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Development of A Finite Element Model for Spherical Roller Bearings Under Static Load

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***Abstract* – Spherical Roller Bearings (SRBs) are now being used extensively in industries where there is a need for carrying high loads from rotating machinery. SRB's capability to support the same load for the same shaft diameter by consuming less radial space, as compared to other bearings is one of the most useful characteristic of them. Their ability to carry loads in different axes is one other notable trait. These advantages make it necessary to study their behaviour. This study dealt with the response of SRB (with no lubrication) under a pure static load. In this model, non-linear Hertzian contact theory was used to analyse the contacts between rollers and raceways. The results from the methodology formulated were reproduced for generation of force-displacement curve using MATLAB. The curve generated shows the non-linear characteristics of the SRB under loading. Crowning of the rollers, material non linearity, etc leads to a non-linear behaviour of these bearings when loaded. The MATLAB output was used to model the rollers using spring elements in a Finite Element (FE) model. The developed FE model was validated by comparison with a reference paper and an existing SRB model, for which results were already available. The new model was found to be less complex than the previous model. The new model is also much more efficient for computational analysis purposes compared to existing bearing model.**

***Keywords* – Spherical Roller Bearings, Downsizing, Force Displacement Curve, Hertzian Contact, Bearing Model.**

I. Introduction

Roller bearings are commonly used for high load carrying applications due to their much higher stiffness compared to other types of bearings. Moreover, they have much higher endurance strength as compared to other type of bearings of same size [1]. Roller bearings are much needed when it comes to reducing friction which is mandatory in rotating machineries. These bearings can have two types of rolling elements, balls or rollers. Roller bearings are of many types (shown in Fig. 1).Spherical roller bearings have an outer raceway which is a portion of a sphere, due to which they are internally self-aligning. Due to good osculation between the rollers and raceways, these bearings have high load carrying capacity.

These bearings have barrel shaped symmetrical rollers. These rollers are held in the raceways by a cage or retainer or separator. Due to sliding between the roller and raceway, these bearings are not suitable for high speed operations. SRBs are most commonly used in heavy duty applications such as rolling mills, power transmissions, marine applications etc. They can carry combination of radial as well as axial load, but cannot support moment loadings. Radial roller bearings are designed for high radial load carrying capacity. Needle roller bearings differ from cylindrical roller bearings in that they require less radial space. Spherical roller thrust bearings have high load carrying capacity but can operate at a low speed only due to sliding friction between rollers and guide flange. Cylindrical and tapered roller thrust bearings can operate only at slow speed due to sliding friction. Needle thrust bearings can operate only at light load and low speed. Tapered roller bearings can carry a combination of radial as well as axial loads or just axial load only.

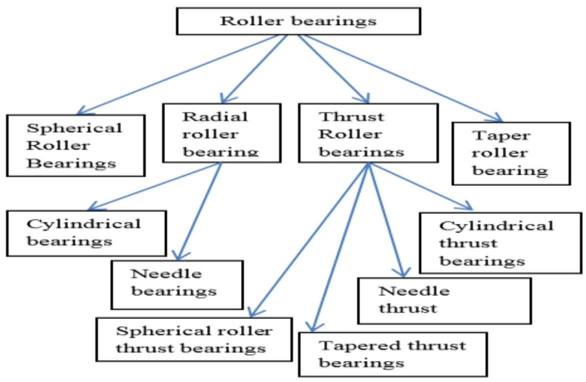

Figure 1: Classification of Roller Bearings

This paper deals with the double row spherical roller bearings and their behaviour when subjected to static radial load. The reason behind doing such a study is to get more knowledge of SRBs when subjected to this type of load, and once having that knowledge, use it to model the bearing which comes in various simulations and try to improve the efficiency of computational analysis at a lesser cost

II. Static Model Of The Spherical Roller Bearing

Spherical Roller Bearing consists of a number of parts like inner and outer races, rollers, cage, etc. Only the inner and outer races and the rollers are modelled here to reduce the complexity of the model as well as to improve the computational efficiency. This bearing model is expected to exactly simulate the behaviour of a bucket test set up under static load. The SRB modelled here is valid for static radial load conditions only. Some assumptions made during modelling of the bearing are:

i. The bearing is subjected to a static radial force only.
ii. The bearing is not lubricated.
iii. Cage movement is not considered.
iv. The rollers are equally spaced.
v. Isothermal conditions prevail during the operating period.

Interaction of contacting solids:

Since the load acting on the bearing is taken up by the rollers the rollers make contact with the races above and below it. Bothe these solids have different radii of curvature in different planes. In the no-load condition, the contact between these solids is of point type. But once a load acts upon the bearing, the point contact gets disturbed and the contact junction becomes an elliptical one. The various curvatures for the solids are assumed as positive for convex surfaces and negative for concave surfaces. Contact between two solids can be defined by two parameters, which are: Curvature sum (*R)* and curvature difference (R_d) and given by [2, 3],

$$\frac{1}{R} = \frac{1}{R_x} + \frac{1}{R_y} \tag{1}$$

$$R_d = R\left(\frac{1}{R_x} - \frac{1}{R_y}\right) \tag{2}$$

The curvature sums in each of the two planes can be written with the help of the radius of curvatures associated with the solids as:

$$\frac{1}{R_x} = \frac{1}{r_{Ax}} + \frac{1}{r_{Bx}} \tag{3}$$

$$\frac{1}{R_y} = \frac{1}{r_{Ay}} + \frac{1}{r_{By}} \tag{4}$$

R_x and R_y represent the effective radii of curvature in x and y planes. If "*a"* is the semi-major axis and" *b"* is the semi minor-axis then a number called ellipticity parameter can be defined as [4]:

$$k = \frac{a}{b} \tag{5}$$

Elliptic parameter can be written in terms of curvature difference and elliptic integrals of the first (ξ) and second (ζ) kind as:

$$k = \left[\frac{2\xi - \zeta(1+R_d)}{\zeta(1-R_d)}\right]^{1/2} \tag{6}$$

ζ and ξ are calculated as follows:

$$\zeta = \int_0^{\frac{\pi}{2}} \left[1 - \left(1 - \frac{1}{k^2}\right)\sin^2\varphi\right]^{-1/2} d\varphi \tag{7}$$

$$\xi = \int_0^{\frac{\pi}{2}} \left[1 - \left(1 - \frac{1}{k^2}\right)\sin^2\varphi\right]^{1/2} d\varphi \tag{8}$$

Since these integrals can't be evaluated using any of known integration techniques, curve fitting techniques were used to approximate the values of these two integrals and the ellipticity parameter. These relations can be given as [2]:

$$k = 1.0339 * \left(\frac{R_y}{R_x}\right)^{0.6360} \tag{9}$$

$$\zeta = 1.5277 + 0.6023 * \ln\left(\frac{R_x}{R_y}\right) \tag{10}$$

$$\xi = 1.003 + \left(0.5968 * \left(\frac{R_x}{R_y}\right)\right) \tag{11}$$

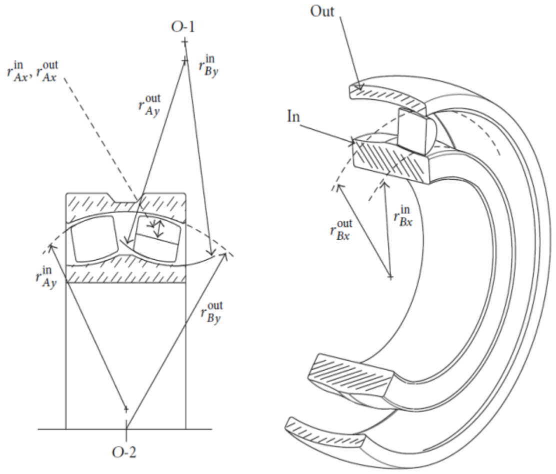

Figure 2: Radii of curvature between roller, inner raceand outer race

Geometry of SRB

The various dimensions, curvatures associated with the SRB are extracted from the cad geometry of the model. These are shown in Figure 2. The most important dimensions for a Spherical Roller Bearing are shown in Figure 3. All the quantities required for the formulation are derived from these dimensions. Osculation is the parameter which shows the conformity of the race and rollers.

Krzeminski-Freda and Warda [5] focused in their study on determining a proper ratio of osculation coefficients for both races to obtain self-stabilization of the barrel shaped roller and to minimize friction losses. Good osculation number means less contact pressure, but more fatigue life and less frictional heating. The radii of curvatures for roller-to-inner race contact can be calculated by the following set of equations("12") as:

$$r_{Ax}^{in} = \frac{d_r}{2}$$

$$r_{Ay}^{in} = r_r \tag{12}$$

$$r_{Bx}^{in} = \frac{d_e - d_r\cos\phi_0 - \left(c_d/2\right)\cos\phi_0}{2\cos\phi_0}$$

$$r_{By}^{in} = -r_{in}$$

Similarly the radii of curvature for roller–to-outer race contact can be calculated by following set of equations

(“13) as:

$$r_{Ax}^{out} = \frac{d_r}{2} \qquad (13)$$

$$r_{Ay}^{out} = r_r$$

$$r_{Bx}^{out} = -\frac{d_e + d_r \cos\phi_0 + \left(\frac{c_d}{2}\right)\cos\phi_0}{2\cos\phi_0}$$

$$r_{By}^{out} = r_r$$

Displacement in SRB

Cao and Xiao [6, 7] developed and applied a comprehensive spherical roller bearing model to give quantitative performance analyses of SRBs. They had considered point contact. Gargiulo [8] had given empirical relations for bearing radial deflections. Since deflections occur between inner raceway and roller as well as deflection between outer raceway and rollers, the deflection of an SRB is taken as the relative of these two

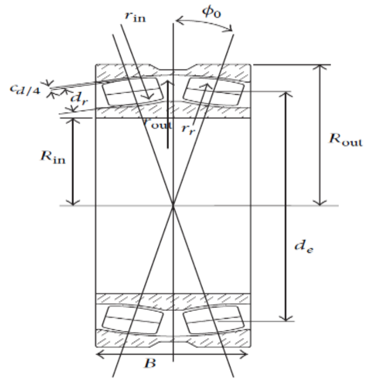

Figure 3: Dimension of Spherical Roller Bearing

R_{in}: Bore radius
R_{out}: Outer radius
r_{in}: inner raceway contour radius
d_r: Roller diameter
r_r: roller contour radius
B: bearing width
r_{out} : outer raceway sphere radius
d_e: bearing pitch diameter
c_d: diametral clearance
ϕ_0: free contact angle

interactions. Since contact is considered, when a load comes onto the bearing, we will consider the contact stiffness coefficient, which is related to the dimensions of the ellipse at the conjunction. Since the dimensions of the ellipse changes with the load, contact stiffness is expected to vary. If F denotes the radial load, K_c denotes the contact stiffness coefficient, and then radial deflection between race and roller can be approximated as (shown in Fig.4, [1]):

$$\delta_r = \left(\frac{F}{K_c}\right)^{2/3} \qquad (14)$$

The effective modulus of elasticity E′ can be defined as:

$$E' = \left[\frac{2E_a E_b}{E_b(1-\upsilon_a^2) + E_a(1-\upsilon_b^2)}\right] \qquad (15)$$

Using elliptical integrals and the material properties, the contact stiffness coefficient can be calculated as:

$$K_c = \pi k E'\left(\frac{R\xi}{4.5\zeta^3}\right) \qquad (16)$$

III. MATLAB Numerical Calculations

To verify the bearing model, the double row SRB having symmetrical rollers model was subjected to a simple static radial load. Numerical calculations were performed using MATLAB-2016a. MATLAB code was written in such a way that once the user inputs the required geometric parameters of the SRB, the code calculates all other required quantities for the formulation on its own. The Program is written for a maximum force of 100, 000 N, which can be varied. This way, the force-deflection curve for any SRB is generated readily.

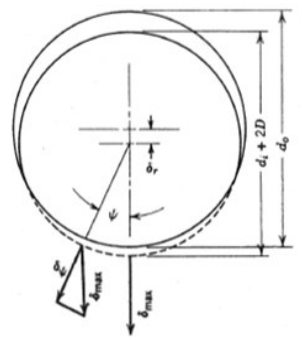

Figure 4: Radial Displacement of SRB

IV. Result and Validation

The results of the MATLAB code was compared against the results from a reference journal [9], in which the parameters of the SRB used (Table I), were exactly the same as that of those used in MATLAB code.The comparison is shown below:

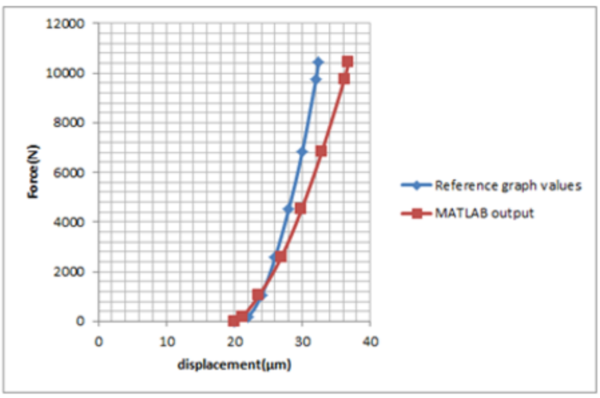

Figure 5: Comparison of Force-Displacement curve Obtained using MATLAB with Reference Curve

From the curve, it is obvious that the displacement increases as the applied radial load increases. The curve is found to be closely matching in the initial stages and is found to deviate as it progresses. Possible sources of error are:

- Since this paper considers static loading conditions and the reference paper is dealing with dynamic conditions, some error is to be expected due to the influence of various other forces in dynamic condition.
- The effect of elastic deformation of rollers and raceways are not considered in this paper.

V. FE Model and Application

The stiffness curve can be used to model the rollers, which are the load bearing elements in a bearing. Hyper mesh pre-processor was used to prepare FE model of the bearing. The stiffness curve was applied to various elements and necessary load was applied so as to verify that the resulting deflection of the rolling element is in agreement with the stiffness curve.

Finally the most suitable element was found out to be springs. The outer and inner races were modelled with general purpose linear brick element with reduced integration (C3D8R, Figure 6), with hourglass control.

Table I: Parameters Used as Input for the Code and Reference Graph

Parameter	Symbol	Value	Units
Clearance	c_d	41	µm
Pitch diameter	d_e	175	mm
Number of rows	n_z	2	_
Number of rolling elements in one row	N	16	_
Modulus of elasticity	E	206	GPa
Poisson's ratio	ν	0.3	_
Free contact angle	φ_0	7.92	degree
Roller diameter	d_r	29	mm
Inner raceway contour radius	r_i	106.61	mm
Outer raceway contour radius	r_o	106.61	mm
Roller contour radius	r_r	103.95	mm
Bearing width	B	50	mm

This element gives good accuracy in results at a considerably low cost [10].

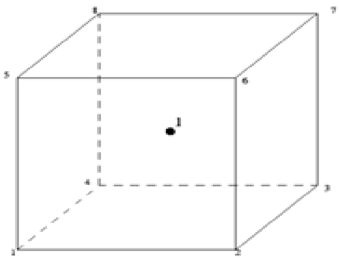

Figure 6: Integration Point Scheme in C3D8R Element

The rollers are modelled using spring elements. Since the rollers in bearing can take only compressive load, tensile force should be null. To satisfy this condition, the rollers are modelled using spring elements of Abaqus. These spring type elements have their line of action in the direction between its two nodes only [11]. Each roller is modelled using ten springs in parallel. More number of springs brings more accuracy in the results. The springs used should not undergo any distortion when some torsional load acts on the shaft and they should be able to transfer the load normally. This is taken care of by using suitable cards of Abaqus/Standard which always keep the springs in radial direction. Once the bearing is fully modelled, it can be used in any simulation application involving an SRB. This model is used in a differential bucket testing set up in which an SRB is used to transfera radial load coming onto the bearing. The test set up, loading and boundary conditions are all shown in the following figures.

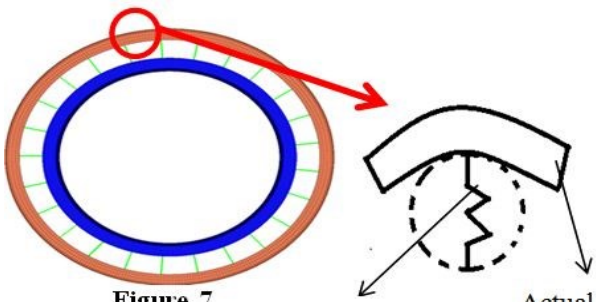

Figure 7

Figure 8

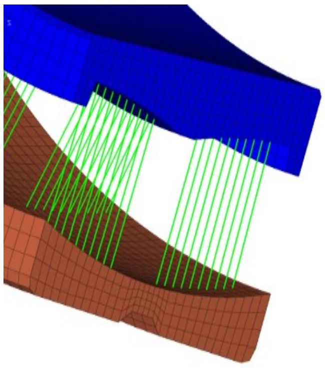

Figure 9

Figure 7: The bearing with its inner and outer races modelled using hexahedral elements and rollers modelled using spring elements. **Figure 8:** Representation of how the springs substitute the rollers. **Figure 9:** Section view of the races with the springs in between.

Fig.10 shows the test set up as well as the SRB location. The SRBs were first modelled using solid elements (C3D8R) and the analysis was conducted for which the results are available. Fig.11 shows the Von Mises stress results for the flange in both SRB models.

Fig.12 shows the Von Mises stress results for the bucket component. Fig.13 shows the bending displacement of the bucket beam element (B31) with respect to the node positions on the beam element and Fig.14 shows the correlation graph for the bending displacement.

VI. Conclusions

In this paper, Spherical Roller Bearings have been studied for their behaviour under static loading conditions. The nature of contact of rollers and raceways was considered using Hertzian theory. Also it can be deduced from the result that point contact is indeed the type of contact occurring in SRB for the load range used. A method was successfully established to formulate SRB's load-deflection curve, which was successfully converted to a MATLAB code for automation of this process. A new FE modelling technique was established to model the rollers. Table II shows the difference in computational time using the existing model and new model, from which it is clear that the new model can give accurate results at comparatively less computational cost.

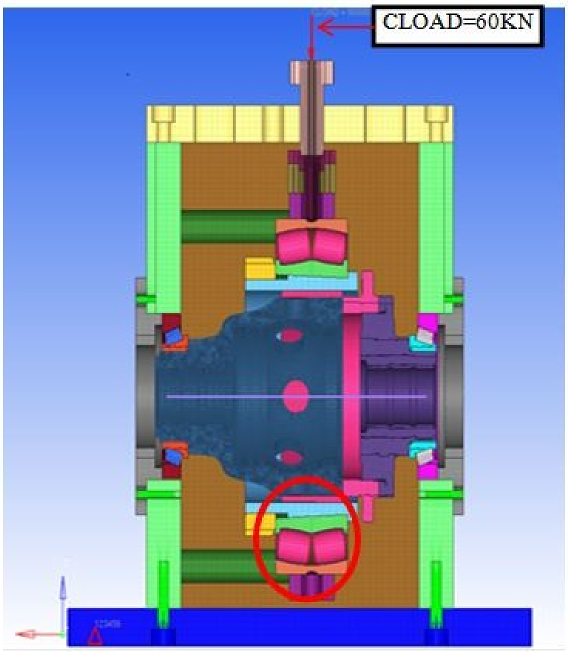

Figure 10: Schematic of the differential bucket test set-up with the loading conditions and boundary shown. The location circled red is where the SRB sits.

Table II: Computational time for Analysis using Both Models

Computational time (in sec)	With Solid model (existing)	With New model	Time saved
	27720	7992	19728

The reduction in computational time has been made possible by reducing the number of nodes in the model, as well as by reducing the contact interactions and hence the complexity.

For the flange and the bucket components in the test set up, Von-Mises stress results using both the models of SRB's was compared at the most critical regions. The correlation obtained was 96.9% and 93.2% respectively. The bending displacements for the bucket beam element in the test set up using both SRB models was plotted, whose correlation graph is shown in figure 14. From the Figures (11, 12, 13, 14), it can be understood that the new model is suitable for use in any simulation applications and that the methodology and FE modelling technique used is valid for SRBs.

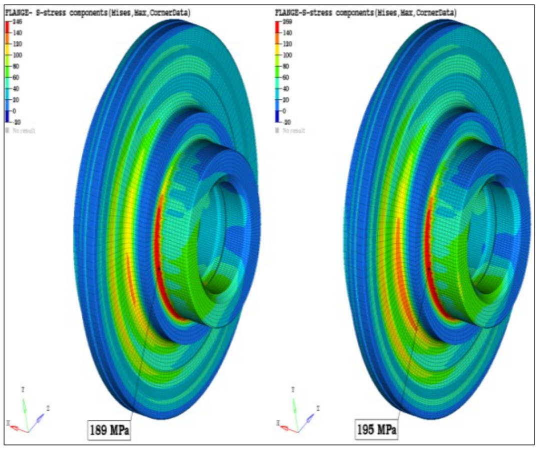

Figure 11: Von-Mises stress Results for Flange

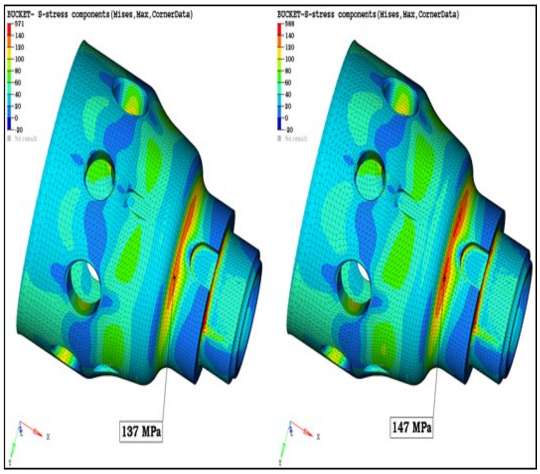

Figure 12: Von-Mises Stress results for Bucket

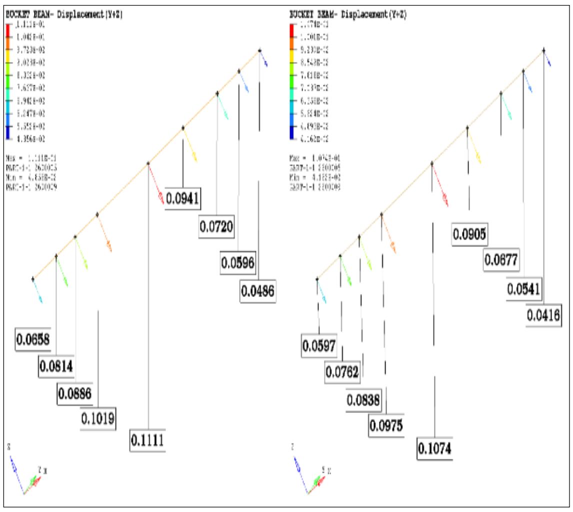

Figure 13: Bending Displacement values at Different node Positions of Bucket Beam Element

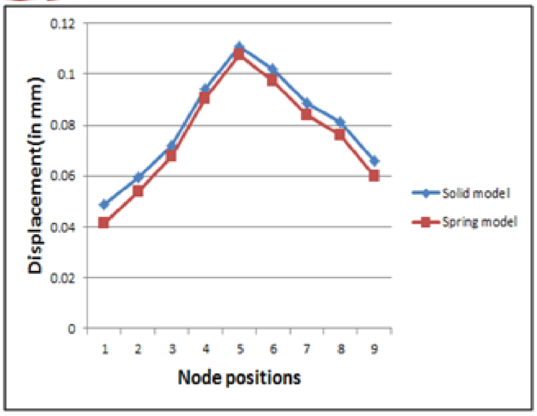

Figure 14: Correlation between Displacement Values in the Beam Element for Both Models

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