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Research Article

Gear Meshing Transmission Analysis of the Automobile Gearbox Based on the Software MASTA

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Abstract: As the main drive components of the automobile manual gearbox, the effect of gear meshing plays an important role on transmission performance. Aiming at the existing problems of the traditional gear meshing analysis, the study take a five-speed gearbox as an example, based on the MASTA software, a professional CAE software for simulating and analyzing the gearbox, to accomplish the gear mesh analysis of the automobile gearbox. Further more, the simulation modeling of the gearbox is built to simulate the actual load conditions and complete the process of analysis for the gear. It is indicated that a new design concept is put forward, that is, using specialized software MASTA for transmission modeling and simulation analysis can heavily improve the design level of the gearbox, reduce the test times and shorten the period of research and development as well. Finally, it can provide references for the development and application of new transmission gear.

Keywords: Gearbox, gear mesh, MASTA, simulation model

INTRODUCTION

The gearbox has a direct impact on the automobile assembly, including its power performance, economical efficiency, reliability and portability of the manipulation, stability and efficiency of the transmission. A gearbox is a speed and power changing device installed at some point between the engine and driving wheels of the vehicle (Tian *et al*., 2010; Xu *et al*., 2011). A gearbox is designed for changing the torque transmitted from the engine crankshaft to the propeller shaft, reversing the vehicle movement and disengaging the engine from the drive line for a long time at parking or coasting (Salgado and Alonso, 2008). The traditional manual mechanical gearbox is still the most widely used automotive gearbox. The practical structure of automobile gearbox is quite complex, distributed with a housing, an input shaft and gear, an output shaft and gear, a counter shaft, a reverse gear, a cluster of gears and a gear shift mechanism and so on, wherein, the gear, as one of the key parts of the gearbox, is used to transmit the motion and torque in the gearbox. In order to ensure that the precision and interchangeability of gear transmission, it is very necessary to put forward the higher requirements for the accuracy and stability of the gear transmission motor.

Recent studies show that the gear mesh has crucial effect on the transmission performance (Liu and Robert, 2008). Previously, the gear meshing analysis is always performed by directly calculate formulas from ISO standard to obtain the analytical results, for each coefficient selection of ISO standard formulas is more complex, the calculation process is correspondingly more complicated. This study puts forward a new design concept, that is, how to use the MASTA software to analyze the gear meshing of the existing gearbox, thus reducing the test cost, shorten the development cycle of the product and has a guiding significance on the design of new gearbox.

CALCULATION OF GEAR STRENGTH

When performing the gear meshing analysis, in order to avoid the pitting of gear surface and gear tooth fracture, the strength of gear must firstly be checked, that is to say, both the gear contact strength and the gear bending strength should meet the requirements; secondly, the gear contact status should be considered (Ge *et al*., 2009). The repeat meshing of the gear will produce a periodic vibration, if the meshing frequency is the same as or close to the vibration frequency of the transmission system, the resonance occurs, thereby producing vibration and noise (Zhu *et al*., 2009). Gear contact strength calculation and bending strength calculation are the basic calculations of gear design and the main theoretical formulas are as follows.

Contact strength of tooth surface:

- **Contact stress σ_H (N/mm^2):**

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$$\sigma_H = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}}\sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \quad (1)$$

where,

- Z_H = The coefficient of node zone
- Z_E = Elasticity coefficient
- $Z\varepsilon$ = Coincidence coefficient
- Z_β = The coefficient of helical angle
- F_t = Nominal tangential force of vertical pitch circle, N
- d_1 = Pitch circle diameter of the driving gear, mm
- b = The tooth face width, mm
- u = The transmission gear ratio
- K_A = The usage coefficient of contact strength calculation
- K_v = Dynamic load coefficient
- $K_{H\beta}$ = The distribution coefficient of longitudinal load
- $K_{H\alpha}$ = The distribution coefficient of load among gear teeth

- **Allowable contact stress σ_{HP} (N/mm^2):**

$$\sigma_{HP} = \frac{\sigma_{H\lim} Z_N}{S_{H\min}} Z_L Z_V Z_R \quad (2)$$

where,

- σ_{Hlim} = The contact fatigue limit of gear test, N/mm^2
- S_{Hmin} = Minimum safety factor of contact strength calculation
- Z_N = The life coefficient of contact strength calculation
- Z_L = Lubrication oil coefficient
- Z_V = Velocity coefficient
- Z_R = Roughness coefficient

- **Strength condition:** σ_H from the formula (1) should be between the upper and lower limits of σ_{HP}. If σ_H is above the upper limit of σ_{HP}, then the contact strength is not enough; If σ_H is below the lower limit of σ_{HP}, then the strength is too safe; So strength condition can be expressed as intensity coefficient S_{TH},

$$S_{TH} = \frac{\sigma_{HP\max} - \sigma_H}{\sigma_{HP\max} - \sigma_{HP\min}} \quad (3)$$

The value of S_{TH} should be between 0 and 1, if close to 1, indicating that strength reserve is too large; if close to 0, indicating that strength reserve is too small; if more than 1, indicating that strength is too safe; if less than 0, indicating that strength is not enough, so as to modify the design.

Gear tooth bending strength:

- **Root stress σ_F (N/mm^2):**

$$\sigma_F = \frac{F_t}{b \cdot m_n} Y_F Y_S Y_\beta K_A K_V K_{F\beta} K_{F\alpha} \quad (4)$$

In the above formula,m_n is normal module; Y_F andY_S are, respectively tooth profile parameters and stress correction coefficient while the loads act on upper bound point of a single pair of meshing zone; Y_β is the coefficient of helical angle; K_A is the usage coefficient of bending strength calculation, K_V is dynamic load coefficient, $K_{F\beta}$ is the distribution coefficient of longitudinal load and $K_{F\alpha}$ is the distribution coefficient of load among gear teeth.

- **Allowable root stress σ_{FP} (N/mm^2):**

$$\sigma_{FP} = \frac{\sigma_{F\lim} Y_{ST} Y_{NT}}{S_{F\min}} Y_{\delta relT} Y_{\mathrm{Re}alT} \quad (5)$$

In the above formula, σF_{lim} is the bending fatigue limit of gear test, N/$_{mm2}$; S_{Fmin} is minimum safety factor of bending strength calculation; Y_{NT} is the life coefficient of bending strength calculation; Y_{ST} is stress correction coefficient of gear test, generally equal to 2; $Y_{\delta relT}$ is sensitivity coefficient of relative tooth root fillet; Y_{RealT} is status coefficient of relative tooth root surface.

- **Strength condition:** σ_F should be between the upper and lower limits of σ_{FP}. If σ_F is above the upper limit of σ_{FP}, then the bending strength is not enough; If σ_F is below the lower limit of σ_{FP}, then the strength is too safe; similar to σ_H, strength condition can also be expressed as intensity coefficient S_{TF} to indicate the degree of σ_F close to the Upper and lower limit of the allowable stress. The value of S_{TF} should be between 0 and 1, if close to 1, indicating that strength reserve is too large; if close to 0, indicating that strength reserve is too small; if more than 1, indicating that strength is too safe; if less than 0, indicating that strength is not enough, so as to modify the design (Wu and Lv, 2011).

GEAR MESHING ANALYSIS

With the help of MASTA software, the study will put a five speed, manually shifted, automotive gearbox as the research object, to perform the transmission gear meshing analysis. In general, the geometry entity model of the gearbox is firstly established in the MASTA

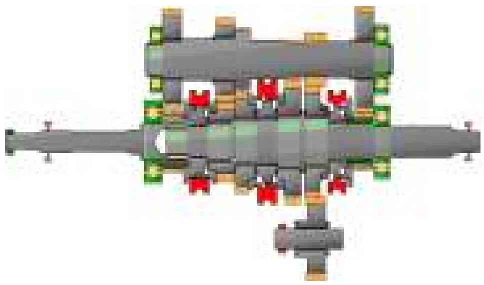

Fig. 1: 2D model of the gearbox

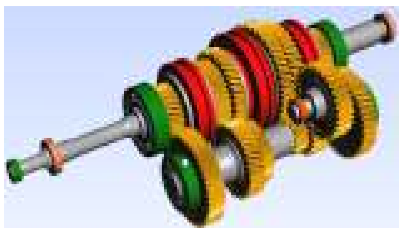

Fig. 2: 3D model of the gearbox

software and then the load is applied to every gear of the gearbox in accordance with the requirements of the fatigue life test of the gearbox assembly (Wang, 2008). The transmission ratios of each stop position of the gearbox are $i_1 = 4.0714$, $i_2 = 2.466$, $i_3 = 1.424$, $i_4 = 1$, $i_5 = 0.787$, $i_R = 3.006$, the center distance of the gearbox is 92 mm.

Establishment of gearbox simulation model: According to the detailed parameters of the gear, shaft and bearing and so on, the 2D and 3D simulation models of the gearbox are established in the MASTA software as shown in Fig. 1 and 2.

Loading of gearbox simulation model: When the gearbox is in the actual working condition, the loading born by the gearbox is changing, embodying that both the torque and the speed are changing, in addition, the usage frequent rate of the different stop positions, that is, the acting time of each stop positions, is not same also. The corresponding relationship among three variables is called the load spectrum. After clarifying the load spectrum of the gearbox in actual working conditions, we can get the actual stress condition of every component in the whole transmission system under the load spectrum, so as to obtain the accurate calculation result.

The load spectrum used by the study is consistent with the one in the fatigue life test of the gear assembly, as shown in Table 1

Gear meshing analysis:

Gear safety coefficient: Gear safety coefficient can be expressed as the ratio of the allowable stress to the actual stress, including bending safety coefficient and contact safety coefficient. Theoretically speaking, as long as both bending safety coefficient and contact

Table 1: The load spectrum of the gearbox

Design state	Load case	Duration (h)	Power load	Speed (rev/min)	Torque(N mm)	Power (Qy)
1st	100%	15	Out power load	42	-4.9999.9991	-2.2201
			Input power load	42	50000	2.22201
2nd	100%	8.3333	Out power load	526.8249	-155140.7309	-8.559
			Input power load	834	98000	8.559
3rd	100%	15	Out power load	562.5	-4.39999.9496	-24.2594
			Input power load	1404	165000	24.2594
4th	100%	6.9	Out power load	381.8182	-1230952.0243	-49.2183
			Input power load	2000	235000	49.2183
5th	100%	15	Out power load	2901.1458	-260.8007	-0.0792
			Input power load	2539	298	0.0792
Rev	100%	15	Out power load	199.0576	-260.5748	-0.0054
			Input power load	665	78	0.0054

Table 2: The gear safety coefficients under all load conditions

	Bending safety coefficient		Contact safety coefficient	
Gears	Left tooth surface	Right tooth surface	Left tooth surface	Right tooth surface
Constant mesh gear pair/pinion	-	2.183	-	1.3042
Constant mesh gear pair/wheel	-	2.0351	-	1.3495
Third gear pair/pinion	3.3776	-	1.7778	-
Third gear pair/wheel	3.7176	-	1.7818	-
Second gear pair/pinion	2.6068	-	1.1837	-
Second gear pair/wheel	1.661	-	1.2233	-
First gear pair/pinion	1.1082	-	0.7235	-
First gear pair/wheel	1.1418	-	0.8135	-
Fifth gear pair/pinion	39.7307	-	6.043	-
Fifth gear pair/heel	43.6841	-	6.043	-
Reverse gear pair/pinion	-	-	8.211	-
Reverse gear pair/wheel	-	-	9.0863	-

safety coefficient should be more than 1, can the requirement of gear strength be satisfied, but considering the connecting relations between the gearbox and the clutch, gear bending safety coefficient should be more than the reserve factor of the clutch, the gearbox can bear larger torque shock and avoid the breaking of gear tooth. Therefore, it is generally recognized that the bending safety coefficient should be more than 1.1.

Table 2 lists the gear safety coefficients under all load conditions. As can be seen from Table 2, the contact safety coefficients of first gear pair are 0.7235 and 0.8135, respectively, less than 1. In theory, it does not meet the requirements. But considering that the straight gear was used in first gear pair, coincidence degree is small, the rotation speed of first gear pair in the actual application condition is low, bearing torque is large, usage frequency is low, so the assessment of contact safety coefficient becomes more loose and only the bending strength of first gear pair should be need to meet the requirements. Finally, the fatigue life test of the gear and road test indicate that the safety coefficient can completely satisfies the requirements.

Gear damage rate: Gear damage rate can be expressed as the ratio of the gear actual cycle periodic number to the allowable cycle periodic number, directly reflects the service life of the gear, which comprises bending damage rate and contact damage rate. Gear damage rate should be as small as possible.

Table 3 lists the gear damage rates under all load conditions. As can be seen from Table 3, the damage rate of first gear pair is larger, which corresponds to the result that the safety coefficient is less than 1. For the same reason, only the bending damage rate of first gear pair should be needed to meet the requirements.

Gear meshing misalignment: The misalignment refers to the deviation between the actual relative position of two parts in transmission system and the relative

Table 3: The gear damage rates under all load conditions

	Bending damage rate (%)		Contact damage rate (%)	
Gears	Left tooth surface	Right tooth surface	Left tooth surface	Right tooth surface
Constant mesh gear pair/pinion	0	0	0	0.13
Constant mesh gear pair/wheel	0	0	0	0.08
Third gear pair/pinion	0	0	0	0
Third gear pair/wheel	0	0	0	0
Second gear pair/pinion	0	0	10.75	0
Second gear pair/wheel	0	0	6.96	0
First gear pair/pinion	0.59	0	4155.86	0
First gear pair/wheel	0.48	0	1532.61	0
Fifth gear pair/pinion	0	0	0	0
Fifth gear pair/wheel	0	0	0	0
Reverse gear pair/pinion	0	0	0	0
Reverse gear pair/wheel	0	0	0	0

Table 4: The cylindrical gear meshing misalignment

		Gear pair					
Gear pair		First gear	Second gear	Third gear	Fourth gear	Fifth gear	Reverse gear
	Pinion	17.797	12.7167	11.7507	-0.1666	1.3934	0.6185
	Wheel	31.7486	14.6594	9.6688	-0.0035	2.0396	1.8078
Constant mesh gear pair	Totality	-13.9517	-1.9427	-25.97	-0.1631	-0.6462	-1.1893
	Pinion	-0.0959	-3.2669	5.0487	8.6091	-0.6966	-1.3249
	Wheel	-5.9748	-6.1224	-5.6281	-30.5401	-1.633	-0.8682
Third gear pair	Totality	5.8789	2.8555	10.6768	39.1491	0.9364	-0.4568
	Pinion	3.9535	3.4595	3.1766	20.2625	1.2178	0.6736
	Wheel	1.1662	4.8357	2.0974	-0.9839	0.5088	0.9481
Second gear pair	Totality	2.7873	-1.3762	1.0792	21.2464	0.709	-0.2746
	Pinion	-2.013	-4.0628	-1.7132	-7.0027	1.2287	0.645
	Wheel	3.5499	5.5985	6.4544	9.6371	0.5265	1.0044
First gear pair	Totality	-5.5629	-9.6613	-8.1676	-16.6398	0.7022	-0.3594
	Pinion	-10.7382	-11.7601	-12.4973	-41.3046	-0.6869	-1.2987
	Wheel	10.8867	10.8234	6.0982	46.8173	-1.5785	-0.8387
Fifth gear pair	Totality	-21.6249	-22.5836	-18.5955	-88.1218	0.8916	-0.46
	Gear 1	-5.1539	-6.2369	-3.0943	-21.5783	1.4063	0.7839
	Gear 2	5.7221	7.6745	8.5171	19.4180	0.5877	1.0859
Reverse gear pair	Totality	-10.8761	-13.9115	-11.6114	-40.989	0.8186	-0.302

Table 5: The results of data after gear optimization

Damage					
Normal module (m)	Kelix angle (°)	Normal pressure angle (°)	Root gear addendum modification factor	Contact damage for worst gear (%)	Bending damage for worst gear (%)
Original design					
2.8	25	20	0.1	4155.86	0.02
14:15 (1 results)					
2.787	26.247	21.6352	0.3287	722.19	0

position under the absolute ideal conditions. In transmission system, two shafts should be parallel to each other, so the corresponding gear center lines are also parallel to each other, however, due to the presence of machining and assembly errors, as well as the deformation of the shaft, bearings, casing and other parts in the case of loading, the relative position of two shafts in the working state is not actually parallel, that is, the gear center lines are not in parallel, but have a small angle. This tiny angle is the so-called misalignment between gears. The amount of misalignment has a decisive influence on the service life of gears, the vibration and noise of gearbox, thereby the bearing capacity and the driving stability of transmission system.

The gear misalignment reflects the stiffness. If the misalignment is too large, the non-uniform force will be exerted on the gear, leading to large deformation of gears and then to large deformation of the gear shaft, thereby causing vibration and noise. Table 4 lists the gear meshing misalignment. As can be seen from Table 4, the larger misalignment of fifth gear occurs under the static load working condition of constant mesh gear, which is mainly caused by the structural form of the gearbox. Hence, we can improve the stiffness of the gear and the gear shaft, as well as improve the stiffness of the bearing to reduce or eliminate the misalignment of the constant mesh gear

Gear optimization: Apart from the above analysis, the study can also use the MASTA software to undertake macroscopic optimization of gear. In the case that the speed ratio is kept constant, the gear optimization can be carried out through adjusting the basic parameters of gears, such as helix angle, normal module, normal pressure angle and the weight value between bending strength and contact strength, to improve the gear damage rate. In general, the helix angle should be kept in the range of 0 to 35 degrees; normal pressure angle should be kept in the range of 15 to 30 degrees; the range of normal module should be kept between 0.5 and 20; the system will automatically perform iterative calculation of gear optimization and give the corresponding gear parameter optimization value after each iteration calculation. After calculation, the optimal results are automatically listed below the original calculation results, in order to compare gear normal modulus, helix angle, normal pressure angle, root gear addendum modification factor, contact and bending damage rate before and after optimization. The results of data after gear optimization are shown in Table 5 as follows. By comparison with the data before optimization, gear contact damage rate for worst gear after optimization are greatly improved from the original 4155.86 to 722.19.

CONCLUSION

In this study, through the gear meshing analysis of the existing gearbox with the MASTA software, we can obtain the gear bending and contact safety coefficient and damage rates and the amount of the gear meshing misalignment and so on. Based on the optimization and comparison of the series of gear parameters, the different analytical results will be secured, which has a very good guide for the future design of new product. If the gear safety coefficient of newly designed gearbox is more than that of the existing gearbox, it is determined that the related gear parameters of newly designed gearbox are reasonable and the newly designed gearbox meets the strength requirements; If damage rates of newly designed gearbox is less than that of the existing gearbox, it indicates that the newly designed gearbox meet the stiffness requirements. Therefore, using MASTA software can effectively perform the gear tooth contact simulation analysis and optimization design of gear instead of the gear tooth profile modification in the actual manufacturing process, which can effectively reduce the test cost, shorten the product development cycle.

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