Analysis of the Position of the Reducer Intermediate Shaft’s Influence on Bearing Selection

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Abstract. The proper configuration of the bearing not only affects the internal structure and service life of the gearbox, but also determines the service life of the bearing itself. In this paper, through changing the angle formed by the input shaft and the intermediate shaft of two-stage reducer in the transverse position, the model was built for several times. the changes of bearing force, bearing damage degree and fatigue life under different conditions were compared and analyzed, and the optimal type of the intermediate bearing was selected according to the results.

Keywords: Reducer; Intermediate shaft position; Bearing selection.

1. Introduction
Reducer is one of the core parts of the automobile transmission system. Bearing, as one of the key parts of the reducer, who’s performance index has a decisive influence on the performance and life of the reducer [1]. Due to the limitation of the space in actual work condition, the size of the reducer is also strictly limited [2]. Therefore, in order to reduce the radial size of the reducer, the input shaft and intermediate shaft usually form a certain angle in transverse position. But different angles would change the force on the intermediate shaft, so would the bearings on it, which would also lead to the changes of bearings’ damage degree and fatigue life. The proper configuration of the bearing not only affects the internal structure and service life of the gearbox, but also determines the fatigue life of the bearing itself [3, 4]. Therefore, for the reducer models of different transverse angles, we need to choose different types of bearings to bear different forces.

2. Establishment of the Simulation Model

2.1. Basic parameter
Taking a domestic electric vehicle as the example, the design maximum speed is 220km/h and the maximum output torque is 3000N/m. The rated power of the motor is 80kW, and the maximum rotate speed is 12000rpm. The reducer transmission ratio is selected as 12, in detail, the first level is 3.3, and the second level is 3.6.
2.2. Establishment of model

The reducer is composed of box, gears, shafts, bearings and connecting parts, etc [5].

The gear pairs are designed to ensure that it can meet the fatigue strength required by the working condition. The basic design parameters of the gears are shown in Table 1 and Table 2.

Table 1. Basic design parameters of the first gear pair

| Normal module [mm] | 1.7500 | Gear1 | Gear2 |
|--------------------|--------|-------|-------|
| Pressure angle at normal section [°] | 14.0000 | Number of teeth | 23 | 77 |
| Rotation direction of gear1 | Helix left hand | Face width [mm] | 28.0000 | 28.0000 |
| Helix angle at reference circle [°] | 31.0000 | Profile shift coefficient | 0.5000 | -0.5457 |
| Center distance [mm] | 102.0000 | Quality (ISO1328:1995) | 6 | 6 |

Table 2. Basic design parameters of the second gear pair

| Normal module [mm] | 2.5000 | Gear1 | Gear2 |
|--------------------|--------|-------|-------|
| Pressure angle at normal section [°] | 16.0000 | Number of teeth | 21 | 76 |
| Rotation direction of gear1 | Helix right hand | Face width [mm] | 40.0000 | 40.0000 |
| Helix angle at reference circle [°] | 29.0000 | Profile shift coefficient | 0.4000 | -0.4524 |
| Center distance [mm] | 138.5000 | Quality (ISO1328:1995) | 6 | 6 |

Then, the software SolidWorks was used to build the three-dimensional model of gear pairs, build the shafts and select the bearings. The model built was shown in Fig.1 and Fig.2.

Fig.1 Reducer model (Angle 45°)  
Fig.2 Cross section of intermediate shaft

The left bearing on the intermediate shaft is 32307JR, and the right one is 303/28R.

By adjusting the position of the shafts, the model was rebuilt twice, so that the intermediate shaft and the input shaft respectively formed angles of 45°, 30° and 0° in transverse position. The side views of the three models are as shown in Fig.3.
Fig.3 Side views of reducer models (Angle 45°, 30°, 0°)

3. Analysis of Simulation Result

3.1. Fatigue life

The life of bearing is generally expressed by ISO damage degree, ISO TS 16281 damage degree, ISO life and ISO TS 16281 life [6].

The calculation formula of linear cumulative fatigue damage refers to Eq.1.

\[ D = \sum_{i=1}^{n} \frac{1}{N_i} \]

\( N \) - The corresponding fatigue life under the particular load.

The calculation formula of the ISO bearing’s basic rating life refers to Eq.2.

\[ L_k = \frac{10^6}{60n} (\frac{f_i C}{P})^\varepsilon \]

\( n \) - Rotate speed of bearing; \( f_i \) - Temperature coefficient; \( C \) - Basic dynamic load rating; \( P \) - Input power of bearing; \( \varepsilon \) - Life index.

The calculation formula of the ISO TS 16281 bearing’s basic rating life refers to Eq.3 [7].

\[ L_{0t} = \left( \sum_{i=1}^{n} \left[ \left( \frac{q_{cei}}{q_{kee}} \right)^{4.5} + \left( \frac{q_{cei}}{q_{kee}} \right)^{4.5} \right] \right)^{\frac{8}{5}} \]

\( n_s \) - Number of laminae;

\( q_{cei} \) - Basic dynamic load rating of a bearing lamina at the outer ring or housing washer contact;

\( q_{ei} \) - Basic dynamic load rating of a bearing lamina at the inner ring or shaft washer contact;

\( q_{ce} \) - Dynamic equivalent load of a bearing lamina at the outer ring or housing washer contact;

\( q_{ei} \) - Dynamic equivalent load of a bearing lamina at the inner ring or shaft washer contact.

Through the static analysis of the intermediate shaft of the reducer, the damage degree and fatigue life of the two bearings on the 45° model’s intermediate shaft were worked out, which are shown in Table 3.

Table 3. Damage degree and fatigue life of 45° bearings

| Bearing Designation | ISO Damage [%] | ISO TS 16281 Damage [%] | ISO Life [hrs] | ISO TS 16281 Life [hrs] |
|--------------------|----------------|-------------------------|----------------|-------------------------|
| 32307JR            | 0.108          | 0.175                   | 4609.916       | 3857.761                |
| 303/28R            | 0.123          | 0.107                   | 4074.198       | 4690.316                |
As can be seen from the table above, the ISO damage degree and ISO TS 16281 damage degree of the two bearings are both very low and far less than 1. According to Miner's law, it can be determined that the failure probability of the two bearings are very small, in other words, the bearings have high reliability and high safety [8]. At the same time, the fatigue life of the two bearings are similar, which is in accordance with the design principle of equal life [9].

Then, the static analysis of 30° and 0° models’ intermediate shaft bearings were carried out, and the results are shown in Table 4 and Table 5.

Table 4. Damage degree and fatigue life of 30° bearings

| Bearing Designation | ISO Damage [%] | ISO TS 16281 Damage [%] | ISO Life [hrs] | ISO TS 16281 Life [hrs] |
|---------------------|----------------|-------------------------|---------------|-------------------------|
| 32307JR             | 0.141          | 0.295                   | 3534.888      | 1695.639                |
| 303/28R             | 0.297          | 0.320                   | 1683.722      | 1563.405                |

Table 5. Damage degree and fatigue life of 0° bearings

| Bearing Designation | ISO Damage [%] | ISO TS 16281 Damage [%] | ISO Life [hrs] | ISO TS 16281 Life [hrs] |
|---------------------|----------------|-------------------------|---------------|-------------------------|
| 32307JR             | 0.190          | 0.474                   | 2630.388      | 1055.604                |
| 303/28R             | 0.775          | 1.1                     | 645.593       | 440.657                 |

As can be seen from the comparison of the three tables above, with the decrease of the shafts’ transverse angle, the damage degree of bearings increased and the fatigue life decreased gradually. The last two situations don’t meet the work requirements anymore.

3.2. Contact stress distribution

For models of the three different situations, respectively, we picked the bearing close to the bigger gear on the intermediate shaft to analyze. As a result, the contact stress between rollers and inner race, outer race are shown in the six figures below.

Fig.4 Inner race contact stress of 45° bearing. Fig.5 Outer race contact stress of 45° bearing.
Fig. 6 Inner race contact stress of 30° bearing. Fig. 7 Outer race contact stress of 30° bearing.

Fig. 8 Inner race contact stress of 0° bearing. Fig. 9 Outer race contact stress of 0° bearing.

By comparing the contact stress of the inner and outer races of the bearing in the three models, it can be seen that the contact stress of the 45° model’s bearing is the minimum, followed by the 30° model, and the maximum is the 0°.

4. Reason Analysis and Model Improvement

4.1. Primary reason

The comparison of static results shows that, with the decrease of the transverse angle, the reaction force, generated by the bearings on the intermediate shaft, in vertical direction increased, so did the radial force, which led to the increase of the dynamic load. Besides, the contact stress between rollers and inner race, outer race of the bearing both increased, all these give rise to the increase of bearing damage degree and the decrease of fatigue life.

4.2. Improvement of bearing selection

Since the bearing strength of the 30° and the 0° models doesn’t meet the requirements, they need to be re-selected. In 30° model, bearings were respectively changed to TR0708-1R and 332/28JR. In 0° model, they were respectively changed to TR0708-1R and 323/28R.

The results of static analysis after the re-selection are shown in Table 6 and Table 7.
Table 6. Damage degree and fatigue life of 30° bearings after correction

| Bearing Designation | ISO Damage [%] | ISO TS 16281 Damage [%] | ISO Life [hrs] | ISO TS 16281 Life [hrs] |
|---------------------|----------------|--------------------------|----------------|-------------------------|
| TR0708-1R           | 0.098          | 0.074                    | 5765.312       | 7045.639                |
| 332/28JR            | 0.077          | 0.139                    | 6235.287       | 3572.518                |

Table 7. Damage degree and fatigue life of 0° bearing after correction

| Bearing Designation | ISO Damage [%] | ISO TS 16281 Damage [%] | ISO Life [hrs] | ISO TS 16281 Life [hrs] |
|---------------------|----------------|--------------------------|----------------|-------------------------|
| TR0708-1R           | 0.092          | 0.117                    | 5934.587       | 4587.654                |
| 323/28R             | 0.105          | 0.158                    | 5082.764       | 3363.975                |

It can be seen that after using higher-strength bearings, the damage degree of the bearings decreased and the fatigue life were significantly improved, making it up to work requirements.

5. Conclusion

In this paper, through changing the transverse angle (45°, 30°, 0°) formed by the input shaft and intermediate shaft of the two-stage reducer, modeling was done three times. On the intermediate shafts of three different models, the changes of bearing force, damage degree and fatigue life were compared and analyzed, then the optimum type of bearings were selected for different models according to the results. By comparison, it can be found that with the increase of the transverse angle formed by the input shaft and the intermediate shaft, the contact stress of the intermediate shaft bearing decreased, the damage degree of the bearing decreased, and the fatigue life of the bearing increased. Besides, increasing the transverse angle would reduce the horizontal dimension of the gearbox, but the vertical dimension will increase at the same time. Therefore, when designing the reducer, the transverse angle formed by the input shaft and the intermediate shaft should be set reasonably, and the corresponding bearing type should be selected appropriately at the same time.

References

[1] Lianggui Pu, Guoding Chen, Liyan Wu, Mechanical Design[M]. Higher Education Press, Beijing, 2013:326-330.
[2] Shaolong Zhang, Fei Gao, Gaofeng Wang, Zeqiang Li, Static Analysis and Selection of Main Reducer Bearing[J]. Chassis Technology. 2013(05):74-76.
[3] Xin Liao, Xianwen Zhou, Juncong Gao. Optimal Axial Preload of Tapered Roller Bearings for Automobile Main Reducer[J]. Bearing. 2019(04):9-13.
[4] Ming Zhong, Zhongliang Liu, Jingwei Xin, Research on Parallel Gearbox Oil Groove Structure in Drilling Platform[J]. Marine Engineering, 2013,35(S2):90-92.
[5] Xiaoli Fu, Chao Han, Yong li, Fang Chen, Research Present Status and Future Trend of Gear Reducer Design Method[J]. Mechanical Drive. 2012,36(10):112-114.
[6] Guangze Zheng, Junxiang Peng, Xiupeng Huang, Research on the Gear Shaft Parameters on Reliability of Intermediate Bearing of Reducer[J]. Journal of Chongqing University of Technology. 2020,34(03):29-34.
[7] ISO/TS 16281:2008, Rolling Bearings - Methods for Calculating the Modified Reference Rating Life for Universally Loaded Bearings[S].
[8] Young-Jun Park, Jeong-Gil Kim, Geun-Ho Lee, Effects of Bearing Characteristics on Load Distribution and Sharing of Pitch Reducer for Wind Turbine[J]. International Journal of PEMGT. 2016(03):55-65.
[9] Zhongchun Yang, Yuan Luo, Failure Analysis and Improvement of the Bearing in Gearbox[J]. Journal of Xidian University. 2015,39(03):179-180.