have made possible concurrent engineering, where all the groups that contribute to the product development process can share information real-time.

4.1.1 USING THE USER INTERFACE:

1. Staring the program :
   - Click the start button
   - From the start menu, click all programs, solidworks.
2. Exit the program :
   - To exit the appilication program, click file, exit.

3. Opening an existing file :
   - We can open the file by selecting file, open and typing or browsing to a file name or by selecting a file name from the file menu in solidworks.
4. Copying a file :
   - Click ‘file’, ‘save as’ to save a copy of a file with a new name.
   - The ‘save as’ window appears, this window shows you in which folder the file is currently located, the file name and the file type.
   - In the ‘ file name’ field, change the name and click ‘save’.
5. Resizing windows :
   - Solidworks, like many applications, uses windows to show the work done.
   - you can change the size of the window by moving the cursor along the edge of the window until the shape of cursor appears to be a 2 headed arrow while the cursor appears so, hold down the left mouse button and drag the window to a different size.
6. Solidworks windows:
   - Solidworks windows have two panels. One panel provides non graphic data. The other panel provides graphic representation of the part, assembly or drawing.
   - The left most panel of the window contains the feature manager design tree, property manager and configuration manager.
   - The right most area panel is the graphic area, where you create and manipulate the part, assembly and drawing.
7. Tool bars :
   - Tool bar buttons are shortcurts for frequently used commands.
   - You can set tool bar placement and visibility based on the document type (part, assembly or drawing).
   - Solidworks remembers which toolbars to display and where to display them for each document type.
8. Command manager :
   - The command manager is a context sensitive tool bar that dynamically updates based on the tool bar you want to access.

- By default, it has tool bars embedded in it based on the document type.
- When you click a button in the control area, the command manager updates to shows that tool bar.
- Use the command manager to access tool bar buttons in acentral location and to save space for graphical area.

4.1.2 DOCUMENTS IN SOLIDWORKS :

There are the kinds of documents in solidworks:

- Parts
- Assemblies of the parts
- Drawings of the parts or assemblies.

1.Part file :

- The first and most basic element of solidworks model is apart with extension SLDPRT.
- Part consists of primitive geometry and features such as extrudes, revolutions, lofts, sweeps, etc.
- Parts will be the building blocks for all of the models that you will create.

2.Assembly file :

- The second component is the assembly. Assemblies are collections of parts which are assembled in a particular fashion using mates (constraint).
- Any complex model will usually consist of one or many assemblies.
- The assembly may be static or dynamic with extension SLDASM. In dynamic assembly the parts are allowed to move as they would in the actual assembled part and assembly may be exploded to show how all parts go together.

3. Drawing file :

- The third and final component in solid works is the drawing which has an extension SLDDRW.
- A drawing is the typical way to represent a 3D model such that any engineer can recreate your part.
- Drawing are important because they provide a standard way of sharing your design where the part section views, detail views and availability views are all included in actual engineering drawing.

4.2 SOLIDWORKS PROCEDURE:

Step 1 : Inorder to draw the outer race of the bearing open a new file in solidworks and select the part drawing. Now select front workplane, click on sketch and draw centerline horizontally. select line option to draw the width, height and groove.

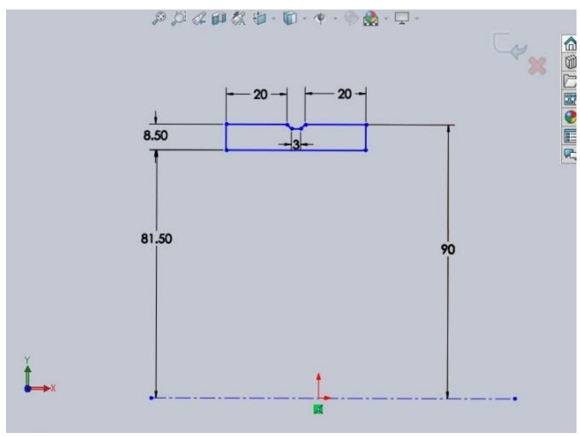


Fig 4.1 Sketch of Outer race of 23024CC bearing

Step 2: Exit the sketch, open isometric view and go to features. Select revolved boos/base and then select the axis about which the sketch should be revolved. Click on ok.

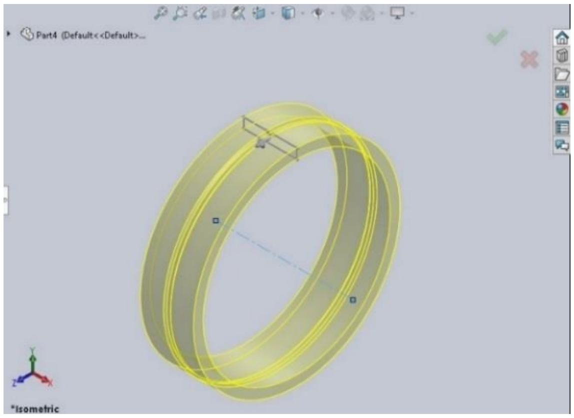


Fig 4.2 Sketch after revolving

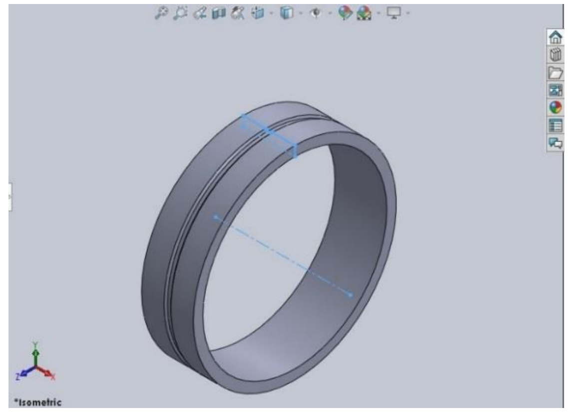


Fig 4.3 Solid part of Outer Race

Step 3: To draw the inner race of the bearing repeat the same procedure as that of the outer race according to the respective drawing of inner race.

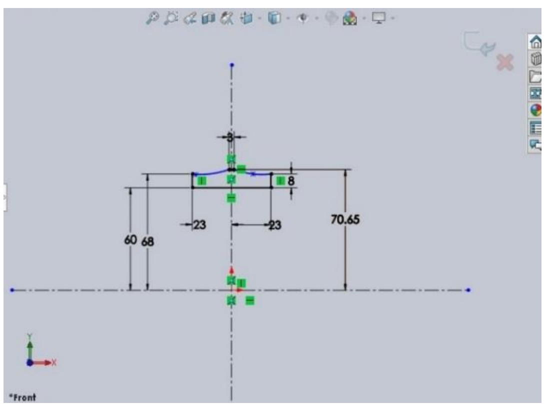


Fig 4.4 Sketch of Inner Race of 23024CC bearing

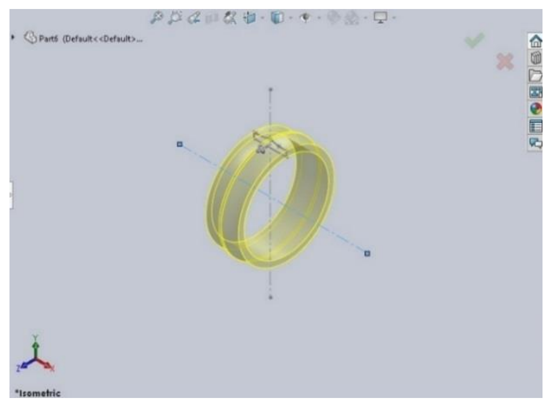


Fig 4.5 Sketch after revolving

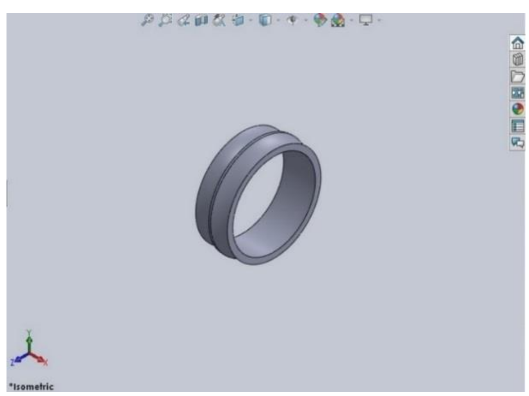
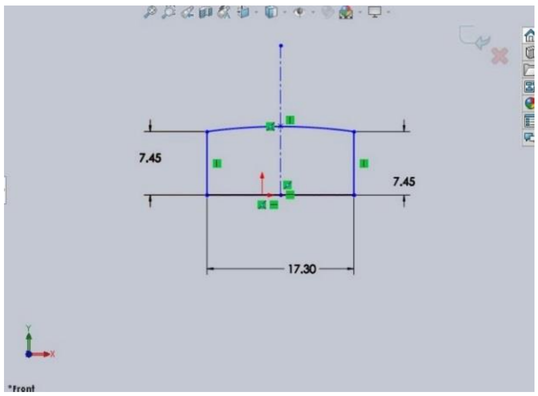
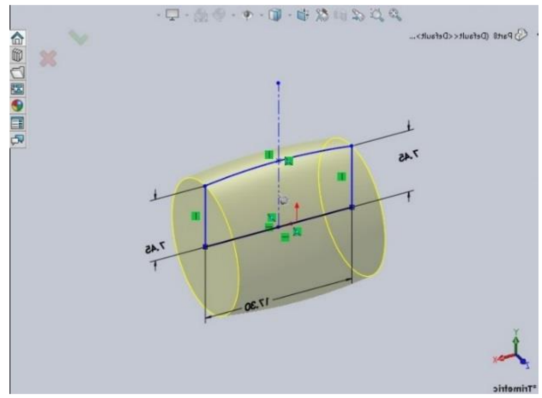


Fig 4.6 Solid part of Inner Race

Step 4: To draw the spherical rollers select the front workplane and draw the centerline horizontally. Now draw two vertical line of length equal to the radius of the roller and are separated by a distance of the length of the roller. Join these vertical lines by using spline option.





Fig

4.7 Sketch of Roller

Fig 4.8 Sketch of roller after revolving

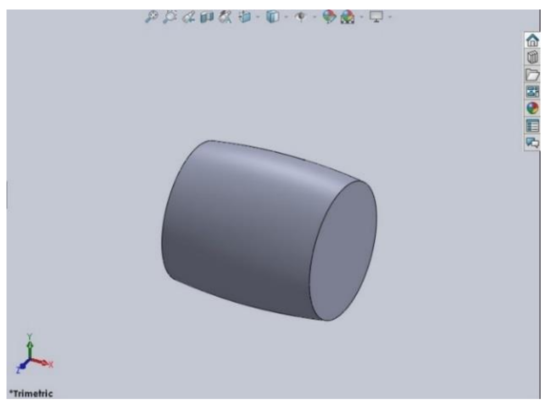


Fig 4.9 Solid part of roller

Step 5: Now to get the array of these rollers, go to features and select circular pattern to arrange the rollers in designated places.

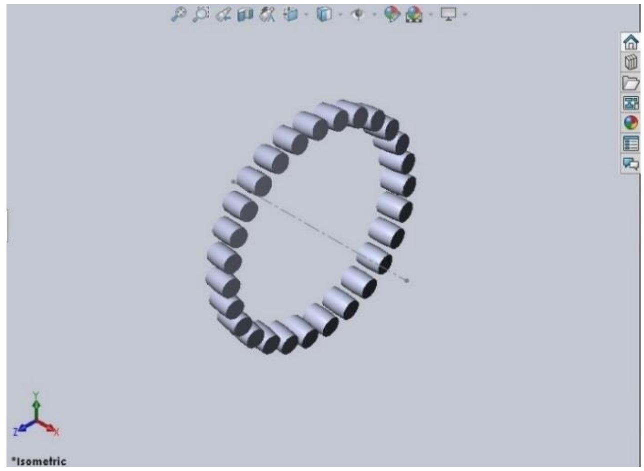


Fig 4.10 Circular Pattern of Rollers

Step 6: For the assembling of parts click on new file and select assembly drawing. Now import all the parts one by one. Now go to mate option, select the outer and inner race and click on concentric. Select the sides of the races and then click on parallel so that they get mated.

Step 7: Finally select the outer surface of the rollers and the inner surface of the outer race and click on coincident. Now select the vertical surface of the roller and side surface of the outer race so they get mated and hence we get the final assembly of the bearing.

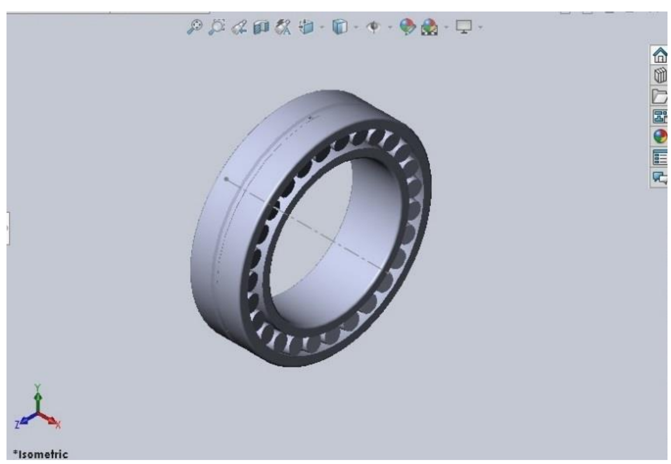


Fig 4.11 Assembly of 23024CC bearing

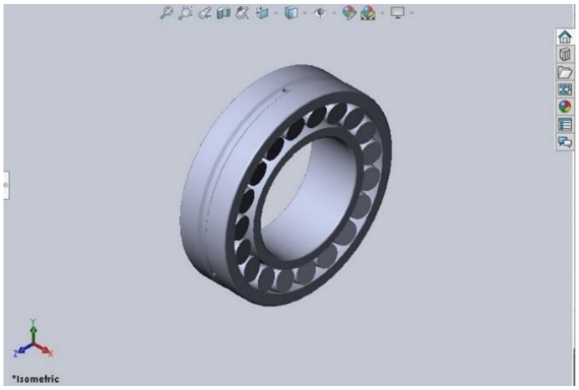


Fig 4.12 Assembly of 22224CC bearing

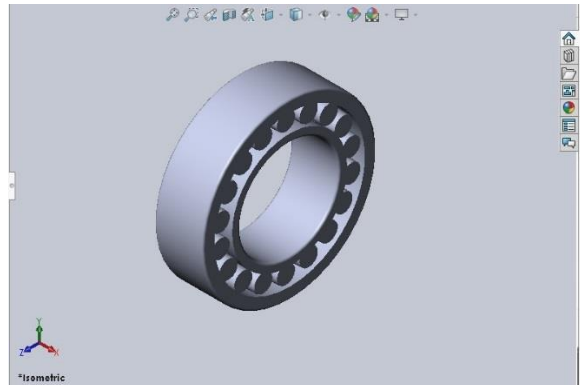


Fig 4.13Assembly of 2226CC bearing

4.3 INTRODUCTION TO ANSYS:

Ansys develops and markets finite element analysis software used to simulate engineering problems. The software creates simulated computer models of structures, electronics or machine components to simulate strength, toughness, elasticity, temperature distribution, electro-magnetism, fluid flow, and other attributes. Ansys is used to determine how a product will function with different specifications, without building test products or conducting crash tests.

Most Ansys simulations are performed using the Ansys workbench software, which is one of the company’s main products typically. Ansys user’s breakdown larger structures into small components that are each modeled and tested individually. A user may start by defining the dimensions of an object, and then adding weight, pressure, temperature and other physical properties. Finally the Ansys software simulates and analyses movement, fatigue, fractures, fluid flow, temperature distribution, electromagnetic efficiency and other effects over time.

4.3.1 FACITIES IN ANSYS:

- Structural Linear
- Structural Non Linear
- Structural Contact or Common Boundaries

- Structural Dynamic
- Structural Buckling
- Thermal Analysis
- CFD Analysis
- Electromagnetic – Low Frequency
- Electromagnetic – High Frequency
- Field and Coupled –Field Analysis
- Solvers –Iterative, Sparse, Frontal, Explicit
- Pre -Processing
- Post – Processing
- General Features

4.3.2 INTRODUTION TO ANSYS WORKBENCH:

ANSYS Workbench combines access to ANSYS applications with utilities that manage the product workflow.

4.3.3 Applications that can be accessed from Workbench include:

ANSYS Design Modeler (for geometry creation); ANSYS Meshing (for mesh generation); ANSYS Polyflow (for setting up and solving computational fluid dynamics (CFD) simulations, where viscous and viscoelastic flows play an important role); and ANSYS CFD-Post (for postprocessing the results). In Workbench, a project is composed of a group of systems. The project is driven by a schematic workflow that manages the connections between the systems. From the schematic, you can interact with workspaces that are native to Workbench, such as **Design Exploration** (parameters and design points), and you can launch applications that are data-integrated with Workbench (such as Polyflow). Data-integrated applications have separate interfaces, but their data is part of the Workbench project and is automatically saved and shared with other applications as needed. This makes the process of creating and running a CFD simulation more streamlined and efficient.

Workbench allows you to construct projects composed of multiple dependent systems that can be updated sequentially based on a workflow defined by the project schematic. For instance, you can construct a project using two connected Polyflow-based systems where the two systems share the same geometry and mesh; and the second system uses data from the first system as its initial solution data. When you have two systems connected in this way, you can modify the shared geometry once and then update the results for both systems with a single mouse click without having to open the Meshing application or Polyflow. Some examples of when this is useful include: performing a non-isothermal flow calculation starting from the solution obtained from an isothermal one; performing a transient calculation starting from the solution obtained from a steady-state analysis; and performing a blow molding simulation using the parison obtained from an extrusion calculation.

Additionally, Workbench allows you to copy systems in order to efficiently perform and compare multiple similar analyses. Workbench also provides parametric modeling capabilities in conjunction with optimization techniques, which can allow you to investigate the effects of input parameters on selected output parameters; however, it is recommended that you use Polyflow's internal parameterization and optimization capabilities if possible, in order to minimize the computational expense.

4.3.4 DIFFERENCE BETWEEN MECHANICAL APDL AND WORKBENCH:

1.Graphical User Interface:

Workbench(WB) has a better graphical user interface compared to APDL.
Workbench(WB) is also more intuitive as compared to APDL. The user can just go through the different options and understand what the different icons mean easily.

2. Use of memory:

Since WB has higher graphical user interface compared to APDL it takes up more computer memory for the same operation performed. Thus we find that WB is more resource intensive.

3.Ease of use:

The interface of workbench is intuitive and much easier to use. It has many of the settings already built in and there is no reason to provide these. For example many of the generally used materials are readily available and can be just selected but in APDL we have to give in the properties of the most common materials even. The other use is that we do not have to select the element type according to the problem type. WB automatically selects them according to the type of analysis, but in APDL we have to know the best element that suits our purpose and select it accordingly.

4.Amount of control:

APDL gives a very high degree of control to the user regarding the problem. The user has to specify each and every step of the problem manually thus the user is aware of everything that APDL does. But in WB as seen in point 3 many of the parameters are chosen by the software in the background. So we are not able to control every aspect of the problem.

5.Undo option:

APDL does not have an undo function as workbench. If an error is committed in giving boundary conditions or forces we have to manually delete it using the delete tool in the menu. The delete tool is very difficult to use. But in workbench we can just press Ctrl+Z or right click on the boundary condition or force and just select delete from the drop down menu.

6.Interacts easily with other CAD packages:

APDL allows importing of only step or iges geometry files for analysis. But workbench supports files of catia, solidworks , etc also. Thus we do not have to waste our time converting a model into iges or step. This is particularly helpful when we have to bring about corrections in the model.

7.Possibility of error:

Workbench has higher probability for error since it gives results even if we do not know what we are doing. It solves the problem even if the input values are not correct. For example even if complete boundary conditions are not given it gives weak springs in certain cases to solve the problem.

8.Complex problems:

Although WB is easier to use APDL helps to solve complex problems. Problems where the material has a very particular behaviour. Example it is very easy to specify the properties of different layers of laminates using APDL but in workbench we have to use an additional module called ACP for the same. Another example may be defining Tertiary creep equations for materials. APDL is the only one that can do the user defined non-linear tables, or equations for the material definition within the simulation. WB is useful for general problems that are defined already in WB as components but to simulate out of the box problems APDL is the go to person.

9.Materials Repository:

WB has a dedicated repository of general materials and materials that are used for specific applications. But APDL does not have any such thing. In APDL we have to define each and every material we use in a particular simulation even if it is structural steel. In this case WB is a real time saver. The point 8 also highlights the advantages APDL presents material wise.

10.Coupled analysis:

APDL does not allow coupled analysis. Thus after a particular analysis we have to manually take down the values and input these for further analysis. But WB allows coupling of different analysis like thermal and structural etc. This helps us to seamlessly share the results from one analysis to the other as input without effort. It helps to simulate real world problems without manual effort from user part saving a lot of time.

4.4 ANSYS STEP BY STEP PROCEDURE:

Step 1: Initially save the assembly of the bearing in "IGES" file and then Open a new project in Ansys workbench and double click on static structural. Then the static structural window appears on the screen. Now right click on the "Engineering Data" and click on Edit.

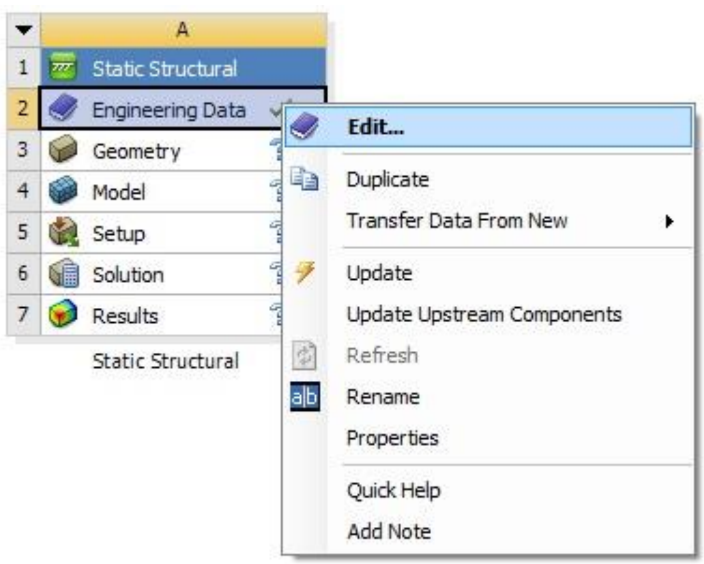
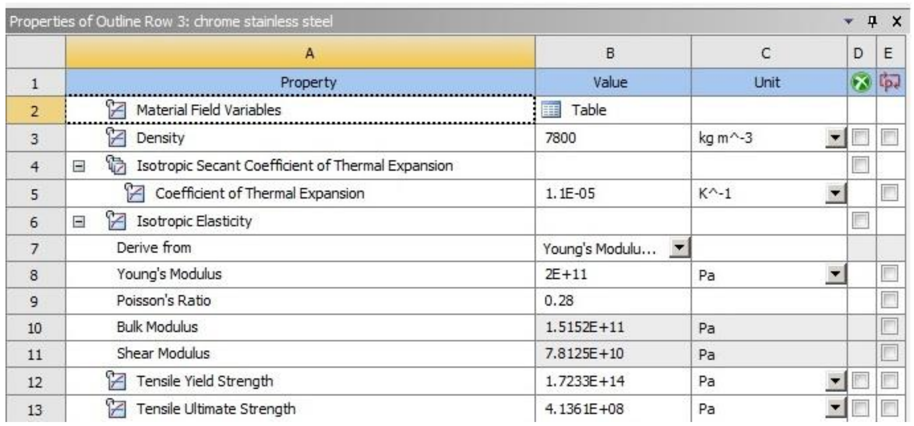


Fig 4.14 Static Structural Module

Step 2: Now type the name of the material as “chrome Stainless Steel” and apply the properties to it.



Properties of Outline Row 3: chrome stainless steel

	A	B	C	D	E
1	Property	Value	Unit		
2	Material Field Variables	Table			
3	Density	7800	kg m^-3		
4	Isotropic Secant Coefficient of Thermal Expansion				
5	Coefficient of Thermal Expansion	1.1E-05	K^-1		
6	Isotropic Elasticity				
7	Derive from	Young's Modulu...			
8	Young's Modulus	2E+11	Pa		
9	Poisson's Ratio	0.28			
10	Bulk Modulus	1.5152E+11	Pa		
11	Shear Modulus	7.8125E+10	Pa		
12	Tensile Yield Strength	1.7233E+14	Pa		
13	Tensile Ultimate Strength	4.1361E+08	Pa		

Fig 4.15 Selection of Material

Step 3: Go back to the static structural window and right click on the geometry, import geometry and browse the “IGES” file of the 23024cc bearing.

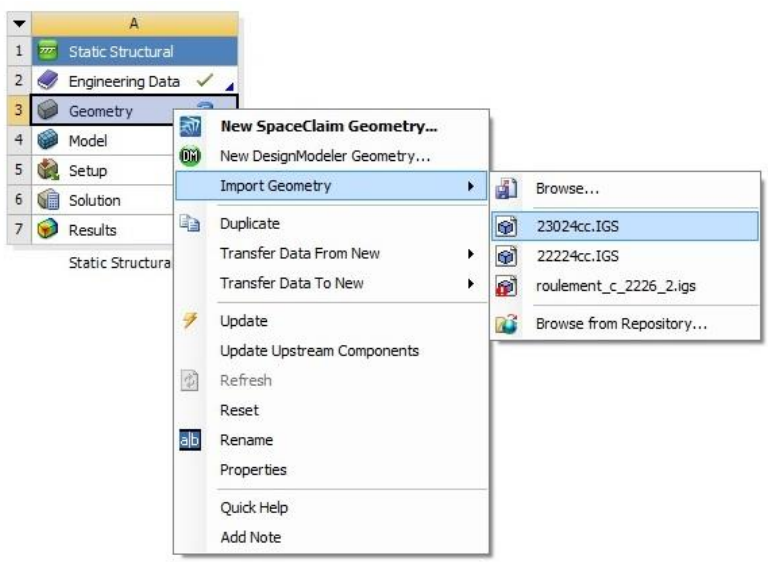


Fig 4.16 Importing Geometry

Step 4: Again go back to the static structural window and double click on the "Model", a new window along with the solid model of the bearing appears on the screen.

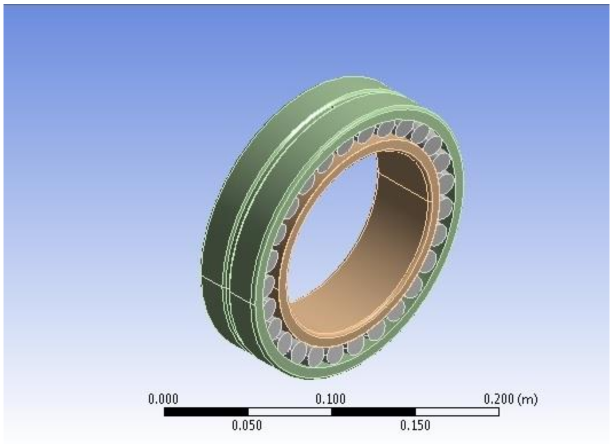


Fig 4.17 23024CC Bearing

Step 5: Go to geometry and select all the parts of the bearing and change the material to "Chrome Stainless Steel".

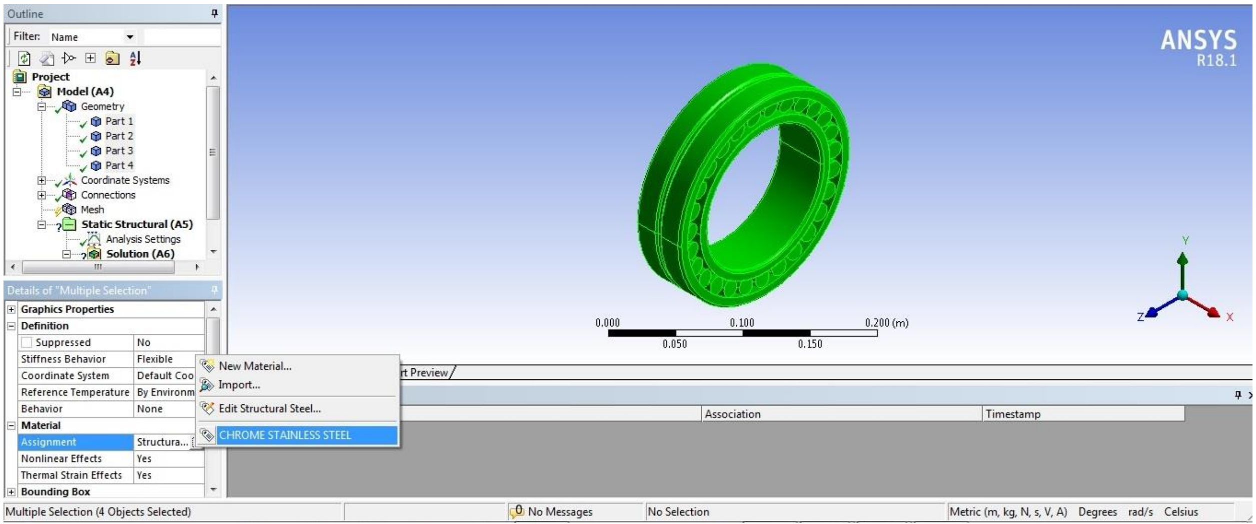
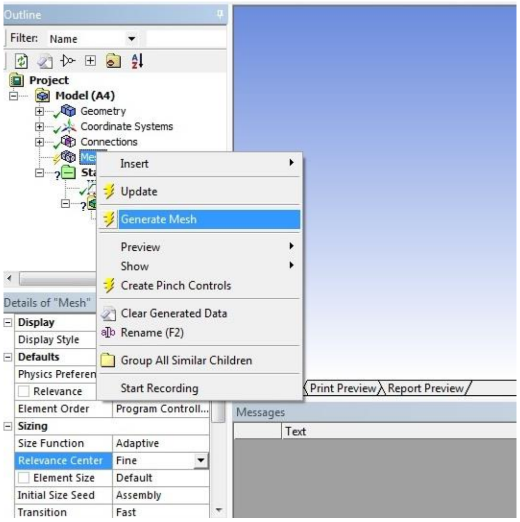


Fig 4.18 Appling the material to the bearing

Step 6: Now click on “Mesh” and change the sizing from “Coarse” to “Fine”. Right click on the “mesh” and click on “generate mesh”.



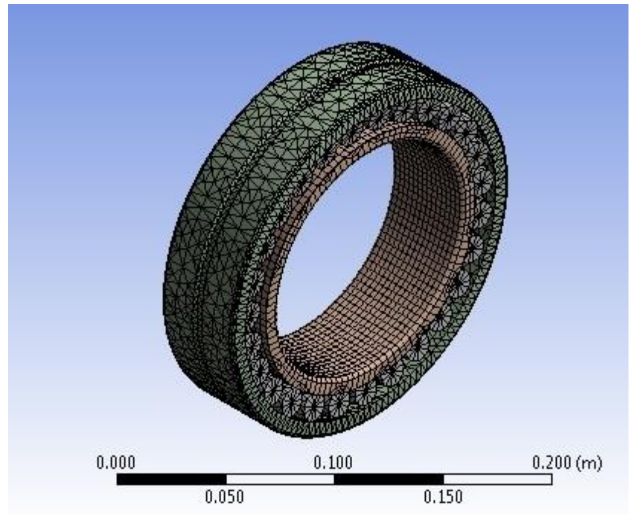


Fig 4.19 Generating Mesh

Fig 4.20 After Meshing

Step 7: Inorder to restrict the motion of the “Inner race”, Go to “static structural” → insert → fixed support.

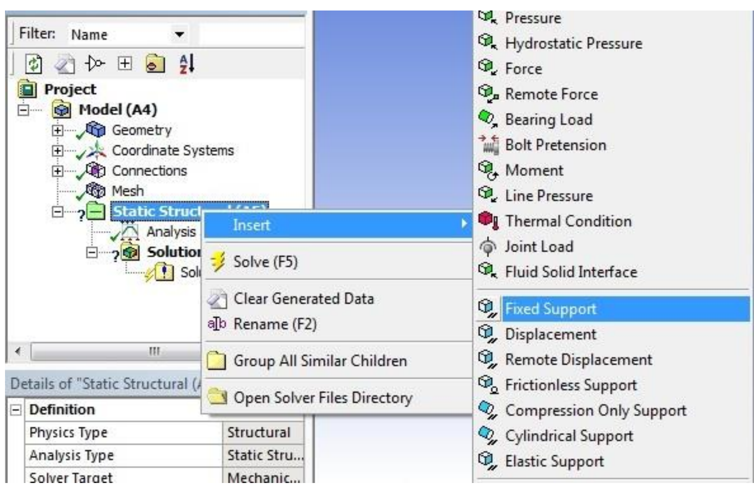


Fig 4.21 Applying the constraints

Step 8: To apply the radial force on the outer race of the bearing again go to static structural → insert → Force. Select force definition as 'Components' and enter the tri-axial force component values and select the faces on which the force is acting.

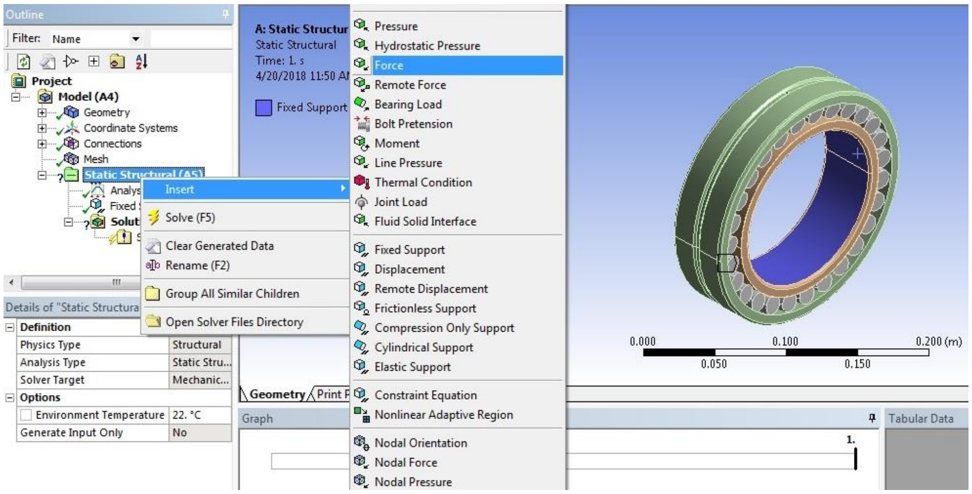


Fig 4.22 Applying the Force

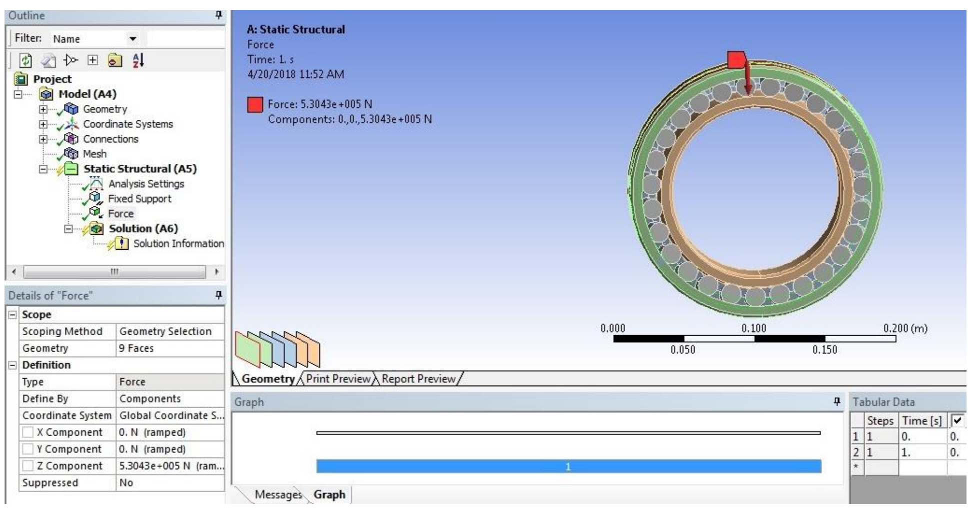


Fig 4.23 Bearing with Static Load

Step 9: Click on Solutions → insert → Stress→ Equivalent Von-mises .

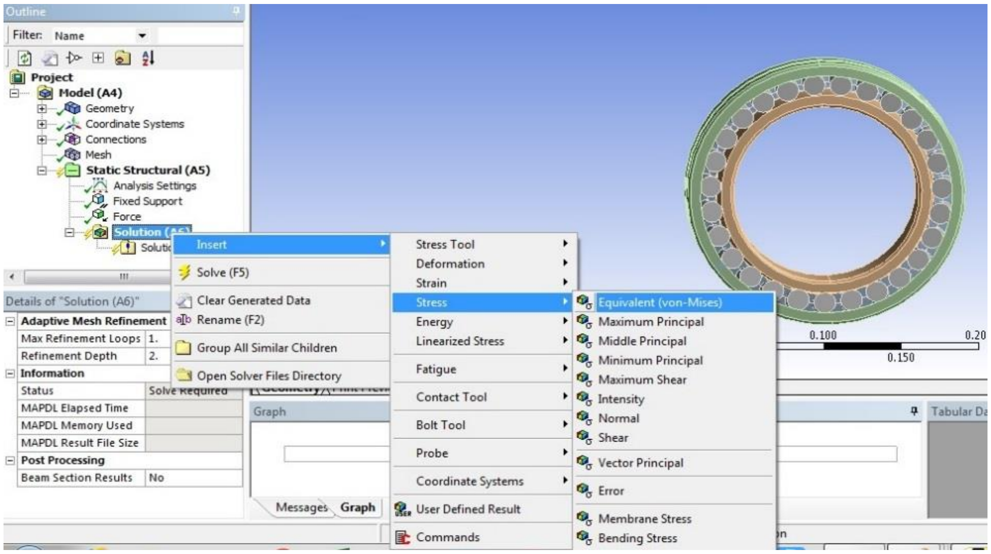


Fig 4.24 Selection of output parameters

Step 10: By opting various parameters in Solutions, Graphical representation of the values are obtained.

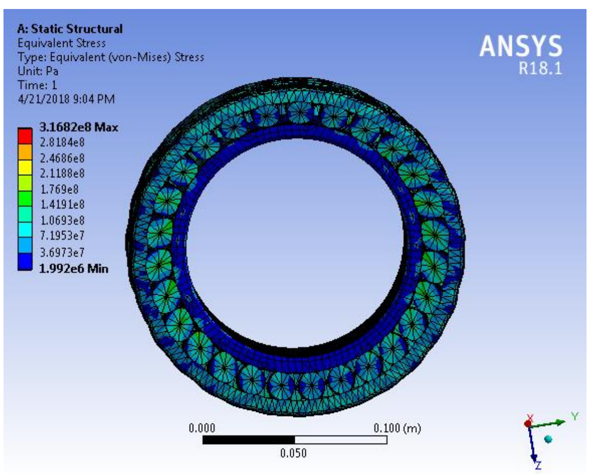


Fig 4.25 Equivalent von-Mises Stress on 23024CC bearing

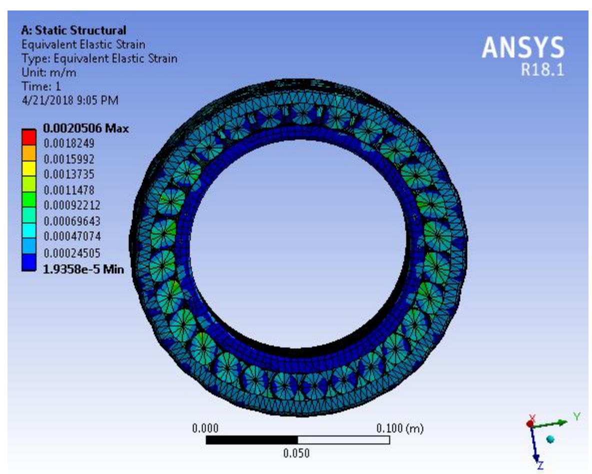


Fig 4.26 Equivalent Elastic Strain on 23024cc bearing

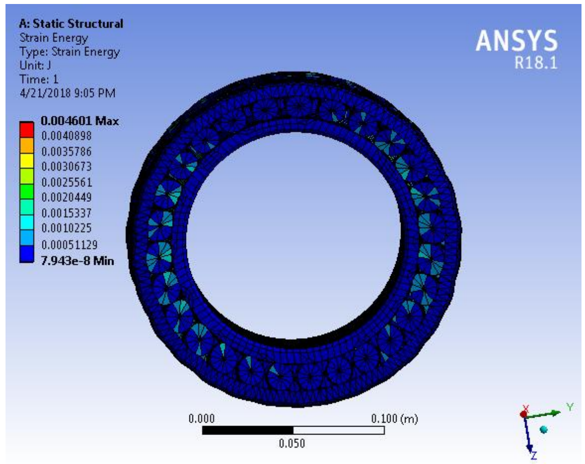


Fig 4.27 Strain Energy on 23024CC bearing

Table 4.1: Results of output parameters of 23024CC bearing

Results			
	Equivalent Stress	Equivalent Elastic Strain	Strain Energy
Minimum	1.9920e+006 Pa	1.9358e-005 m/m	7.943e-008 J
Maximum	3.1682e+008 Pa	2.0506e-003 m/m	4.601e-003 J
Minimum Occurs On	Part 4	Part 2	Part 3
Maximum Occurs On	Part 4		

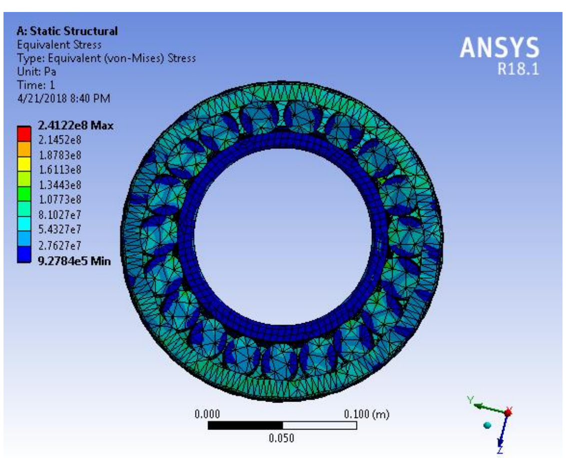


Fig 4.28 Equivalent von-Mises Stress on 22224CC bearing

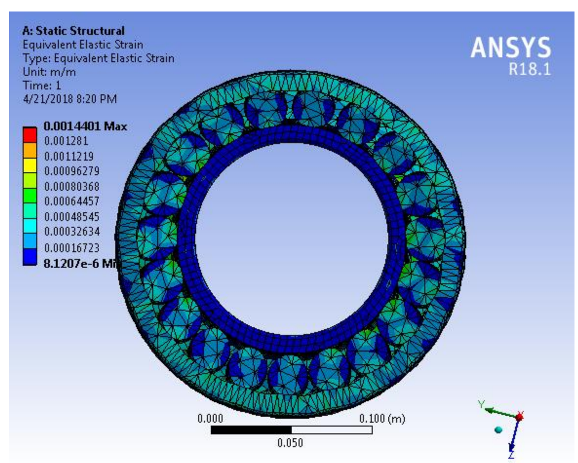


Fig 4.29 Equivalent Elastic Strain on 22224ccc bearing

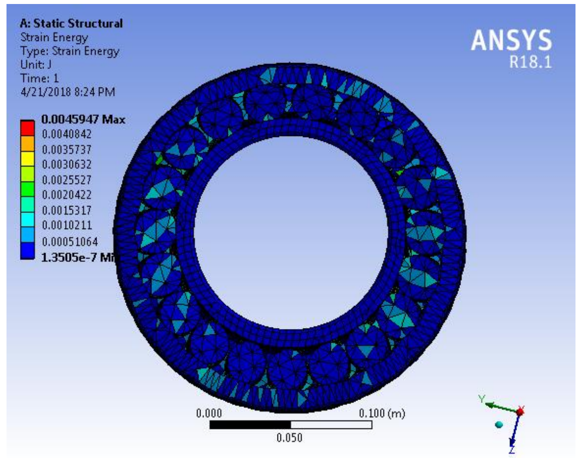
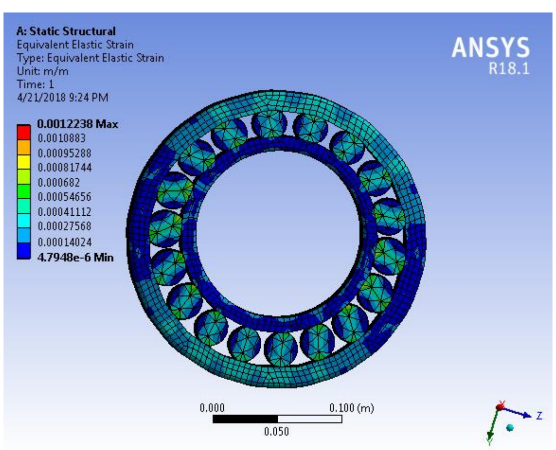


Fig 4.30 Strain Energy on 22224CC bearing

Table 4.2 : Results of output parameters of 22224CC bearing

Results			
	Equivalent Elastic Strain	Equivalent Stress	Strain Energy
Minimum	8.1207e-006 m/m	9.2784e+005 Pa	1.3505e-007 J
Maximum	1.4401e-003 m/m	2.4122e+008 Pa	4.5947e-003 J
Minimum Occurs On	Part 2	Part 4	Part 1
Maximum Occurs On	Part 4		



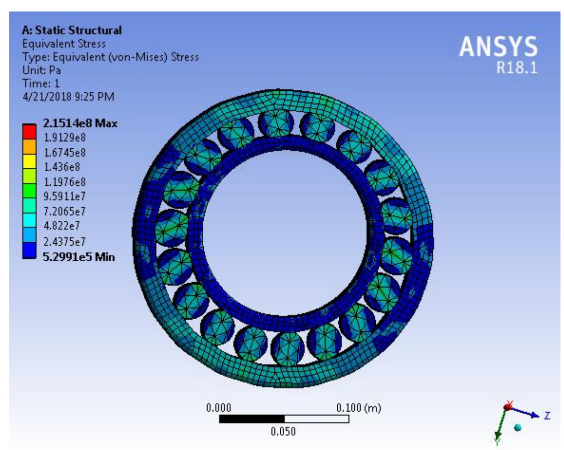


Fig 4.31 Equivalent stress on 2226CC bearing

Fig 4.32 Equivalent Elastic Strain on 2226CC bearing

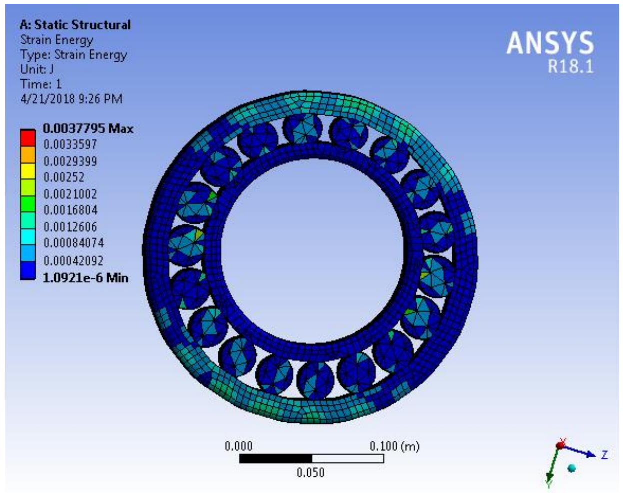


Fig 4.33 Strain Energy on 2226CC bearing

Table 4.2 : Results of output parameters of 2226CC bearing

Results			
	Equivalent Elastic Strain	Equivalent Stress	Strain Energy
Minimum	4.7948e-006 m/m	5.2991e+005 Pa	1.0921e-006 J
Maximum	1.2238e-003 m/m	2.1514e+008 Pa	3.7795e-003 J
Minimum Occurs On	Outer Race		
Maximum Occurs On	Inner Race		

CHAPTER 5

RESULTS AND DISCUSSION

The variation in the theoretical and practical values of static load and dynamic load of three different bearings 23024CC, 22224CC and 2226CC is depicted in the following table given below:

Table 5.1: Static and Dyanamic loads of bearings in theoretical and practical conditions :

`	23024 CC		22224 CC		2226CC	
Theoretical	Static Load(KN)	Dynamic Load (KN)	Static Load (KN)	Dynamic Load (KN)	Static Load (KN)	Dynamic Load (KN)
	500	366	765	652	930	735
Practical	530.43	468.84	530.43	468.84	530.43	468.84

The theoretical values of the Static load of these three bearings are calculated by Stribeck's equation.So by using Stribeck's equation, the max permissible static load capacities of the three bearings 23024CC,22224CC and 2226CC are determined as:

For 23024CC = 500KN

For 22224CC = 765KN

For 2226CC = 930KN

Using free body diagram and force balancing equations, the actual static load exerted on the bearings is found to be 530.43 KN.

The maximum permissible values of the dynamic load capacity of bearings are:

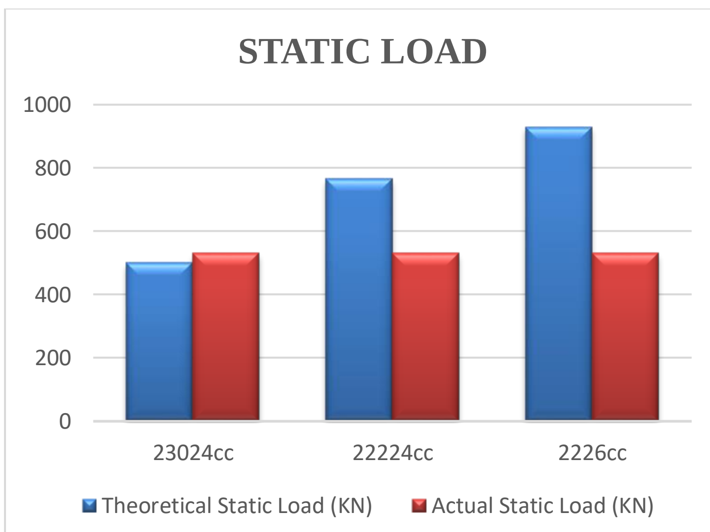
For 23024CC = 366 KN

For 22224CC = 652 KN

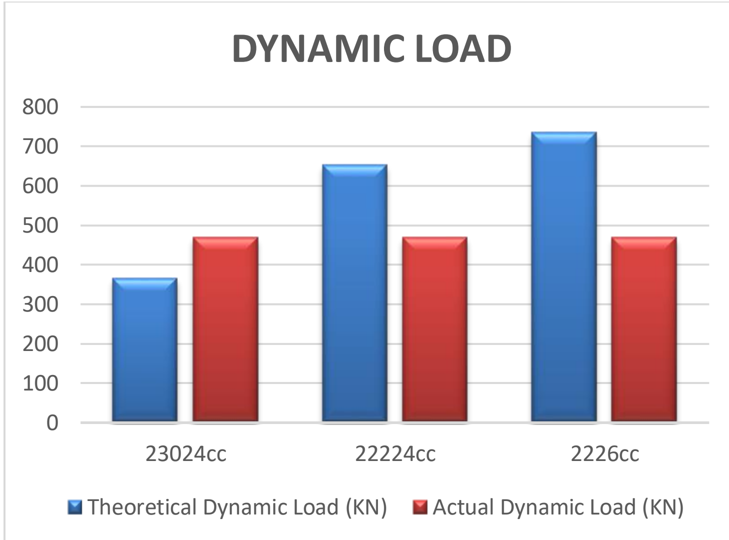
For 2226CC = 735 KN

But the actual dynamical load exerted on the two bearings is found to be 468.84 KN. Here, the mathematically calculated static load is exceeding the max permissible static and dynamic load capacity of 23024CC bearing. So due to this reason the stock bearing (23024CC) is unable to fulfill the actual requirement in the blender reclaimer.

As these calculated loads are with in the permissible limits of 22224CC bearing, 23024CC bearing is replaced by 22224CC bearing. But when we consider the 2226CC bearing which is the advanced model in spherical roller bearing called as 'Toroidal Roller Bearing' has the highest load carrying capacity when compared to 22224CC bearing.



Graph 5.1: Static Load

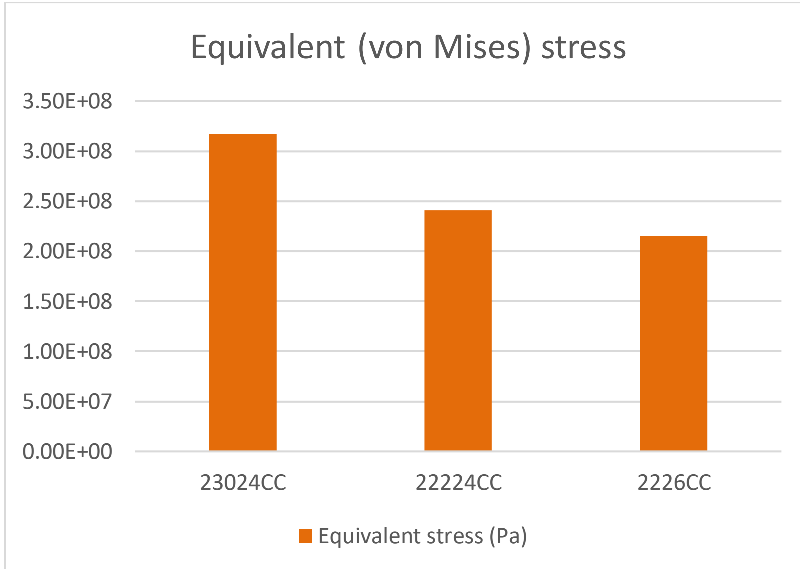


Graph 5.2 Dynamic Load

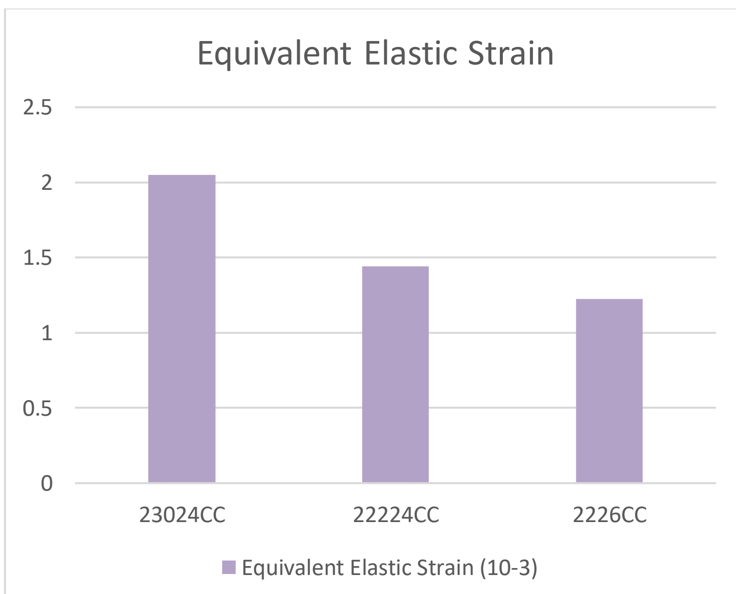
By using Ansys Workbench software, static structural load analysis is done on above mentioned bearings. We considered three parameters for the analysis of the bearings and by comparing the graphical representation of these parameters we came to a conclusion that the 2226CC can withstand the actual working conditions than the 22224CC and 23024CC. Hence we finally attained the same result as that of the mathematical approach through simulation.

Table 5.2: Different output parameters on bearings

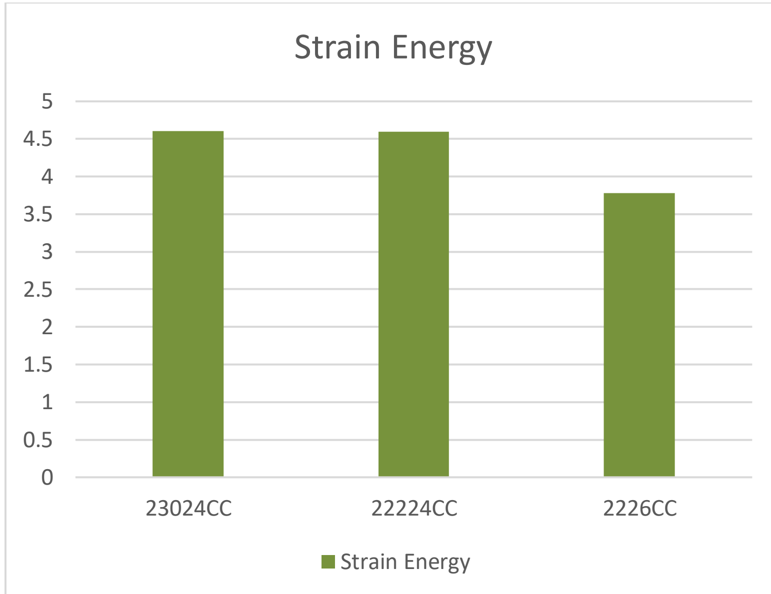
Parameters	23024CC	22224CC	2226CC
Equivalent Stress (Pa)	3.1682 x 10^{8}	2.4122 x 10^{8}	2.151 x 10^{8}
Equivalent Elastic Strain	2.0506 x 10^{-3}	1.4401 x 10^{-3}	1.2238 x 10^{-3}
Strain Energy (J)	4.601 x 10^{-3}	4.594 x 10^{-3}	3.7795 x 10^{-3}



Graph 5.3: Equivalent von-Mises Stress



Graph 5.4: Equivalent Elastic Strain



Graph 5.5: Strain Energy

CHAPTER – 6

CONCLUSION AND FUTURE SCOPE

CONCLUSION :

From the results and analysis, it is evident that though the 23024CC bearing is opted for the job, it couldn’t withstand the actual working loads and working conditions. The 22224CC bearing falls within the working load conditions henceforth its use can be continued. In any case of failure of these two bearings, a suitable and advanced choice like 2226CC can be considered. Proper lubrication, frequent maintenance and suitable covering and protection can increase the life of the bearing significantly.

FUTURE SCOPE :

Static structural load analysis is done for the three different bearings 23024CC ,22224CC and 2226CC and the reason for the replacement of the stock bearing with other bearing which is in permissible limits of actual static load is attained. The dynamic structural load analysis for the bearings to analyze the different parameters acting on the bearings and also we can apply different materials to the bearings for this dynamic structural analysis.

BIBILOGRAPHY

1) Aditya et al.2nd International Conference on Innovations in Automation and Mechatronics Engineering, ICIAME 2014, Published by Elsevier Ltd .

2) Boness, R.J. “Cage and roller slip in high speed roller bearings”, Journal of Mechanical Science, Vol. 2, pp.181-188, 1969.

3) Behnam Galamchi, Dynamic Analysis Model of Spherical Roller Bearing with Defects, Lappeenranta University of Technology, 2014.

4) Eschmann et al. “Ball and Roller Bearings: Theory, Design and Applications” published by Wiley, 1958

5) Jafar Takabi, M. M.Khonsari ., “Experimental testing and thermal analysis of ball bearings . Tribology International, published by Elsevier Ltd, 2012

6) . Poplawski, J.V. “Slip and cage forces in a high speed roller bearing”, Journal of Lubrication Technology, Vol. 94, pp. 143-152, 1972.

7) Riddle, Jhonny, “Ball bearing maintenance” published by University of Oklahoma Pr, Norman 1955.

8. Tedric A. Harris, Rolling Bearing Analysis, fourth edition, Published by John Wiley and Sons,1966.

9). V.B.Bhandari , Design of Machine Elements, third edition, published by Tata McGraw-Hill Education Pvt Ltd.,2012.

10) R.K. Upadhyay, L.A. Kumaraswamidhas and Md.Sikandar Azam, An Investigation and Analysis on Failure for Bearings of Casting Shakeout Used In Foundry Industries, vol – 3, Issue – 1

11) http://www.skf.com/in/products/bearings-units-housings/roller-bearings/spherical-roller-bearings/index.html

12) https://en.wikipedia.org/wiki/Spherical_roller_bearing