

# Dynamic characteristics and experimental study of rigid-flexible coupling for rail heavy-duty maintenance vehicle

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**Abstract.** Railway large-scale maintenance vehicles are key equipment for track maintenance. To investigate their dynamic operational characteristics, a dynamics model incorporating multi-rigid-flexible coupling components was developed, and its validity was verified through full-scale vibration tests. The results indicate that: gear system excitation has a significant influence on the vibration characteristics in the  $X$ -direction and  $Y$ -direction of the vehicle system, so in the dynamic simulation model of heavy-duty maintenance vehicles, gear system excitation load must be considered. As the center distance between the drive gearbox and axle decreases, the root-mean-square value value of vibration of the axleboxes on both sides are reduced, adjusting the center distance while ensuring adequate transmission space of the bogie power axle can effectively control the vibration of the axleboxes on both sides; The vibration response of the left axlebox is always larger than that of other positions under all working conditions, indicating that the side wheel and shaft far from the driving gearbox are more prone to encounter slackness failure induced by dynamic excitation. Therefore, during design stage, it is recommended to increase the axle interference fit.

**Keywords:** rail heavy-duty maintenance vehicle, rigid-flexible coupling, dynamic characteristics, vibration test, wheelset slackness.

## 1. Introduction

China's railway industry has experienced rapid development, with the total operational mileage reaching 162,000 kilometers by the end of 2024. Ballast track, as the most predominant form of railway track structure, gradually suffer from performance degradation under the coupling effects of cyclic mechanical loads from trains and complex ambient excitation [1], posing a significant threat to train operation safety. To ensure the safe operation of ballast tracks, regular maintenance by dedicated railway maintenance vehicles is essential.

Rail heavy-duty maintenance vehicles play a vital role in ensuring the safe operation of trains in China [2]. Domestic and foreign scholars have conducted extensive research on railway vehicles. In terms of experimental testing, tests were carried out on the operation process of engineering vehicles to obtain the lateral resistance characteristics of ballast beds [3, 4], and the relationship among sleeper lateral resistance, lateral load and the displacement of track slab were fitted [5]. Based on test data, an optimal operation parameter matching prediction model was constructed [6]. In terms of rigid-flexible coupling models, establishing a coupling model of the stabilizing device and track [7, 8], a dynamic model of flexible bogie for high-speed train [9, 10] and a coupling dynamics model of flexible gearbox and wheelsets [11, 12], analyzing the interaction between the stabilizing device and the rail, the damage of the bogie of high-speed train [13] and the vibration characteristics of drive components [14]. In addition, researchers analyzed

the influence of heavy train gear excitation [15-20] and engineering vehicle operation parameters [21] on vehicle and sleeper vibration [22] responses, and investigated the influence of different tamping unit [23, 24] on ballast beds and the track safety after the operation [25].

To sum up, domestic and foreign experts and scholars have conducted extensive research on high-speed trains, heavy trains, and track ballast beds, etc. However, there are few studies on the service characteristics of rail heavy-duty maintenance vehicles, and no in-depth research has been conducted, from the perspective of multi-rigid-flexible coupling dynamics, on the relevance of complex disturbances, such as driving gear drive excitation, track irregularity excitation, and rail line characteristics, to dynamic operation characteristics of rail heavy-duty maintenance vehicles.

Therefore, considering track irregularity excitation, drive system excitation, etc., establishing a multi-rigid-flexible coupling components dynamics model of rail heavy-duty maintenance vehicle, incorporating gear pair, axle, bogie and car body, conducting vibration test on these maintenance vehicles to investigate the influence laws of driving gear, arrangement of driving gearbox, and rail line on system vibrations. This study reveals the dynamic mechanical characteristics of rail heavy-duty maintenance vehicles in service, and provided theoretical support for railway track maintenance and operation.

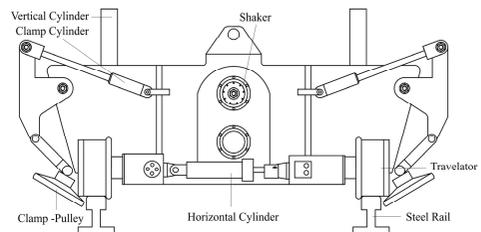
## 2. Dynamic modeling of rigid-flexible coupling systems for rail heavy-duty maintenance vehicles

### 2.1. Rigid-flexible coupling dynamic modeling for rail heavy-duty maintenance vehicles

The stabilizer, as critical equipment in rail heavy-duty maintenance machinery, is shown in Fig. 1(a). During track maintenance, the walking wheels and clamping wheels of the stabilizing device grip the rails, while hydraulic cylinders apply vertical downward pressure to induce overall track sinking. The vibrator generates lateral excitation forces that transmit sequentially through the walking wheel-rail-sleeper-ballast chain. Under external forces, ballast particles gradually compact, increasing ballast bed density and enhancing stability. The stabilization device is illustrated in Fig. 1(b).



a) Ballast stabilizer



b) Stabilization unit

**Fig. 1.** Rail heavy-duty maintenance stabilization vehicle.

Photo was taken by the author Liu Zhiyang in Chengdu, June 2024

Fig. 2 (a) shows the solid model of the large-size maintenance vehicle, with the bogie illustrated in Fig. 2(b). The bogie primarily consists of 2 wheelsets, 2 traction gearboxes, 4 axleboxes, and a bogie frame. Each wheelset is equipped with 1 traction gearbox and 2 axleboxes. The input gear shaft transmits power through spiral bevel gear meshing to drive the wheelset rotation, propelling the maintenance and stabilization vehicle during operation. The gear system is detailed in Fig. 2(c).

### 2.2. Multi-body rigid-flexible coupling model for rail heavy-duty maintenance vehicles

The multibody dynamics model of the large maintenance vehicle was established in

SIMPACK, with the connections between components shown in Fig. 3. The wheels and the track are connected through a rolling contact pair. Both ends of the axle are rigidly connected to the axle boxes, with only the rotational degree of freedom about the axle released. The drive gearbox is rigidly connected to the axle, also releasing only its rotational degree of freedom about the axle. The upper part of the gearbox housing is connected to the bogie frame via a stiffness-damping composite unit, which simulates the gearbox suspension device. The meshing of the spiral bevel gear pair inside the gearbox is simulated using a stiffness-damping model. The bogie frame is connected to the axle boxes through hydraulic dampers and rubber springs, and the frame is connected to the car body through the car body springs.

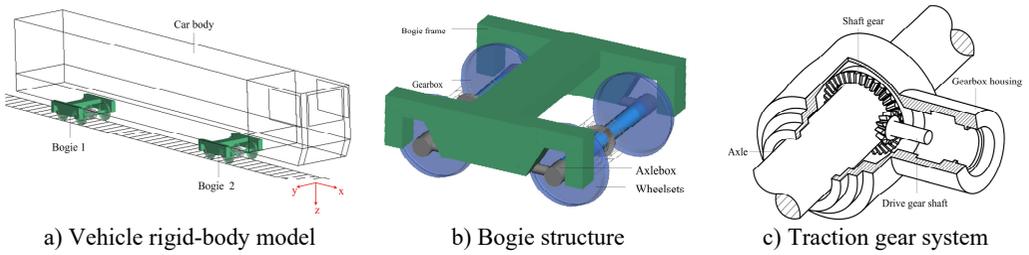


Fig. 2. Solid model of rail heavy-duty maintenance vehicle

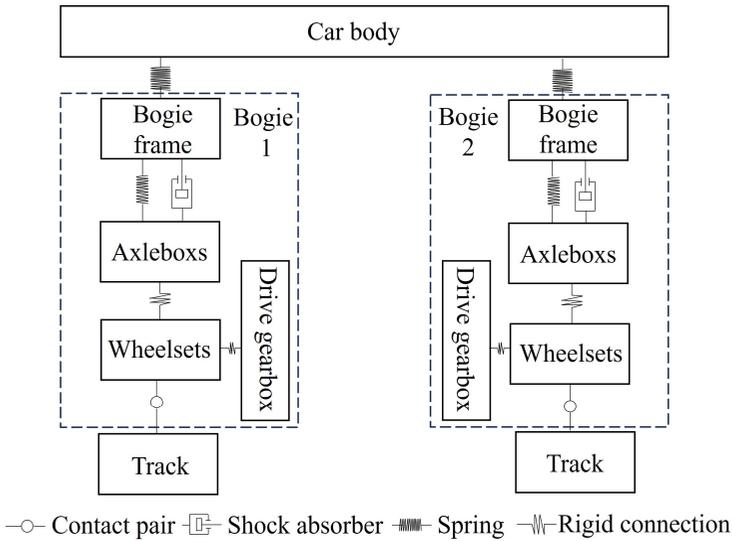


Fig. 3. Connection relationships of vehicle main components

Track irregularity excitation based on measured railway track spectra was applied to the simulated track model. Track irregularity excitation is as shown in Fig. 4.

Wheel-rail rolling contact constraints were enforced to ensure continuous wheel-rail kinematic coupling, while a driving torque was applied at the pinion shaft to propel the wheelset. This approach yielded a complete multibody dynamics model of a Rail heavy-duty Maintenance Vehicles, capturing its essential dynamic characteristics.

The structural components, including the bogie frame, wheelsets, and gearbox housing, were discretized into finite element meshes using HYPERMESH software (Fig. 5). Following rigorous mesh quality verification, the resulting mesh models comprised 24,494, 96,816, and 139,621 elements respectively, meeting the convergence criteria for subsequent dynamic analysis.

The yellow reference points indicated in the figure were selected from the flexible models of the bogie frame, wheelset, and gearbox housing to generate flexible substructures via master nodes. These substructures were subsequently imported into SIMPACK to develop flexible-body

models of the bogie frame, wheelset, and gearbox housing, which replaced their rigid counterparts. This process ultimately yielded a comprehensive rigid-flexible coupled dynamics model of the rail heavy-duty maintenance vehicles. A comparison of the frequencies of the non-rigid body modes between the reduced substructure of the frame and the finite element model is shown in the Table 1.

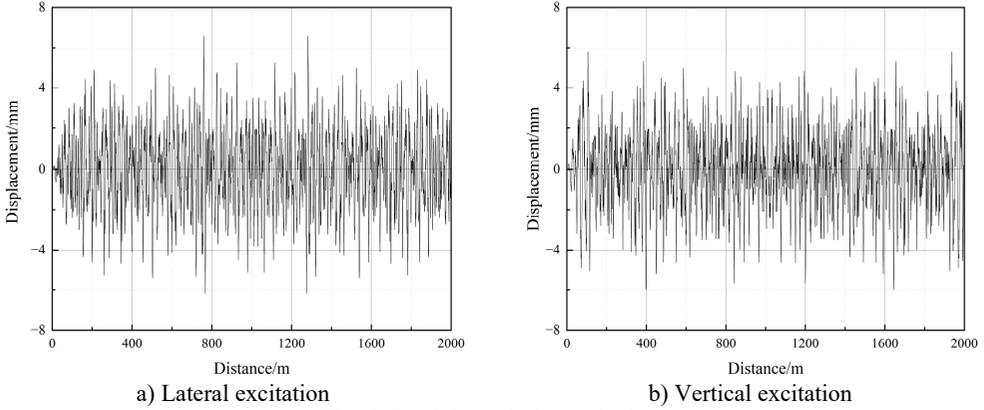


Fig. 4. Track irregularity excitation

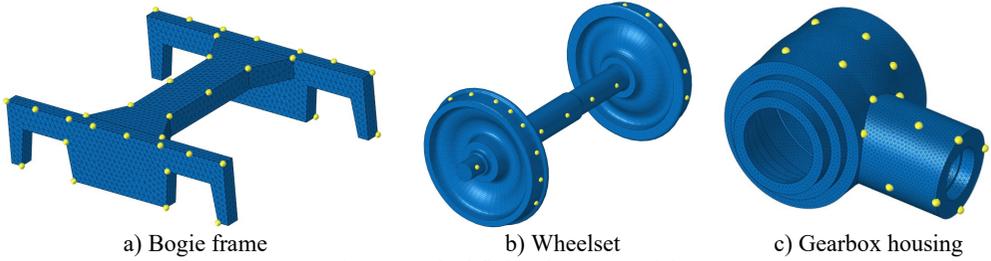


Fig. 5. Meshed finite element model

Table 1. Frequencies of the non-rigid body modes

Order	Reduced substructure modal frequency / Hz	Finite element model modal frequency / Hz	Error / %
1	139.98	139.31	0.48
2	140.19	139.51	0.49
3	268.17	266.96	0.45
4	268.95	267.75	0.45
5	292.72	290.78	0.66
6	294.08	292.13	0.66

As shown in Table 1, the comparison of the elastic modal frequencies between the reduced substructure and the global finite element model shows an error of less than 1 %, demonstrating excellent agreement and validating the applicability of the substructure model for rigid-flexible coupled dynamic simulations.

### 3. Multi-flexible-body coupling analysis and testing of rail heavy-duty maintenance vehicles

#### 3.1. Multi-flexible-body coupling dynamics analysis

Applying a driving torque at the gearbox input shaft, the large railway maintenance vehicle accelerated from rest to 70 km/h and then maintained at that speed. The simulation torque was set to match the actual motor output torque. The dynamic responses of the gearbox, bogie frame, and

axle box were computed using the SIMPACK multibody dynamics software. A schematic of vehicle dynamics analysis points is given in Fig. 6.

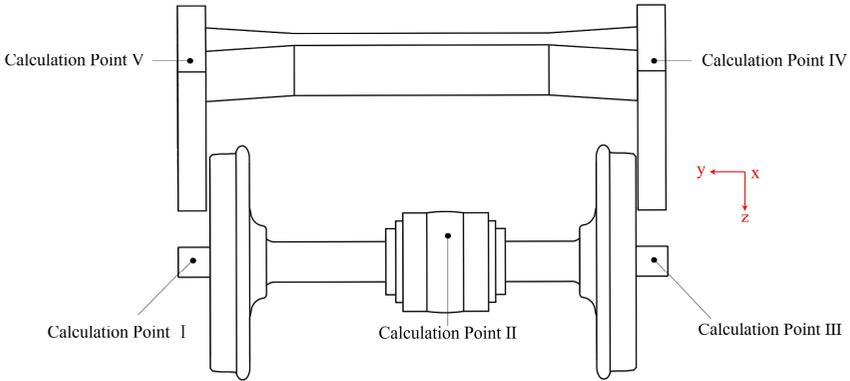


Fig. 6. Computational points for multi-flexible-body coupled dynamics analysis of the vehicle

Point I is located on the left axlebox surface, Point II is located on the drive gearbox housing surface, Point III is located on the right axlebox surface, Point IV is located on the right side of the bogie frame, Point V is located on the left side of the bogie frame.

Due to space limitations, only the X-direction, Y-direction, and Z-direction vibration acceleration responses of Point II (gearbox housing) are presented.

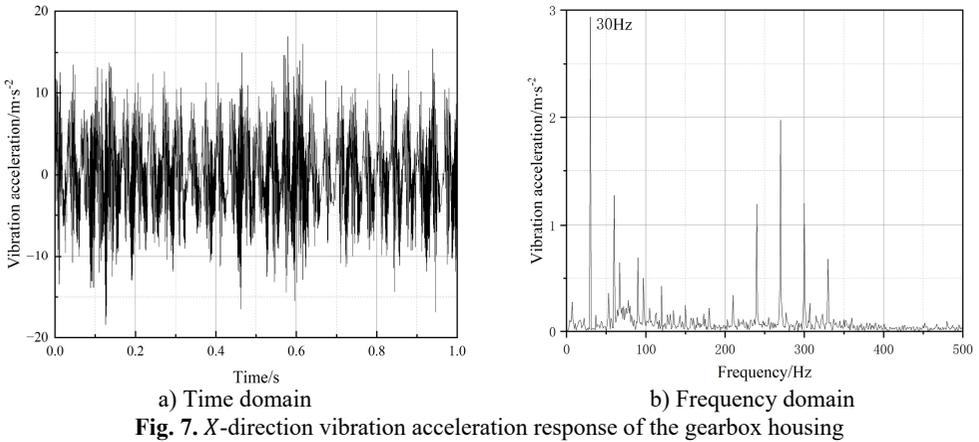


Fig. 7. X-direction vibration acceleration response of the gearbox housing

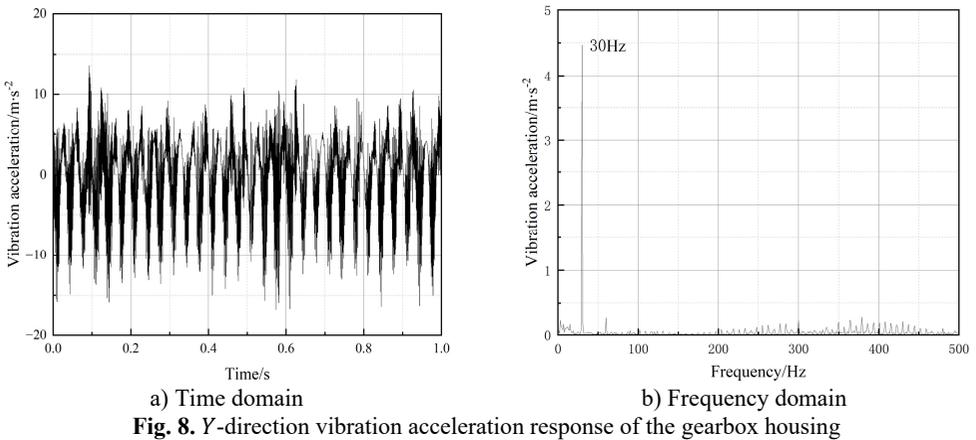
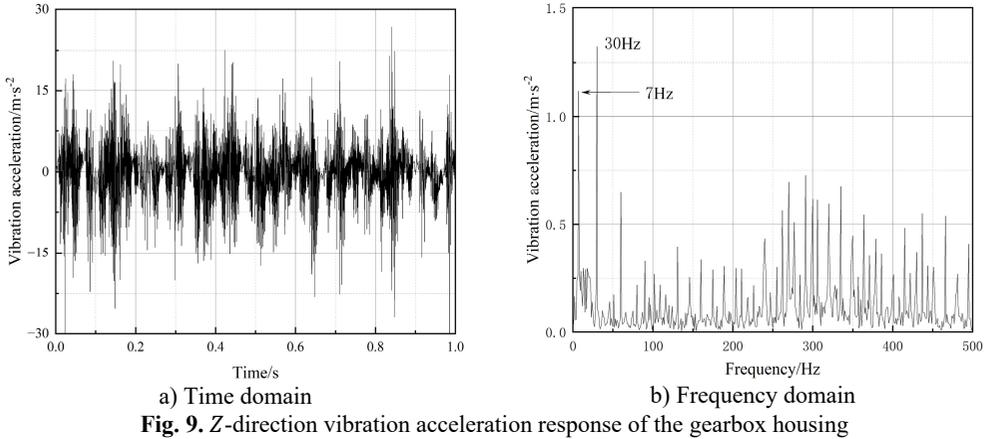


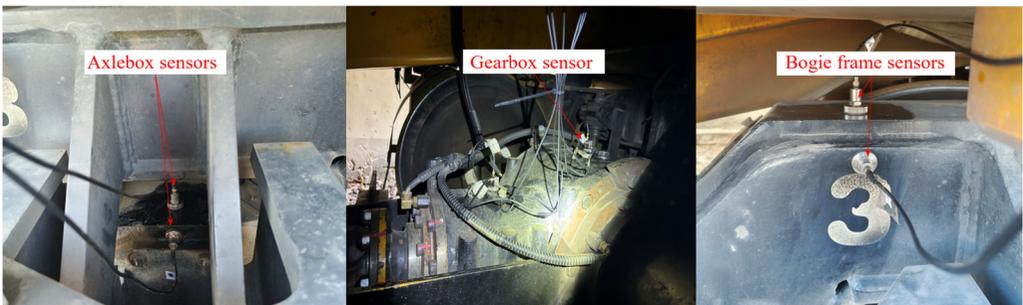
Fig. 8. Y-direction vibration acceleration response of the gearbox housing



The analysis indicates that among the vibration acceleration responses of the axleboxes, gearbox housing, and bogie frame, The maximum root-mean-square (RMS) value occurs in the Z-direction of the left axlebox. Both the left axlebox and right side of the bogie frame exhibit higher vertical vibration RMS values compared to their right and left counterparts, respectively; The frequency domain responses of the gearbox housing’s X-direction, Y-direction, and Z-direction vibration accelerations all contain a dominant 30 Hz component, matching the gear meshing frequency; A 7 Hz spectral component appears in the Z-direction frequency response of the housing, corresponding to the wheel rotational frequency. This primarily results from significant and asymmetric Z-direction vibrations between the left and right wheel-axle assemblies, which induce shaft misalignment and consequently generate rotational frequency components.

### 3.2. Vibration testing and analysis of rail heavy-duty maintenance vehicles

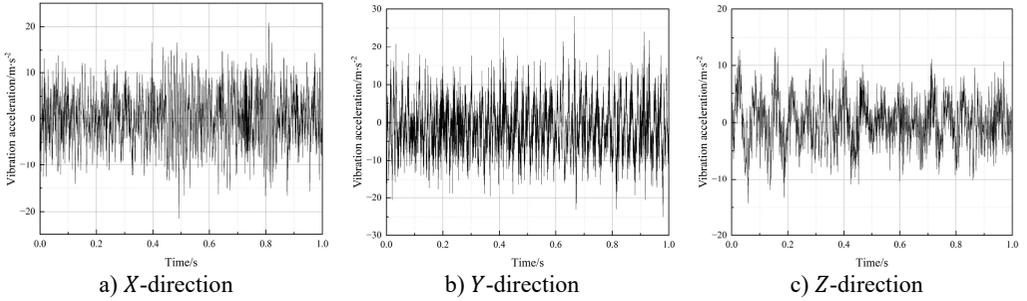
Vibration response testing was conducted at the Chengdu Railway Maintenance Depot in China to validate the multi-flexible-body coupled dynamics model of rail heavy-duty maintenance vehicles. Fig. 10 shows the test point arrangement, where the measurement locations correspond exactly with the computational points.



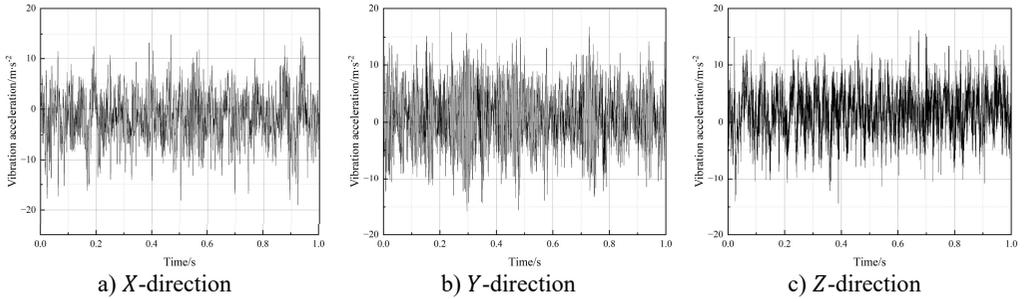
**Fig. 10.** Measurement point arrangement. These photographs were taken by the author Liu Zhiyang in Chengdu, June 2024

Prior to testing, all measurement points on the large maintenance vehicle were ground and marked. The magnetic bases of the sensors were then placed at each measurement point. During testing, the vibration signals (acceleration vibration signals) measured by the triaxial accelerometers under various working conditions were amplified by a charge-to-voltage filter-integration amplifier. These signals were then fed into an intelligent signal acquisition and

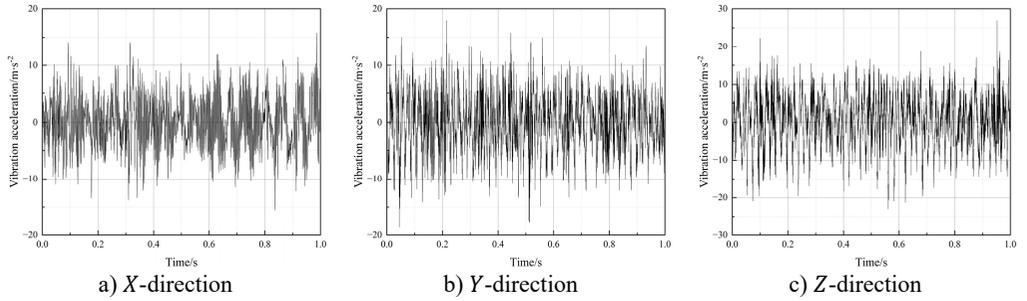
processing analyzer for collection. Through the intelligent data acquisition and signal analysis system, vibration acceleration and the frequency spectrum at each measurement point were recorded and analyzed. The tri-directional vibration responses of each measurement point obtained after analysis by the signal analysis system are shown in Figs. 11 to 15.



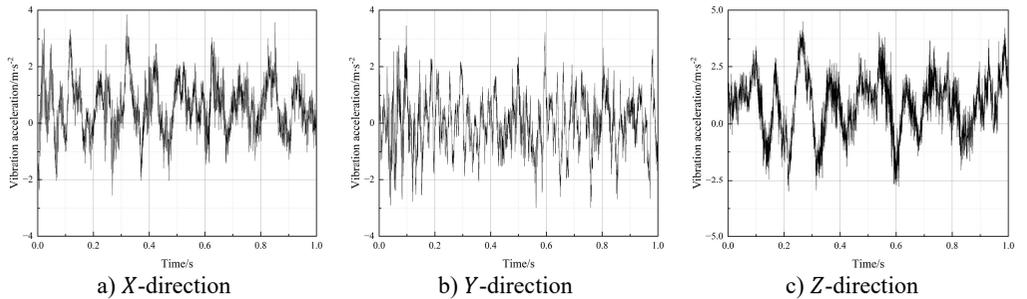
**Fig. 11.** Vibration acceleration response of the left axlebox (Calculation Point I)



**Fig. 12.** Vibration acceleration response of the gearbox housing (Calculation Point II)



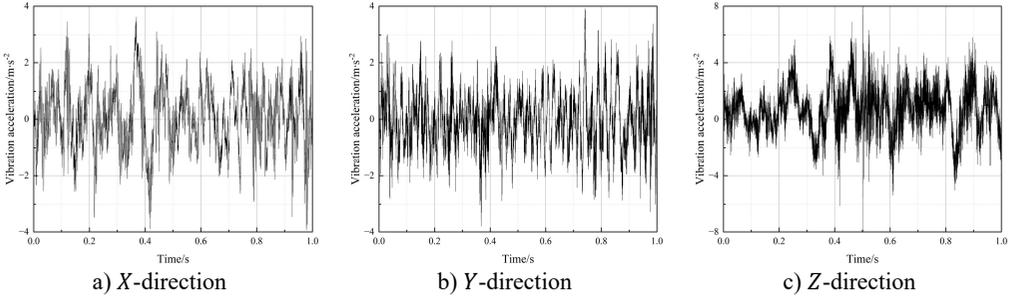
**Fig. 13.** Vibration acceleration response of the right axlebox (Calculation Point III)



**Fig. 14.** Vibration acceleration response of the bogie frame right side (Calculation Point IV)

Table 2 presents the comparison between measured and simulated RMS values of vibration

responses for the axleboxes, gearbox housing, and bogie frame. The analysis shows that the measured and simulated longitudinal vibration accelerations of the gearbox housing, bogie frame, and axleboxes exhibit good agreement, validating the rationality of the dynamic model. Subsequent analysis will further investigate the influence patterns of drive configuration, load distribution, and operating routes on system vibration characteristics.



**Fig. 15.** Vibration acceleration response of the bogie frame left side (Calculation Point V)

**Table 2.** Comparison of measured and simulated RMS values of vibration responses for calculation points

Calculation Points	Directions	Test value / $m \cdot s^{-2}$	Analysis value / $m \cdot s^{-2}$	Error
Left axlebox (Calculation Point I)	X-direction	3.53	4.01	13.60 %
	Y-direction	5.49	4.73	13.84 %
	Z-direction	7.35	6.21	15.51 %
Gearbox housing (Calculation Point II)	X-direction	5.00	4.67	6.60 %
	Y-direction	5.33	4.61	13.51 %
	Z-direction	4.56	5.12	10.94 %
Right axlebox (Calculation Point III)	X-direction	3.85	3.15	18.18 %
	Y-direction	4.47	4.73	5.82 %
	Z-direction	7.01	5.50	21.54 %
Bogie frame right side (Calculation Point IV)	X-direction	1.16	1.27	8.66 %
	Y-direction	1.06	0.92	13.21 %
	Z-direction	1.65	1.81	9.70 %
Bogie frame left side (Calculation Point V)	X-direction	1.04	0.96	7.69 %
	Y-direction	1.08	0.90	16.67 %
	Z-direction	1.89	1.90	0.53 %

#### 4. Dynamic response of vehicle system considering gear drive excitation, gearbox arrangement and track characteristics

##### 4.1. Influence of gear drive excitation on vehicle system vibration characteristics

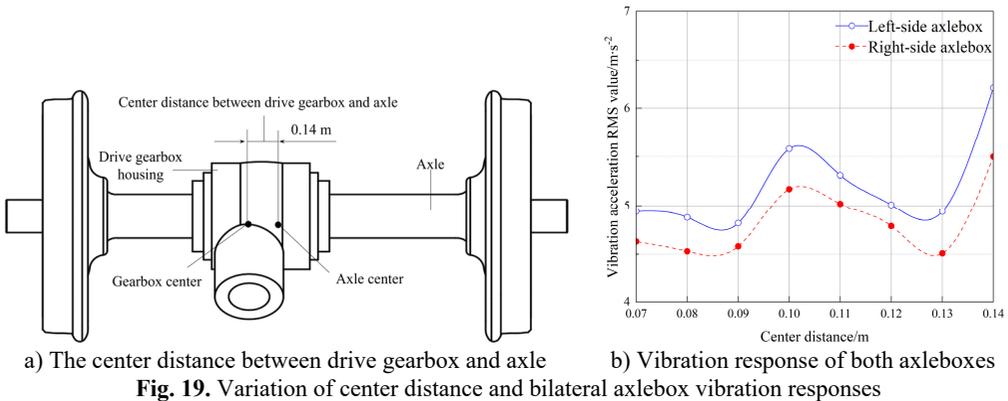
Study on vibration characteristic variations of vehicle system with/without gear drive excitation based on multi-flexible-body coupling dynamics modeling approach. A model of a heavy-duty Maintenance Vehicle without a gear transmission system was established. Torque was applied around the axial rotation direction at the center of the axle to drive the vehicle forward, as shown in Fig. 16(a). Fig. 16(b) illustrates the traction gear system where torque is transmitted through the meshing action between the input shaft gear and the axle gear to drive the vehicle forward.

Fig. 17 presents the vibration response curve of the left axle box (calculation point I) when a vehicle model with or without a gear transmission system is traveling at a constant speed of 70 km/h.

The analysis indicates that the gear excitation has the most significant influence on the X-direction and Y-direction vibration characteristics of the vehicle system. Under the effect of gear excitation, the root-mean-square (RMS) value of X-direction vibration acceleration of the left



and axle.



Adjusting the center distance between the drive gearbox and axle, the influence of changes in the center distance between the drive gearbox and axle on the vertical vibration responses of both sides of the axleboxes is analyzed as shown in Fig. 19(b). As the center distance decreases, the RMS values of vibration at both left and right axleboxes will reduce, with a maximum reduction amplitude reaching 20.45 %.

Table 3 presents the root mean square values of the vertical and lateral wheel-rail forces corresponding to different gearbox center distances. Analysis of the table shows that the RMS values of the vertical force exhibit no significant change under different center distances, while the variation trend of the lateral force is consistent with that of the axle box vibration shown in Fig. 20(b), with a maximum reduction of 15.8 %. This indicates that the change in the gearbox position affects the vibration amplitude of the wheelset, which in turn leads to a variation in the contact area between the wheel flange and the rail, ultimately influencing the lateral wheel-rail force. While ensuring the transmission space of the bogie's power axle, reasonably adjusting the center distance between the drive gearbox and axle helps control vibration at both axleboxes.

**Table 3.** RMS values of vertical and lateral wheel-rail forces under different center distance

Center distance / m	Vertical force / kN	Lateral force / kN
0.07	182.09	12.89
0.08	182.08	12.73
0.09	182.10	12.49
0.10	182.26	14.43
0.11	182.11	13.59
0.12	182.15	12.15
0.13	182.09	12.60
0.14	182.27	14.41

### 4.3. Influence of operational track characteristics on vehicle system vibration characteristics

To investigate the influence patterns of different operational routes on system vibration characteristics, three types of operating routes were established: a straight line, a curved section with a radius of 1200 m, and a gradient section with a slope of 6 %. Fig. 20 presents the RMS values of vibration acceleration responses at various computational points of the vehicle when traveling at constant speed through curved and sloped track sections. Fig. 21 presents the wheel-rail forces as the vehicle passes through different route conditions.

Analysis shows that when the vehicle is traveling uphill, the longitudinal vibration of the drive gearbox housing increases by 4.61 % compared to straight-line operation, the vertical vibration of the left axlebox increases by 10.90 % compared to straight-line operation, and the vertical

vibration of the right axlebox increases by 6.00 % compared to straight-line operation; When a vehicle negotiates a left-hand curve, centrifugal force causes the flange of the outer wheel to have increased contact area with the rail, leading to a rise in the lateral force on the wheel. Compared to straight-track operation, the peak lateral force increases by 147.34 %. Concurrently, the vertical vibration of the left-side axle box amplifies by 23.19 %. In contrast, the contact area between the inner wheel flange and the rail diminishes, resulting in a 3.82 % reduction in the vertical vibration of the right-side axle box compared to straight-line travel. The vibration response of the left axlebox (Calculation Point I) is consistently greater than other locations under all operating conditions, indicating that wheels and axles on the side opposite the gearbox are more susceptible to dynamic excitation effects leading to loosening failures, therefore the wheel-axle interference fit should be increased during design.

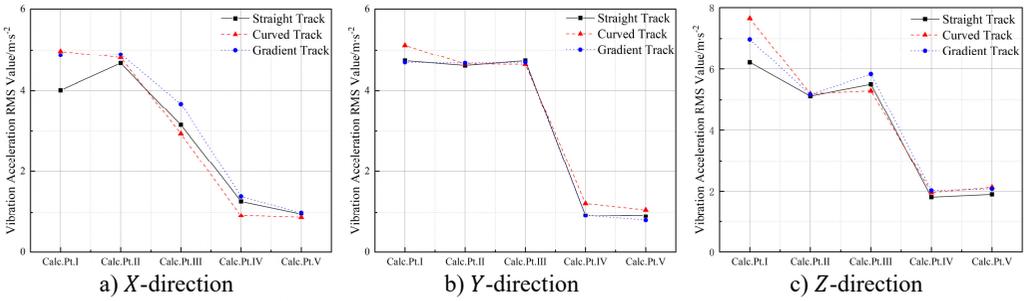


Fig. 20. Vibration responses at vehicle measurement points

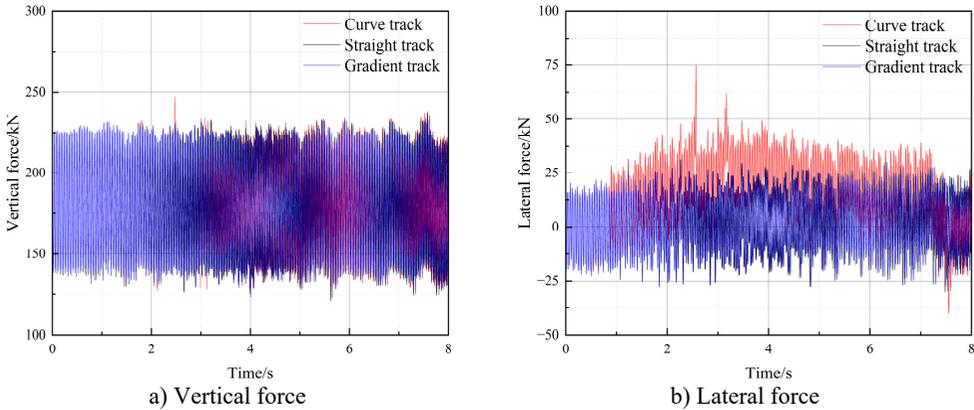


Fig. 21. Wheel-rail forces under different route conditions

## 5. Conclusions

1) A multi-flexible-body coupled dynamics model for railway large-size maintenance vehicles was established, and the rationality of the dynamic model was verified through comparative analysis between experimental test data and simulation results, demonstrating that this model can be effectively applied to analyze the vibration characteristics of vehicle systems.

2) Under gear excitation conditions, the RMS values of X-direction and Y-direction vibration accelerations at the vehicle's left axlebox increased by 591.38 % and 605.97 % respectively, while the peak values of wheel-rail vertical and lateral forces increased by 23.68 % and 236.79 % respectively. These results demonstrate that gear system excitation loads must be included in the dynamic simulation models of large-size maintenance vehicles.

3) Under gear excitation conditions, the RMS values of X-direction and Y-direction vibration accelerations at the vehicle's left axlebox increased by 591.38 % and 605.97 % respectively, while

the peak values of wheel-rail vertical and lateral forces increased by 23.68 % and 236.79 % respectively. These results demonstrate that gear system excitation loads must be included in the dynamic simulation models of large-size maintenance vehicles.

4) Compared to straight-line operation, when the vehicle is traveling uphill, the *X*-direction vibration of the drive gearbox housing increases by 4.61 %, while the *Z*-direction vibrations of the left and right axleboxes increase by 10.90 % and 6.00 % respectively; during curve negotiation, the *Z*-direction vibration of the outer-side (left) axlebox increases by 23.19 %, whereas the *Z*-direction vibration of the inner-side (right) axlebox decreases by 3.82 %. The vibration response of the left axlebox remains greater than other locations under all operating conditions, indicating that wheels and axles on the side opposite the gearbox are more susceptible to dynamic excitation effects leading to loosening failures. Therefore, the wheel-axle interference fit should be increased during design.

## Acknowledgements

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## Data availability

The datasets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

## Author contributions

Zhiyang Liu: investigation, writing - original draft preparation, visualization, methodology. Zeyin He: conceptualization, funding acquisition, supervision, project administration. Runjie Duan: investigation. Junjie Yang: resources. Gaohua Hu: resources.

## Conflict of interest

The authors declare that they have no conflict of interest.

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