Moan Noise Diagnosis and Research of Certain E-axle BUS

Jin Wenhui1, Deng Xin1,2*, Duan Longyang1,2, Zhong Chengping1,2, Yu Xianzhong1,2
1Jiangling Motors Co., Ltd, Nanchang 330001
2Jiangxi Vehicle Noise and Vibration Key Lab., Nanchang 330001
*Corresponding author’s e-mail: xdeng6@jmc.com.cn

Abstract. As the mainstream power system of new energy commercial vehicles, the E-axle system directly drives the wheels. Since there is no suspension, the drive motor and reducer are directly assembled on the axle, connected to the body or frame through leaf springs and shock absorbers. Without the traditional engine noise masking, the NVH problem of E-axle poses a great challenge to NVH engineers. This article is aimed at the obvious Moan problem in a commercial passenger vehicle with E-axle under medium and high speed conditions, with the "source-path-response" theory used in analysis. Based on the investigation and analysis of test results and CAE methods, the main reason is that the imbalance excitation of the first axis of the E-axle is too large. By changing the speed ratio of the reducer, the first axis rotating speed of the E-axle corresponding to the same vehicle speed is reduced, then the high-speed Moan noise of this vehicle can get reduced.

1. Introduction
In recent years, with the national policy supporting new energy vehicles and driven by the gradually stricter environment emission standard and negative fuel consumption credits, automobile manufacturer are vigorously developing BMEC(Battery, Motor and Electric control) system, and BEV(Battery Electric Vehicle) has become the main trend of future auto industry. Motor, battery and electric control technology are developing rapidly. Many commercial vehicles are developing BEV models. As an integrated product of motor, reducer and axle, E-axle has been widely utilized in BEV commercial vehicles.

In term of structure, E-axle can be categorized to the following five types: rear-mounted rear-drive half shaft output E-axle, central-driven system E-axle, coaxial / parallel-axis E-axle [1], wheel rim E-axle and wheel hub E-axle. The central-driven system E-axle replaces the original engine and transmission. It is easy to develop and its manufacturing cost is also low, however its efficiency is not high and it is difficult to package power battery, with ordinary vehicle NVH (Noise, Vibration and Harshness). Coaxial E-axle has similar structure with parallel-axis E-axle, and both of them are integrated by motor and traditional axle. The wheels are directly driven by motor with higher torque through reducer. The main difference is that the motor shaft of coaxial E-axle shares the same axis with reducer output axis, but the motor shaft of parallel-axis E-axle is parallel with reducer output axis. Coaxial / parallel-axis E-axle does not have driveshaft or suspension, so it owns small weight, low assembly cost and high transmission efficiency. It’s easy for battery pack package thanks to its small volume. Since there is no suspension, the driving motor and reducer are directly assembled on the axle, and connected with the vehicle body or frame through leaf spring and shock absorber, without the
traditional engine noise masking, so the NVH performance of coaxial / parallel-axis E-axle is not good.

Moan noise is also one of the descriptions of roar, and mainly happen under a condition of the medium and high speed, which may exist both in traditional vehicles and new energy vehicles. Moan issue of traditional vehicles mainly happens on the powertrain system, intake & exhaust system and the braking system. For the roaring noise of certain front-mounted rear-wheel drive sedan at high speed, Zhang Qiang, etc., made a systemic analysis on the critical X from source, path and response. They studied the influence of transmission calibration, exhaust system, powertrain imbalance and body system on Moan, with effective and feasible optimization proposals to reduce Moan by 8dB(A)[2]. Sun Jia, etc., studied roaring noise caused by intake system, reduced the source by optimizing system pressure fluctuation and adding Helmholtz resonator, thus significantly mitigating the cabinet roaring noise[3]. Chang Qingbin, etc., studied out that the critical X in braking system that caused Moan was the brake caliper. They considered fixing the eccentric wear issue of the braking lining in design. The re-designed braking caliper eliminated Moan noised caused by braking[4].

This article is aimed at the cabinet low-frequency Moan issue in certain light bus with E-axle at 60-100Km/h, with the "source-path-response"[5] theory used in analysis. With CAE (Computer Aided Engineering) simulation, a reasonable solution to the main cause of Moan was raised, which provided great technical support to the low-frequency NVH issues of parallel-axis E-axle.

2. E-axle Moan Overview

2.1 E-axle Parameters

| Table 1 E-axle Parameters |
|---------------------------|
| E-axle Type | Parallel-axis |
| Drive Mode | Rear-mounted rear-drive |
| Motor, Reducer Connection | Spline fit |
| Reducer Type | Double reduction |
| Gear | First gear |
| Reducer ratio | 14.1 |

Parallel-axis E-axle is also referred to offset E-axle and its structure was shown as in Figure 1. Motor is connected with reducer by spline fit, and reducer transfers power to axle housing differential through gears, thus driving the wheels. E-axle was connected to vehicle body through leaf spring and shock absorber.
2.2 Issue Description
The vehicle model with E-axle in this article has obvious low-frequency Moan noise in the rear row when the speed is within 60-100Km/h. Subjective driving evaluation there is obvious abnormal vibration in the rear floor when the speed is within 60-100Km/h. After analysis on motor rotation order and noise filtering, it was found that Moan noise mainly has order 1, with harmonic frequency between 2nd order and 3rd order, as shown in Figure 2.

As shown in Figure 3, through analysis of order spectrum on vibration test points of E-axle housing, it was found that the cabinet Moan noise order was correspondent with E-axle housing vibration, and it was located at the central surface of the first axis of reducer end cover.

Motor shaft Excitation Frequency Formula:
\[ f_m = \frac{n_m \cdot \text{order}}{60} \]  

Notes: \( f_m \) represents motor shaft excitation frequency, \( n_m \) represents motor rotation speed, order represents 1st order, 2nd order and 3rd order, etc. Based on Figure 1, 2, and the frequency formula (1), the main order frequency of Moan is 100-165Hz, 2nd order harmonic frequency is 200-330Hz, and 3rd order harmonic frequency is 300-495Hz.

3. Test Analysis Diagnosis
3.1 Moan Fishbone Analysis
According to the classic theory of “source-path-response” in NVH issue analysis, this article made a fishbone analysis on the potential causes for Moan, as shown in Figure 4, with Siemens LMS Test. lab test analysis system.

Excitation source analysis in the fishbone analysis on Moan critical X mainly includes spline fit phase of motor and reducer, dimension chain assembly tolerance [6] motor shaft imbalance, reducer first axis imbalance and E-axle assembly model analysis. Path analysis mainly includes shock absorber and rear leaf spring paths. Response analysis includes rear floor model analysis. The following content makes analysis and sorting out one by one based on the mentality above. Due to word limit, this article only makes detailed analysis on key critical X.
3.2 Source Analysis

3.2.1 Diagnosis and analysis on E-axle dimension chain assembly tolerance, imbalance of motor shaft and reducer first axis

1st Order excitation is mainly related to imbalance of alternator matched with reducer first axis through the spline when the E-axle was assembled [7]. The optimization analysis made an assumption: the imbalance of motor shaft and reducer first axis is equal to E-axle assembly’s imbalance. Figure 5 is the original imbalance requirement of motor and reducer dimension chain and shafts. Reducing the imbalance of the E-axle dimension tolerance and shaft by 50%, the test result is not optimized obviously on physical vehicle. See shown in Figure 6.
3.2.2 Analysis on motor and reducer spline fit phase impact

Based on 2.2.1 result and targeting to reduce E-axle imbalance, adjust motor shaft and reduce first axis spline fit phase several times (shown as Figure 7), the coincidence of red mark is 0 phase (subjective driving assessment acceptable), 180° of each other is opposite phase.

Through Figure 8 and Figure 9 analysis, Moan noise optimization can be acceptable by adjusting motor and reducing spline fit phase. Compared with the opposite phase status, Phase 1 noise reduced by 5dB(A) at high speed driving condition after adjusting spline fit phase. The test analysis proved the conclusion again that first phase excitation is mainly related to imbalance of motor matched with reducer first axis through the spline when the E-axle was assembled.
Figure 9 Comparison of E-axle Z Direction Vibration with Adjusted Spline Fit Phase

3.2.3 E-axle Model Analysis

Siemens LMS Test. Lab model test module is used to establish E-axle experiment model. 60 measuring points are to ensure the model shape readability, which use multi-input and multi-output exciter to test. PolyMax Model Analysis calculates the model vibration shape of the whole system within 50-200Hz, as shown in Table 2, Figure 10 and Figure 11.

![Figure 10 E-axle Model Comprehensive Frequency Response Function SUM](image)

| Test System | Frequency (Hz) | Model shape                      |
|-------------|----------------|----------------------------------|
| E-axle      | 54.59          | overall first phase bending      |
|             | 117.47         | motor partial bending            |
|             | 137.98         | overall second phase bending     |
|             | 152.72         | Motor + reducer bending          |
E-axle model[8] test result showed that E-axle has three models in 100-200Hz, which are close to 1st order excitation frequency of the moan, making E-axle is sensitive in structure.

3.3 Analysis on Transfer Path

E-axle is assembled on body through left and right leaf springs, and connects with body cross beam through shock absorbers. The transfer path are shock absorbers and leaf springs.

3.3.1 Path1 analysis: disconnecting shock absorber

Disconnect shock absorbers, E-axle and body, noise in cabinet reduced by around 2dB(A) around 65-70Km/h in objective test, no obvious subjective optimization. See Figure 12, it can be judged that shock absorber is not the main transfer path.

3.3.2 Path2 analysis: increasing leaf spring weight

As the main load-bearing structure, the leaf spring adopts leaf spring counterweight, the left and right leaf spring are evenly weighted with 15KG, and test results were shown in Figure 13. 1st Order noise is reduced by around 4dB(A)A at 65-70Km/h, and deviates at 80Km/h. It can be concluded that the leaf spring is the main path of the transferring.

3.4 Response Analysis on Rear Floor

Based on body interior Catia model, CAE is applied to establish vehicle body model in Hypermesh[9], MSC Nastran after treatment model is also used to analyze body sheet metal model’s contribution between 100-180Hz. CAE shows the main contributor is rear floor sheet metal between 100-180Hz,
which are high order model, just see Figure 14 for rear floor 145Hz model as example due to word limit.

Figure 13 Cabinet 1st Order Noise Comparison after Increasing Leaf Spring Weight

Figure 14 Body Rear Floor Vibration Model at 145Hz

The advantages of damper patch are reducing sheet metal vibration sensitivity and change sheet metal model, not affecting by space. The article investigates the damping’s effect on Moan on rear floor. Only 3dB(A) is reduced within 75-90Km/h on physical vehicle with an area of 1.5 m² damping, see shown in Figure 15.
Figure 15 Cabinet 1st Order Noise Optimization with Damper patch on Rear Floor
Through source-path-response analysis, this article found that the main cause of the E-axle commercial vehicle Moan at high speed is that the imbalance is too large. Due to the motor shaft match with reducer first shaft through the spline, resulting in the E-axle’s 1st order excitation large, and transfer to body through leaf spring, making low frequency moan and low frequency floor vibration.

4. Research on Optimization Scheme
Based on the above analyses, common optimizations adopt the following methods: reducing the excitation source vibration, path transfer, and resonance response [10]. The main transmission path here is the leaf spring. The transfer path of the leaf spring can be decreased by reducing the stiffness of the leaf spring bushing or enhancing the natural frequency of the leaf spring, which will extend the development cycle of the leaf spring, increase the development cost and reliability risk. Therefore, this article will not take the optimization of leaf spring into consideration. The common method to reduce the body response is to paste a damper plate on the floor of the vehicle body, which has been verified previously that such proposal had limited space for optimization and was not cost-efficient. As a result, the high-speed Moan noise can only be optimized by reducing the vibration of the E-axle excitation source.

There are two ways to reduce the vibration of the E-axle excitation source: 1. Decreasing the imbalance of the system when the motor shaft is matched with the reducer input shaft through the spline; 2. Lowering down the reducer speed ratio. The conventional three-point method [11]cannot be used for imbalance phase measurement and have low feasibility since the motor shaft and the reducer input shaft are both inside the E-axle; the imbalance excitation will go up with the speed rise. In consequence, the reduction in speed ratio of reducer can decelerate the motor at the same speed and achieve the purpose of reducing E-axle excitation.

When selecting a reducer gear with the gear ratio closest to 14.1 from the supplier’s existing gear products, the 11.6gear ratio was found. Comparing Figure 16 and Figure 17, after reducing the speed ratio of the reducer from the initial 14.1 to 11.6, the motor rotation corresponding to 60Km/h is reduced from about 6400rpm to about 5400rpm; the motor rotation corresponding to 100Km/h is reduced from 10600rpm to about 9100rpm; so rotation corresponding to 60-100Km/h is reduced by 1000-1500rpm.

After switching low-speed ratio, the sound pressure level in the vehicle of order 1 Moan noise is reduced by 5-6dB(A), which is acceptable in the subjective Moan driving evaluation, and the corresponding vibration of the E-axle is reduced by about 30-50%, as shown in Figure 18 and Figure 19.
Figure 16 Motor Rotation and Vehicle Speed Curve at Initial Ratio

Figure 17 Motor Rotation and Vehicle Speed Curve at Small Ratio

Figure 18 Cabinet Order 1 Noise Comparison before and after Ratio Switch

Figure 19 E-axle Order 1 Vibration Comparison before and after Ratio Switch
Table 3 Reducer Power Economy Comparison at Different Ratios

| Load state | Performance | 14.1 Speed Ratio | 11.6 Speed Ratio |
|------------|-------------|------------------|------------------|
|            | 0-100kph acceleration time | 16.5s | 16.8s |
|            | 0-50kph acceleration time | 4.8s  | 5.3s  |
|            | 50-80kph acceleration time | 5.6s  | 5.5s  |
| Full load  | Maximum speed | 109kph | 111kph |
|            | Maximum gradient | 30%   | 25%   |

Changes in the speed ratio of the reducer will bring about changes in power economic performance. As shown in Table 3, after reducing the speed ratio of the reducer, the acceleration performance of the vehicle at 0-100kph under full load basically has no effect, but the low-speed acceleration performance at 0-50kph deteriorated by 0.5s. When maximum speed increased from 109kph to 111kph, the overall acceleration performance had no obvious changes; the maximum grade ability at full load was reduced from 30% to 25%, considering the common standard slopes are 10% and 16.6% and 20%, hence the 25% climbing performance can also satisfy the actual working conditions.

5. Conclusion
E-axle with parallel axis has poor NVH performance due to their own design and layout. Therefore, the vibration target of E-axle is higher than that of general E-axle. For the low-frequency Moan issue in a light bus with E-axle in parallel-axis feature and in 60-100Km/h acceleration working condition, this article applies "source-path-response" analysis theory to carry out experimental investigation and analysis, and combines the CAE simulation analysis method to find the root cause for Moan. Starting from the engineering practice, an optimization plan with low cost, short cycle, and strong feasibility was proposed. The NVH issue in the vehicle caused by the imbalance of the E-axle was successfully solved, which is important for identifying NVH risks of the vehicle with E-axle in advance, putting forward an optimization plan, and accumulating engineering experience.

References
[1] Li Hui, Ma Pengfei. Domestic E-Axle Technology Patent Analysis [J]. Heavy Truck, 2020(02): 41-42.
[2] Zhang Qiang, Li Shoukui, Bi Jinliang, Zhao Jingbao, Chen Xiaomei. The Study on Improving Booming Noise in Back Row of Vehicle at High Speed [J]. AUTO SCI-TECH, 2019(06): 14-18.
[3] Sun Jia, Zhou Dandan, Luo Enzhi. Study on Vehicle Booming Noise Induced by Intake System [J]. Noise and Vibration Control, 2016, 36(6): 202-205.
[4] Chang Qingbin, Pu Xinyu, Zhang Zhijian, Zhan Guangrong. Study on the Solution of Braking Moan Noise Induced by Brake Shoes Taper Wear [J]. Noise and Vibration Control, 2017, 37(1): 68-71.
[5] Ruan Shisong, Wang Yuanming, Pei Yongsheng, Xu Xiaoahan. Noise analysis and optimization of electric drive assembly based on transmission path[J]. Auto Parts, 2019(04): 82-85.
[6] Xu, Y., Development of Commercial Vehicle E-Axle System Based on NVH Performance Optimization[C]. SAE Technical Paper, 2020-01-1421,2020.
[7] Shachar Tresser, Amit Dolev, Izhak Bucher. Dynamic balancing of super-critical rotating structures using slow-speed data via parametric excitation[J]. Journal of Sound and Vibration, 2018, 415:59-77.

[8] Rong Guo, Xiao kang Wei, JunGao. Experimental NVH evaluation of a pure electric vehicle in transient operation modes[J]. VibroengineeringPROCEDIA ,2016, 10:364-368.

[9] Liang Lin. CAE method apply to the development of the NVH performance of the automobile white body [J]. Automobile Applied Technology, 2016(12): 53+65.

[10] Pang Jian. Vehicle noise and vibration [M]. Beijing: Beijing Institute of Technology Press, 2006.

[11] Zhang Junhui. Three Point Field Balancing Method in Automobile Application in NVH [J]. Guangxi: Equipment Manufacturing Technology, 2012(10).136-137.