HIGHLY ACCURATE ELASTIC CONTACT IN MOVING STRUCTURES

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THEME

FE analysis with contact update

SUMMARY

Drive trains contain several joints which have to transmit high loads. Therefore, not only the motion of joints is of interest but also stresses and strains have to be predicted during development. Inside joints contact is of particular interest, because contact surfaces are transmitting loads and motion. In addition, load history with assembly and several load cases should also be investigated.

In order to perform all these effects in one analysis without additional modeling effort, updating of contact plays the leading part. An industrial universal joint is taken as an example to show the relevance of updating in contact of moving structures. Moreover, adequate post-processing to visualize both motion and transmission effects will be highlighted. Finally, efficiency of the updating process and detailed result comparison will be presented.

KEYWORDS

FE analysis, contact analysis, contact update, contact partner changes, contact direction update, contact pressure, analysis process, non linear load history, modelling and computational performance

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1: Introduction

In the virtual development of structures based on FE analysis the combination of efficient high accurate contact analysis with contact update is strongly needed. This method promises more accurate results without additional modelling effort.

During the past 20 years the FE based contact analysis was established in the industry to analyze the behaviour of assemblies with several parts. These parts are connected in many assemblies by contact. The special feature of contact never transmits tensile forces but can transmit unlimited compressive forces between two bodies. For detailed analysis of the contact behaviour also algorithms for the evaluation of frictional forces are implemented.

The stresses and strains of flexible parts are of major interest to improve the part performance. During taking contact into account and checking of stresses and strains in more detail, the models also become finer. From this follows, that the number of unknowns increases. The two boosters behind the model enlargement are the higher accuracy of finer meshes [1] and the less human resources that are needed for automatic meshing. In the past a lot of human working time was spent to create special geometry adaption for FE analysis and for meshing with special emphasis to reduce the number of degree of freedom (DOF) for universal joint models (see Figure 1).

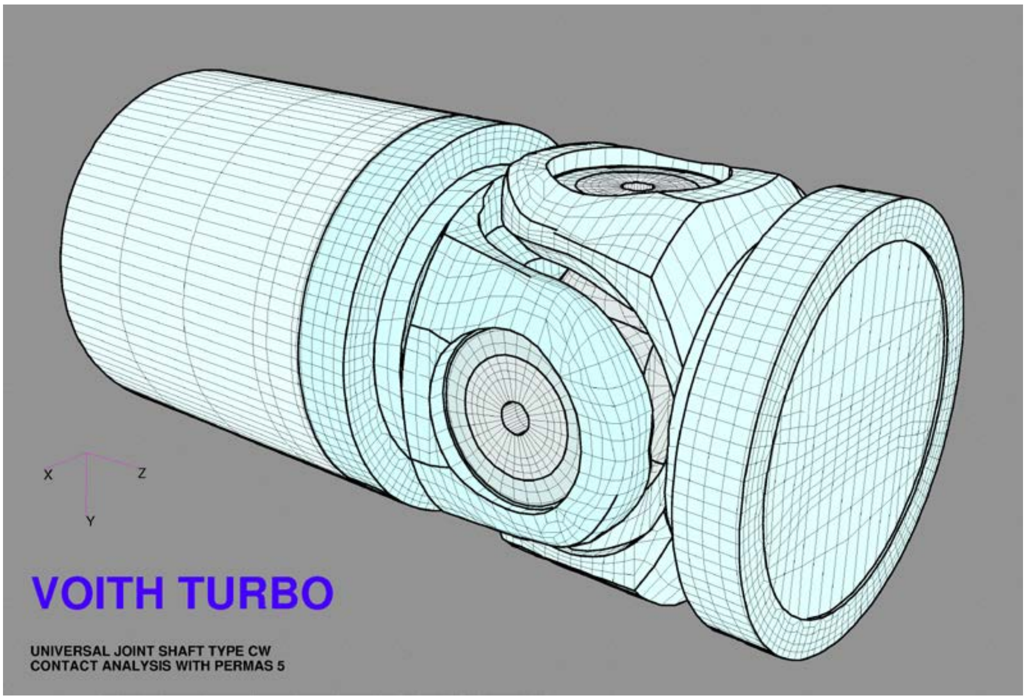

Figure 1: Universal joint shaft model from 1995 [2].

Today the original CAD geometry and automatic meshing in combination with manual meshing is used for universal joint [2]. Only with usage of the incompatible coupling feature this results in drastically reduced human working time to create a mesh (see figure 2).

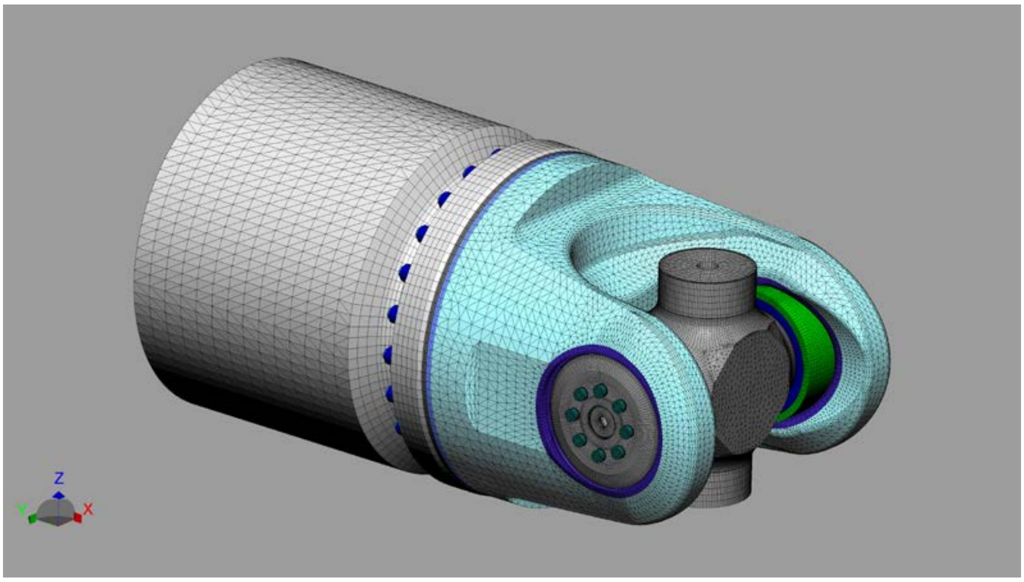

Figure 2: Current universal joint shaft model.

The quality of the results depends on the fine mesh and on the accuracy of the contact algorithms. The next step to raise the quality of the results is from the static analysis based on only one given configuration of the assembly to an investigation of the assembly of the different configurations during rotation. For the universal joint shaft the relative position between the journal cross and the yoke changes.

The precondition for this kind of analysis is to deliver the results in the same high quality as for an assembly in one single position. From this it follows that the FE analysis software has to be extended with this capability.

In the following sections a new solution based on the existing flexibility method for contact in PERMAS will be shown. For the example of an universal joint shaft the basic kinematics, the process of the FE analysis with contact update and comparison of the results are demonstrated.

2: Contact Analysis with Contact Partner Update

Several methods of solution schemas for the numerical treatment of contact boundary conditions have been developed in the past, e.g. Lagrangian para-meters, penalty functions or staggered u/p iterations [3, 4]. PERMAS uses a slightly modified flexibility method which exactly simulates the discontinuity of the contact area. Furthermore, the method shows an excellent efficiency [5].

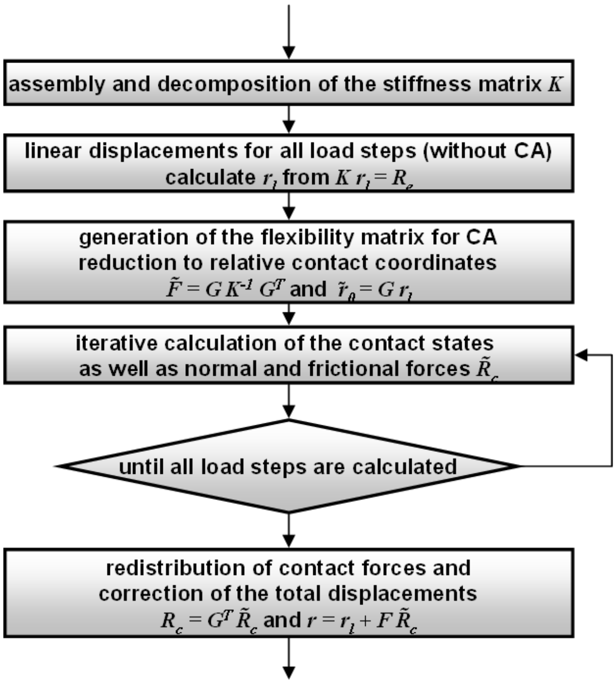

Figure 3: Flexibility method in PERMAS [6].

Figure 3 shows the complete algorithm for a linear static analysis of several load steps with contact. Starting with the global stiffness matrix $\boldsymbol{K}$ and the applied forces $\boldsymbol{R_e}$ a linear-elastic solution $\boldsymbol{r_l}$ is calculated in a first step without consideration of the contact boundary conditions. This solution is then transformed to a significantly smaller system which contains only the relative displacements of the potential areas of contact. A condensed flexibility matrix $\boldsymbol{F}$ is then built for the contact system. During subsequent iterations the contact is closed or opened at all potential locations, respectively, until penetration is compensated by reaction forces and a state of equilibrium is reached. Finally the contact forces are transformed back into the original displacement coordinate system and the global displacements are corrected by the relative displacements of the contact zones.

Part of the classic flexibility method is the calculation of contact partner and contact normal direction from the original undeformed state. Figure 4 presents in the upper part a two dimensional example for an undeformed configuration. Contact is defined between two nodes of an element and the smoothed surface as contact partner. Nadir points, the points on the surface with the shortest distance in normal direction of the surface, are computed once at beginning, before the time consuming solution process starts. Any solution based on this configuration is correct for small relative displacements $\boldsymbol{r_i}$, only.

The lower picture of figure 4 shows the loaded configuration with respect to the initial contact geometry of the contact partner for a relative displacement $\boldsymbol{r_i}$ that is too big for the assumption of small relative displacements.

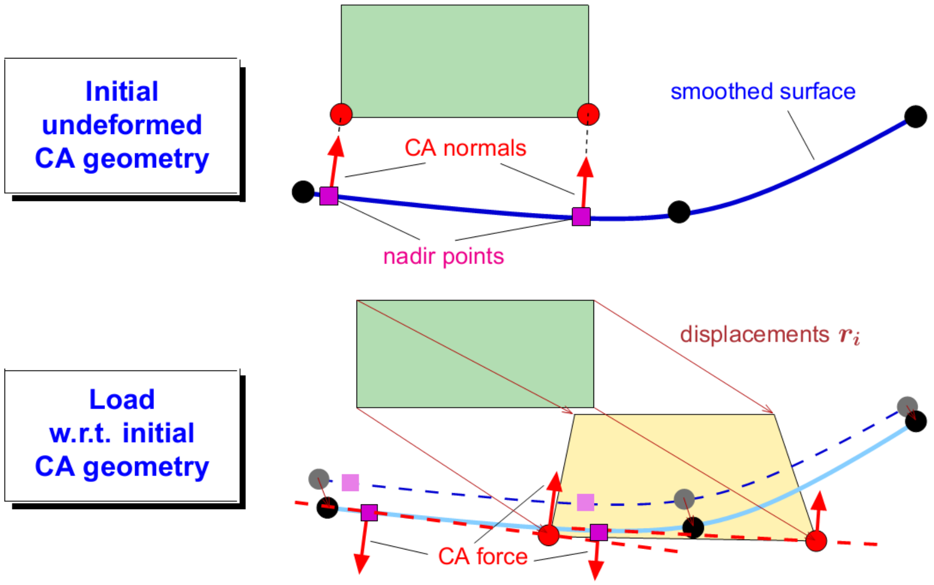

Figure 4: **Undeformed contact geometry and load [7].**

Here, neither the normal direction nor the location of nadir points matches the deformed configuration anymore. In the given example the deformed configuration is represented by the yellow element and the light blue continuous line for the surface. The inaccurate initial assumptions lead to a considerable penetration with contact forces in wrong directions at the wrong location. Also, the pairs of contact normal forces have the same value and opposite direction, this is correctly solved. But they are no longer in-line.

Figure 5 shows one single iteration of the contact update with two major steps. In the upper picture the configuration is based on the first displacement result with respect to the initial contact geometry. Then all nadir points and the normal direction for the contact of the nodes on the surface are updated according to this first intermediate result. After that the displacements and contact are recalculated with respect to the new contact geometry. In the picture the new configuration is represented by the light red element and the little bit darker continues blue line. The old surface keeps the same colour and is now dashed. In this step the difference in the contact normal directions between this new configuration and the configuration of the step before is much smaller.

This demonstrates the convergence of the contact geometry update. In general the updates are done until a convergence criteria is fulfilled or the maximum number of allowed contact update steps is reached. The convergence rate depends on the geometry of the surface, especially on changes of the normal direction, and on the load.

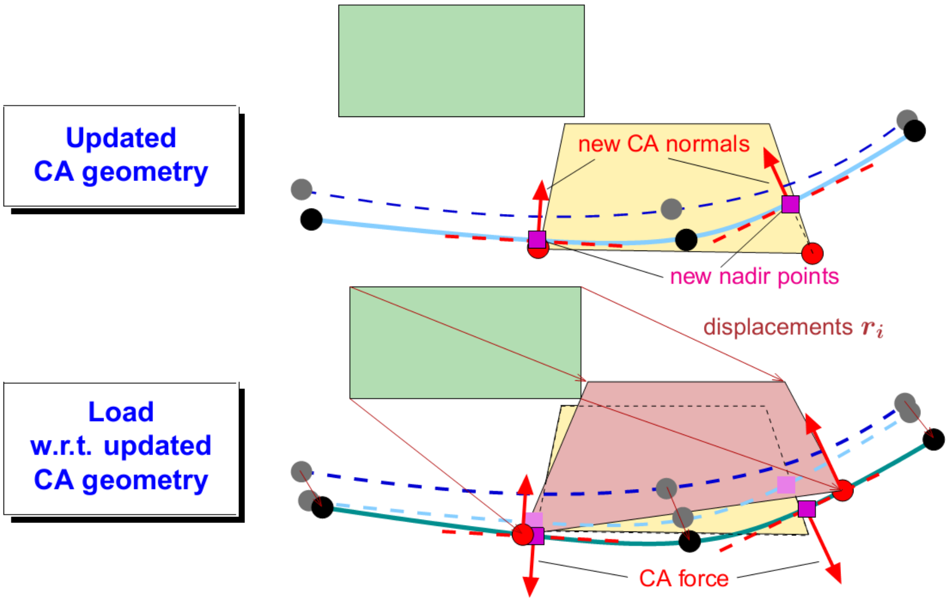

Figure 5: Contact geometry update and load [7].

This process doesn't exclude the other features of FE analysis in PERMAS. Contact geometry update is possible in combination with:

- Friction (isotropic and anisotropic),
- Non linear load history for process definition,
- Non linear geometry and
- Non linear material.

With contact update a relative movement of parts in an assembly is allowed and the high accuracy of FE analysis with contact is fully kept. In combination with the flexibility method of PERMAS drastic algorithmic advantages appear, because for linear static analysis with small displacements the stiffness matrix can be reused and only the smaller flexibility matrix has to be updated and solved several times until the contact geometry update has converged.

3: Load and Kinematics of Universal Joints

As introduction for the example the main kinematic effects and the loads of the universal joint are summarized here. The task of a shaft is to transmit high torque. But, if a deflection angle or a parallel offset is required, universal joints are needed. Universal joints are used only in pairs, because of the kinematics.

Technical terms for the explanation of the kinematics are defined in figure 6.

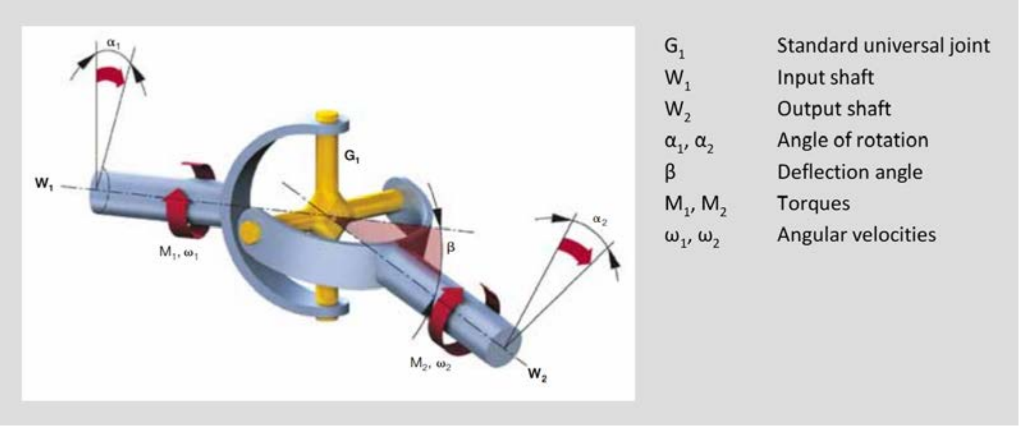

Figure 6: Name definitions for universal joint [8].

With a deflection angle ß > 0° the rotation speed of input and output shaft is no longer the same. Only at four singular points during one full rotation the speed is the same. If the input shaft W_1 is rotated with a constant angular velocity ω_1, the angular velocity ω_2 of the output shaft W_2 has a sinus characteristic. Hence the angles of rotation are also different. This difference is known as the differential angle $\varphi=\alpha_1-\alpha_2$. The differential angle has sinus characteristics, too.

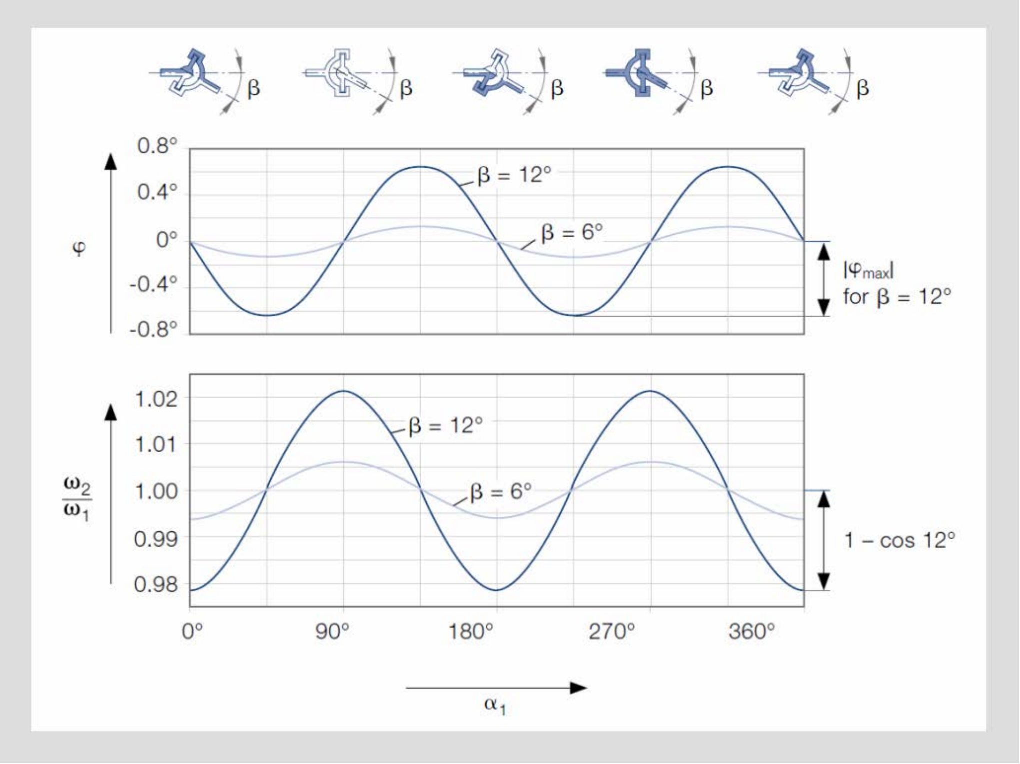

Figure 7: Movement relation between input and output shaft [8].

Differential angle change and difference between two angular velocities are shown for a full rotation of 360° in figure 7. This so called gimbal error has to be taken into consideration for the analysis of universal joints. For a deflection angle of 0° one single static analysis contains all strain and stress distributions during operation. But for all deflection angles ß > 0° the strain and stress distribution changes during one rotation. The size of the differential angle φ and the difference between the angular velocities grows with the deflection angle.

All effects of the deflection angle are described here until now only quasi-static, only based on the geometry and without taking into account the amount of the angular velocity. But the real angular velocity has to be taken into account, because the size of angular acceleration and of angular deceleration depends in addition on the velocity. Thus the amount of variation in torque during one rotation depends on the angular velocity.

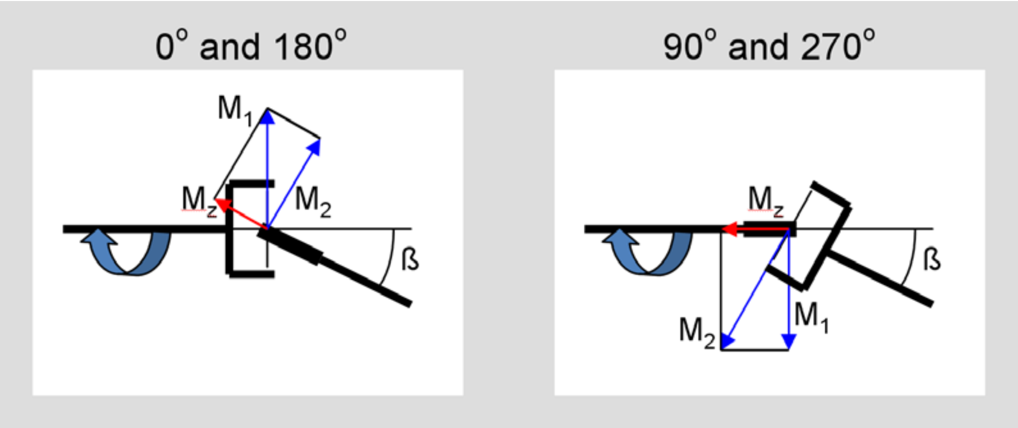

Figure 8: Extrema of the additional bending moment M_z.

Additional bending moment occurs from the deflection of the torque in the joint. This bending moment also has a sinus characteristic during one rotation of the shaft and acts as additional load for the shaft (see figure 8).

4: Example: Universal Joint Shaft

For the FE analysis in PERMAS with contact update an industrial universal joint shaft FE model from Voith Turbo GmbH & Co. KG is used. The characteristic of the model is as follows:

- Nodes 1,168,947
- Elements 954,884
- Multipoint Constraints 23,603
- No.of.DOFs 3,332,339
- Contact DOFs 49,141 (in 53 contact regions)

In figure 2 the model is presented with element edges. State of the art static analysis with high accuracy contact and pretension of two different groups of bolts as two assembly steps is established for this kind of model since several years. As boundary conditions the shaft has realistic displacement constraints and the torque is applied at the free boundaries of the journal cross. In figure 9, 10 and 11 the reference results are pictured. All results are scaled by confidentiality reasons to values between zero and one.

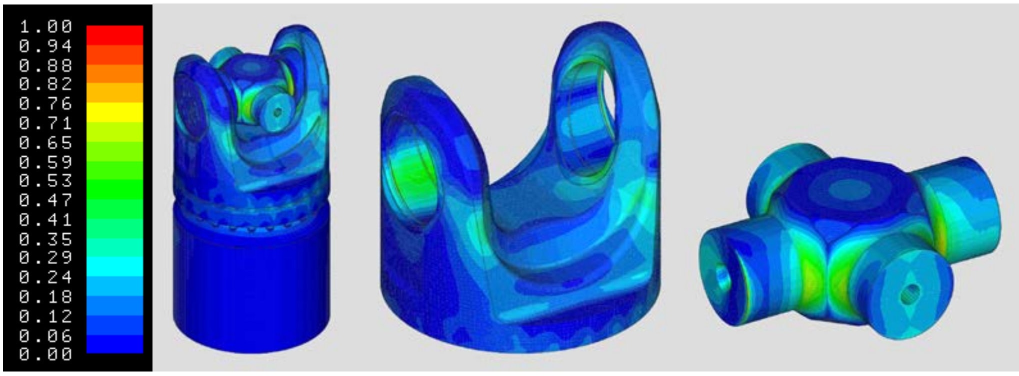

Figure 9: Reference v. Mises stress distribution: complete model, yoke and journal cross.

Important high stress areas are mainly at the outside of the yoke in the transition from the bores to the more solid flange area. There are considerable differences between the more tension and more compression loaded sides. High stresses appear at the journal cross in the area between the cylindrical connection regions and especially in the transition to the cylindrical areas.

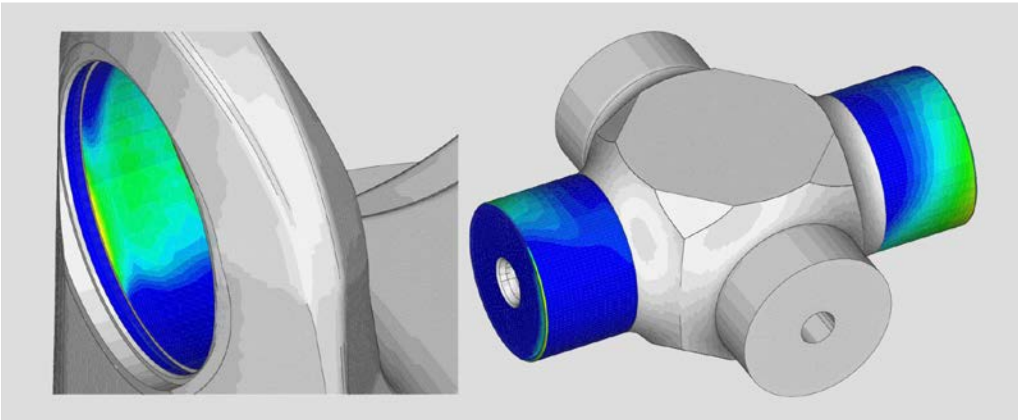

Figure 10: Reference contact pressure: yoke and journal cross.

Contact pressure, figure 10, clearly shows the high loaded areas. Edge pressure and distribution inside the bore of the yoke are very characteristic. Also the cylindrical contact area of the journal cross has contact pressure distribution that is expected from the torque load. Both parts of this figure have the same colour scale.

Normal contact forces, figure 11, prove this behaviour. They give a visual feedback of the direction, but the forces must be analysed carefully, because their size is discretization dependent. The support of the journal cross in axial direction becomes apparent.

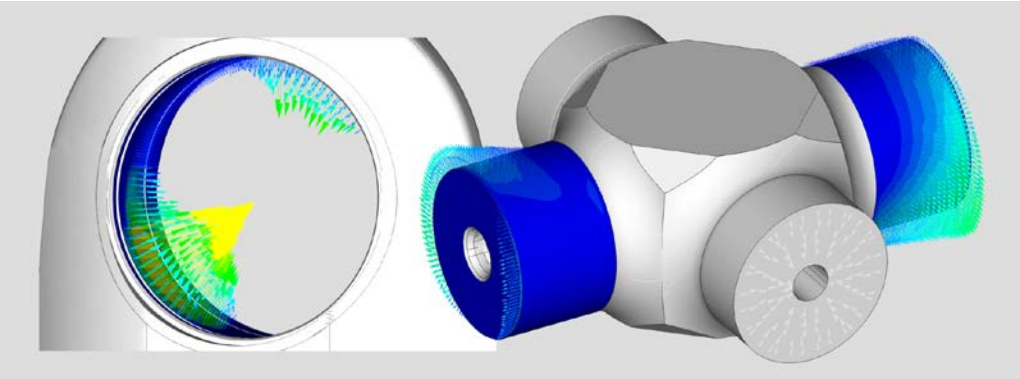

Figure 11: Reference contact normal force: yoke and journal cross.

The next step of improvement for the FE analysis is to add the movement of structures to the high accurate contact analysis. For the universal joint the movement is rotation of the shaft. With a rotation of 180° the complete spectrum of conditions is covered. The exact and correct analysis of the rotation is only possible with contact update.

The same assembly steps as in the reference analysis are done at the beginning of the analysis with contact update. Figure 12 shows the nonlinear load history for the complete process with assembly, torque load, pretension and rotation of the shaft. The rotation is added as additional time steps in the artificial time space from 3 to 5. Analysis angle steps of 3° are used to check all intermediate configurations.

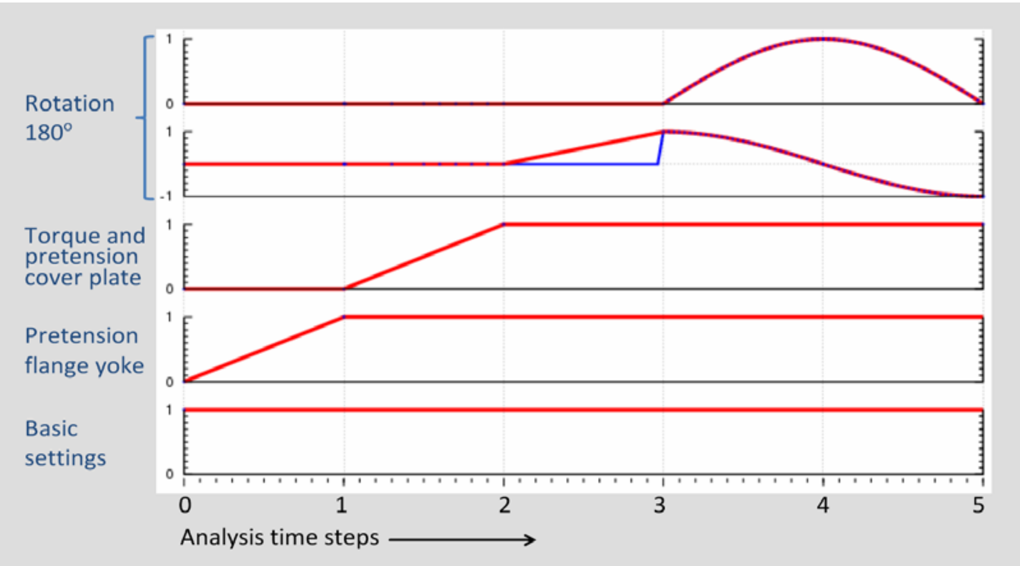

Figure 12: Nonlinear load history for universal joint shaft with 180^0 rotation.

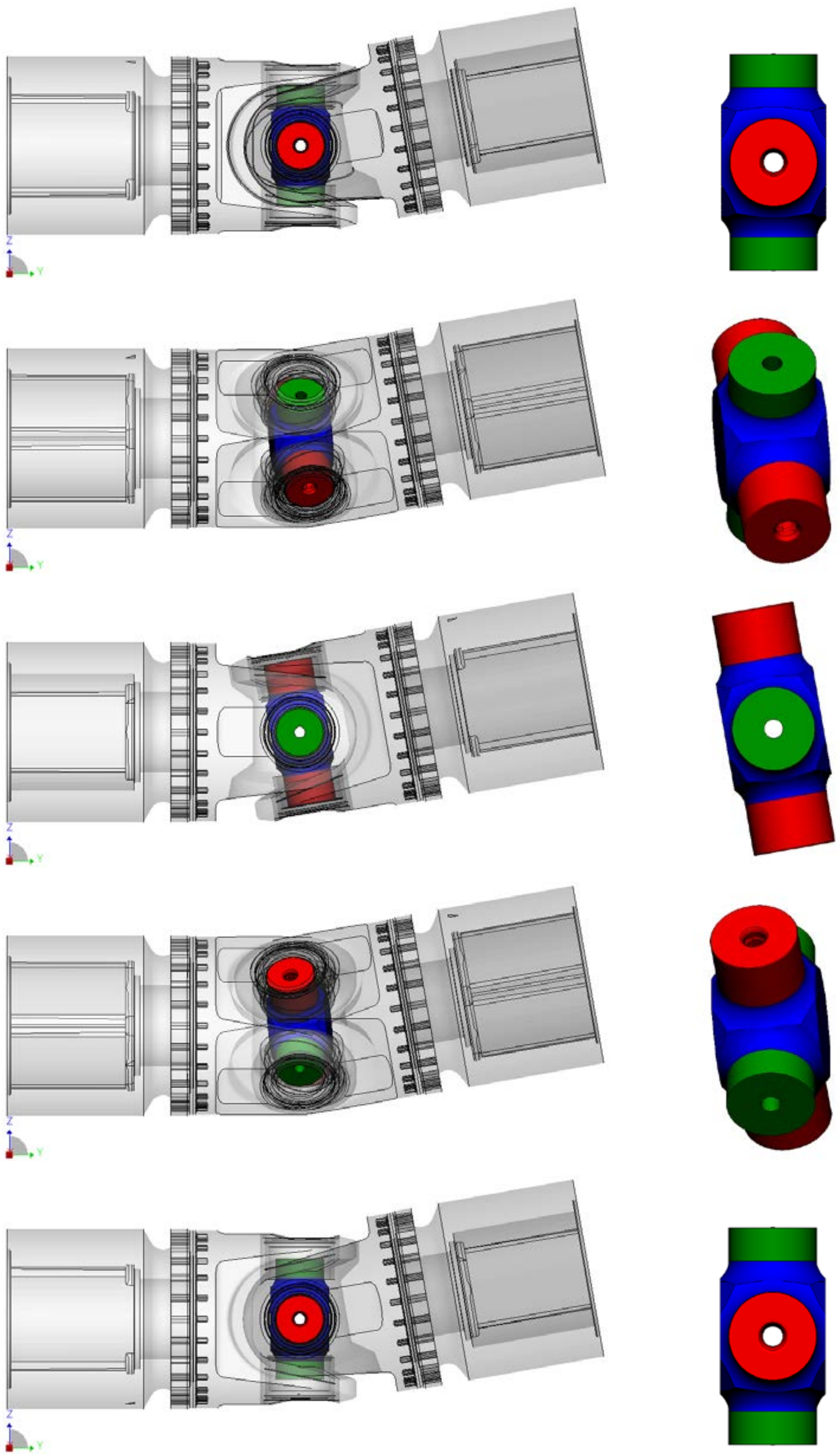

Figure 13: Universal Joint Shaft (transparent) and journal cross with deflection angle ß=10° in five positions 0°, 45°, 90°, 135° and 180°.

Between independent single analysis of several positions and the here described process with contact update there are several considerable differences. Analysis with contact update has the following advantages:

- One model for the complete analysis of several positions,
- Assembly process has to be calculated only once at the beginning (like in reality),
- Correct non linear load path,
- Influence from former position to next position,
- Friction effects from rotation process and
- Elastic-plastic deformations can be investigated meaningful only with rotation process.

As explained in the section about the kinematics of universal joint shaft, the rotation changes the load conditions, if the deflection angle ß is greater than 0°.

For the given universal joint the maximum allowed deflection angle is ß = 10°. Hence, this is used for the analysis, because the maximum deflection angle changes the conditions during the rotation most.

Figure 13 shows the different positions of the journal cross during 180° rotation in 45° steps. The deflection angle is constant during the rotation and fixed to 10°. A journal cross is connected to two shafts. Both connections are marked with different colours (connection to the left in green, to the right in red). Both coloured parts rotate on their own circular path, but the circular paths, that the green and the red part describe, are, according to their shafts, inclined by 10°. For the analysis the yoke and shaft connected to the green areas are replaced by a rigid element. With the rigid element the results can easy be compared with the reference solution and the efficiency of the analysis is very high.

For the detailed analysis of the results relative results are shown in the figures 14 to 16. All changes are depicted with reference to the static result that is without rotation and without deflection angle. Again the results are scaled to values between zero and one.

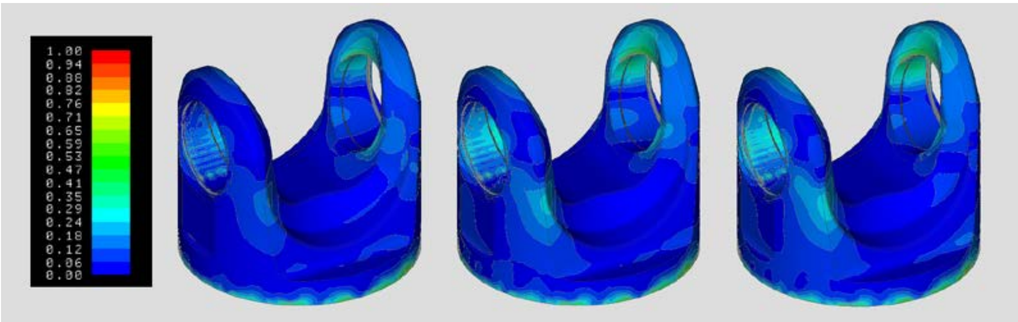

Figure 14: Change of v. Mises stress in yoke, 30°, 60° and 90° rotation.

The change in v. Mises stress of the yoke gives information about variations in stresses during the rotation (see figure 14 and 15). Considerable changes of stress are at the inside of the top of the yoke, on the outside at the transition from the bore to the flange and at the flange in the region of the bolts with pretension to the shaft. The difference also grows from 30° to 90°, because 90° is the position where the journal cross reaches the maximum 10° deflection angle.

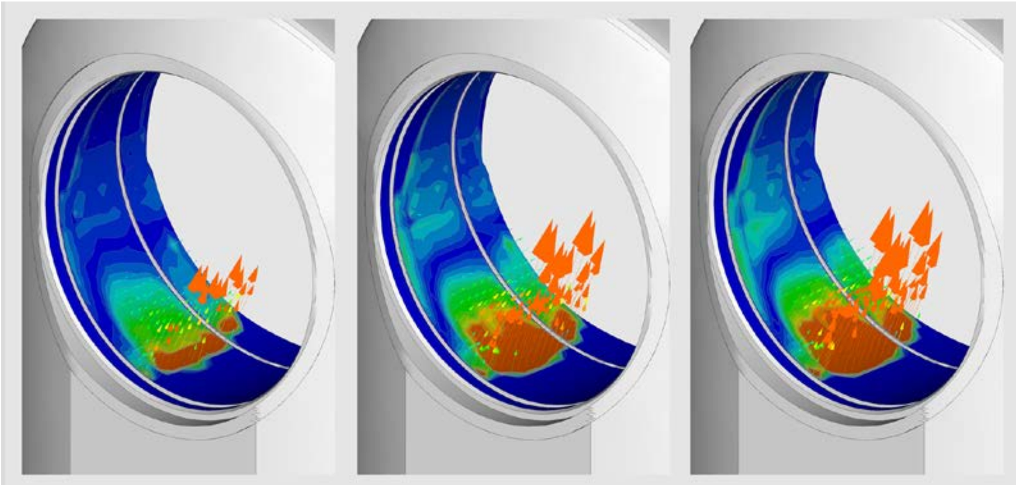

Figure 15: Change of contact force in journal cross, 30°, 60° and 90° rotation.

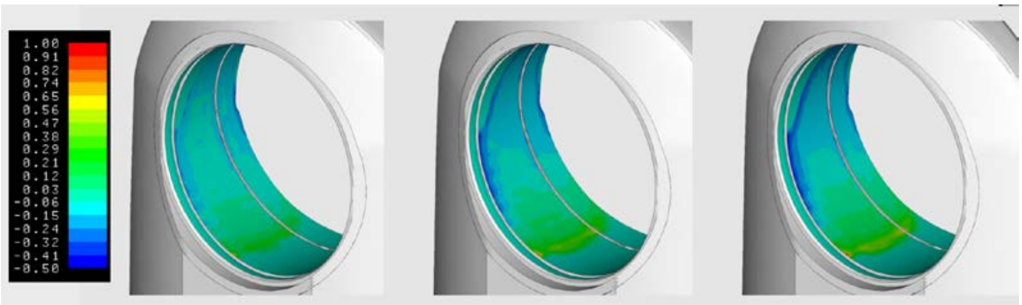

Figure 16: Change of contact pressure in bore of yoke, 30°, 60° and 90° rotation.

In addition the contact behaviour inside the bore of the yoke is in of interest. Contact force and contact pressure show both considerable growing during the rotation. For scalar values, like contact pressure, also negative values appear in regions with relief. Here a different colour table is used. The reason for the variations is the change of contact partner in the contacts between bore, bushing, roller bearing and journal cross.

The results demonstrate that considerable variations in geometric relations between parts are covered by contact update. The complete elastic model without any simplification and thus with the full accuracy from FE analysis results can be used.

5: Conclusion and Outlook

Moving structures in combination with highly accurate elastic contact improve the quality of results. Maxima and minima can be located for different states. Additional effects that only occur by moving structures with contact update, like changes of stresses and stress amplitude during rotation give additional information. The quality of the behaviour prediction is raised to a new level.

At the same time the modelling effort is reduced in comparison with the analysis of several configurations with several models. Only one single model is required, if contact update is used.

This kind of analysis is a complete new class of analysis, because this is the first time that complete FE models of assemblies from industry with a lot of contact regions and with big relative displacement can be analysed with movements without losses in elastic behaviour. Before only static analysis of this detail level with a lot of contact areas could be analysed in this way.

For the future there are more physical effects that could be analysed in detail. E.g. the effect of the real velocity and the resulting acceleration and deceleration is not taken into account until now. But it is easy to add this by a sinus function as additional torque load.

A detailed investigation of the effect from “beginning rotation failure” is planed also. The reason for this failure is that at the beginning the analysis starts the rotation from a configuration without any rotation before. It is expected that this effect depends on the friction coefficients and will disappear for the universal joints after an angle less than 10 degree of rotation.

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