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Analysis of the Influence of Second-Order Excitation in the Transmission System on Acceleration Vibration

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***Abstract* – Light truck models generally adopt the front engine and rear drive scheme. The powertrain and rear drive axle are usually connected by two transmission shafts, and the power is transmitted by cardan joint. In actual automotive research and development projects, there are often problems with the design or assembly process of the transmission system, resulting in a large equivalent angle of the transmission system, resulting in low-speed driving vibration of the entire vehicle. This paper analyzes the low-speed vibration problem of a certain light truck during the engineering prototype stage, and obtains the mechanism of low-speed vibration by analyzing the order characteristics of vibration and the modal characteristics related to the entire vehicle system. Targeted optimization has been carried out, and the vibration of the entire vehicle has been significantly improved.**

***Keywords* – Vehicle, Equivalent Angle, Vibration, NVH, Transmission Shaft, Cardan Joint.**

I. Introduction

The automobile transmission shaft transmits the torque from the powertrain to the drive axle, thereby driving the vehicle forward [1]. For front engine and rear drive vehicles, there is a certain angle between the powertrain output shaft and the rear drive axle in space, and the transmission shaft needs to adapt to the changes in power transmission distance and angle caused by wheel runout [2]. In actual automotive research and development projects, if there are problems with the design or assembly process of the transmission system, resulting in a larger equivalent angle of the transmission system, there will be a larger second-order excitation of the transmission system, with a frequency generally in the range of 5-30Hz. The various systems of the entire vehicle have many modals within this range, which are prone to low-frequency resonance caused by second-order excitation of the transmission system. At the same time, the vibration sensitivity frequency band of the human foot is 9-15Hz, so passengers are extremely sensitive to low-frequency vibration in this frequency band [3].

Guangqiang Wu summarize the difficulties of NVH issues related to automotive transmission systems, the characteristics and generation mechanisms of typical vibration and noise phenomena, the NVH analysis method of the transmission system was discussed from the perspectives of experiments and simulations [4]. Qingshuang and Bo Ma analyzed the solutions to the shaking problem of the entire vehicle caused by dynamic balance issues in commercial vehicle transmission systems [5, 6].

This paper analyzes the low-speed vibration problem of a certain light truck during the engineering prototype stage, and obtains the mechanism of low-speed vibration of the entire vehicle by analyzing the order characteristics of vibration and the modal characteristics related to the entire vehicle system. Targeted optimization have been carried out, and the vibration of the entire vehicle has been significantly improved.

II. Description of Vibration Issues

During the subjective driving evaluation process of a light truck under research, it was found that the entire vehicle experienced severe vibration within the range of 5-30 km/h under the first and second gear of tip in conditions. The vibration phenomenon is related to vehicle speed and has obvious characteristics of the order of the transmission shaft. The order of the transmission shaft in 1st and 2nd gears is shown in Table 1.

Table 1. Power Train Order.

Transmission Case		Transmission System	
Gear Position	**Speed Ratio**	**First Order Transmission System**	**Second Order Transmission System**
Gear 1	4.415	0.23	0.45
Gear 2	2.547	0.39	0.79

The vibration amplitude of the first gear is as high as 0.072g, as shown in Figure 1. The order characteristics of the whole vehicle vibration problem correspond to the second order of the transmission system, so it is first suspected that the vibration problem is caused by the transmission system.

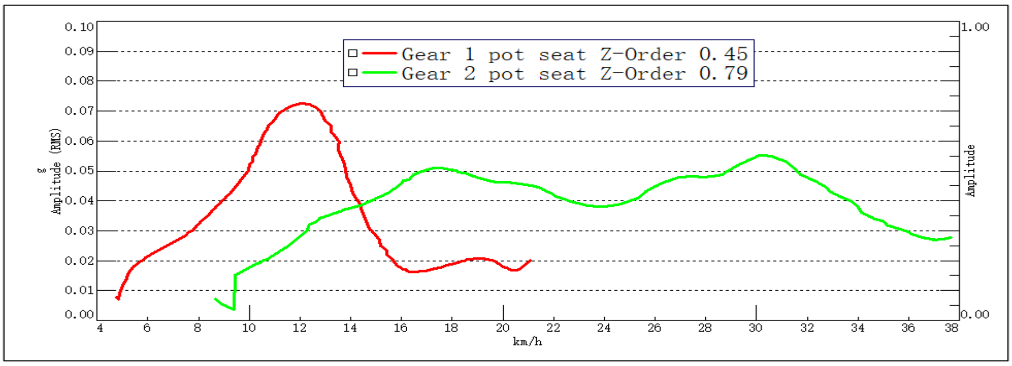

Fig. 1. 1st and 2nd gear pot seat vibration.

III. Diagnostic Analysis

For NVH problems, it is necessary to use the source path response diagnostic approach for problem diagnosis and troubleshooting, identify the main factors that affect the problem, and optimize it to efficiently solve the NVH problem.

A. *Transmission Path Analysis-Source*

Due to the fact that the vibration problem is related to vehicle speed, in order to extract the signal characteristics of the whole vehicle vibration, three axis acceleration sensor is arranged on the passive of powertrain rear mount bracket, the middle support bracket of the transmission shaft, and the bracket of front mounting point of the rear axle spring, as shown in Figure 2.

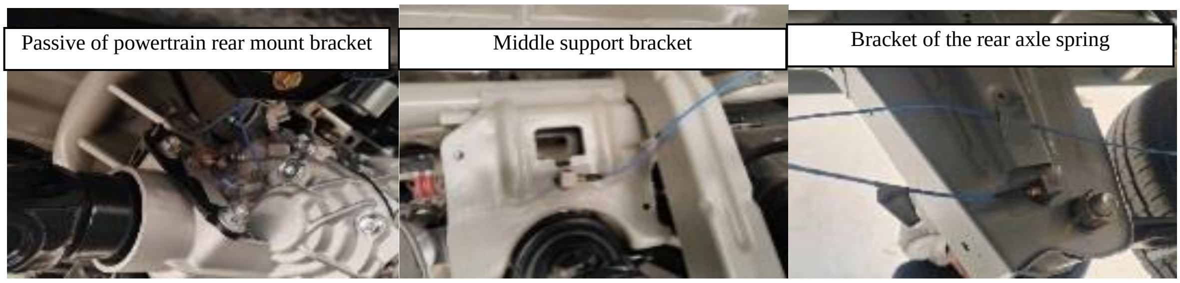

Fig. 2. Location of Sensor Measurement Points.

The test conditions are 1st and 2nd gear pot conditions, and the vehicle speed characteristics and order characteristics of the vibration at each measuring point are shown in Figure 3 and Figure 4.

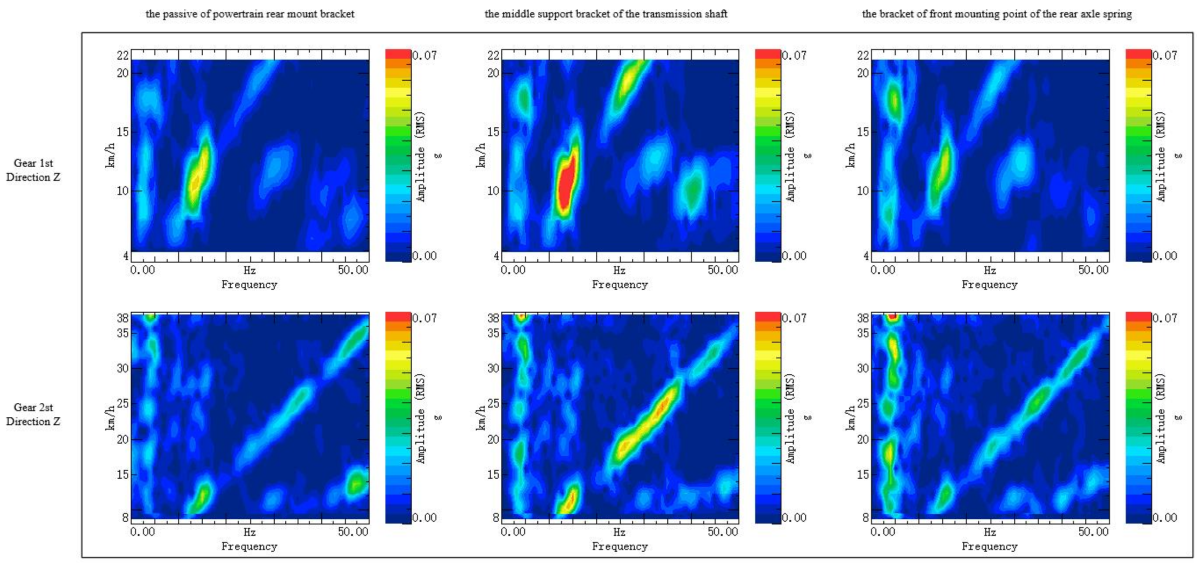

Fig. 3. Vehicle speed characteristics of vibration at each measuring point.

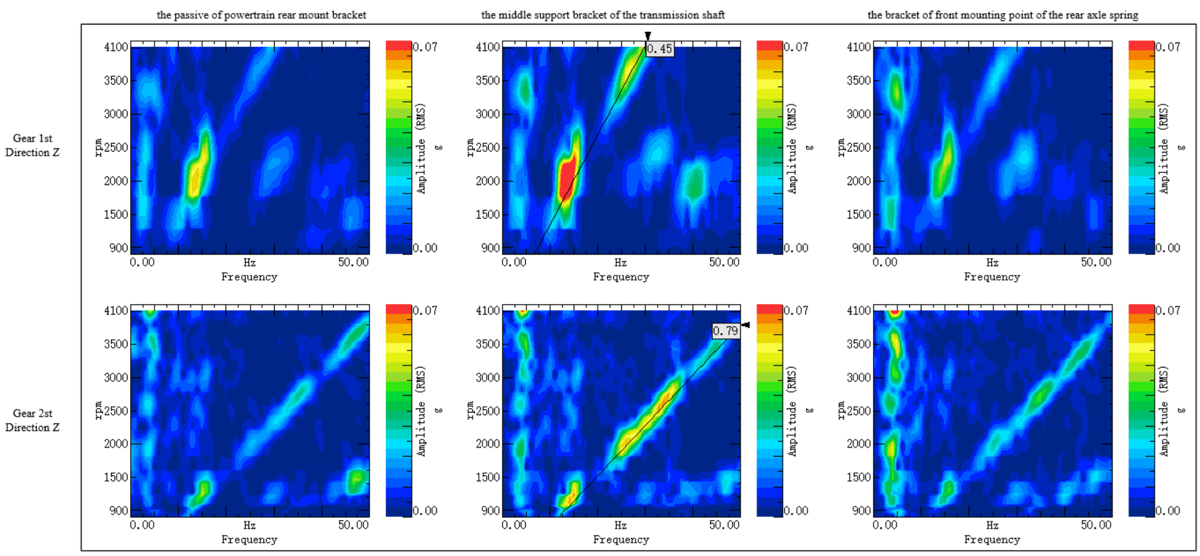

Fig. 4. Vibration order characteristics of each measurement point.

The experimental results show that the positions with the highest vibration response in the first and second gear working conditions are all middle support bracket of transmission shaft, with orders of 0.45 and 0.79, respectively, corresponding to the second-order excitation of the transmission system. This indicates that the excitation source of this vibration problem is the second-order excitation of the transmission system.

B. *Transfer Path Analysis - Transfer Path*

The vibration test results of middle support bracket of transmission shaft show that the main vibration energy of the entire vehicle is concentrated at 14Hz and 24-32Hz, indicating that the second-order excitation energy of the transmission system generates low-frequency resonance during the transmission to the body. Through investigation of various main systems, it was found that the first order torsional modal frequency of the entire vehicle is 13.5Hz, as shown in Figure 5. The rigid body modal frequency of the middle support of the transmission shaft is 28.3Hz, with a damping ratio of 5.34%. The main peak energy of the IPI at the origin of the middle support is 24-32Hz, as shown in Figure 6. By comparing the vibration response characteristics of each measurement point of the entire vehicle and the modal test results of each system, it can be determined that the vibration mechanism is the resonance caused by the middle support modal and the first order torsional mode of the entire vehicle being excited by the second order excitation energy of the transmission system, resulting in the vibration of the entire vehicle.

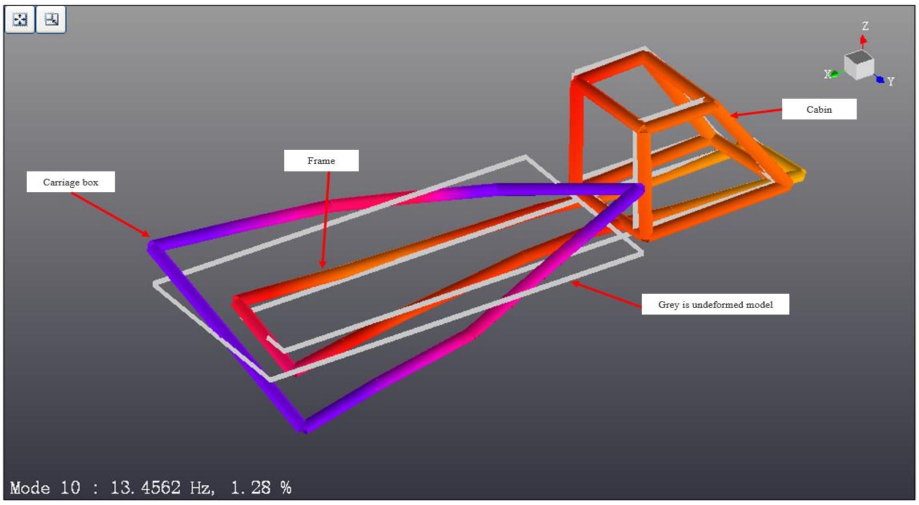

Fig. 5. First order torsional mode of frame.

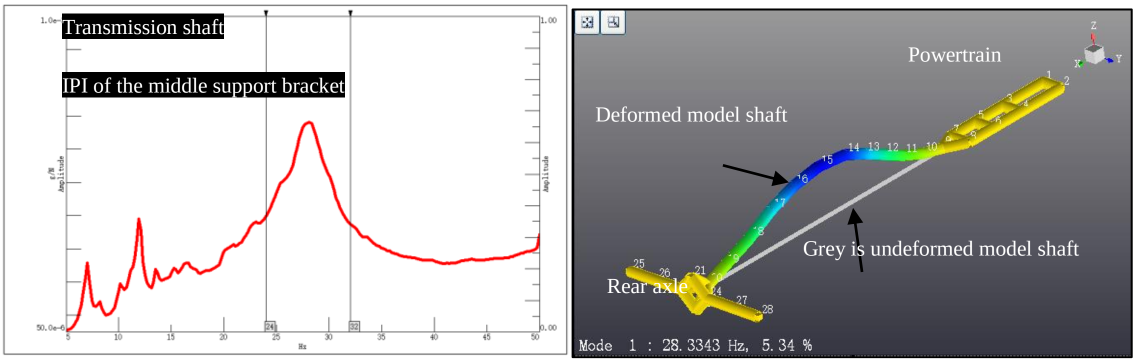

Fig. 6. IPI of middle support bracket and rigid modal of transmission shaft.

C. *Diagnosis Summary*

From the above transmission path analysis, it can be seen that the mechanism of causing vehicle body vibration at 5-30 km/h is that the second-order excitation energy of the transmission system generates strong low-frequency resonance between the middle support and the body. Due to the fact that the modal frequencies of the middle support rigid body and the first order torsion of the vehicle body meet the design specifications, and the high cost of adjustment in the later stage of vehicle development, the optimization control strategy adopted in this paper is to reduce the excitation energy of the excitation source.

IV. Optimization Plan

When the transmission shaft system is connected by a cross shaft type rigid universal joint, the speed of the driven shaft changes twice in a rotation cycle, and the angular speed of the driven shaft is uneven, so the second order excitation of the transmission system will be generated. The magnitude of the second-order excitation energy of the transmission system is related to the equivalent angle of the transmission system θ_e [1]. The cross shaft universal joint commonly used in engineering is required the equivalent angle $\theta_e < 3°$ to reduce the impact of rotational non-uniformity on vibration and noise during universal joint transmission.

A. *Introduction to Equivalent Angle of Transmission System*

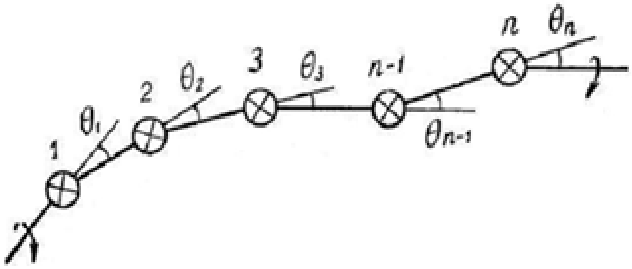

Fig. 7. Schematic diagram of transmission shaft arrangement angle.

As shown in Figure 7, θ_1, θ_2, θ_3,…… θ_N are the angle between the input and output shafts of each universal joint of the transmission shaft. The calculation formula for the equivalent angle of the transmission shaft is as follows: $\theta_e = \sqrt{\left|{\theta_1}^2 \pm {\theta_2}^2 \pm \cdots \pm {\theta_{n-1}}^2 \pm {\theta_n}^2\right|}$

The positive and negative signs in the formula are defined as follows: the plane where the active fork of the first universal joint is located is defined as A, the plane where the active fork of the other universal joint coincides with plane A is positive, and the vertical plane is negative [3]. It can be seen from the calculation formula of equivalent angle of the transmission system that the equivalent angle is related to the fork phase of each universal joint and the angle of each axle, so it is necessary to determine the equivalent angle of the real vehicle transmission system with vibration problem.

B. *Confirmation of Actual Vehicle Equivalent Angle*

The transmission system layout scheme of the real vehicle is shown in Figure 8. The first to the third universal joints are vertical, vertical and horizontal respectively. According to the calculation formula of the equivalent included angle of the transmission system, the first and second universal joint take positive value, and the third universal joint takes negative value.

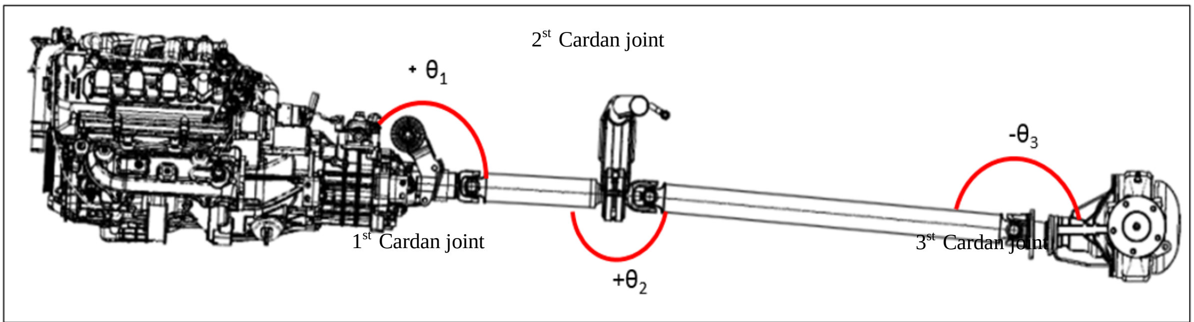

Fig. 8. Transmission system layout plan.

Perform a three-coordinate scanning of the actual vehicle transmission system and compare the angle differences between the actual vehicle and the design state of the transmission system, as shown in Table 2.

After calculation, the equivalent angle of the actual vehicle is 6.08°, which is much greater than the design value of 0.71°, especially the first angle θ_1 and the second angle θ_2. There is a significant deviation from the design value.

Table 2. Actual Vehicle Transmission Angle and Design standard.

Empty State				
	θ_1	θ_2	θ_3	**Equivalent Angle**
Design Value	0°	1.37°	1.19°	0.71°
Real Vehicle Scanning	3.4°	5.05°	0.29°	6.08°

After comparing the differences between the three-coordinate scanning data of the actual vehicle and the design data, as shown in Figure 9, it was found that the bottom surface of the middle support bracket (yellow) of the actual vehicle is 17.46mm higher than the bottom surface of the design state (gray), which is the reason for the excessive equivalent angle of the transmission system.

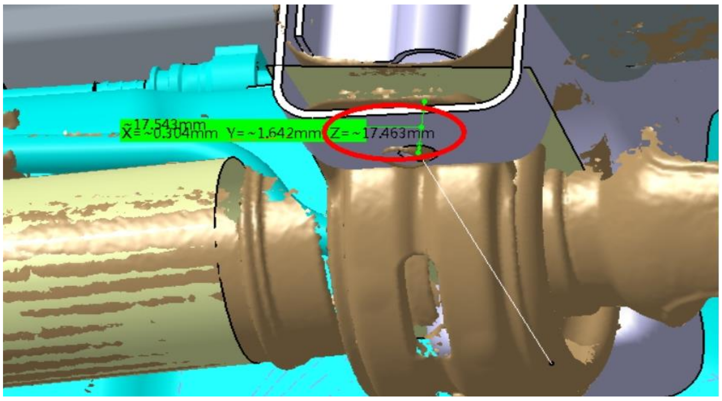

Fig. 9. Comparison between the actual vehicle and design state of the middle support.

C. *Verification of the Effectiveness of Manual Solutions*

To quickly determine the cause of the vibration, an 18mm shim was added at the position of the connecting bolt between the middle support and the transmission shaft to reduce the height of the middle support and bring it closer to the design angle of the transmission system. At this time, the equivalent angle is 1.5° (meeting the requirement of less than 3°), as shown in Table 3.

Table 3. Equivalent angle after adding shims to the middle support of the transmission shaft.

Empty State				
	θ_1	θ_2	θ_3	**Equivalent Angle**
Design State	0°	1.37°	1.19°	0.71°
Adding Shim	0.63°	0.5°	1.7°	1.50°

The measured maximum value of seat vibration decreased from 0.072g to 0.024g, as shown in Figure 10, and the subjective evaluation of vibration completely disappeared. At this point, it can be confirmed that the cause of the vibration of this vehicle is due to the deviation of the middle support position, which leads to the deviation of the equivalent angle of the transmission system, resulting in a larger second-order excitation of the transmission system.

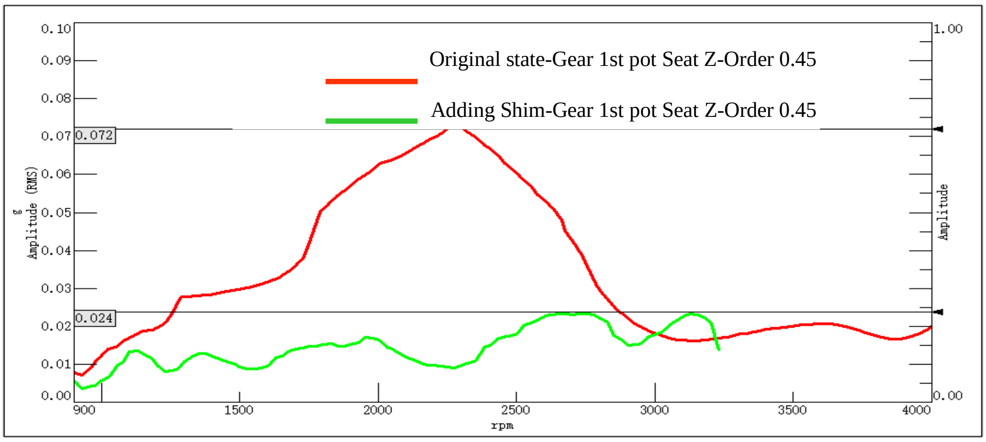

Fig. 10. Effect of manual solution.

A detailed traceability investigation was conducted on the middle support bracket, and it was found that the middle support bracket and the bracket tube beam were welded, as shown in Figure 11. The welding between the bracket and the pipe beam is significantly out of tolerance, with a measured value of 17.46mm (standard ± 2mm). Therefore, it is confirmed that the vibration problem of this vehicle is caused by the unqualified manufacturing of components, resulting in the vibration problem. It is necessary to strictly control the welding accuracy of the middle support in the later stage to ensure that the equivalent angle of the transmission system meets the design requirements, in order to avoid the vehicle vibration problem caused by the large second-order excitation of the transmission system.

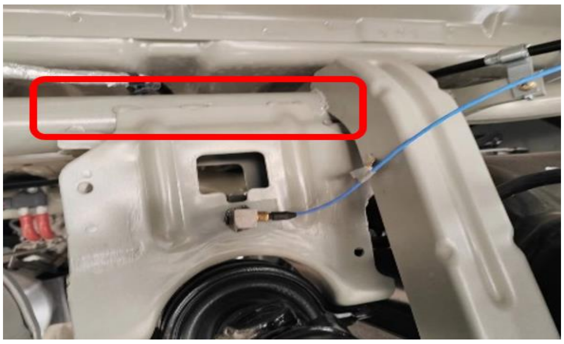

Fig. 11. Welding status of the middle support of the transmission shaft and the pipe beam.

V. Conclusion

(1) Based on the diagnostic analysis of the acceleration vibration problem of a light truck, the transmission path analysis was used to determine the mechanism of vibration generation, which is that the second-order excitation of the transmission system is too large, and the resonance is caused by the excitation of the middle support mode and the first-order torsional mode of the entire vehicle, resulting in the vibration problem.

(2) This paper theoretically introduces the equivalent angle of the transmission system and its influencing factors. By conducting a three-coordinate scanning of the actual vehicle's transmission system and manually rectifying it, it was confirmed that the manufacturing of components was not qualified, resulting in the transmission system equivalent angle exceeding the standard, which led to the vibration problem of the entire vehicle.

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