

Formula One Car Wheel Bearings: an FE Approach

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Abstract: This report concerns the Finite Element modeling of a complete Formula One car wheel group. The main goal of such a model was to develop a design tool for wheel ball bearing optimization, both in terms of their layout and their dimensions. The project was a collaboration with the bearing manufacturer with the goal of best performance in conjunction with best reliability.

Keywords: Contact, Contact pressure, Hub, Non linearity, Upright, Suspension, Wheel Bearings.

1. Introduction

In the competitive environment of Formula One, Finite Element simulation plays a determinant role not only in reliability issues but also in performance. In this report a Finite Element approach to handle the wheel bearing modeling is reviewed. Some results are also presented.

2. An F1 Wheel Group

After the binomial tire-rim, which is the primary interface between the track and the vehicle, the hub-wheel bearings-upright assembly is probably the most important part in a racing car suspension. Generally hub and upright are designed by the car constructor, following some criteria such as lightness, strength, heat dissipation (from brakes) and stiffness. Wheel ball bearings are commercial products, in the sense that they are designed and produced by third parties. Nevertheless, although a wheel bearing can be considered as a “black box”, the mileage of which is given by its supplier, it is still possible to introduce in a model some elements that guarantee the correct force flow (load path) between hub and upright. To do so it is clearly necessary to have the collaboration of the bearing supplier because some information, such as number of balls, their diameter and their material, contact angle and geometry of the rollers is necessary. Figure 1 shows a layout example of the wheel bearings. The bearings are assembled following the “O” scheme, which is classical for an F1 application. The bearings are preloaded by tightening the nut causing the preload force to spread through the spacer, the bearing and the upright, as shown in the stiffness scheme of Figure 2. Generally the hub, upright and spacer stiffness are transmitted to the bearing supplier who performs calculations, and then provides feedback about the necessary preload force to be applied by the nut. This preload is a compromise among different needs:

stiffness of the assembly, bearing endurance and behavior under working temperature. This procedure can be iterated for a few cycles, which of course requires time. By having the bearing behavior in one assembly model, design time can be reduced.

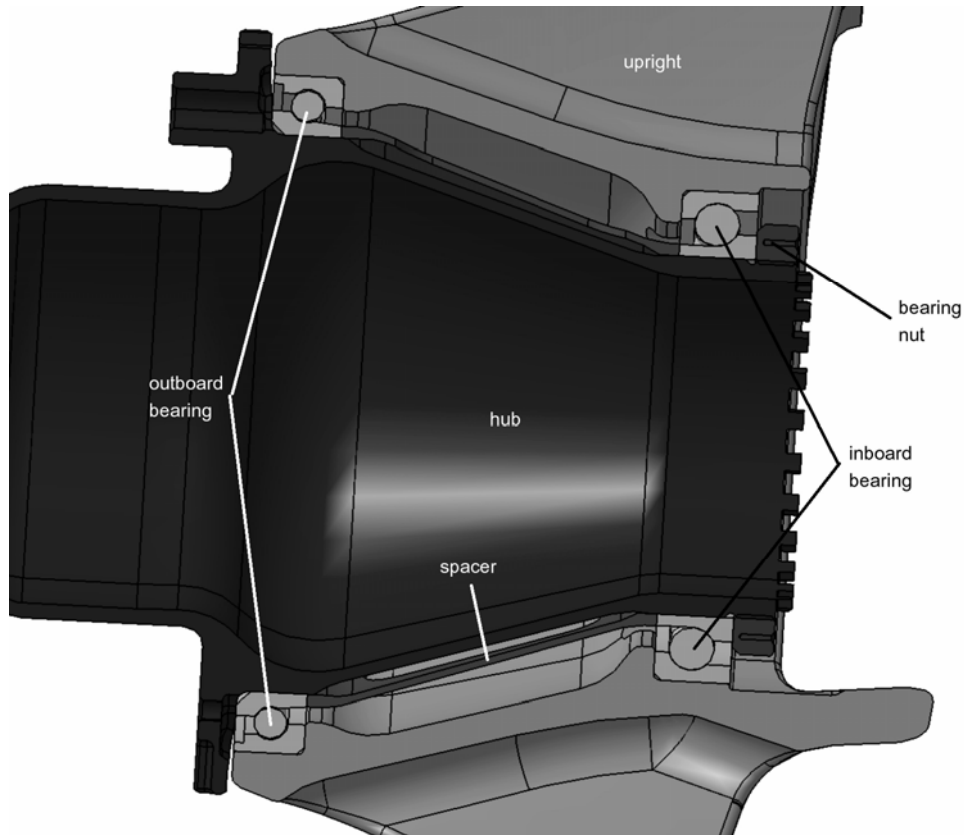


Figure 1. Wheel bearing installation.

3. Bearing FE model

In order to correctly model the structural behavior of the bearing it is necessary to represent its stiffness, which is basically located in the balls. The stiffness of a ball is non linear, which has been deduced from Hertz theory. If for example we take the relation giving the deformation of a sphere pressed against an infinite rigid planar surface (Gianini, 2006) we have:

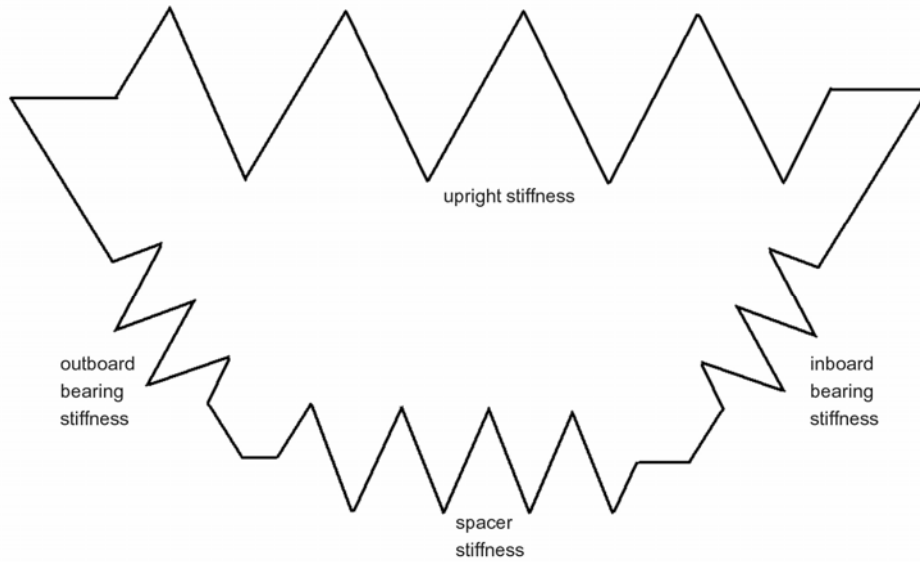


Figure 2. Wheel bearing assembly stiffness scheme.

$$u = 1.55 \cdot \sqrt[3]{\frac{Q^2}{4 \cdot d \cdot E^2}} \quad (1)$$

where u is the displacement, Q is the applied load, d is the diameter of the sphere and E the Young's modulus of its material. If we graph Equation 1 for some given input data we obtain the curve shown in Figure 3.

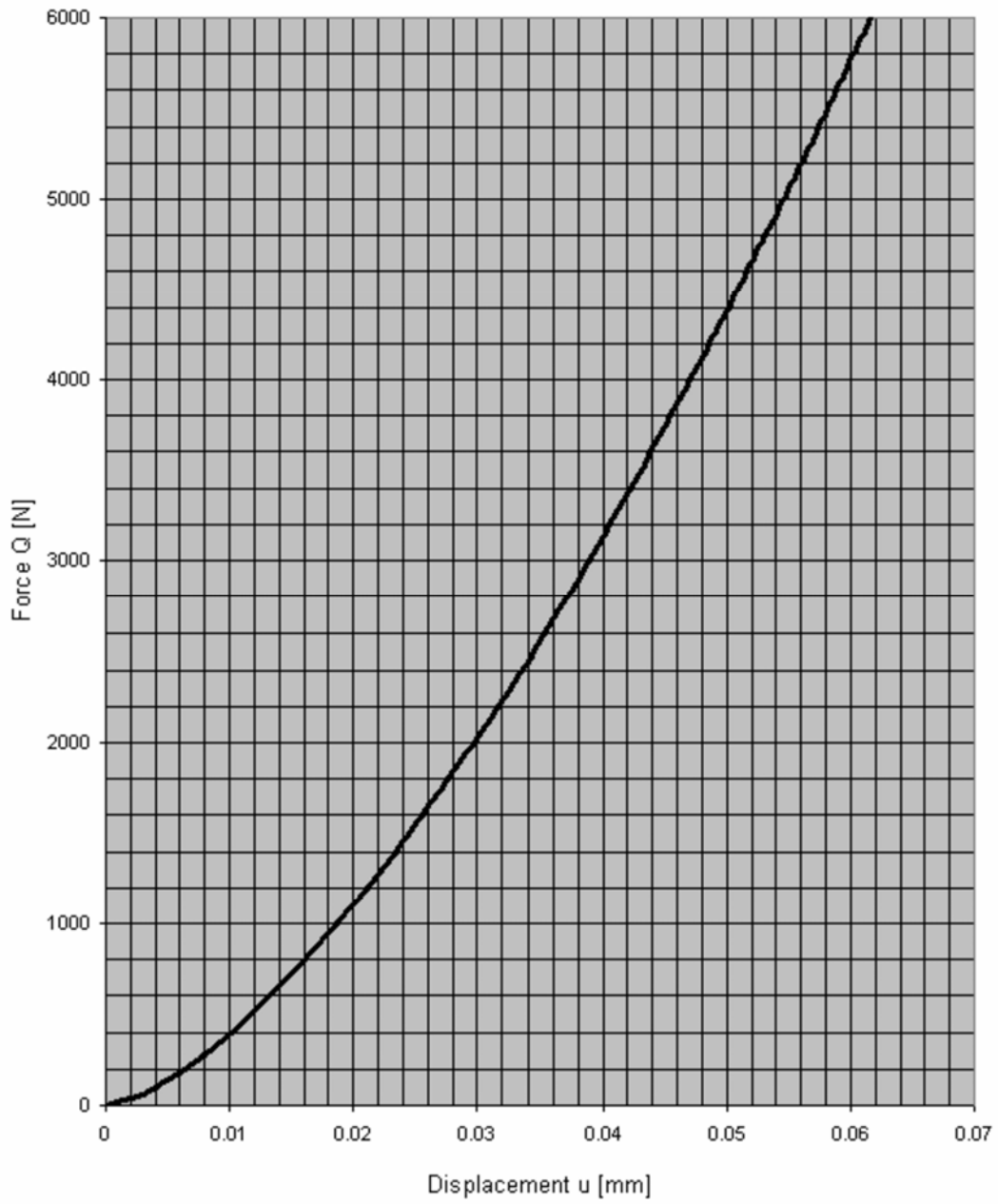


Figure 3. Stiffness of a ball pressed against an infinitely rigid planar surface.

It can be noticed that the sphere becomes more rigid as the load acting on it increases. Therefore, once the physical data of the balls is known, it is possible to use a non linear element, such as SPRINGA, to model the behavior of the ball. As an example the following lines illustrate the ABAQUS input for one of the bearings.

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**
*SPRING, ELSET=Small_Balls, RTOL=.03, NONLINEAR

-6000., -.0646
-5000., -.0583
-4000., -.0502
-3000., -.0415
-2500., -.0367
-2000., -.0316
-1500., -.0261
-1000., -.0199
-750., -.0164
-500., -.0126
-250., -.0079
-100., -.0043
-50., -.0027
0., 0.
1., 10.

**
** SPRING ELEMENTS
**
*ELEMENT, TYPE=SPRINGA, ELSET=Small_Balls
34978, 19024, 78741
34979, 19001, 78712
34980, 78683, 18978
34981, 18955, 78654
34982, 78625, 18932
....., ....., .....

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Once complete for all bearings in the assembly, such data can be introduced in the complete model. The rollers (inner and outer) have also to be modeled in order to get the correct connection of the “spring balls” to the other parts, as shown in Figure 4, where K_b represents the non linear stiffness defined above.

4. The Wheel Group FE model

Figures from 5 to 7 show the Wheel Group model in which the wheel bearings have been introduced. The upright was constrained at the suspension pick-up points, i.e. no suspension arms are present and the boundary conditions were set in such a way that the correct load path was respected (statically determined constraint conditions). The preloads given by the wheel nut and the bearing nut have been introduced using the CLEARANCE option in the CONTACT card (negative clearance). The same negative clearance method has been used to model the radial interference fittings of the bearings on the hub and on the upright. A second step has been applied to the model to impose a uniform temperature distribution on the parts to account for differences in thermal expansion of the materials used. Finally, the forces coming from a high speed corner

(approximately 5Gs of lateral accelerations), specifically vertical and lateral forces plus the self aligning torque, were applied at the theoretical position of the tire contact patch through a DISTRIBUTED COUPLING element. This acts to simulate the part of the rim that was cut to reduce the size of the model.

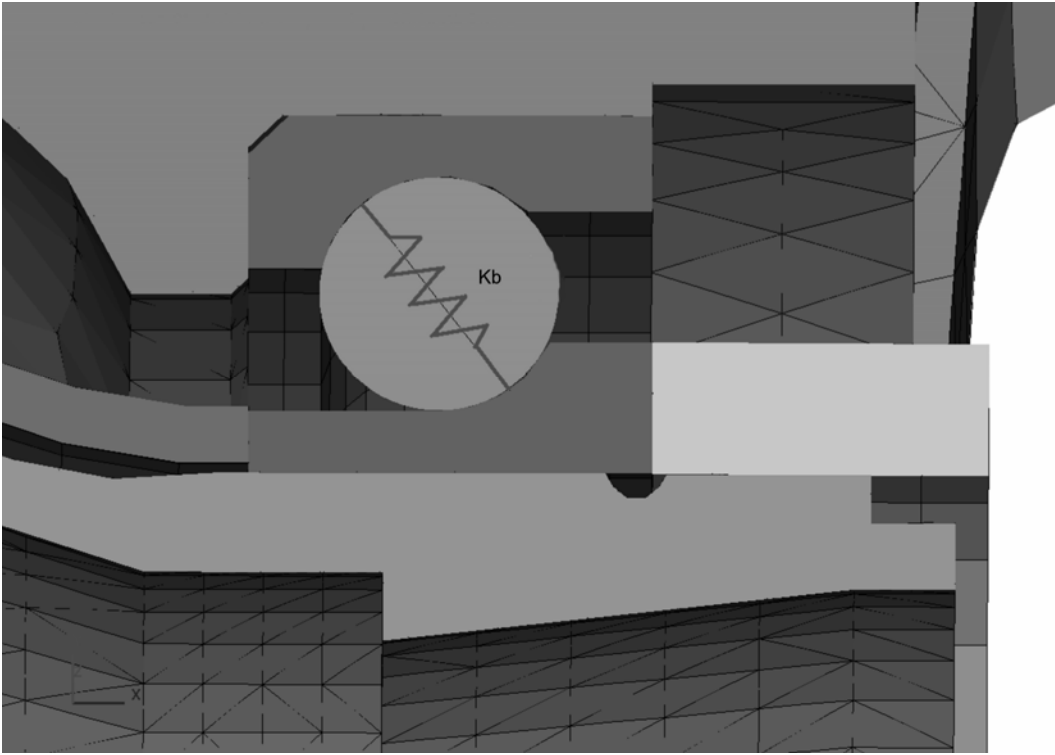


Figure 4. The “spring ball” superimposed to the actual ball.

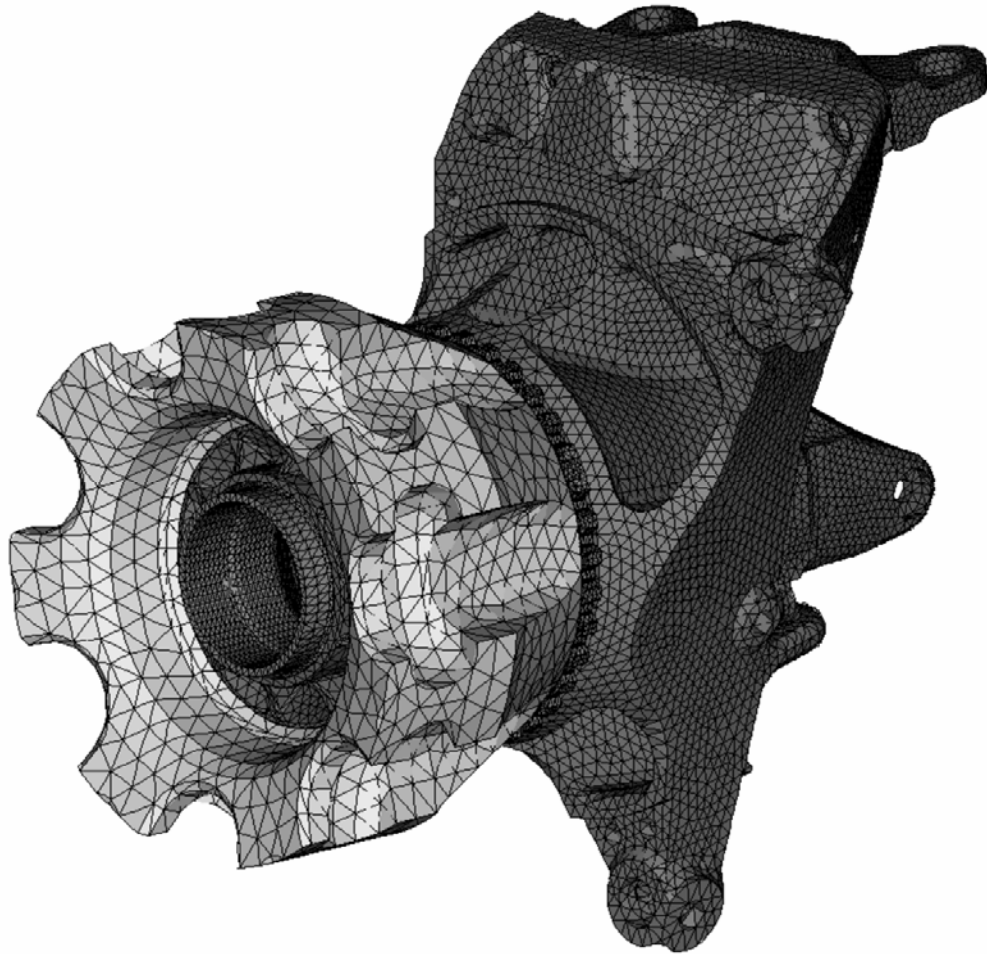


Figure 5. FE model of the complete wheel group.

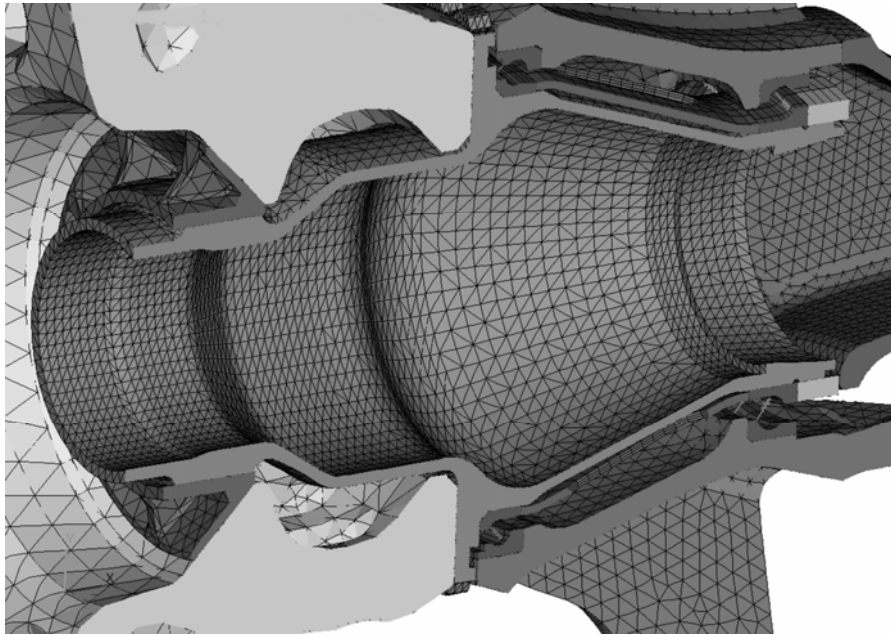


Figure 6. FE model of the complete wheel group – Section.

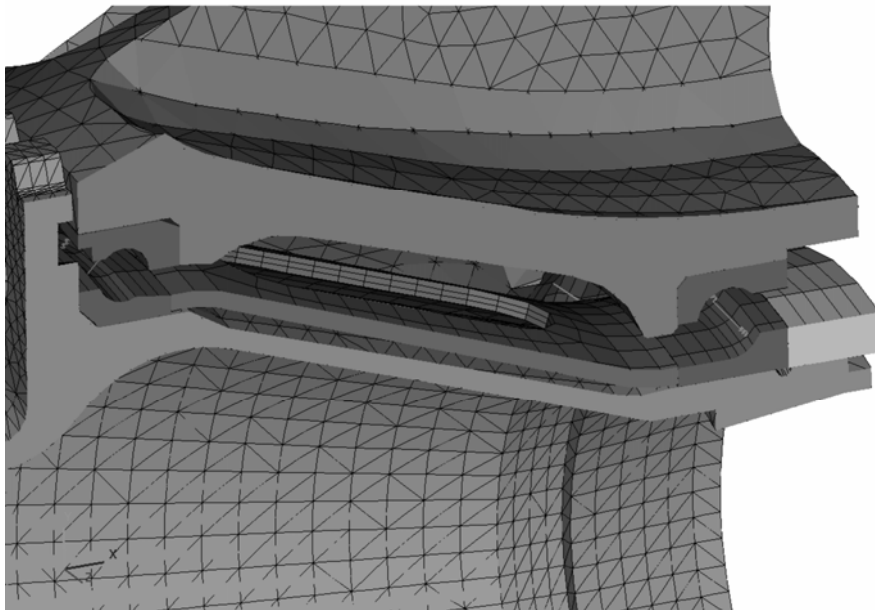


Figure 7. FE model of the complete wheel group – Bearing detail.

5. Results

The implementation of such a model has been required to check the feasibility of the bearing layout, to establish their mileage limit and to evaluate the stiffness of the assembly. Moreover it was possible to tune the value of the preload necessary to obtain the best compromise between stiffness (more preload) and durability (less preload). All this information was based on the post-processing of the forces acting on the “spring balls” in the load conditions of interest (i.e. high speed cornering). In fact, once again from Hertz theory, from the force acting on a sphere it is possible to extract the contact pressure (Gianini, 2006):

$$p_H = \sqrt[3]{\frac{K \cdot E^2}{4.28}} \quad (2)$$

$$K = \frac{Q}{d^2} \quad (3)$$

$$E = \frac{2 \cdot E_1 \cdot E_2}{E_1 + E_2} \quad (4)$$

where E_1 and E_2 are the Young's modulus of the materials in contact. The experience of the bearing supplier was used to judge the allowable limits for the contact pressure. It was possible to check in precise way the stress level of the various components; clearly the stress levels in the inner and outer rollers are not reliable due to the fact that the spring element has a node-to-node connection. Figures from 8 to 10 illustrate the stress level for the three step of the calculation: preload, thermal effect, cornering. Further Figures 11 and 12 show the forces acting on the balls of the inboard and outboard bearings. As it can be noticed some balls of the inboard bearing see no load, which means that there is no contact; this fact is generally detrimental for bearing endurance and should be minimized as much as possible, for example by increasing the preload level. Finally Figures 13 and 14 show the deformation of the bearing rollers, magnified by a factor of 20.

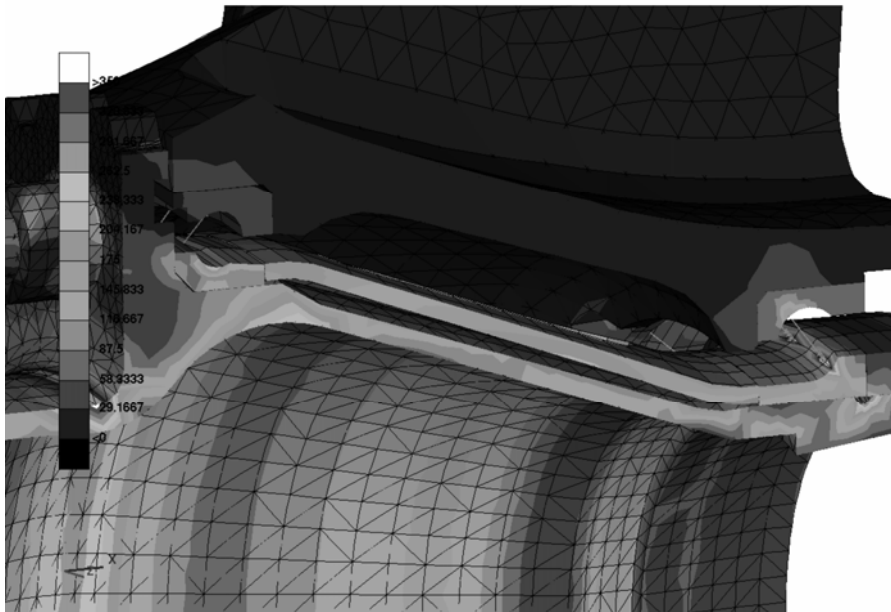


Figure 8. Von Mises stress distribution – Preload.

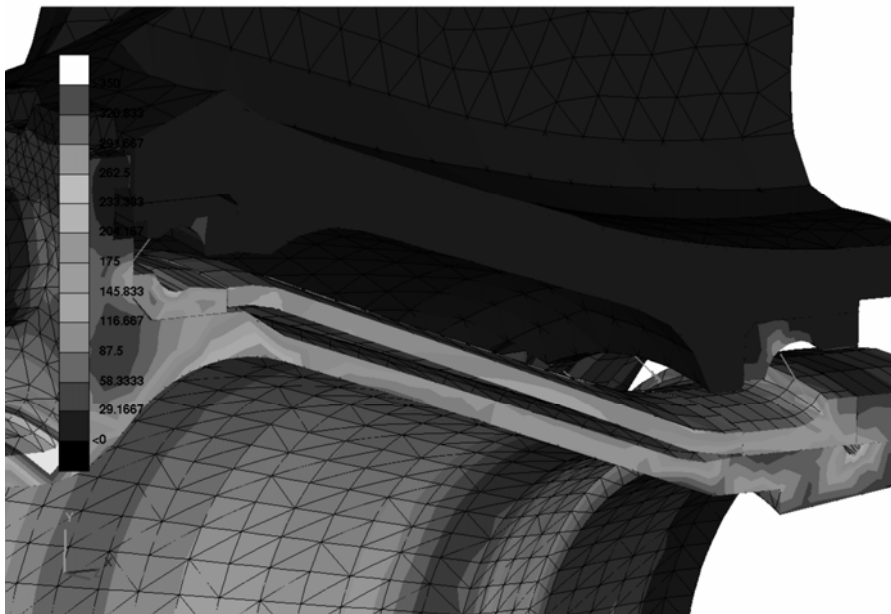


Figure 9. Von Mises stress distribution – Preload + Temperature.

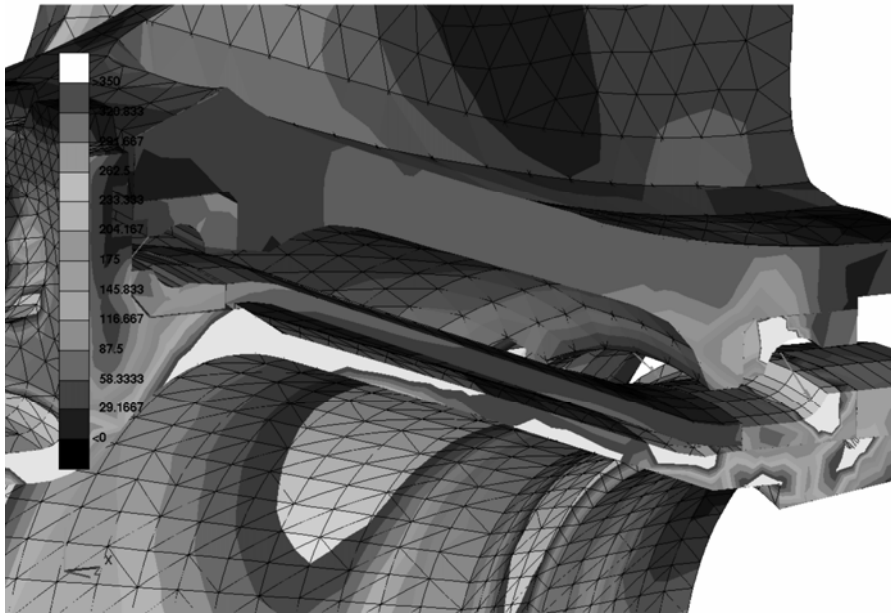


Figure 10. Von Mises stress distribution – Preload + Temperature + Cornering.

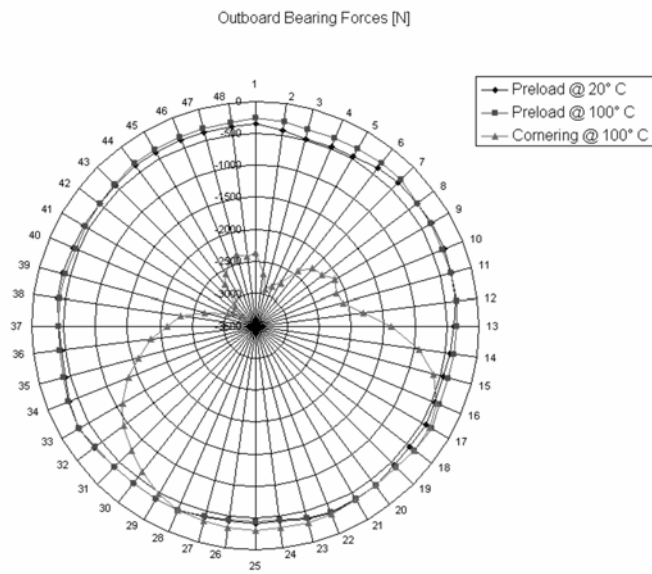


Figure 11. Forces acting on the balls of the outboard bearing.

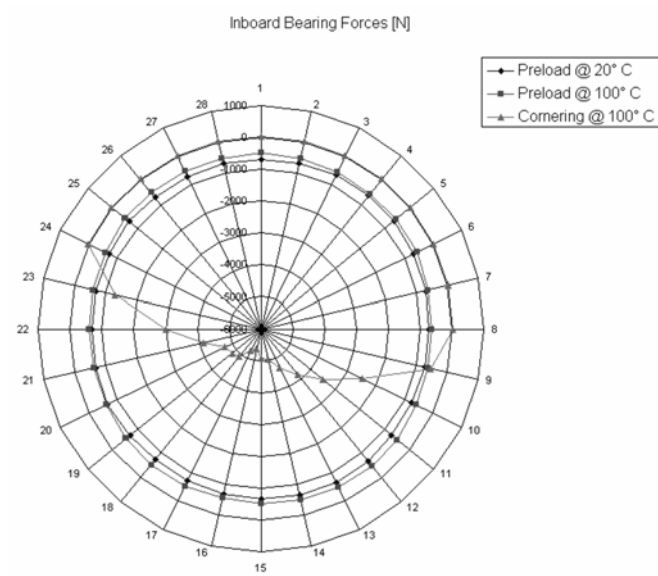


Figure 12. Forces acting on the balls of the inboard bearing.

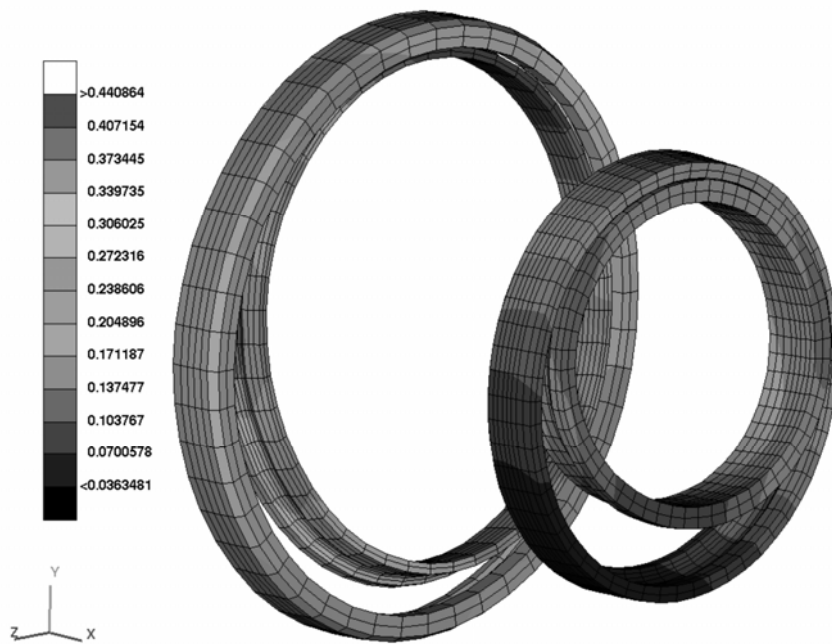


Figure 13. Deformed shape of the bearing rollers (amplification factor = 20).

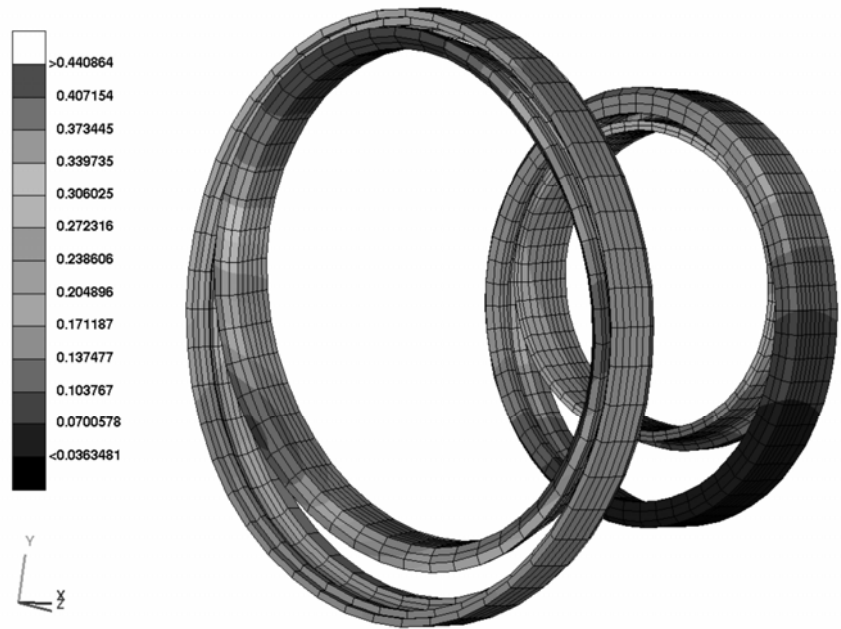


Figure 14. Deformed shape of the bearing rollers (amplification factor = 20).

6. Conclusions

Through this modelling of the wheel group, it was possible to achieve improvements in the understanding of the wheel bearing behaviour under the various load conditions. The synergy of the bearing supplier experience together with the structural knowledge of the racing car constructor gave the best compromise in terms of performance vs. reliability. Strong points of this approach are:

- the wheel bearing model is quite quick to be obtained;
- the results are reliable, as demonstrated by experimental data comparison and by the complete different approach used by the bearing supplier;
- the model runs quite quickly on actual machines and therefore different preload values can be checked rapidly.

On the other hand some weak points are present in this approach:

- the model is static, i.e. dynamic effects are not accounted for;

- the node-to-node connection of the spring element to the brick elements of the rollers is less than optimum;
- the ball contact angle (see Figure 15) cannot vary, apart from geometrical deformations, while in actuality this angle can change (i.e. the ball moves).

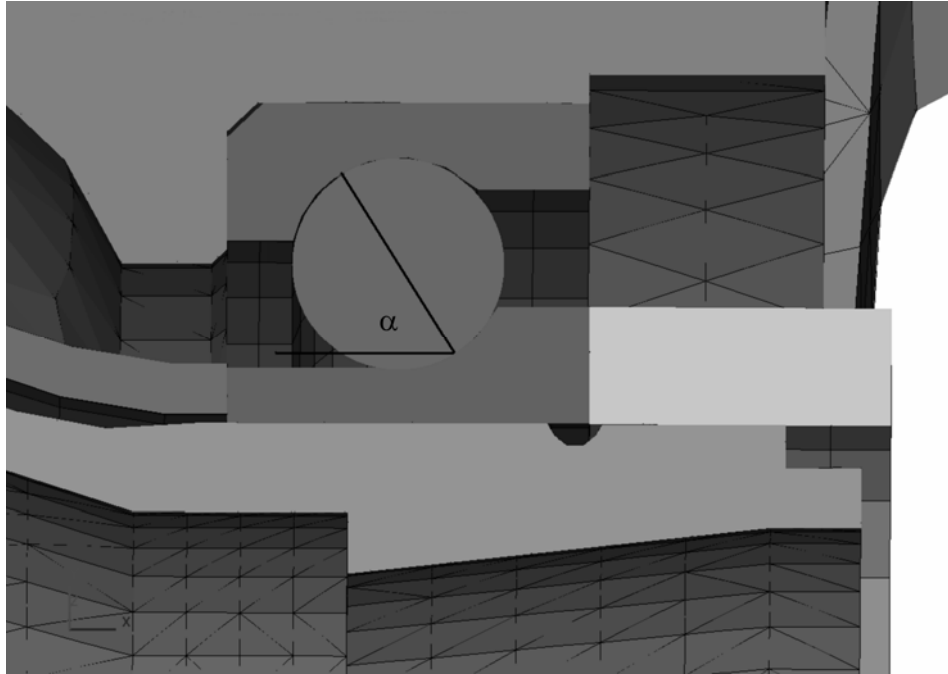


Figure 15. Constant contact angle.

7. References

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