The driveshaft is the part that transmits the vehicle’s engine torque and rotation to the tires, and predicting the stress of each component in the CVJ, a major part within the driveshaft, is important to creating a strong CVJ design. However, because contact between each such component is complex, a formula for effectively calculating such stress has not been established. This paper introduces a stress analysis model that, with the aim of developing an effective design tool, we constructed utilizing the finite element method and taking contact between components into consideration. Validation results are also presented.

Key Words: constant velocity joint, computer aided engineering, finite element method, stress analysis, contact

1. Introduction

The constant velocity joint (CVJ), one of which is mounted at each end of the driveshaft, has the functions of transmitting engine torque and rotation from the differential gear to the tires and also expanding and contracting in order to absorb steering-caused power transmission shaft bending and axial displacement (Fig. 1). In order to design adequate CVJ strength, it is necessary to predict the stress situations of each component during vehicle operation. However, the contact load exerted on each component varies in accordance with rotation of the power transmission shaft. This makes it important in regard to designing to clarify maximum load conditions. In theoretical studies, it has not been possible to study the influence of clearances between components or elastic deformation, and such studies have not conformed with actual phenomena, which has been a problem.

On the other hand, the representative CAE method, the Finite Element Method (FEM) is an approximation method in which object characteristics are shown by a number of representative points. Recent advances in calculation technology have made it possible for FEM to handle nonlinear issues such as contact because of no restriction on model shape.

This report presents examples of efforts to analyze contract stress by FEM with the aim of simulating CVJ load situations during rotation, which has been a problem in designing, and constructing a design tool for evaluating component stress. As a study subject, a cross groove type VL series, used mainly in high-class vehicle rear wheels, has been selected from among JTEKT driveshaft products.

2. Outline of CVJ

Figure 2 shows the structure of a VL series CVJ, the analysis subject. The VL series transmits torque and rotation from the outer-race to the inner-race (differential gear side) or from the inner-race to the outer-race (tire side) through contact between the inner-race and outer-race ball grooves and balls. Ball sliding in the inner-race and outer-race ball groove direction enables an angle to be formed between the inner-race shaft and outer-race shaft, which enables bending of the rotation transmission shaft. Constant velocity between the input shaft and output shaft is maintained by maintaining the positions of multiple balls on the plane bisecting the rotation transmitting shaft bending angle (crossed axes angle). As shown in Fig. 3, on the bisection plane of the crossed axes angle, the distance $r_i$ between the effective load center and input shaft, regardless of crossed axes angle, is always equal to the distance $r_o$ between the effective load center and output shaft with rotational speed of both shafts equal.
Contact load between the balls and other components during rotation is shown in the schematic diagram of Fig. 5. Contact load generated between the inner-race and outer-race during rotation is exerted in the direction bisecting the angle formed between the inner-race and outer-race, in which each groove is tilting, so as to push out the balls. This ball push-out load is exerted on the retainer window area. The direction of the load applied to the retainer changes for every other window because six pairs of ball grooves are crossing each other obliquely. As a result, the two load sets whose directions oppose each other are balanced. Each CVJ component transmits rotation in this equilibrium condition.

3. Outline of Analysis

3.1 Model Profile

As FEM is an approximation method, solutions derived by calculations are preconditioned on micro-deformations, and large behaviors such as rotation cannot be handled. Therefore, our study in this report was conducted by the static deformation analysis method. As shown in Fig. 6, for the analysis model, a shape created by a 3D CAD comprising an inner-race, an outer-race, a retainer and six balls was used. The crossed axes angle is 5 degrees clockwise against the X-axis in the figure with the outer-race arranged at a slant. Balls are arranged such that the center of two balls is located on the Y-Z face and also at even intervals on the bisection plane of the crossed axes angle (the X-Y face is rotated 2.5 degrees clockwise around the X-axis). The retainer is also inclined at 2.5 degrees around the X-axis to adjust to the ball position.

Although the above arrangement is standard, this can correspond to any crossed axes angle and rotation phase angle against the Z-axis by preparing profile data with coordinate conversion function of 3D CAD.
3.2 Contact Areas

Contact between the inner-race/outer-race ball grooves and balls and between the retainer window and balls taking friction into account is defined. Figure 7 shows an example of a pair of contacting surfaces. In actual modeling work, at the time of CAE data reading, contact is automatically defined on the adjacent surface by the CAE software. The friction coefficient was adjusted with stress measurement results described in the following section.

Fig. 7  Inner-race and ball contact surface

3.3 Load and Boundary Conditions

The model is fixed so that the cylindrical surface of the outer-race edge is fully constrained, and inner surface of the inner-race is constrained so as to rotate freely. At the same time, torque is loaded on the constraining surface of the inner-race (Fig. 8).

As the retainer and balls are positioned by their contact, they are not constrained in the CAE model. Therefore, when they are arranged in non-contact with any other components or conversely in a state of excess interference, calculation is hard to converge. To avoid this, when making the profile by 3D CAD, profile data is prepared by checking the interference, etc.

Fig. 8  Boundary condition

3.4 Element Meshing

The FEM model was meshed as shown in Fig. 9. In the future, in order to be usable as a design tool by designers with little experience in CAE software, the model is meshed by tetrahedrons using the automatic meshing function of CAE software, and a high-order element is used so as not to decrease analysis accuracy. The high-order element is such that nodes are at the top of each tetrahedron and the center of each edge, and stress, displacement, etc. are calculated there. (The lower-order element has nodes only at the top of each tetrahedron.)

Fig. 9  Finite element mesh
4. Analysis Result

The test equipment shown in Fig. 10 was used so that values at stress concentration areas of an actual product became almost equal. Strain under torsional load was measured with strain gauge put on the position shown in the figure. The retainer bar is one of the weakest positions of the CVJ and most susceptible to breakage.

Figure 11 shows stress distribution on the retainer inner peripheral surface due to loaded torque. Stress at the strain gauge position after adjustment by the friction coefficient explained in the previous section is as shown in Fig. 12. Values are generally consistent, although slight deviation is seen in the high-torque regions.

Figure 13 shows the distribution of surface pressure caused by contact between the balls and retainer window. The points in the ball middle areas with slightly different coloring are positions where surface pressure is high due to the contact with the retainer. As described in Section 2 hereof, of the six balls, every other ball contacts the facing retainer window surface, and each contact load pulls the retainer bar. As seen from this result, this model can simulate the contact condition of a cross-groove type CVJ.
5. Confirmation with CVJs of Different Sizes

Friction coefficient adjustment enabled the establishment of a model that can simulate actual product stress situations within about 10% of torsional test results. Then, a study was conducted to investigate whether this model can be applied to other models of different sizes and how this analysis method can be utilized in designing. By torsional fatigue testing, the relationship between loaded torque and the number of loaded cycles when the retainer breaks (T-N diagram) was investigated. Generated stress and the time of breakage were plotted (S-N diagram) in correspondence with the relation between torque and generated stress in this analysis. The result is shown in Fig. 14. The T-N diagram shows the smaller size model A breaks at lower torque than the larger model B, which shows that the strength of model A is lower. The S-N diagram shows the stress of models A and B is almost on the same line, indicating that for two types of CVJs with differing sizes, the same stress situations as in the case of actual products can be calculated.

If stress is calculated by this analysis method and applied to the S-N diagram shown in Fig. 14 to estimate the retainer’s fatigue strength limit, optimal designing becomes possible without repeated sample manufacturing and testing, as well as reduction of size and weight.

6. Further Activities

The study of the previous section was carried out in low cycle regions, but this study will be expanded in the future to high cycle regions and actual product behavior investigated.

While we studied a cross groove type CVJ this time, we hope to continue our studies focusing on a ball joint type CVJ, a more general type. The ball joint type, which has a structure wherein the ball grooves are parallel with the rotation transmission shaft and form a crossed axes angle larger than that of the cross groove type, is used mainly in the front wheels of FF vehicles. It is experimentally known that as the crossed axes angle increases, retainer stress likewise increases, and therefore needs for evaluation by CAE are increasing.

7. Conclusion

Retainer stress analysis has conventionally been carried out on individual retainers under theoretically assumed load conditions, and the main method of evaluating results was comparison with the analysis results of products whose performance evaluation had been judged acceptable.

In this study an assembly model was used, and by accurately considering load transmission between components and comparing with actual product testing, a possible means of stress absolute value evaluation using CAE was found. While striving to strengthen the calculation environment, we hope to establish an analysis method for evaluating the characteristics of CVJs and other driveline products.

Fig. 14 VL type driveshaft T-N and S-N diagrams