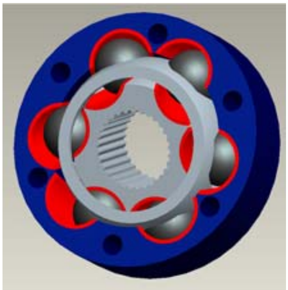
Contact stress analysis and optimization design of ball plunging CVJ



Lohrengel, A.; Shen, L. J.

Most of the studies of ball type CVJ in the past are focused on the contact strength and fatigue of the ball-track contact zone, this paper will pay more attention to the area between ball-track contact interface and spline shaft-hub connection interface, i. e. the inner race of the joint. Some new ideas that how to optimize it's design will be presented in this paper.

1 Introduction

The annual production of constant velocity joint(CVJ) for use in drive shafts currently totals more than 2 billion, and CVJ remains a highly profitable business in the near future. With growing demands to improving fuel economy, performance, durability and drivability of vehicles, demands for CVJ to be made smaller and more lightweight, more durable, and better NVH performance are increasing rapidly/1-3/. Among all kinds of CVJ, ball plunging CVJ are often used as inboard joints in side shafts of FWD cars, side shafts of RWD cars with independent suspension and other industrial machines to accommodate the change in axle length, at the same time, transmit torque uniformly with joint articulation. Much attention has paid to the ball-track dynamic contact interface in previous studies, Carsten Bauer investigated the dynamic contact stresses, ball movement, internal friction and the influence of induction annealing to the contact fatigue of ball plunging CVJ via analysis as well as experiments/4/.Current static design criteria is that the plastic deformation near the ball-track contact interface due to Hertzian stress do not exceed d*0.01% (P_0<4000N/mm^2) /5/.But, for the inner race, failures may occur rather than the ball-track contact area because of the combination effects of ball-track contact stress and spline shaft-hub connection contact stress, and which becomes very concerned when people want to make it smaller and more lightweight.

2 FEM analysis of inner race

Tracks of the joints are case hardened or induction hardened, the highest yield strength of the hardened layer can reach 2500N/mm^2, much higher than the core, which is about 900 N/mm^2. T.Harris/6/ and Paland E.-G./7/ have drawn guideline that the thickness of hardened layer must enough to avoid plastic deformation. And vice versa determine the static torque capacity of the joint according this guideline. Because the designed rating torque capacity is usually 50% of the static torque capacity of the joint, which make sure the deformation of under the contact interface within the elastic limit. So, the homogeneous assumption of the hardened layer makes no difference even the hardened layer is plastically graded (the yield strength decrease with the distance from the surface). The unarticulated joint is cyclic symmetrical, so only 1/3 of the whole joint FEM model is set up to proceed contact analysis in order to alleviate the difficulty of calculation, the cyclic symmetrical FE model shown in **Fig 1**.

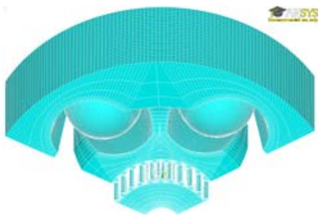


Fig. 1: Cyclic symmetry FEM Model

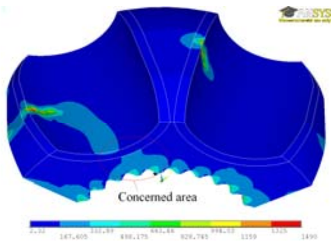


Fig. 2: Von-missies stress of inner race

The FE analysis results showed that the maximum contact pressure in ball-track contact zone is 2285N/mm^2, which is proved by the Hertzian analysis results. Though the maximum stress always occurred in the contact zone, but it remains within the strength limit because of case hardening or induction hardening. As showed in **Fig. 2**, failure may occur in the weakest area between spline and tracks (concerned area), but it greatly depends on the geometry parameters of the joint.

3 Optimization design of inner race

3.1 Ratio of out/inner effective radius

Main geometry parameters of ball plunging CVJ include: ball diameter **d**, effective radius **R**, track radius $\mathbf{r_L}$, pressure angle α, skew angle γ, reciprocal conformity value ψ, number of balls **z** and pitch circle diameter of spline **DP**. In most often used design: $\alpha = 40^\circ$, $\psi = 16^\circ$, $z = 6$. Among remaining parameters, $\mathbf{r_L}$, **R** and **DP** will affect the stress of the concerned area, but $\mathbf{r_L}$ is closely related to **d**. Here, we define a non-dimensionalised parameter **f**:

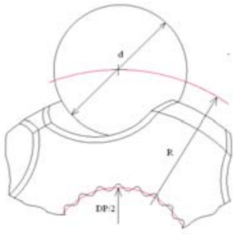


Fig3. Parameters to be optimized

$$f = \frac{\dfrac{R}{d/2} - 1}{\dfrac{DP/2}{d/2}} = \frac{2R - d}{DP}$$

Then, a threshold of **f** will be determined via FEM analysis aimed to fully use the load capacity of concerned area. Because the pitch circle diameters of spline are discrete according to DIN5480, **f** is also discrete.

3.2 Profiles other than spline connection

Polygon profiles are a complete replacement for other profiles, splined, keyed etc. Compared with most traditional forms there is no notching effect and hence no influence on inertia. And it is believed that the polygon profile has more than 30% greater fatigue resistance than similarly sized alternative profiles. Along with the achievement of

Fig. 4: Polygon profile (n=6, e=0.3)

Fig. 5: Complex cycloid TO2 profile (n=6, e=0.15)

economic manufacture of polygon profiles, studies and applications of polygon profiles becomes more and more popular recent years/8/. Integrating characters of polygon and traditional spline profiles, Ziaei, etc. presented a complex cycloid profile/9/. In view of structural characteristics of inner race, both of the profiles mentioned above (shown in **Fig. 4** and **Fig. 5**) are possible to improve the strength of concerned area and therefore improve the torque capacity of the joint compared with similarly sized spline connection. At the same time, the mean diameter of the profile is continuous, which gives more freedom in optimization of **f**.

4 Conclusion

Other than the contact zone, more attentions are paid to the wick area between ball-track and spline contact interface, a non-dimensionalised parameter **f** is presented in order to optimize the inner race quantitatively and two kinds of recently popular profiles also introduced to the optimization design of inner race. Future work will verify above ideas via extensive FEM analysis as well as experiments.

5 Literatur

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