Multibody Contact Analysis of an Rzeppa CV-Joint

L. Armellin[1], F. Gatelli[*,1], and G. Tanghetti[1]
[1]Metelli S.p.A., R&D Department
[*]Corresponding author: via Caduti sul lavoro 43, 25033 Cologne (Bs), Italy, fabiogatelli@metellispa.it

Abstract: Ball joints are widely used in many applications. This paper describes the contact and kinematic analysis of an Rzeppa type constant-velocity joint (CV-joint).

Starting from a conveniently simplified 3D model, at fixed joint angle of 45°, a CV-joint made of all "generic steel" components has been studied.

Considering only a "perfect" geometry (i.e. not affected by tolerances), suitable contact pairs have been defined to reflect mutual interaction among joint components.

Rotation has been imposed to inner race and a suitable resisting torque has been applied to outer race to reproduce the working conditions in a typical application.

Keywords: Structural Mechanics, Contact Analysis, Kinematics, CV-Joints

1 Introduction

The need to transmit rotary movement through angled shafts arose since the XII century in construction of clocktowers.

The transmission of rotary movement between two angled shafts by means of two bewel gears is uniform and you can imagine to replace gear teeth with balls and spherical pockets. The last step is to give the joint the ability to articulate; the spherical pockets have to be extended into ball tracks in a bell-shaped outer race and a spherical inner race (see figure 1).

The first example was the 1908 Whitney's ball joint patent, but no one paid attention to his idea due the lack of practical applications at that time.

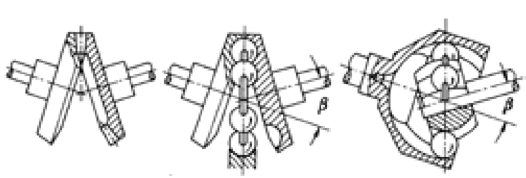

Figure 1: Evolution from bewel gears to ball joint

Significant further steps were made by Rzeppa in the '30s, leading with his patents to a geometry still in use.

Nowadays these joints are widely used in several industries (energy, aerospace, automotive).

Especially in automotive industry every front-wheel drive car has a couple of them to transmit engine torque to the wheels even at high steering angles (up to 48°) without introducing any significant angular velocity ripple.

2 CV-joint Structure and Operation

Figure 2 shows a section of a CAD model of an Rzeppa CV-joint. Starting from the center you can see:

- the inner race (dark gray)
- the balls (red)
- the cage (green)
- the outer race (light gray)

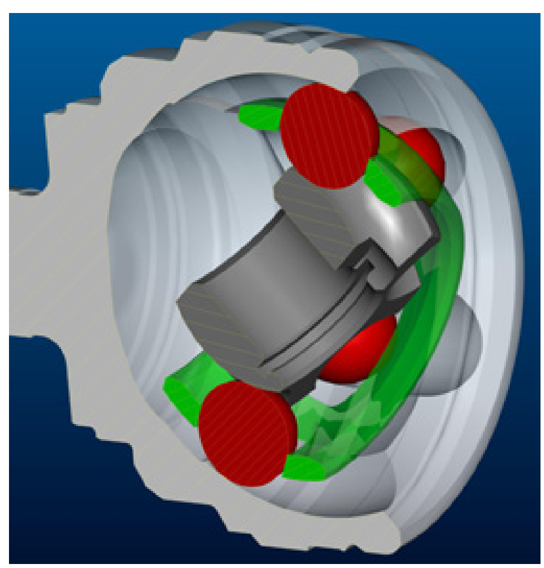

Figure 2: CAD model of an Rzeppa CV-joint

The inner race is engaged through a spline on the shaft which transmits the torque coming from the differential gear. Spline and shaft are not shown in figure 2.

The components shown in figure 2 are lubricated with grease and a rubber boot (not shown in figure 2) is normally clamped over the shaft and the outer race to avoid the loss of grease and preserve inside components against environment (dust, moisture, salt, etc.).

Rotation and torque are transmitted to the outer race through the balls. The cage must keep the balls and the inner race in the proper position, to avoid joint jamming at some kinematically critical angles (e.g. null joint angle, ϕ).

Joint angle ϕ is the angle between inner and outer race axes. In case $\phi \neq 0$, during joint rotation the balls run along relative races, aimed at keeping the joint almost homokinetic; in every instant only 3 or 4 of the balls are transmitting most of the torque between inner and outer race. The plane of ball centers (which is cage simmetry plane) is rotated by exactly $\phi/2$ with respect to inner race simmetry plane (see figure 3).

For a detailed description of Rzeppa CV-joints you can see e.g. [1] and [2].

Due to its complex geometry, CV-joint has several contact areas among all of the components and the relevant stresses are therefore almost impossible to estimate in a closed-loop design.

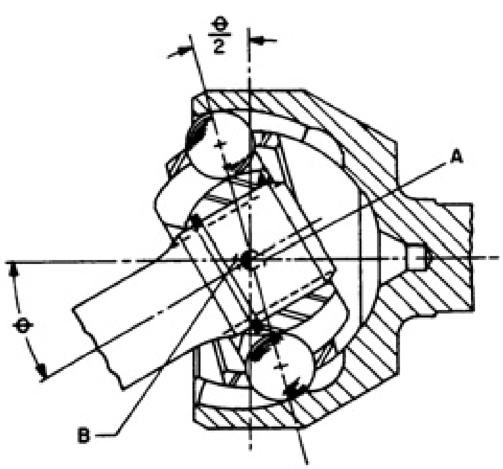

Figure 3: Cross-section of an Rzeppa CV-joint

3 Model Description

A real CV-joint with some simplifications has been used as test case. The simplifications introduced are aimed at keeping the model "simple" without affecting the significance of the obtained results.

A fixed joint angle $\phi = 45°$ has been chosen.

"Solid, Stress-Strain (smsld)" application mode of COMSOL Multiphysics Structural Mechanics Module has been used.

3.1 Geometry

To avoid unnecessary complexity of the model (i.e. to keep the number of degrees of freedom of the mesh as small as possible), most of the outside geometric features of the outer race have been removed, as well as the entire shaft and the boot.

Moreover, nominal geometry is used, i.e. without any effect of misalignments due to geometrical and dimentional tolerances. Figure 4 shows the result of these simplifications.

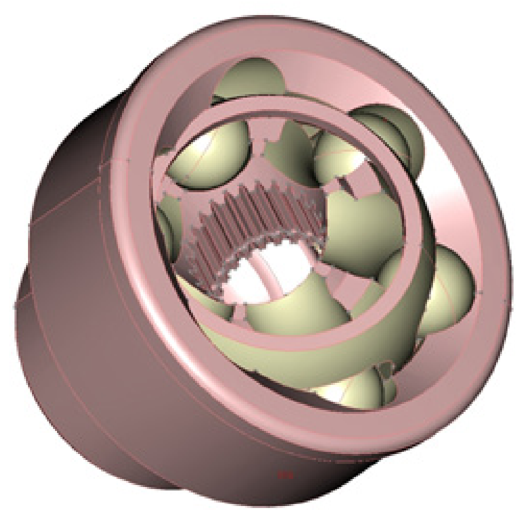

Figure 4: Geometry of simplified Rzeppa CV-joint in COMSOL Multiphysics

3.2 Material

In real CV-joints, each component has its specific material (high-performance steels) and heat treatment, generally:

- *induction hardening* for outer race
- *case hardening* for balls, cage and inner race

In the model a unique uniform isotropic material (a "generic steel") has been used for all joint components, without taking into account any hardening-induced pre-stress.

3.3 Contact Pairs

During joint rotation, the balls are normally in contact with inner race, outer race and cage; in case of non-null joint angle ϕ, the balls run along respective races (inner and outer), moving corresponding contact points.

The cage is normally in contact with the balls and with inner and outer race.

In the model, it's enough to define only 3 contact pairs:

- cage (master) and inner race (slave)
- cage (master) and outer race (slave)
- cage, inner and outer race (master) and balls (slave) (this contact pair is shown in figure 5)

To simplify the model, friction has not been taken into account.

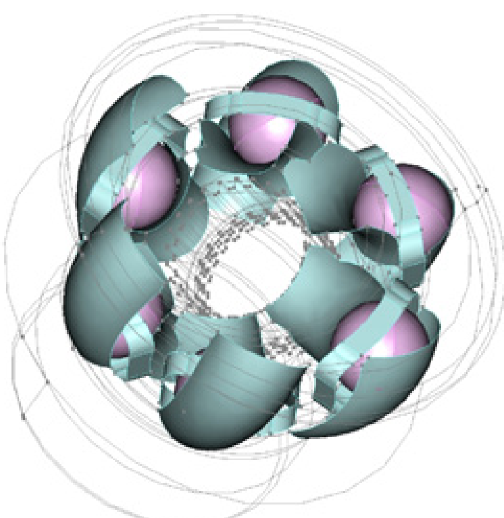

Figure 5: Contact pair between balls, cage, inner race and outer race

3.4 Constraints

A real CV-joint is put in rotation by the shaft which is not present in the model. For this reason, a rotation around its axis has been imposed to the spline of the inner race as prescribed displacement (see figure 6).

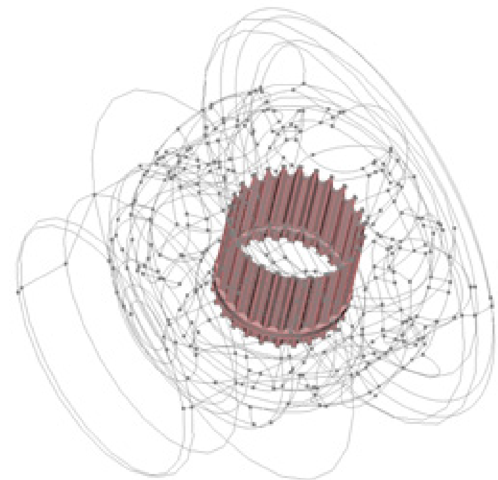

Figure 6: Surfaces with imposed rotation around inner race axis

Outer race is kept at angle ϕ and cannot move along its axis; this is obtained by imposing that surface shown in figure 7 can only act as a roller.

Remaining surfaces are free.

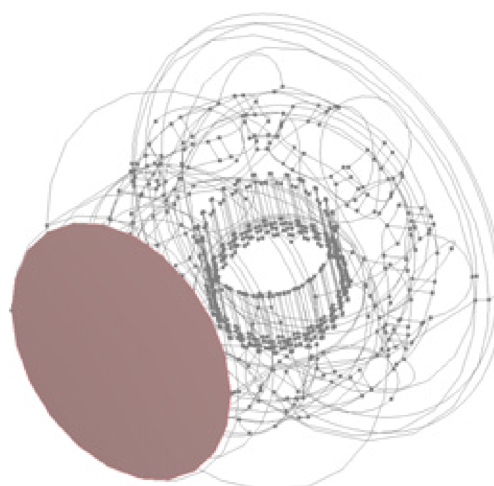

Figure 7: Roller surface

3.5 Loads

The resisting torque of real CV-joints is normally applied to the outer race. In the model, due to the geometric simplifications introduced, it is impossible to apply the resisting torque in a physically correct position, therefore it has been chosen to apply it in an "easy" position (see figure 8).

This last simplification certainly affects the general stress status of the outer race, but not near the contact areas which are normally the areas with the maximum stress values.

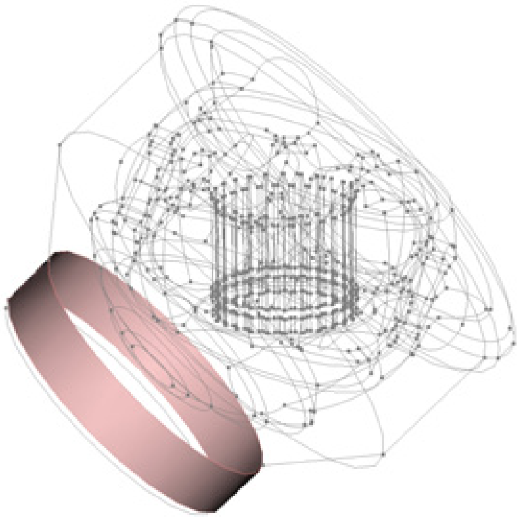

Figure 8: Surface where resistive torque is applied

3.6 Solver Parameters

The model has been solved with a transient time-dependent solver. A time interval $0 \leq t \leq 0.1s$ has been chosen. In this time joint makes one complete revolution. Linear system solver PARDISO has been used.

A quite rough mesh has been used (approximately 106000 degrees of freedom for the whole model). On a 2 dual-core (Xeon 2.33 GHz) 64 bits Linux workstation, solution took about 100 hours.

4 Simulation Results

4.1 Kinematic

Figure 9 shows total displacement of joint components at $t = 0.098s$.

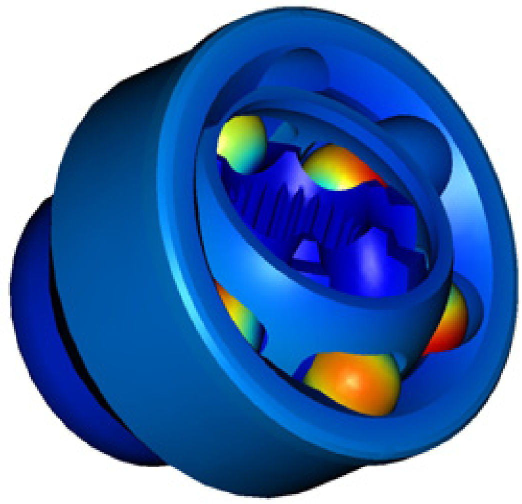

Figure 9: Total displacement of joint components

Figure 10 shows y-displacement of two points during joint rotation. One of the points belongs to inner race and the other to outer race. Both points are initially (i.e. at $t = 0$) in the same x-z plane at different values of y.

As can be seen, the two y-displacements differ only for a scale factor, due to the different radial position.

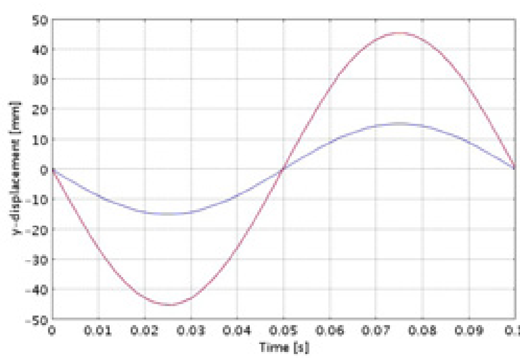

Figure 10: Y-displacement of one point of the inner race (blue) and one point of the outer race (red)

4.2 Stress

Figure 11 shows von Mises stress of one of the balls. It's quite evident that a much finer mesh should be used, but the figure shows anyway the typical fingerprint of Hertzian contact stress.

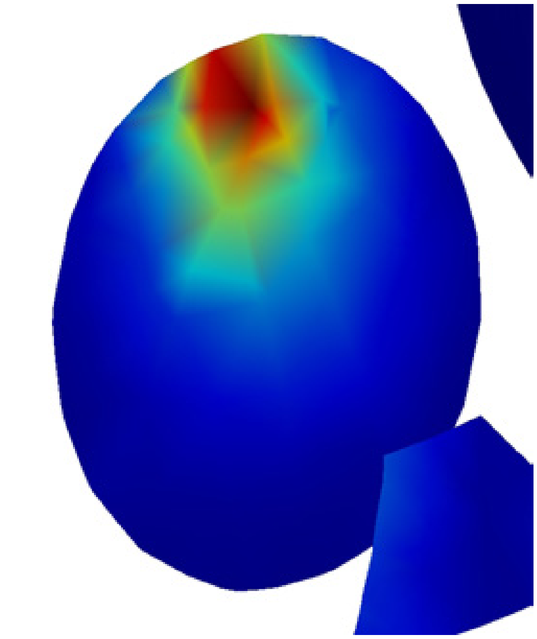

Figure 11: Von Mises stress of a ball

Figure 12 shows von Mises stress of inner race. Again, the fingerprint of Hertzian contact stress is evident.

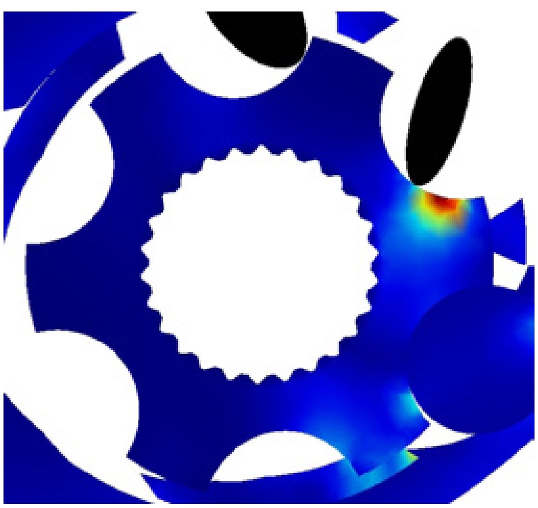

Figure 12: Von Mises stress of inner race

4.3 Contact areas

Figure 13 shows contact pressure over the balls. The effect of a too coarse mesh is evident, but the different values of contact pressure among the balls is also quite evident.

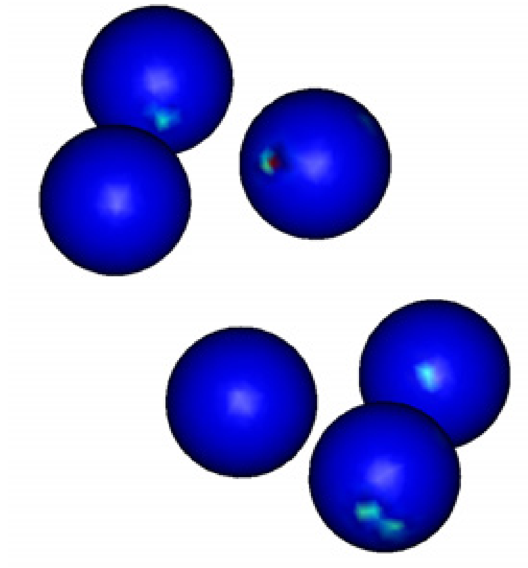

Figure 13: Contact pressure of the balls

Figure 14 shows contact distance of the same contact pair (i.e. the one involving the balls).

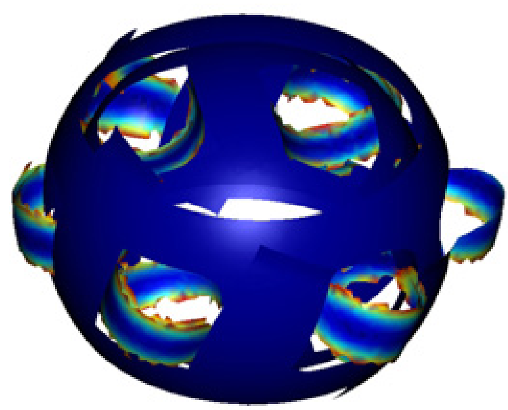

Figure 14: Contact distance of the balls

5 Discussion

As already pointed out (see section 3), the model includes several simplifications which make it quite different from a real CV-joint.

In spite of the above simplifications, kinematic behavior of the model reflects the real one, with the exception of the almost unpredictable effects introduced by production tolerances.

Due to these simplifications, unfortunately, model stress results aren't reliable (at least for the moment) for joint design, but show anyway a realistic stress distribution among the components.

In addition to the above points, some more efforts must still be made to reduce solution time (e.g. using segregated solver) and to refine mesh especially around contact areas.

6 Next Steps

The results shown above prove COMSOL Multiphysics as a valuable tool for the analysis of Rzeppa CV-joints. For a more accurate analysis of a real joint, however, several steps must still be done. We will try to sketch out the most significant ones.

First, material is not the same for all the components. Moreover, as already pointed out, all components of a real CV-joint are hardened (induction or case hardening) and for this reason materials are affected by relevant non-uniform pre-stresses (especially induction hardened components). To have a reliable failure index as a result of simulation, the pre-stress state must be put in the model (e.g. by simulating case and induction hardening) and the model must include the shaft.

Second, in a real CV-joint, friction among components is not negligible (for this reason all components are lubricated with grease) and reduces joint efficiency. An estimation of friction-induced power loss could be of great interest.

Third, components tolerances have an impact on kinematic behavior and on stress conditions. Again, an estimation of this impact could be of great interest.

Fourth, constraints of a real CV-joint are certainly much more complex than those in the model. This affects only slightly the kinematic behavior but has a heavy impact on stress conditions.

7 Conclusion

The results shown in this article are to be considered as very preliminary and they are certainly susceptible of great improvements, but in our opinion they look promising.

The kinematic is already correct and a first significant improvement of components' stress values and in volume distribution could be easily achieved simply by refining the mesh.

A second step, far more complex, is the introduction of real materials (taking care of hardening effects) and more realistic geometry, constraints and loads.

References

[1] *Universal joint and driveshaft design manual*, The Society of Automotive Engineers, Inc., 1979.

[2] Hans Christian Seherr-Thoss, Friedrich Schmelz, and Erich Aucktor, *Universal joints and driveshafts - analysis, design, applications*, Springer-Verlag, 2006.

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