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High-speed BEV Reducer NVH Performance Optimization and Experimentation

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Abstract: To solve the common rattle and whine issues in high-speed BEV reducer, this article aims to calculate and analyze the NVH performance of the reducer by using MASTA. By optimizing gear macro geometry and micro geometry to reduce the transmission error of gear pair which plays a decisive role in the common rattle and whine. By using MASTA, import the macro and micro geometry parameters of the gear pairs of the reducer. And using ABAQUS/CAE to preprocessing reducer housing, differential housing etc. and import them to MASTA. And then, assembly those components together to simulate the reducer as precise as possible. The system stiffness matrix and mass matrix are calculated which is strongly concerned to the NVH performance of the reducer. This article analyzed the assembly vibration model by calculating the assembly stiffness matrix and mass matrix and optimized the gear pairs' parameters to reduce the transmission error of the gear pairs. And compared the optimized reducer with the original reduce in the same conditions, to make sure that the optimization is work. And so get the experience to control the NVH performance from the research. And according to these experiences, instruct engineers to design reducers that meet the harshness requirements of NVH to optimize the performance of speed reducer assembly.

Keywords: High-speed Reducer, NVH, MASTA, FEA, Transmission Error

1. Introduction

With the development of China's economy, car ownership continues to go high. Vigorously develop electric vehicles, can speed up the alternative fuel, reduce vehicle emissions, to energy security, promote energy conservation and emissions reduction, prevention and control of atmospheric pollution. Compared with the developed countries, there is no consensus on the development of BEV or hybrid electric vehicle technology routes. China is weak in the aspect of design development, key parts and components manufacturing is relatively un-advanced, especially in the areas such as the key technology of BEV's NVH ability (noise, vibration and harshness), will become a big obstacle of the industrialization of new energy vehicles.

As the requirement of local laws and regulations of the vehicle NVH is becoming stricter, and demand of vehicles' comfortability and safety from consumers is higher and higher, NVH has become one of the key indicators of performance. Compared with traditional vehicle, BEV's power source is changed from engine to motor, and the noise of the transmission will become more and more obvious, which will become an important source of noise.

In the study of transmission NVH performance, as early as 1967, K.Nakamura [1] studied the nonlinear dynamics of gear system gap. In 2002, J.Lin [2] studied the stability characteristics of dynamic parameters of two-stage gear transmission system considering the meshing stiffness of gears. In 2008, Tang Jinyuan [3] deduced the nonlinear dynamic model of the modified gear pair system in the case of tooth surface friction, time-varying meshing stiffness and tooth side clearance. In 2014, Wang Liansheng [4] studied the nonlinear dynamics and NVH performance of the coupling system of engine and transmission. In 2015, S. Rosbi [5] researched the vibration of the vehicle caused by the powertrain. And in 2017, Pan Xiaodong [6] studied BEV high-speed gear train NVH performance.

2. High-speed Reducer NVH Performance Analyzing

The BEV's powertrain is usually composed of motor and transmission system, and the combination of permanent magnet synchronous motor and two-stage reducer is mostly adopted [7-10]. The motor system is mainly electromagnetic noise, and the radial electromagnetic force is the main excitation source of electromagnetic noise. The noise sources of transmission system are mainly gear rattle, whine and low frequency resonance. For BEV reducer, and the main affect its harshness (NVH) is the performance of whine and the system of low frequency resonance problem. This article mainly uses MASTA and ABAQUS, through calculation and analysis of stiffness matrix, mass matrix and intrinsic mode, to optimize the design of the reducer gear pair, therefore, to achieve the goal of optimal reducer NVH performance.

2.1. Calculation of Dynamic Meshing Stiffness

Dynamic meshing stiffness refers to the displacement of tooth mesh place by a certain frequency unit (torsional angular displacement) under the periodic vibration, mesh in the space needs to exert the same frequency method to the size of meshing force (torque).

According to the definition of dynamic meshing stiffness and the vibration analysis model of gear transmission system, the dynamic meshing stiffness of gears is determined by the torsional flexibility of the driving and driven gears [11-13].

$$K_m(\omega) = \frac{1}{R_{pt}(\omega) + R_{gt}(\omega)} \tag{1}$$

In the formula (1): $K_m(\omega)$ is dynamic meshing rigidity, $R_{pt}(\omega)$, $R_{gt}(\omega)$ are the torsional flexibility of the driving and driven gears, and ω is the frequency of the gear meshing excitation.

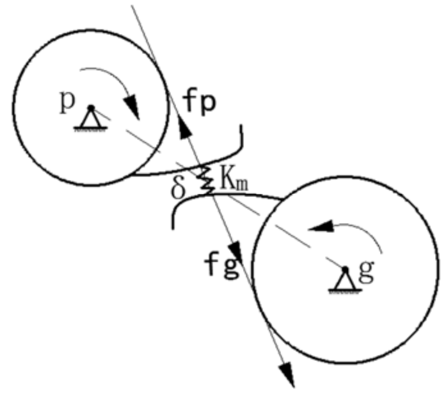

Figure 1. *Dynamic meshing analysis model of gears.*

2.2. Calculation of System Mass Matrix

In calculating the vibration response of the system, a one-dimensional finite element model of the full degree of freedom needs to be established [14-15], as shown in Figure 2.

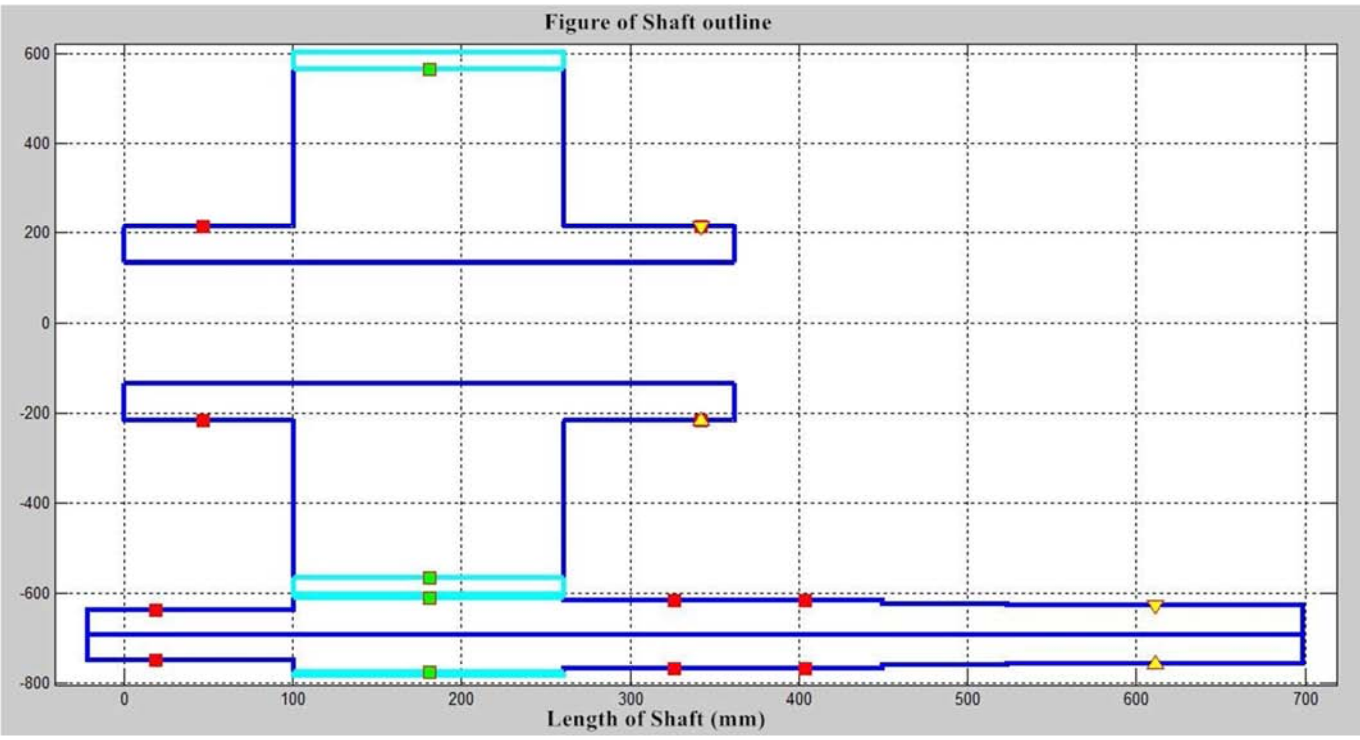

Figure 2. *One-dimensional finite element model of shaft system.*

The system mass matrix is extracted from the model, As shown in formula (2):

$$\begin{bmatrix} [K_1] & & & & \\ & \ddots & & & \\ & & K_p & \begin{bmatrix} k_{p1} & & & \\ & k_{p2} & & \\ & & k_{p3} & \\ & & & k_{p4} \end{bmatrix} & \\ & & & \ddots & \\ & & & & [K_m] \end{bmatrix}_{(6n,6n)} \tag{2}$$

Among them: M_p for the No. p axis mass matrix in the shaft system, m_{p1} for the plane xoz all the nodes in the No. p axis mass matrix (x , θ_y), m_{p2} for the plane yoz all the nodes in the No. p axis mass matrix (y , θ_x), m_{p3} for all the nodes on the No. p shaft axial (z) mass matrix, m_{p4} to the No. p axis on the torsion (ϕ) mass matrix of all nodes.

The system stiffness matrix is extracted from the model:

Among them: K_p for the No. p axis stiffness matrix in the shaft system, k_{p1} for all the nodes of plane xoz in the No. p axis stiffness matrix (x , θ_y), k_{p2} for all the nodes of plane yoz in the No. p axis stiffness matrix (y , θ_x), k_{p3} for all the nodes on the No. p axial (z) stiffness matrix, k_{p4} for all nodes on the No. p shaft to the torsion (ϕ) stiffness matrix.

System mass matrix and stiffness matrix are very important NVH performance indexes, the calculation formula is complex, the accuracy of the calculation directly affect the NVH results of the analysis, the mass matrix and stiffness matrix in this article mainly adopts FE analysis software.

3. FE Analysis Model

The noise map in a certain type of gear reducer is shown in the figure below:

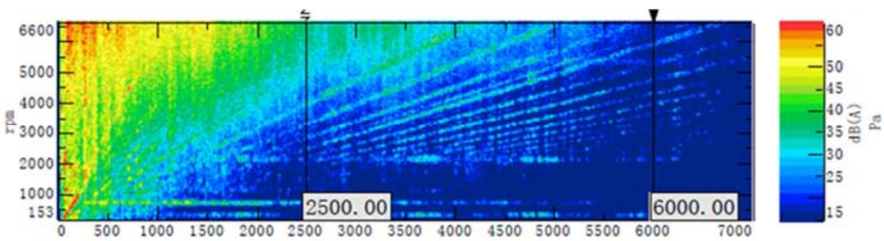

Figure 3. *Noise map before optimization.*

As we can see from the test results in the Figure 3, there are multiple resonant noise zones within the range of 2500Hz~6000Hz.

3.1. Model Analysis

MASTA analysis model as shown below:

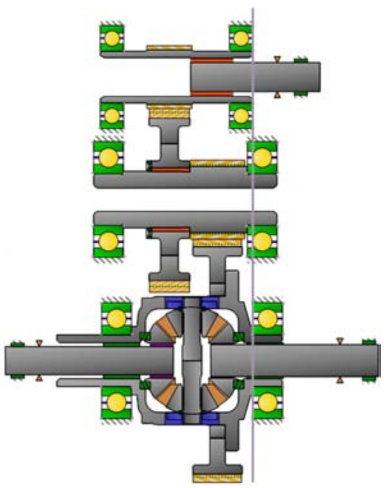

Figure 4. *2D MASTA model.*

Reducer's housing is meshing by ABAQUS/CAE, setting bolt connection, mounting points and mass attributes, and the housing model is imported into the MASTA software, which is shown in Figure 5.

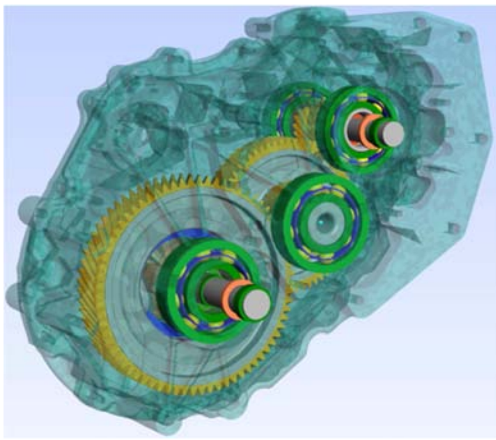

Figure 5. *MASTA model.*

Using the FE import module of MASTA software, the stiffness matrix, mass matrix and the first 40 order modes are calculated as shown in Figure 6, Figure 7 and Figure 8 respectively.

	Node 1 Dx	Node 1 Dy	Node 1 Dz	Node 1 Rx	Node 1 Ry	Node 1 Rz
Node 1 Fx	4042520.8725509644	25106.758960646916	1232644.1338067432	371573.05901571689	23262594.003372192	−1109128.0111999512
Node 1 Fy	25106.758960646916	2710422.723274231	−8949.1865250301471	17179783.521156311	100724.54582443347	−53914930.626976013
Node 1 Fz	1232644.1338067432	−8949.1865250301471	2194231.9650497437	1794108.8595428467	60008354.933784485	174006.14791431921
Node 1 Mx	371573.05901571689	17179783.521156311	1794108.8595428467	2022790471.0976562	6617611.7795410156	−1446097378.4357605
Node 1 My	23262594.003372192	100724.54582443347	60008354.933784485	6617611.7795410156	3688360964.578125	−40223232.917205811
Node 1 Mz	−1109128.0111999512	−53914930.626976013	174006.14791431921	−1446097378.4357605	−40223232.917205811	7758025215.0859375

Figure 6. *Stiffness matrix of the reducer(part).*

	Node 1 Dx	Node 1 Dy	Node 1 Dz	Node 1 Rx	Node 1 Ry	Node 1 Rz
Node 1 Fx	0.00032560305272823708	4.72658955055201E-06	−3.0387530490889534E-05	0.00012633794716382886	−0.0006104020497553304	−0.00095873224413320715
Node 1 Fy	4.72658955055201E-06	0.00039842431961091454	1.4546332711370247E-05	0.0032990936478479249	0.00033545005343405064	−0.0052286718775811942
Node 1 Fz	−3.0387530490889534E-05	1.4546332711370247E-05	0.00032911119618085443	0.00080523202652001418	0.0012249661643591841	−0.00052367474197071 73
Node 1 Mx	0.00012633794716382886	0.0032990936478479249	0.00080523202652001418	0.29532631378152741	0.017516306759132079	−0.03703303219325299
Node 1 My	−0.0006104020497553304	0.00033545005343405064	0.0012249661643591841	0.017516306759132079	0.22016196018884268	−0.0065085300947806349
Node 1 Mz	−0.00095873224413320715	−0.0052286718775811942	−0.00052367474197071 73	−0.03703303219325299	−0.0065085300947806349	0.45845414040135651

Figure 7. *Mass matrix of reducer(part).*

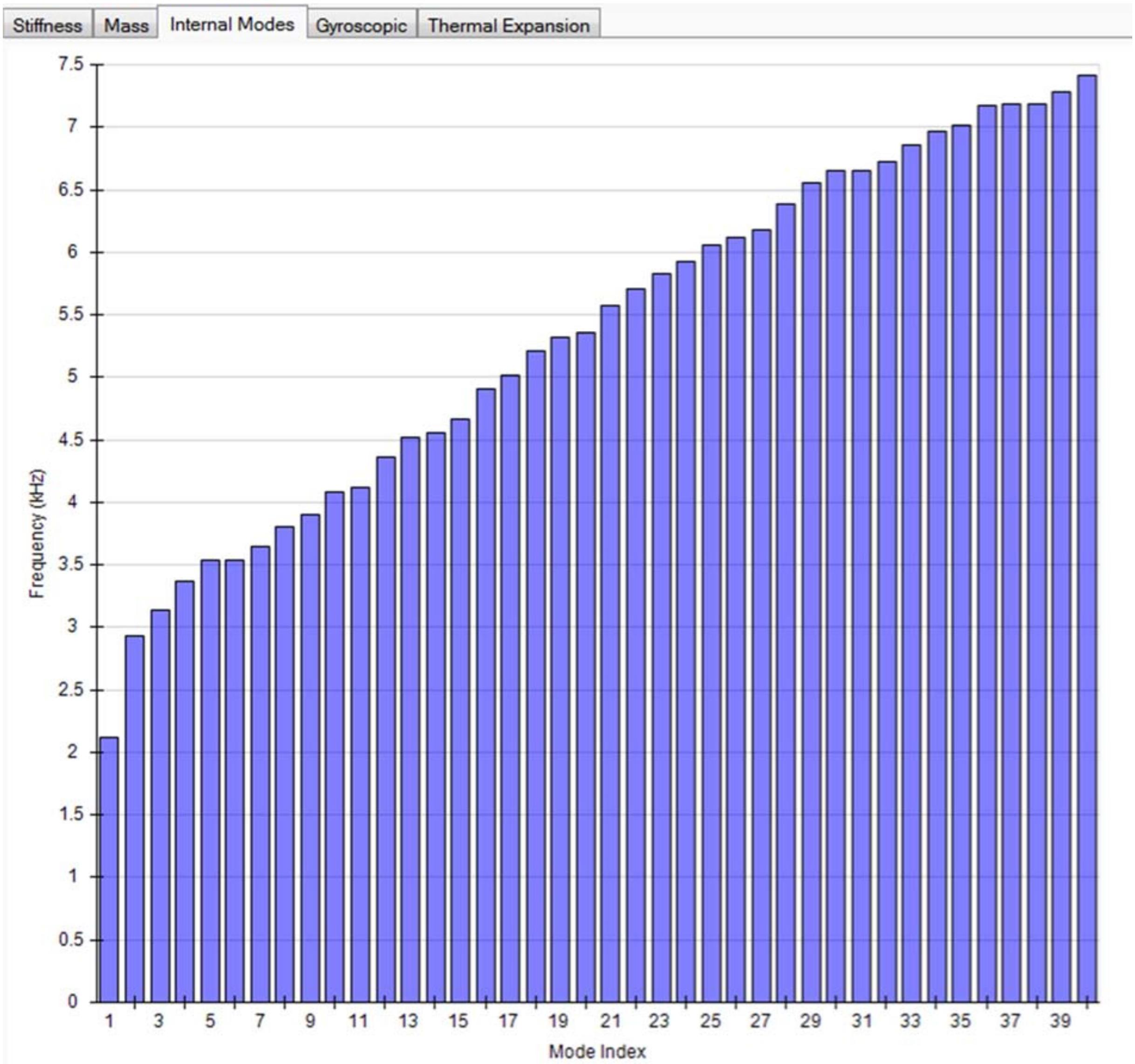

Figure 8. *First 40 order mode of the reducer.*

The parameters before optimization are shown in the following table:

Table 1. *Macro parameters of high-speed gear.*

Basic parameter	Pinion	Gear
Teeth	21	62
Normal module(mm)	2.012	2.012
Width(mm)	35	31
Normal pressure Angle(°)	19.5	19.5
Center distance (mm)	95	
Modification coefficient	-0.03	-0.5892
Hand	Right	Left
Top width coefficient of normal tooth	0.746	0.843

Table 2. *Macro parameters of low-speed gear.*

Basic parameter	Pinion	Gear
Teeth	23	71
Normal module(mm)	2.414	2.414
Width(mm)	42.5	37.6
Normal pressure Angle(°)	19	19
Center distance (mm)	125	
Modification coefficient	0.13	-0.144
Hand	Right	Left
Top width coefficient of normal tooth	0.662	0.796

The noise map of NVH test is shown in the Figure 3 above.

After entering modification of parameters, through MASTA simulation, the input torque of 240 Nm transmission errors are shown in Figure 9 and Figure 10, and Campbell diagrams are shown in Figure 11 and Figure 12.

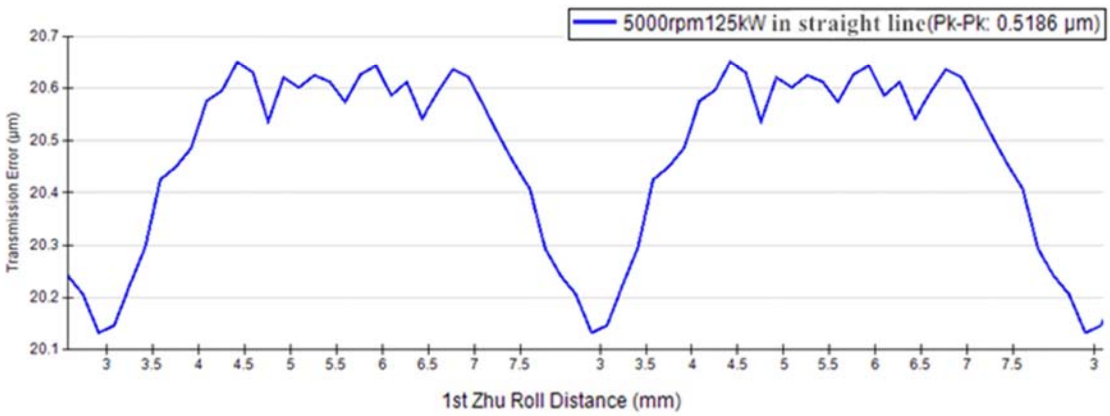

Figure 9. *Transmission error of high-speed gear (peak: 0.5186μm).*

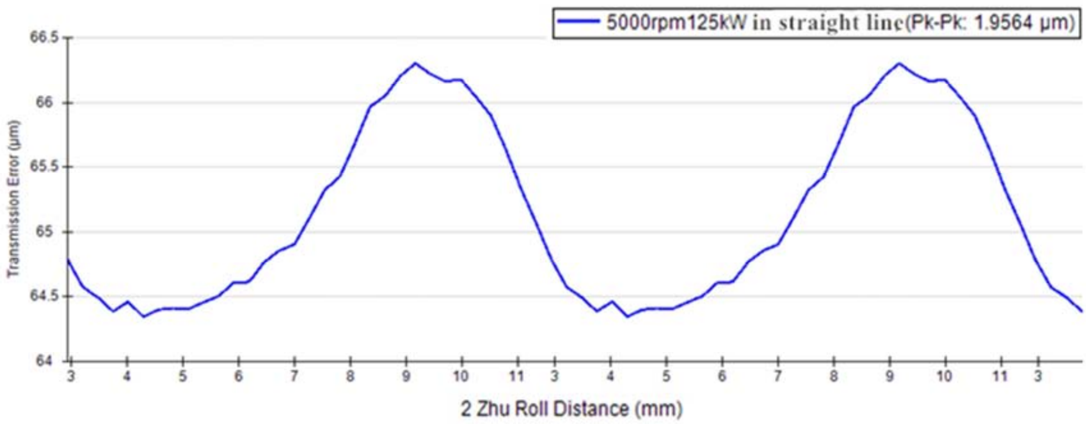

Figure 10. *Transmission error of low-speed gear (peak: 1.9564μm).*

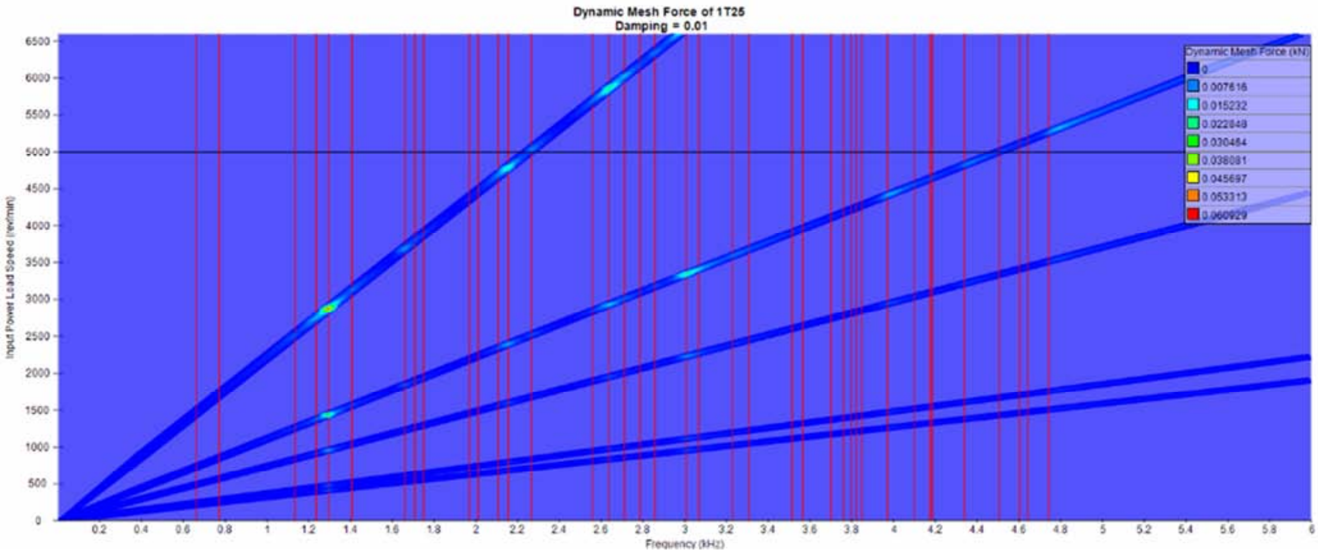

Figure 11. *High-speed Campbell diagram.*

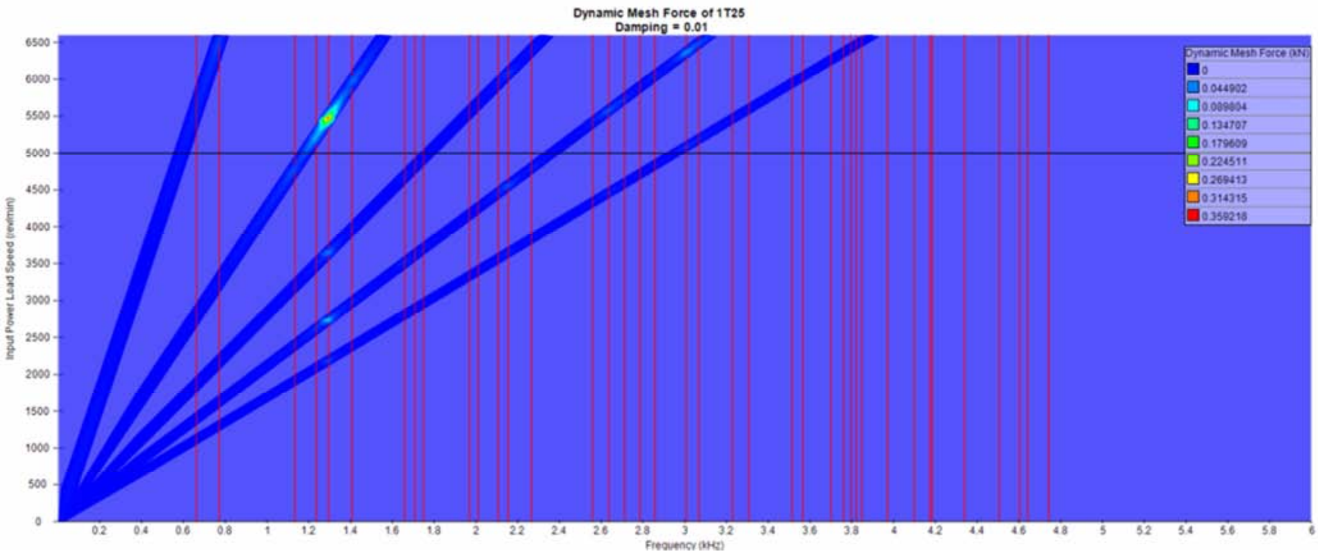

Figure 12. *Low-speed Campbell diagram.*

As can be seen from the Campbell diagram of the assembly, there are multiple resonance points in the range of 2500Hz~6000Hz.

3.2. Parameter Optimization

By optimizing the gear parameters for MASTA, the macro parameters of the optimized gear pair are shown in the table below.

Table 3. *Macro parameters of high-speed gear after optimization.*

Basic parameter	Pinion	Gear
Teeth	27	80
Normal module(mm)	1.552	1.552
Width(mm)	30	30
Normal pressure Angle(°)	16.2	16.2
Center distance (mm)	95	
Modification coefficient	0.155	-0.9494
Hand	Right	Left
Top width coefficient of normal tooth	0.6789	0.8916

Table 4. *Macro parameters of low-speed gear after optimization.*

Basic parameter	Pinion	Gear
Teeth	21	65
Normal module(mm)	2.658	2.658
Width(mm)	45.2	45.2
Normal pressure Angle(°)	17.28	17.28
Center distance (mm)	125	
Modification coefficient	0.3553	-0.4184
Hand	Right	Left
Top width coefficient of normal tooth	0.4058	0.7527

After receiving modification of parameters, through MASTA simulation, the input torque of 240 Nm transmission errors is shown in Figure 13 and Figure 14, and Campbell diagrams are shown in Figure 15 and Figure 16.

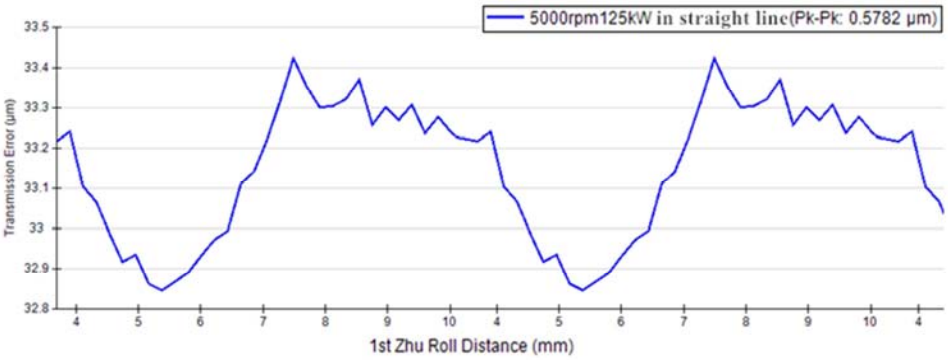

Figure 13. T*ransmission error of high-speed gear (peak:0.5782μm).*

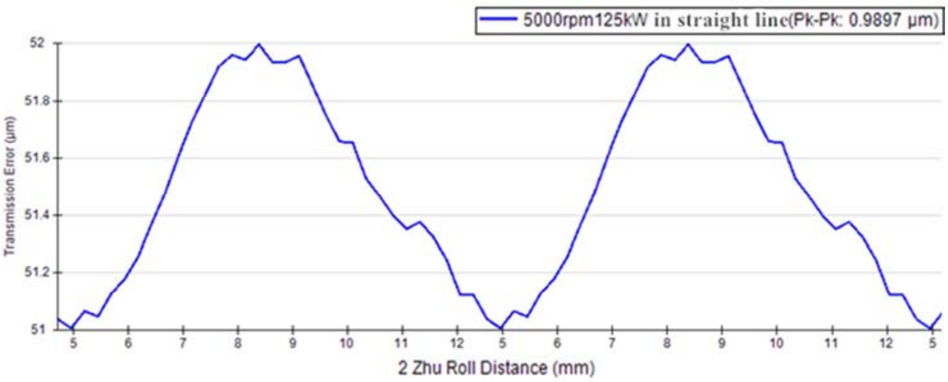

Figure 14. *Transmission error of low-speed gear (peak: 0.9897μm).*

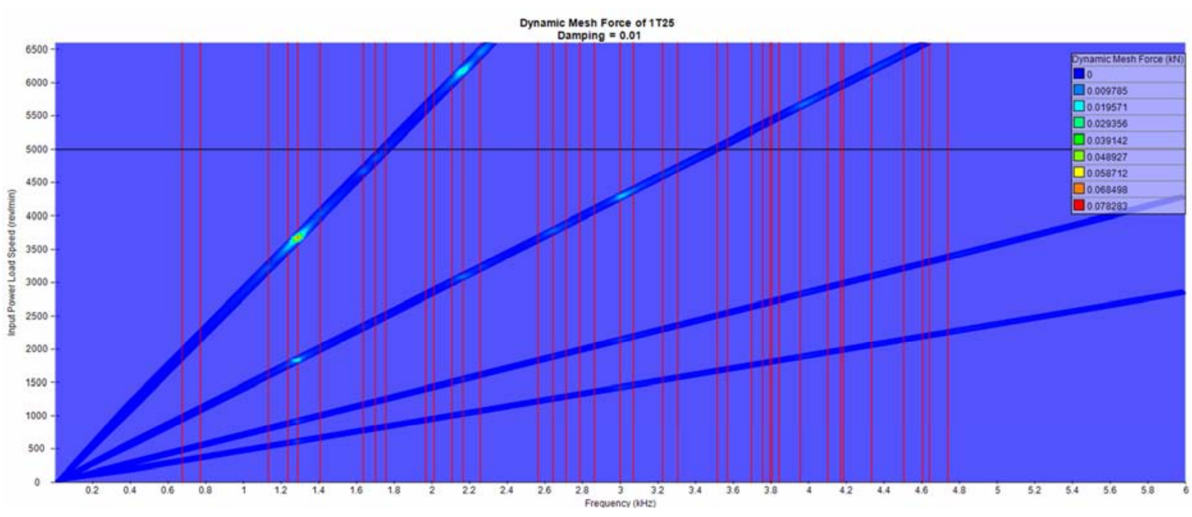

Figure 15. *High-speed Campbell diagram after optimization.*

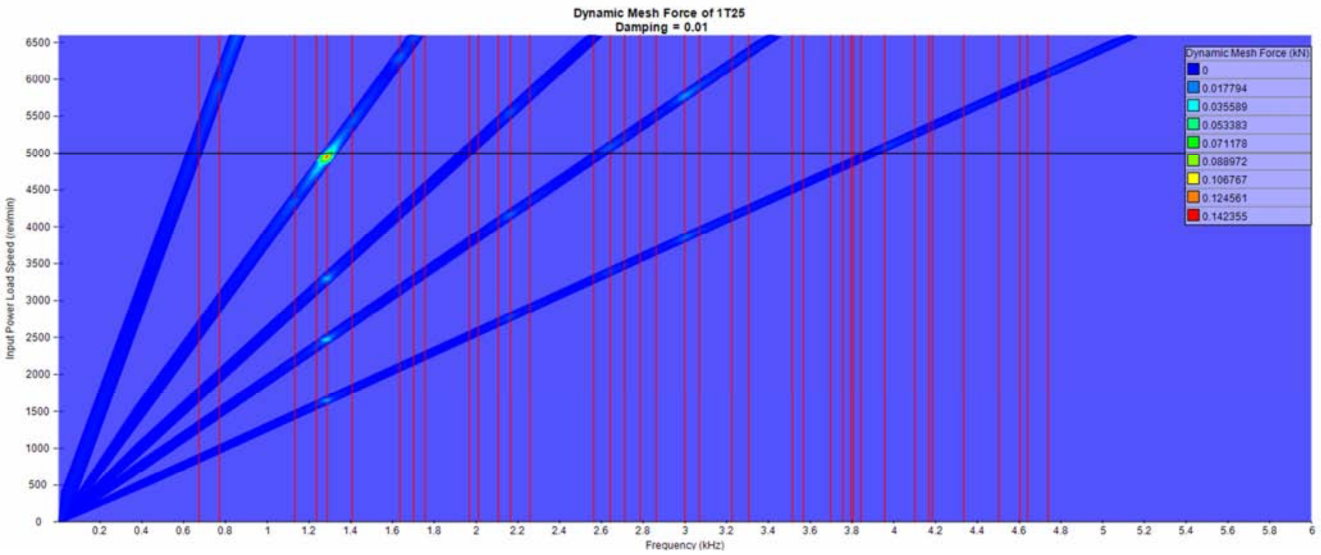

Figure 16. *Low-speed Campbell diagram after optimization.*

It can be seen from the analysis results that, after optimization of the gear shaft parameters are significantly reduced in the range of 2500Hz~6000Hz, and the dynamic meshing force of resonance is obviously reduced.

4. NVH Test After Optimization

To verify the optimization effects, guaranteed that housing and shaft are not factors of the test. We set up a vehicle to test the NVH performance as figure 17.

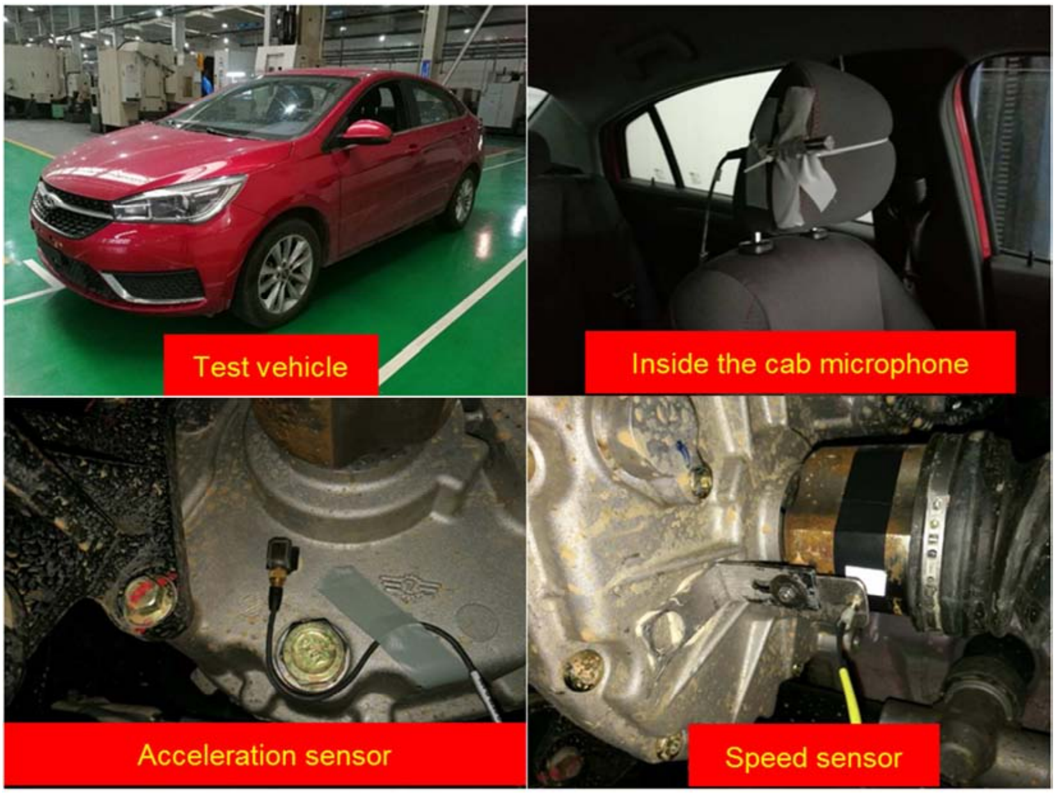

Figure 17. *Sensor layout in test vehicle of NVH.*

Test results collected by using Siemens LMS/NVH portable device, and results shown as follows:

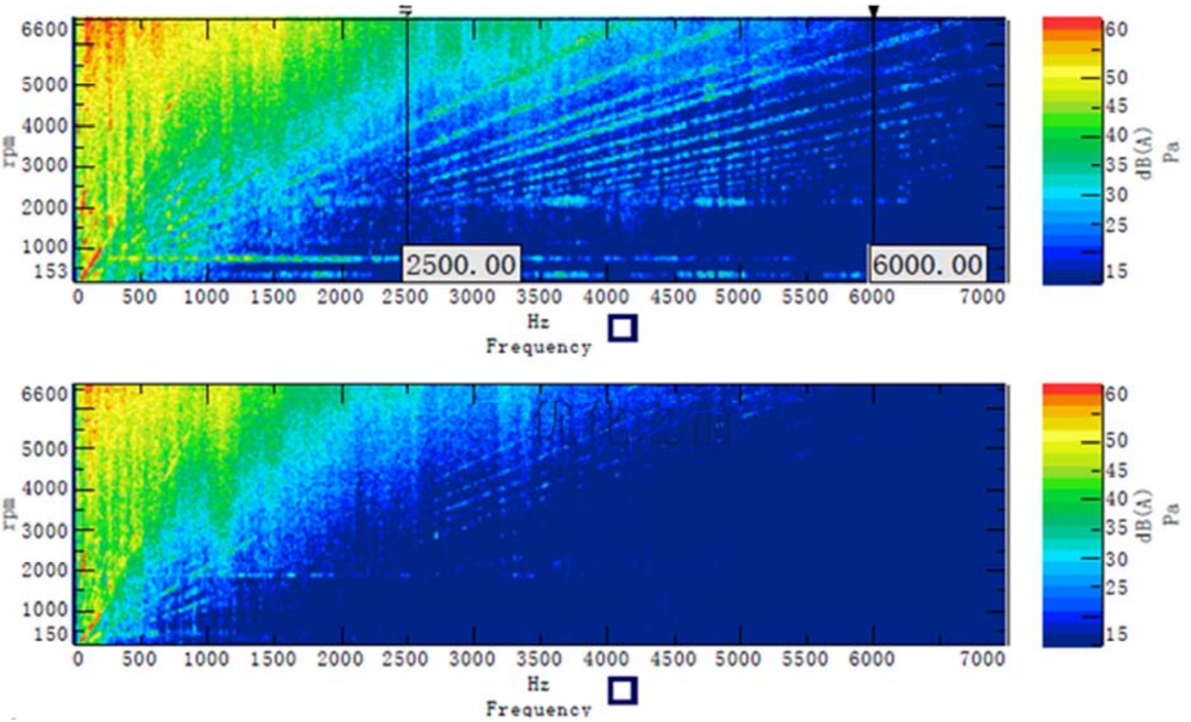

Figure 18. *Noise map before optimization after optimization.*

In Figure 18, top area stands for before the optimization and the bottom stands for the after. It can be seen from the figure that high-frequency noise in the range of 2500~6000Hz is basically eliminated, which is consistent with the subjective evaluation result.

5. Conclusion

From figure 10 and figure 14, we can see that the transmission error peak of the low-speed gear pair reduced from 1.9564um to 0.9897um, and the relative dynamic mesh force peak show in figure. 12 and figure. 16 reduced from 0.359 kN to 0.142 kN . It can be indicated that the transmission error of the gear pair is an important factor of the dynamic mesh force. And from figure. 18 we can see that the reducing of the transmission error can reduce the order noise of the reducer. And from this article, the whine problem can be effectively solved by modifying the macro and micro geometry to improving the contact stress of gear pair and change the NVH excitation spectrum of the reducer. The peak of transmission error is an important index that affects NVH performance of the reducer. So, to solve the NVH problem of the BEV reducer, the transmission error of the gear pair of the reducer should be controlled at a reasonable range, which need a large number of experiments to confirm.

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MAIN RESEARCH FIELD: NVH performance research and control of new energy vehicle gearbox and reducer; Gearbox and reducer design, produce and vehicle matching.