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DETERMINATION OF OPTIMUM HOLLOWNESS IN TAPERED ROLLER BEARING USING FINITE ELEMENT ANALYSIS TO INCREASE THE FATIGUE LIFE

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ABSTRACT: An attempt has been made to obtain the optimum hollowness in the rollers to increase the fatigue life by decreasing contact stress in the tapered roller bearing. In the present analysis it has been proposed that under the applied load hollow rollers deflects more than a solid roller of the same size. By making the balls hollow which are flexible enough, the stress concentration can be reduced to increase the fatigue life of ball bearings. Finite Element Analysis has been used to investigate the contact pressure, contact stress and deformation as available theoretical method is applied only to solid tapper roller bearing. Using FEA package ANSYS the analysis has been carried out considering the equivalent cylindrical roller of actual tapered roller of the test bearing. Analysis have been made for different hollowness percentage ranging from 0% to 90% of the outer diameter of the equivalent cylindrical roller of actual tapered roller in the ANSYS to find the optimum hollowness for bearing's life period point of view. It is noticed that the approximately 65 % hollowness gives the lowest contact stress which finally increases the fatigue life of the bearing.

KEYWORDS: Hollow tapered roller bearing, Finite element analysis; Hertz contact stress, Contact pressure, Fatigue life.

1 INTRODUCTION

In railway wagons, mainly, the tapered roller bearings are used for carrying heavy loads but the failure ratio of the bearing is very high in spite of the regular maintenance. Hence, present research work is mainly based upon increasing the life of the bearings. Hertz [1] first investigated the problem of two elastic bodies subjected to pure normal loading. He used the Newtonian potential function to study the distribution of the stresses in the contact body and to study the distribution of the load over the contact area. The solution was verified by experimental results. Good results for the stresses in the contact region, called Hertzian contact stress were obtained. In the roll supports of sheet, billet, tube-rolling mills and gear housing four-row tapper roller bearing is widely used. Smith and Liu [2] modified and elaborated the work of Hertz by including the tangential component and assuming the Hertzian distribution of the normal and tangential loads over the area of contact. The resulting stresses of applying the normal and tangential loads were represented by a closed form and used as verification for the numerical solution obtained in this work. Goncharov et. al. [3] developed a new bearing design of four-row tapered bearing. The test results of new bearing design showed that their life was 2 to 2.5 times the life of a normal bearing. It deferred from the existing one in the fact that the support collars on the inner rings were movable in the axial direction. Kletzli et. al. [4] analyzed a thermally-induced failure in railroad tapered roller bearings operating at relatively high speeds. It was shown that this failure was caused by an unstable thermal expansion and internal bearing load feedback. It was also found through simulation that at axle rotational speeds equivalent to a train speed of 100 mph , a combination of grease starvation and heat flux from the contact seals caused high rib temperature and subsequent unstable load growth which would lead to failure. Grigorescu and Gafitanu [5] dealt with optimization criteria to increase ball bearing service life. The analytical and experimental results presented in this paper established the correlation between bearing service life and housing stiffness. It was concluded that for each housing shape, an optimum position of the supporting points determined a maximum service life. Further, proper design of housing wall thickness and initial value of gap between the housing and outer ring assured the highest value of the bearing service life. Ferreira, Balthazar and Araujo [6] carried out a performance analysis of double self-aligning roller bearings used in railway ore transportation wagon. Two different methodologies were used to access the bearing nominal life L_{10s}. It was shown that the methodology suggested by the author for estimating the lives L_{10s} was more flexible as it allows the control of the parameters directly involved in the bearing failure phenomenon than methodology proposed by the bearing manufacturer. Gerdun et. al. [7] presented a research article dealt with two cases of failure in freight wagon cylindrical roller bearings and axles. Based on the analysis, it was accomplished that the failure can be prevented through a more frequent replacement of the inner rings of bearings. Further, the axle broke faster than it would if the cages were made of some other material rather than the Brass. Wisam [8] studied a relative fatigue life estimation of cylindrical hollow rollers in general pure rolling contact. Investigators used the finite element package ABAQUS to find the stress distribution and the resulting deformations in the bodies of the rollers. Four main different hollowness percentages were studied namely 20, 40, 60 and 80%. It was found that the hollow rollers have longer fatigue life than solid rollers when subjected to a combination of normal and tangential loading. Also, the fatigue life was improved as the hollowness percentage increased up to 60% and improvement in fatigue life were decreased when the percentage of hollowness was 80% as the bending stress started to affect the stresses in the contact zone significantly, by decreasing the shearing stress value there. Darji and Vakharia [9] dealt with the determination of optimum hollowness for hollow cylindrical rolling element bearing. Different hollowness percentage from 30 to 80% was analyzed in FEA package to find the optimum percentage hollowness for increasing fatigue life of the bearing. It was concluded in this article that 67% hollowness was the most desirable to impart sufficient roller flexibility and load carrying capacity. Tiwari, Sunil kumar and Reddy [10] described an optimum design methodology of tapered roller bearing using genetic algorithms. This investigation proposed an optimum design methodology which could be considered as a basic step towards a more advance design requirement of the tapered roller bearing.

Here, it has been proposed to determine optimum hollowness in tapered roller bearing using finite element analysis to increase the fatigue life as less work has been carried out performing to this aspect.

2 Definition Of Problem

Tapered roller bearing under consideration

Bearing No: 32212

Internal diameter of bearing: 60 mm.

Outside diameter of bearing: 110 mm

Average outer diameter of inner ring: 76.09 mm

Average inner diameter of outer ring: 96.65 mm

Average roller diameter: 12.96 mm.

Length of roller : 19.96 mm

Number of rollers: 19

Applied load on bearing (F): 30 kN

Modulus of elasticity: 2.058 x10^5 N/mm^2

Poisson's ratio: 0.3

3 Finite Element Analysis Of Tapered Roller Bearing

A Major limitation of rolling-element bearing is that they are subjected to very high alternating stresses at the rolling contacts, which leads to a limited fatigue life. In fact, pre-stressing and centrifugal forces at high speed operation significantly increase the contact stresses and further reduce the fatigue life. If stresses are low, fatigue life can be practically unlimited.

In the present analysis, it has been proposed that the contact stresses can be significantly reduced by making the rollers hollow. The contact stresses in solid rollers can be obtained using Hertz equation [10]. The same Hertz equation cannot be applied to obtain the contact stresses in the case of hollow rollers due to some assumptions taken by Hertz. In the past, Finite Element analysis was successfully applied by various researchers to study the contact interaction between the rollers

[11-12]. Thus, Finite Element analysis approach has been carried out to determine the optimum hollowness in the tapered roller bearing.

Analysis has been carried out considering equivalent cylindrical roller of tapered roller taking average diameter of the tapered roller in the finite element package ANSYS. Different cylindrical hollowness percentages ranging from 10 to 90% have been analyzed to obtain the contact stress developed in the heavily loaded roller.

Fig. 1. 3-D model of tapered roller bearing

The configuration of the bearing which is under investigation is presented in figure 1. Considering applied load of 30 kN on the bearing, the load on the heavily loaded roller has been obtained [10]. The Finite Element Analysis has been carried on the heavily stressed equivalent cylindrical roller for the actual tapered roller by applying the external normal load of heavily loaded roller as shown in figure 2.

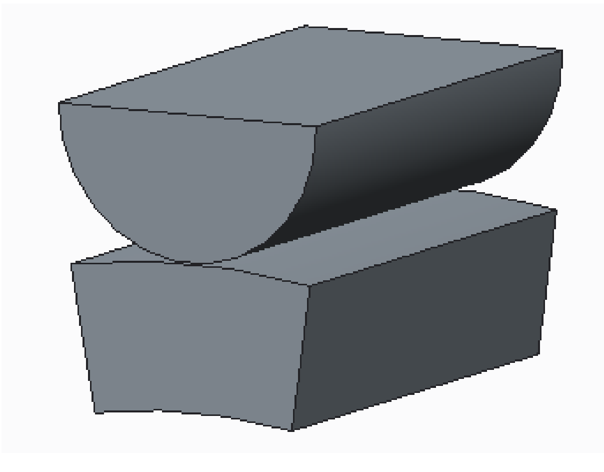

Fig. 2 Single contact of heavily stressed equivalent cylindrical roller bearing between roller and inner race

4 RESULTS AND DISCUSSION

Fig: 3 and 6 show the total deformation of the solid and hollow rollers in contact respectively. It is seen from Fig: 3 and 6 that the total deformation is more in case of hollow rollers as compared to the solid rollers. Fig: 4 and 7 represent the contact pressure generated at the contact point between two heavily loaded solid and hollow rollers respectively. It is noticed that the contact pressure remains considerably less in case of hollow tapered roller bearing as compared to the solid taper roller bearing. Finally, from Fig:5 and 8 it is established that the contact stress is significantly less in the hollow tapered roller bearing as compared to solid tapered roller bearing. FEA results for different hollowness has been presented in the tabular form as well as graphically as shown in the Table:1 and Fig: 9 respectively.

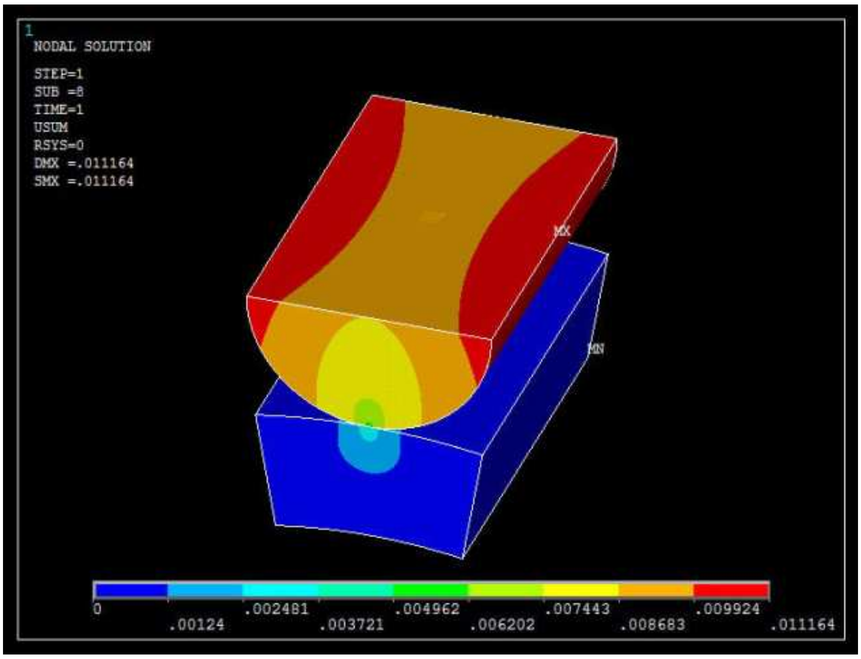

Fig. 3. Total displacement of solid rollers

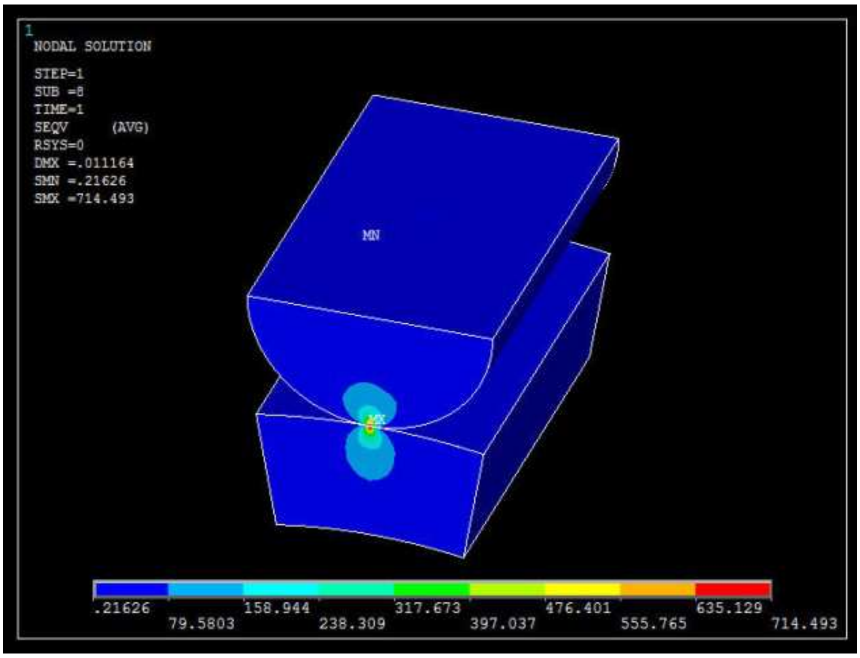

Fig. 4. Von mises stress distribution in solid rollers

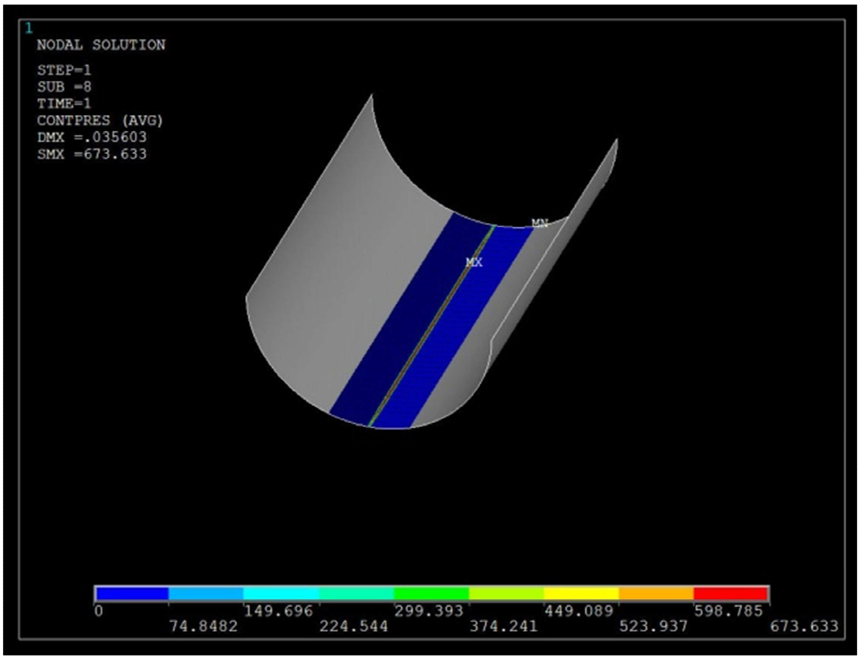

Fig. 5. Contact pressure in the solid rollers

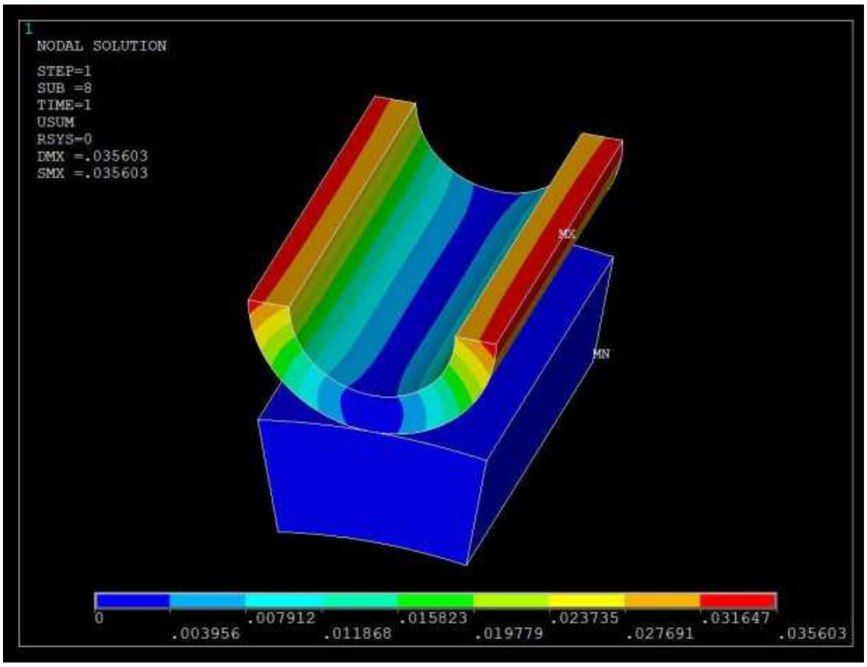

Fig. 6 Total deformations of hollow rollers (67% hollowness)

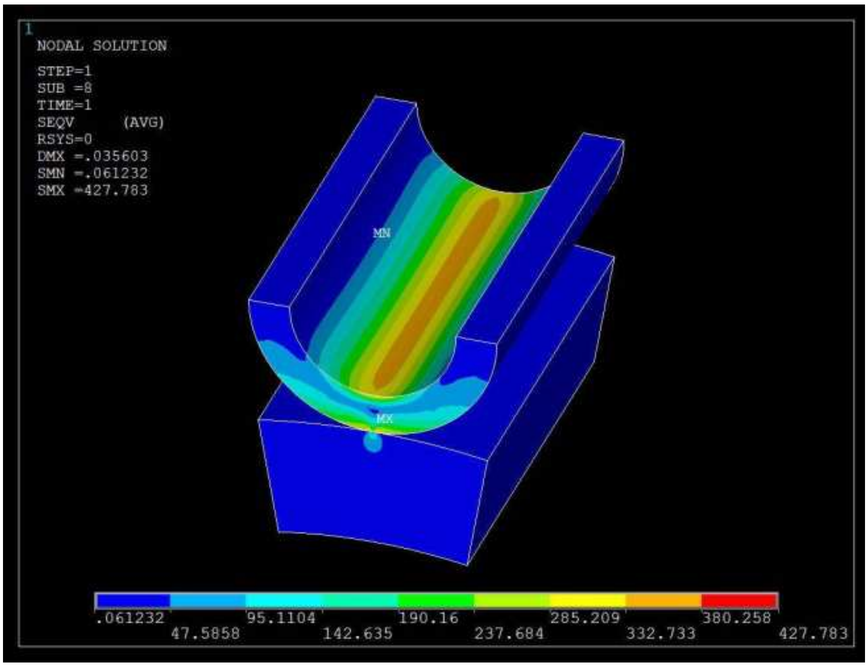

Fig. 7. Von mises stress distribution in hollow rollers (67% hollowness)

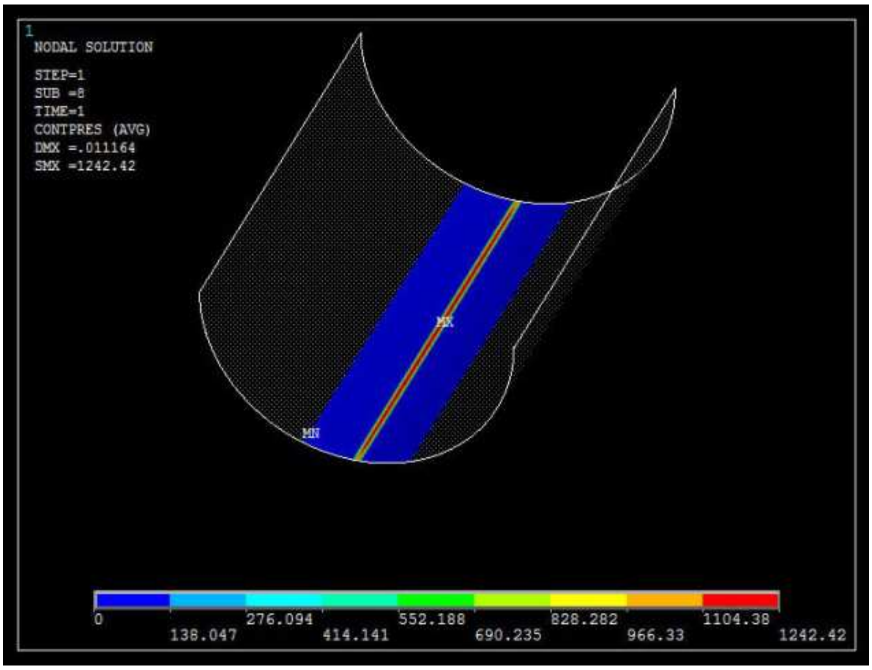

Fig. 8. Contact pressure in hollow rollers (67% hollowness)

Table 1. FEA Results for Different Percentage of Hollowness

% Hollowness	Max Deflection (mm)	Max Von mises stress (Mpa)	Max Contact Pressure (Mpa)
0	0.011164	714.493	1242.420
10	0.010723	660.971	1158.780
20	0.010684	621.961	1094.870
30	0.011212	575.491	1016.970
40	0.012711	523.920	933.628
50	0.016263	469.081	848.029
60	0.024016	433.314	743.757
61	0.025162	437.371	734.877
62	0.026535	425.521	721.416
63	0.028042	423.221	715.340
64	0.029668	433.678	706.251
65	0.031335	420.793	693.420
66	0.033267	429.578	684.732
67	0.035603	427.783	673.633
68	0.037816	428.070	664.612
69	0.040524	432.862	657.624
70	0.043082	434.435	643.924
80	0.102706	572.492	542.375

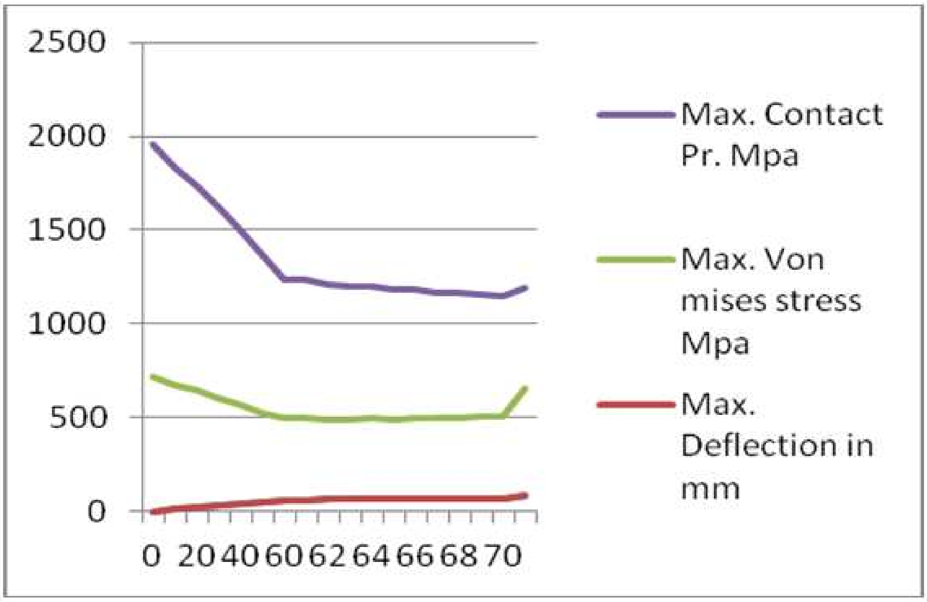

Fig. 9. Graphical representation for FEA results of Different Percentage of Hollowness

5 CONCLUSION

This type of bearing system becomes more favorable in the industry as the hollowness provides more fatigue life due to the decrease in contact stress for almost all geometrical configurations. It has been noticed that the contact stress is minimum for the hollowness between 60% and 70% of the average diameter of tapered roller bearing. More precisely, at 65% hollowness contact stress is found to be lowest that gives optimum hollowness for maximum fatigue life.

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