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FINITE ELEMENT ANALYSIS OF TAPERED ROLLER BEARING: COMPLETE MODEL X SIMPLIFIED

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Abstract. *The goal of this study is to evaluate a tapered roller bearing that supports the differential housing of an automotive manual transmission. A non-linear finite element analysis (FEA) of the stress-strain state of this component was carried out, considering two models: the complete bearing and a simplified model (with only one tapered roller). Comparing the FEA Hertz stress results with the analytical calculation, the simplified model showed a good correlation.*

Keywords: *Tapered roller bearing, finite element analysis, automotive manual transmission.*

1. INTRODUCTION

The most common vehicle power train configuration has an engine and a manual transmission. The engine changes the fuel energy into torque and angular velocity, and these are adjusted for each travel situation by the transmission, as it has a set of gear ratios.

There are a lot of parameters to be studied in a vehicle manual transmission (MT). One of them is the contact between moving elements (gear teeth, synchronizers, shifting components, oil properties, shafts and bearings). The friction involved in these contacts may generate temperature increase and power loss.

According to the chronological development of transmission loss phenomena presented by Lechner and Naunheimer (1999), gears and lubrication were the main objective of study, and little attention was given to bearing losses, which can be correlated to their contact pressures.

Figure 1 presents a MT scheme, and highlights the two tapered roller bearings applied in the differential.

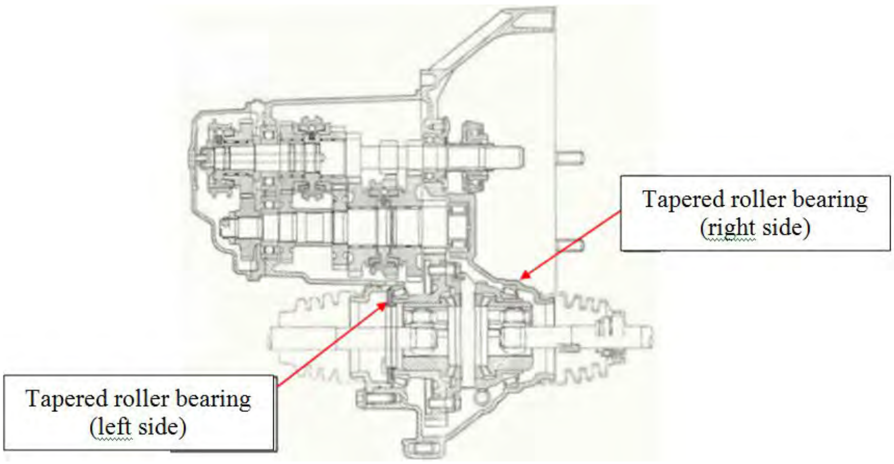

Figure 1. Manual transmission (MT) scheme.

This article shows the evaluation of a tapered roller bearing (TRB) applied in the differential of a manual transmission, by a non-linear finite element analysis (FEA) of its stress-strain state. Yet, a simplified finite element (FE) model (with only one tapered roller) was built and both FE models had their results compared with the analytical calculation.

2. TAPERED ROLLER BEARINGS

The single-row tapered roller bearing is able to carry large radial and thrust loads combined or only the thrust load. Also, as the inner (α_i) and outer (α_o) raceway contact angles are different (see Fig. 2), there is a force component that drives the tapered rollers against the guide flange (Harris and Kotzalas, 2006).

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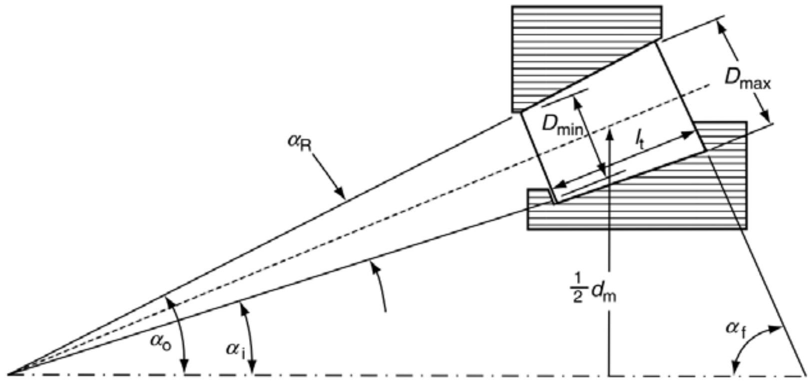

Figure 2. Internal dimensions for tapered roller bearing performance analysis (Harris and Kotzalas, 2006).

The line contact between the roller and the raceway becomes a semicylindrical form, which leads to a maximum normal stress as written in Eq. (1), where Q is the normal force between rolling element and raceway, l is the roller effective length and b is the semiwidth of the contact surface. According to standard ISO 76, allowable values of σ_{max} must not exceed 4 GPa for roller bearings.

$$\sigma_{max} = 2Q/\pi lb \tag{1}$$

3. METHODOLOGY

The TRB studied is identified as 32009 on the manufacturers catalogs, and is made of SAE 52100 steel. The main geometric aspects of this TRB are presented in Fig. 3. Yet, this roller bearing has 23 rollers.

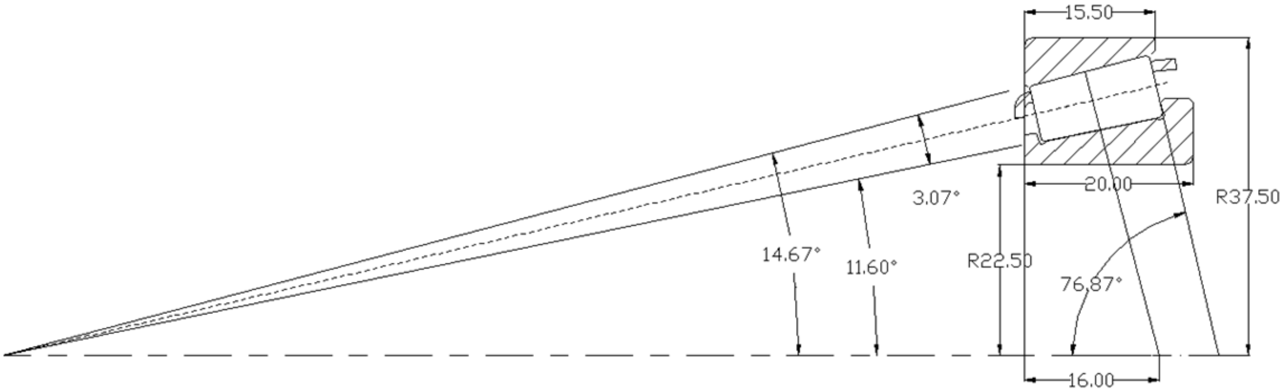

Figure 3. 32009 TRB internal dimensions.

Both FE models (complete and simplified, see Fig. 4) were built with linear tetrahedral (cage and tapered rollers) and hexahedral elements (inner and outer rings). The softwares adopted were *Hypermesh (*pre-processor), *Abaqus* (solver) and *Hyperview* (post-processing). Bearing outer ring is fixed and the thrust load was applied in the center.

The thrust load concerns the maximum torque in 1st gear for a given passenger vehicle, which has 206 N·m maximum engine output torque and 277 mm tire static radius. Knowing the MT gear ratios and dimensions, it may be seen that the maximum thrust load applied on the differential TRB is 4,4 kN. For further information, please refer to Scari (2012).

For the simplified FE model, the cage was not modeled as its main function is to keep the distance among the rollers. Also, a part of the differential housing had to be considered as the connections among the rigid elements and the outer ring results in false stress concentrations.

There are contact elements between the tapered rollers and the cage, inner and outer raceways. The constraints have penalty functions with scale factor equal to 1, and the friction value is 0,2 as recommended by NSK (1992).

When the problem has analytical solution, a good way to validate an FEA result is to compare each other. This way, the contact stresses obtained with the FE models were compared to the analytical calculation presented by Scari (2012).

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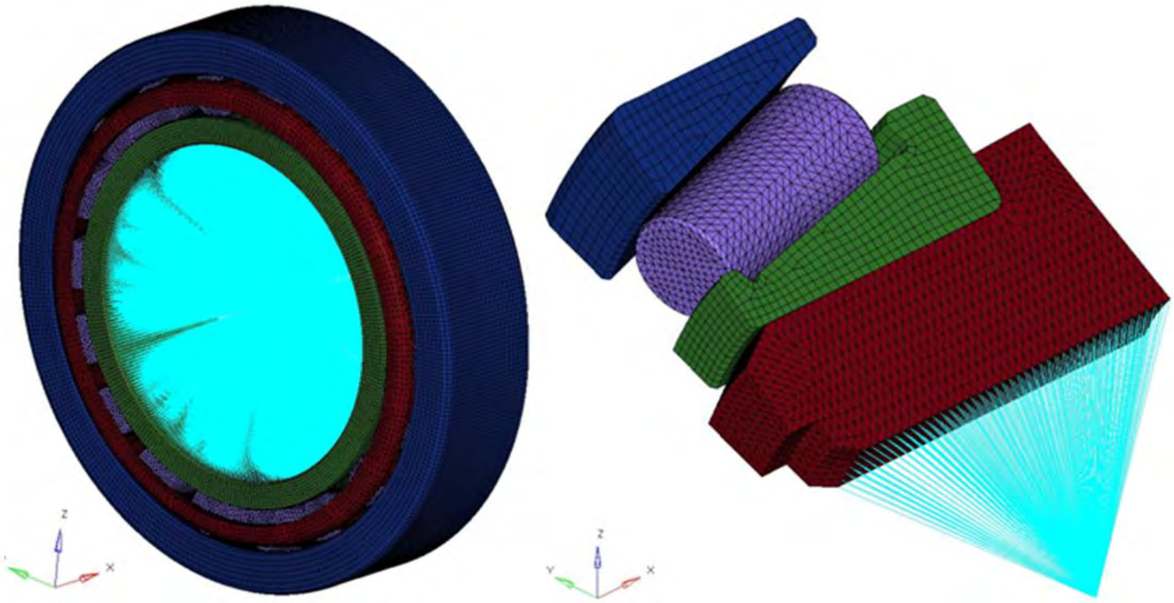

(a) (b)

Figure 4. FE models: (a) complete, (b) simplified.

4. RESULTS AND DISCUSSION

4.1 Complete FE model

Figure 5 and Figure 6 present the results for the complete FE model. For the axial load considered, due to the maximum torque in 1st gear, the contact stresses obtained were 2.829 MPa for the tapered roller end, and 3.155 MPa for the outer raceway. These results are in accordance with the ISO 76 limit (< 4 GPa).

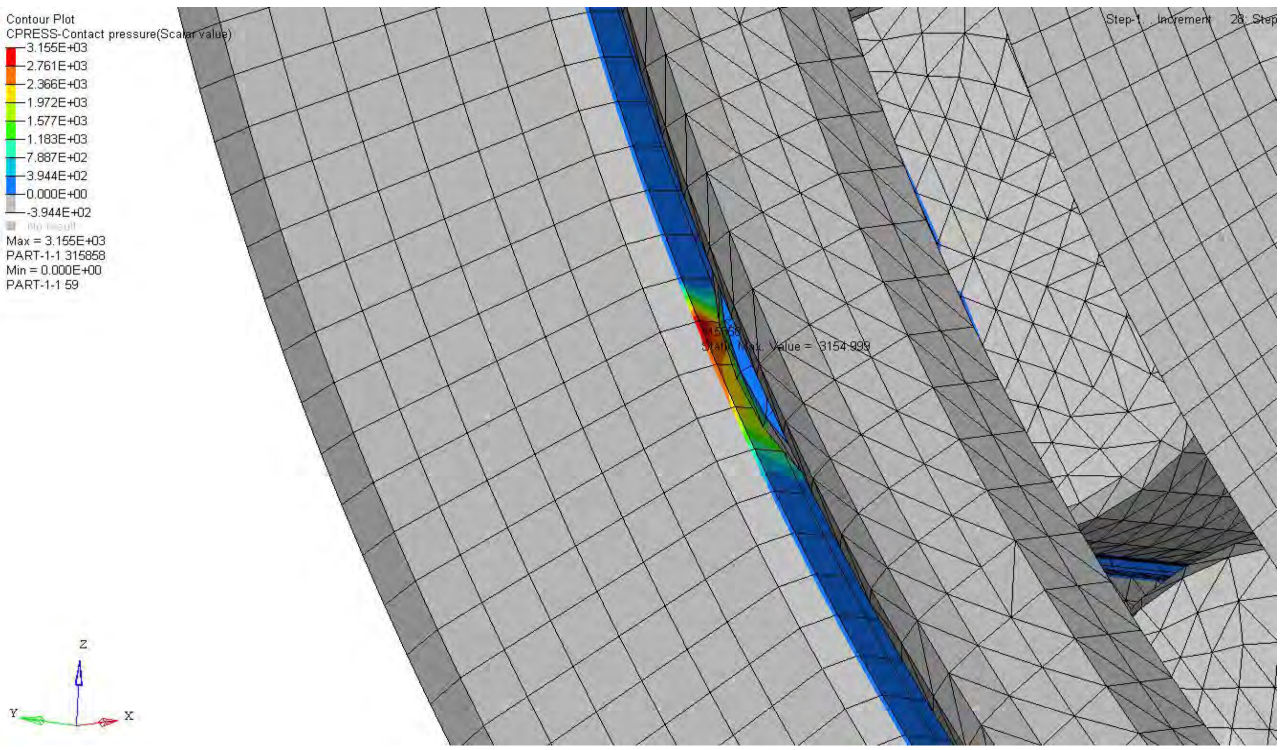

Figure 5. Contact stress results for the complete FE model.

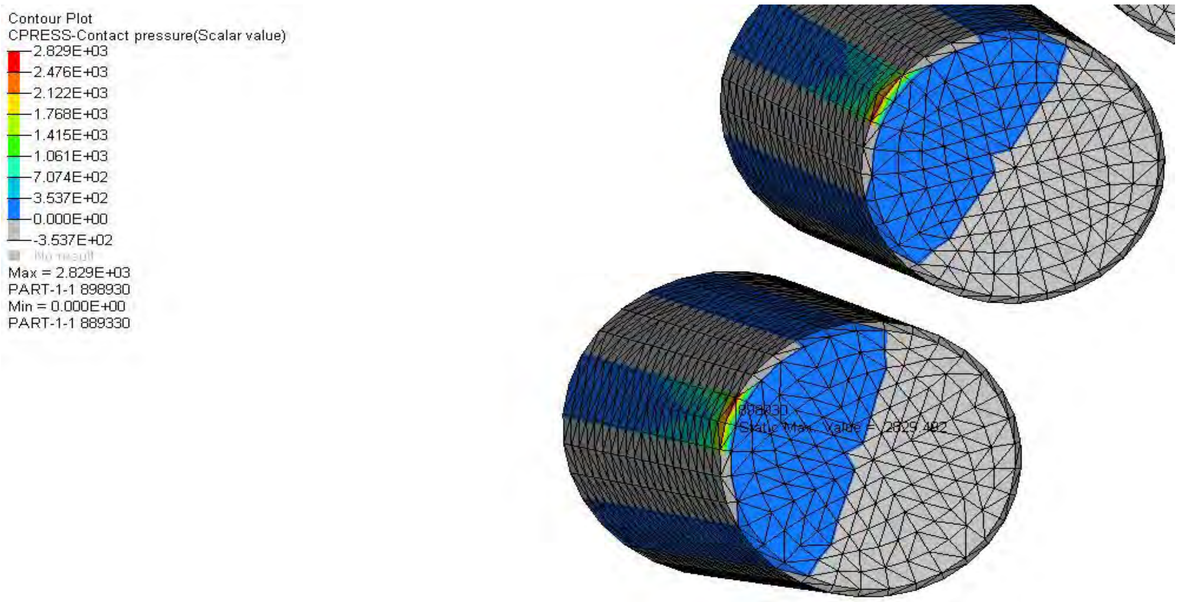

Figure 6. Contact stress results for the tapered rollers - complete FE model.

4.2 Simplified FE model

The results for the simplified FE model are displayed on Fig. 7 and Fig. 8. For the axial load considered, due to the maximum torque in 1st gear, the contact stresses obtained were 3.595 MPa for the tapered roller and 3.752 MPa for the outer raceway. These results are in accordance with the ISO 76 limit (< 4 GPa).

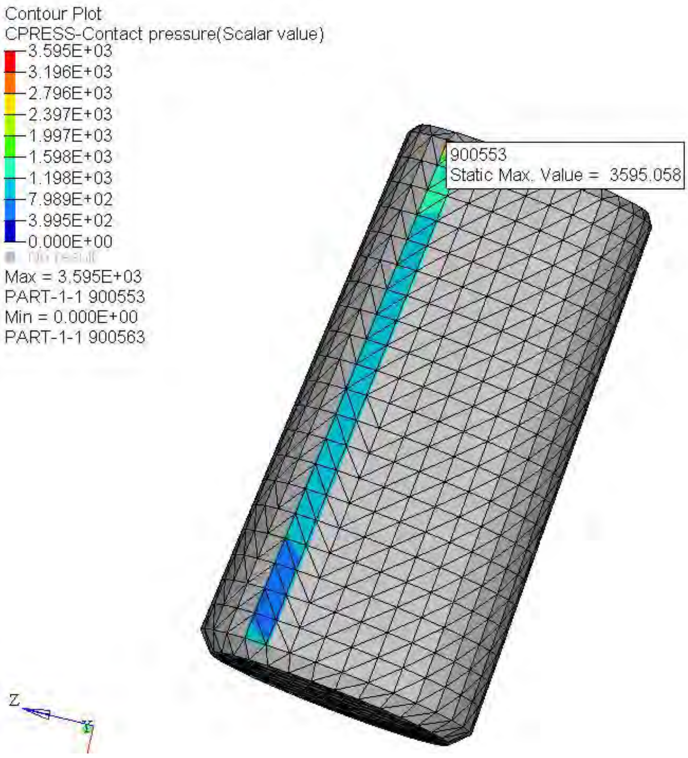

Figure 7. Contact stress results for the tapered roller - simplified FE model.

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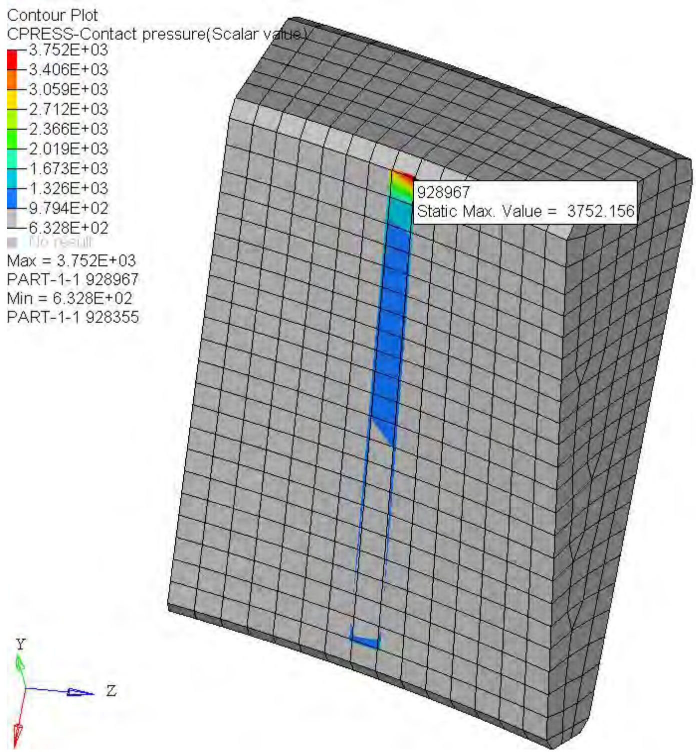

Figure 8. Contact stress results for the outer raceway - simplified FE model.

4.3 Comparison wih the analytical calculation

The analytical contact stress calculation, for the contact between the tapered roller and the outer ring, is 3.679 MPa, considering the axial load due to the maximum torque in 1st gear for a given vehicle. Table 1 presents the results for both FE models considered (complete and simplified), and a comparison among them.

Table 1. Comparison among the FE models results and the analytical calculation for the contact stress.

FE Model	σ_{Hertz} [MPa] Tapered roller	Δ [%] in relation to the analytical	σ_{Hertz} [MPa] Outer raceway	Δ [%] in relation to the analytical
Complete	2.829	-23,1	3.155	-14,2
Simplified	3.595	-2,3	3.752	2,0

From Tab. 1, it can be seen that the simplified model presented the closest result to the analytical calculation, with approximated 2,0 % difference.

5. CONCLUSIONS

The goal of this study was to evaluate the tapered roller bearings that support the differential housing of an automotive manual transmission, by finite element analysis (FEA). Two FE models were considered: the complete and a simplified (with only one tapered roller).

The contact stress is more relevant between the tapered roller and the outer raceway, as expected. Yet, the outer raceway presents the highest contact stress.

Comparing the FEA results with the analytical calculation from Scari (2012), the simplified FE model presented a better correlation, with 2,0 % difference, approximately. It shows that the simplified FE model, besides requiring less computational effort, is most suitable for static roller bearing analysis subjected to thrust loads.

6. REFERENCES

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7. RESPONSIBILITY NOTICE

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