

Article

Flow-Induced Vibration Analysis by Simulating a High-Speed Train Pantograph

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Abstract: This paper investigates the aerodynamic behavior and dynamic characteristics of high-speed train pantographs under various operating conditions using advanced aerodynamic simulations and dynamic analyses. The simulations show significant fluctuations in aerodynamic loads during tunnel entry and exit, heavily influenced by train speed and pantograph position (raised/lowered). Modal simulations reveal distinct low-frequency vibrations in pantographs, significantly impacted by external aerodynamic forces. Importantly, the lowered position exposes the pantograph to upward aerodynamic forces, leading to increased bow-net contact force and off-line rate, ultimately compromising current collection stability. Both maximum contact force and off-line rate further increase with higher train speeds. To improve pantograph design, the paper proposes adjustments to the airbag's equivalent spring stiffness and the bow head's density. These modifications aim to mitigate contact force and enhance the stability and reliability of pantographs at high speeds. This research offers theoretical and practical insights, aiding in the design, optimization, and refinement of future pantograph systems.

Keywords: aerodynamic load; contact force mitigation; flow-induced vibration; high-speed train pantograph; modal simulation



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1. Introduction

With the increase in the operational speeds of trains, the aerodynamic loads experienced by the trains increase proportionally with the square of the speed. The complex environments in which high-speed trains operate lead to significant increases in the vertical, longitudinal, and lateral vibrations of the pantographs under aerodynamic load. This results in an exacerbation of pantograph disengagement and a severe deterioration in current collection quality. Additionally, the dynamic loads make the pantograph components susceptible to fatigue damage, thus making the analysis of flow-induced vibrations a critical consideration in the design of high-speed pantographs. Currently, research on the pantograph-airflow interaction, both domestically and internationally, primarily focuses on three aspects: the aerodynamics of pantographs, structural vibrations, and flow-induced vibrations.

The study of the aerodynamic characteristics of pantographs is mainly conducted through experimental research and numerical calculations. A significant number of scholars have used experimental methods to study the aerodynamic properties of pantographs. For instance, Pombo et al. [1] conducted wind tunnel experiments on the aerodynamic characteristics of Faiveley CX type pantographs under crosswind conditions using sensor technology to measure the aerodynamic force coefficients of pantograph components and

articulations. Zhang et al. [2] tested the aerodynamic drag and lift on Faiveley pantographs at speeds of 400 km/h to 600 km/h under raised and lowered conditions in a $2.4\text{ m} \times 2.4\text{ m}$ wind tunnel. Additionally, numerical calculation methods typically involve creating aerodynamic models of pantographs and using fluid analysis software combined with turbulence models such as RANS to analyze the aerodynamic performance of pantographs. For example, Yang et al. [3] utilized Fluent to establish a steady-state aerodynamic fluid calculation model of pantographs, analyzing the impact of aerodynamic forces on the contact pressure between the pantograph and overhead wires. Liu et al. [4] calculated the unsteady aerodynamic forces on various components of pantographs using Fluent software 6. Song et al. [5] developed a fluid simulation model of pantographs, analyzing the surface pressure and lift force. Based on the N-S equations and k- ϵ turbulence models, Li et al. [6,7] constructed a model for calculating the aerodynamic lift force on pantographs and analyzed the lift force as high-speed trains pass through tunnels at 350 km/h. Zhao [8] analyzed the variation in aerodynamic characteristics of pantograph components under crosswind conditions and optimized the aerodynamic shape of some pantograph structures. Lin et al. [9] analyzed the impact of pantograph structural displacement on aerodynamic lift force. Given the pantograph's complex three-dimensional shape, the high-speed airflow causes numerous vortices to shed, resulting in unsteady vibrations under the aerodynamic load. Since steady methods only provide average flow field information, they cannot adequately capture the fluctuations and spectral characteristics of aerodynamic forces. Therefore, to study the flow-induced vibrations of high-speed train pantographs, it is essential to investigate the unsteady aerodynamic performance of pantographs.

In terms of structural vibration analysis of pantographs, the models used include the following: equivalent mass models, multi-rigid body models, rigid-flexible hybrid models, and flexible body models. The equivalent mass model simulates the pantograph with mass blocks, springs, and dampers, generally categorized into single-mass block models, two-mass block models, and multi-mass block models. By establishing a two-mass lock model of the pantograph, Vesely et al. [10] studied its static and dynamic characteristics, verified through experiments. Multi-rigid body models [11,12] consider all pantograph components as rigid bodies and establish the pantograph system's dynamic theoretical model after obtaining the actual dimensions, mass, moment of inertia, and center of mass position of each component. Rigid-flexible hybrid models [13,14], building on multi-rigid body models, replace components such as the bow head and upper arm with flexible bodies to analyze better the vibration characteristics of deformable parts of the pantograph. Flexible body models typically involve a reasonable simplification of the pantograph structure, then establish models for each pantograph component, and finally create the entire pantograph's flexible body model [15,16] through appropriate hinge and spring constraints. In summary, the equivalent mass models' assumed equivalent mass, spring stiffness, and damping parameters are considerable and do not account for the flexible deformation of pantograph rods, making them inaccurate and unsuitable for the analysis and design of bow-nets at higher speeds. Multi-rigid body models and rigid-flexible coupling models, which consider all or part of the structure as rigid, overlook the rods' flexible deformation, leading to significant errors under high-speed conditions. Flexible models, with their high accuracy, are suitable for dynamic analysis of high-speed pantographs.

Research on the flow-induced vibration of pantographs is broadly categorized into experimental studies and simulation research. Experimental investigations include track tests and wind tunnel experiments. For instance, Zhang et al. [17] conducted aerodynamic force testing on the SSS400 type pantograph on the Wuhan–Guangzhou line, examining the impact of aerodynamic forces on pantograph-catenary current collection. Fu et al. [18] carried out tests on high-speed pantographs under crosswind conditions in the FL-9 wind tunnel of the Aviation Industry Aerodynamics Institute, analyzing the aerodynamic forces and wind-induced vibration characteristics of the entire pantograph. Zhang et al. [19] undertook real-vehicle tests on the high-speed pantograph on the Beijing–Shanghai pilot section, obtaining measured data on the pantograph's aerodynamic drag, lift, surface pressure,

structural stress, and flow-induced vibration. Zhang Hong et al. [20] and Zhang et al. [21] conducted wind tunnel experiments on the pantograph, validating theoretical analysis results, and found that the theoretical outcomes largely agreed with the wind tunnel experimental results. Incorporating the structural model and aerodynamic characteristics of the pantograph, some researchers have also analyzed the flow-induced vibration of high-speed pantographs. Based on quasi-steady theory, Bocciolone et al. [22] performed aerodynamic simulations on the pantograph, determining the impact of aerodynamic forces on the dynamic contact force of the bow-net. Wang [23] analyzed the fluid-structure interaction characteristics of the pantograph as high-speed trains enter tunnels, examining the deformation and stress characteristics of the pantograph during operation on open tracks. Chi [24] analyzed the aerodynamic characteristics and the strength of the fluid-structure interaction of pantograph components under aerodynamic forces. Jiang [25] analyzed the aerodynamic characteristics and stress distribution of high-speed pantographs during open track operation and tunnel passage. Yang [26] investigated the external flow field characteristics of high-speed trains at 400 km/h and the distribution of structural stress and deformation of the pantograph under aerodynamic forces. Mei et al. [27] investigated friction-induced vibration and wire corrugation in metro rigid overhead wire-pantograph systems. A friction coefficient above 0.07 significantly increases self-excited vibration and affects wire corrugation. Wang et al. [28] developed a dynamic coupling model for double-pantograph-catenary systems, validated with field tests, revealing the impact of speed, span, and wind on contact force fluctuations. Qin and Yang [29] created a 17-DOF model to analyze the time-varying characteristics of guide roller-rail contact stiffness in super high-speed elevators, showing significant influence on horizontal vibration. Song et al. [30] evaluated crosswind effects on pantograph-catenary interaction, finding acceptable performance at 20 m/s crosswind, but safety issues at 30 m/s, emphasizing the importance of aerodynamic forces. Jackson et al. [31] analyzed aerodynamic forces on HSX pantograph components, showing the impact of configuration on drag forces and emphasizing the importance of aerodynamic modeling. Peng et al. [32] developed an aerodynamic shape design method for pantograph monitoring devices, including numerical simulation and shape optimization, applicable to other small devices. Song et al. [33] explained how higher train speeds can improve pantograph-catenary performance in certain ranges, establishing a model, defining critical speeds, and proposing an indicator to predict optimal speeds. Wu et al. [34] reviewed the pantograph-catenary electrical contact system in high-speed railways, discussing progress, challenges in harsh environments, and outlooks for ultra-high-speed trains. Wang et al. [35] studied fluctuating loads on pantograph-catenary systems, conducting experiments to measure stability and offline rate, and using support vector machines to predict electrical contact failures. Song et al. [36] developed a spatial coupling model for pantograph-catenary dynamics under vehicle-track excitation, considering rail irregularities' impact on contact reliability. Jin et al. [37] conducted an experimental study on pantograph-catenary discharge, showing that voltage, gap distance, and electrode motion affect discharge characteristics, with the vertical motion causing larger pulse current peaks and stronger radiation.

In summary, as trains continue to increase in speed and as pantograph structures evolve towards lightweight designs, the impact of local deformation of pantograph components on the dynamics of the pantograph-catenary interaction becomes increasingly significant. In order to study further the coupling mechanism between the high-speed operation of the pantograph and the high-speed flow field, the equivalent mass, spring stiffness, and damping parameters in the multi-mass model are artificially assumed to be relatively large, without considering the flexible deformation of the pantograph components, resulting in low accuracy; Moreover, the multi-rigid body model and the rigid-flexible coupling model treat all or part of the pantograph structure as a rigid body, ignoring the flexible deformation of the members, resulting in significant errors under high-speed working conditions. This article adopts a new calculation method to establish a fully flexible simulation model of the pantograph coupled with high-speed flow field aerodynamic

simulation results. This article studies the flow-induced vibration of pantographs under different working conditions, analyzes the effects of operating speed, knuckle-upstream and knuckle-downstream operating condition of pantographs, and structural parameters on flow-induced vibration, and proposes optimization suggestions for pantographs. Figure 1 illustrates the research scope and technical roadmap of this paper.

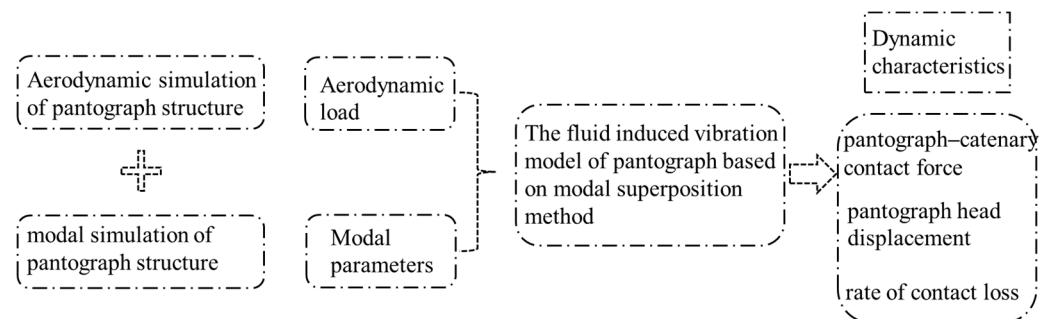


Figure 1. Simulation and analysis structure.

This article is structured as follows: Section 2 introduces the aerodynamic simulation analysis and surface aerodynamic load extraction of high-speed pantographs. Section 3 briefly summarizes the modal simulation analysis of high-speed pantographs. Section 4 discusses the flow-induced vibration simulation method for high-speed pantographs. In Section 5, based on the simulation method proposed in this article, the dynamic response of high-speed pantographs is analyzed, examining their dynamic characteristics under various operational conditions. Finally, Section 6 concludes the study.

2. Aerodynamic Simulation and Analysis of Pantograph

The aerodynamic simulation analysis of pantographs mainly includes the following aspects: First, we introduce the numerical calculation methods used in aerodynamic simulation; then, we establish a pneumatic simulation model and calculation settings for the pantograph; next, we conduct aerodynamic simulation analysis on the aerodynamic load of the pantograph passing through the tunnel and we compare the aerodynamic loads of low-carbon high-speed gimbal under knuckle-upstream and knuckle-downstream operating conditions; finally, we compare the aerodynamic load distribution of pantographs at different operating speeds.

2.1. Numerical Calculation Method

In this article, numerical simulations are performed by Computational Fluid Dynamics (CFD), and the turbulence governing equations chosen is the Realizable k- ϵ turbulence model. This model is suitable for a wide range of flow types, including rotating uniform shear flow, free flow (jet and mixing layer), channel flow, and boundary layer flow. The simulation results for these flow processes are all better than those of the standard k- ϵ model.

We calculate the unsteady N-S equation and use the Boussinesq assumption to obtain the Reynolds equation:

$$\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} = \bar{f}_i - \frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_i} + \nu \frac{\partial^2 \bar{u}_i}{\partial x_i \partial x_j} - \frac{\partial \overline{u'_i u'_j}}{\partial x_j} \quad (1)$$

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (2)$$

where $\tau_{ij} = -\rho \overline{u'_i u'_j}$ is the Reynolds stress.

The Renault stress model closes the equation by establishing the Renault stress transport equation:

$$\frac{\partial \overline{u'_i u'_j}}{\partial t} + \overline{u_k} \frac{\partial \overline{u'_i u'_j}}{\partial x_k} = -\overline{u'_i u'_k} \frac{\partial \overline{u'_j}}{\partial x_k} - \overline{u'_j u'_k} \frac{\partial \overline{u'_i}}{\partial x_k} + \phi_{ij} + D_{ij} - \varepsilon_{ij} \quad (3)$$

where ϕ_{ij} is the Reynolds stress redistribution item, D_{ij} is the Reynolds stress diffusion term, ε_{ij} is the Reynolds stress dissipation item.

The vortex viscosity model was proposed by Boussinesq imitating the viscosity of molecules, namely setting the Reynolds stress as follows:

$$\overline{u'_i u'_j} = \nu_T (U_{i,j} + U_{j,i} + \frac{2}{3} U_{k,k} \delta_{ij}) + \frac{2}{3} k \delta_{ij} \quad (4)$$

where $k = 0.5 \overline{u'_i u'_i}$ is the turbulent kinetic energy, ν_T is the vortex viscosity coefficient. Different vortex viscosity models are formed by adding additional turbulence quantities in Equation (4).

The k equation and the ε equation of the Realizable k - ε model are expressed as follows:

$$\frac{\partial(\rho K)}{\partial t} + \frac{\partial(\rho \overline{u_j} K)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{Pr_K} \right) \frac{\partial K}{\partial x_j} \right] + P_K + G_b - \rho \varepsilon - Y_M \quad (5)$$

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \overline{u_j} \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{Pr_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 \overline{S} \varepsilon - C_2 \rho \frac{\varepsilon^2}{K + \sqrt{\nu \varepsilon}} + C_{\varepsilon 1} \frac{\varepsilon}{k} C_{\varepsilon 3} G_b \quad (6)$$

2.2. Geometric Modeling and Computational Setup for Pantograph

In this article, a simplified model of a certain high-speed train consisting of three cars is utilized, with the train model having a height (H) of 3.1 m, a width (W) of 3.2 m, the length (L) of the lead and tail cars being 25.8 m, and the length (L) of the middle car being 24.9 m. The established high-speed train model is depicted in Figure 2.



Figure 2. Geometric model of train.

Focusing on a specific type of pantograph as the research subject, a simplified model of the pantograph was established. The pantograph principally consists of the bow head, upper arm, balancing rod, lower arm, upper and lower pull rods, base frame, and insulators. The structure is illustrated in Figure 3.

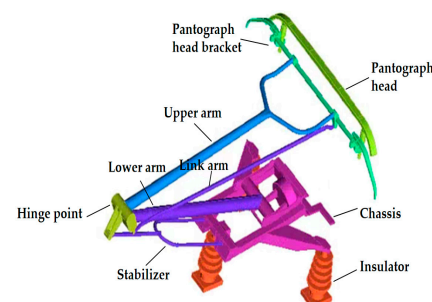


Figure 3. Geometric model of pantograph.

In this article, the tunnel model is 255 m in length, designed as a single-track tunnel. According to the National 8th Five-Year Plan Scientific and Technological Project—“Research Report on the Selection of Design Parameters for High-Speed Railway Line Bridges and

Tunnels”—the cross-sectional area of the tunnel can be adopted as 100 m^2 . The tunnel model and its cross-sectional area are depicted in Figure 4.

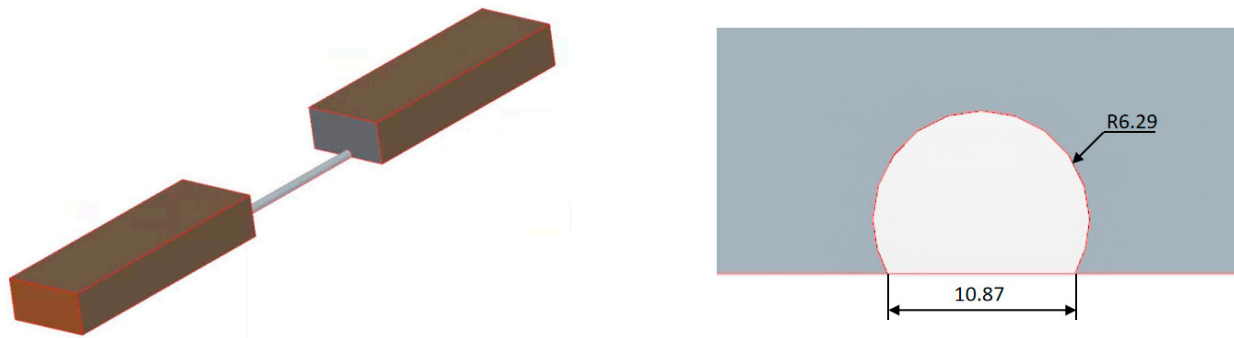


Figure 4. Tunnel geometric model and cross-section.

The calculations were performed using the commercial software Fluent for the simulation of the airflow around a train set operating within a tunnel. The computational domain is illustrated in Figure 5, with the train positioned such that the nose tip of the lead car is 50 m away from the tunnel entrance. A pressure outlet boundary condition is defined, with buffer zones of 310 m set up at both the tunnel entrance and exit to accommodate the acceleration and deceleration phases of the train. The study assumes the gas to be an ideal gas, requiring the definition of a pressure far-field boundary at the exit. The sides of the buffer zone are set as symmetric walls, located $10H$ away from the train's symmetry line, where H represents the height of the train. The top surface of the buffer zone is also designated as a symmetric wall, positioned $15H$ above the ground. The tunnel entrance and exit wall surfaces are defined as stationary walls. To simulate the ground effect during the train's operation, the tunnel ground is set as a stationary wall, while the ground in the train operating domain is defined as a slip wall.

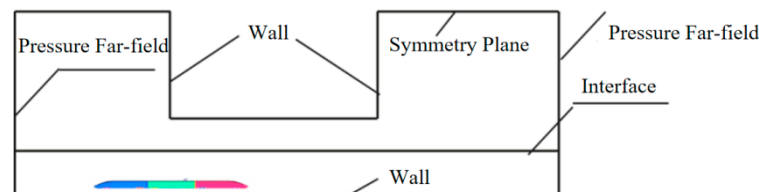


Figure 5. Boundary condition settings.

The global mesh size for the train body surface is set to 60 mm. In this study, a finer mesh resolution of 20 mm is applied in the vicinity of the pantograph to enhance the mesh density in that area. Additionally, boundary layers are established on the surfaces of both the train and the pantograph, with the first layer of the boundary near the pantograph wall being 0.1 mm in height. The sliding mesh method is employed for the simulation, with the mesh count for the sliding block being approximately 11.58 million and the mesh count for the stationary domain being about 3.16 million, totaling a mesh count of 14.74 million. Figure 6 presents a simulation and analysis diagram of the mesh.

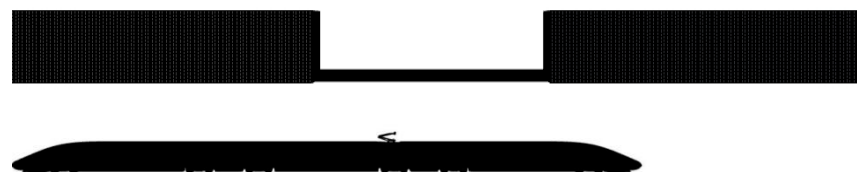


Figure 6. Computational domain.

In this article, 10.55 million, 14.74 million, and 16.87 million grids, are used to compare the calculation results of 10.55 million, 14.74 million, and 16.87 million grids, 10.55 million

grids and two other quantity grids, 16.87 million and 5–10%, considering the influence of grid quantity within the acceptable range. Then, 14.74 million grids are selected in the final project to ensure the calculation cost.

2.3. Aerodynamic Load Analysis for a Single Pantograph Passing through a Tunnel

This section analyzes the aerodynamic pressure distribution and the aerodynamic loads on various components of the pantograph under the knuckle-upstream operating condition at a speed of 400 km/h as it passes through a tunnel on a single track. Figure 7 presents the surface pressure contours of the pantograph at three different moments: before entering the tunnel, inside the tunnel, and upon exiting the tunnel. It is observed that when the high-speed train operates at a speed of 400 km/h, the windward side of the pantograph primarily experiences positive pressure, while the leeward side is subjected to negative pressure. The pressure difference between these two sides is a significant source of aerodynamic drag on the high-speed pantograph. It is evident that before entering and after exiting the tunnel, the windward surface of the pantograph's base frame experiences higher surface pressure values, with the maximum positive pressure being approximately 4000 Pa and the maximum negative pressure being around -3200 Pa. Upon entering the tunnel, the amplitude of the pressure experienced on the surface of the pantograph significantly increases, with the maximum positive pressure reaching about 8000 Pa and the maximum negative pressure reaching -4500 Pa.

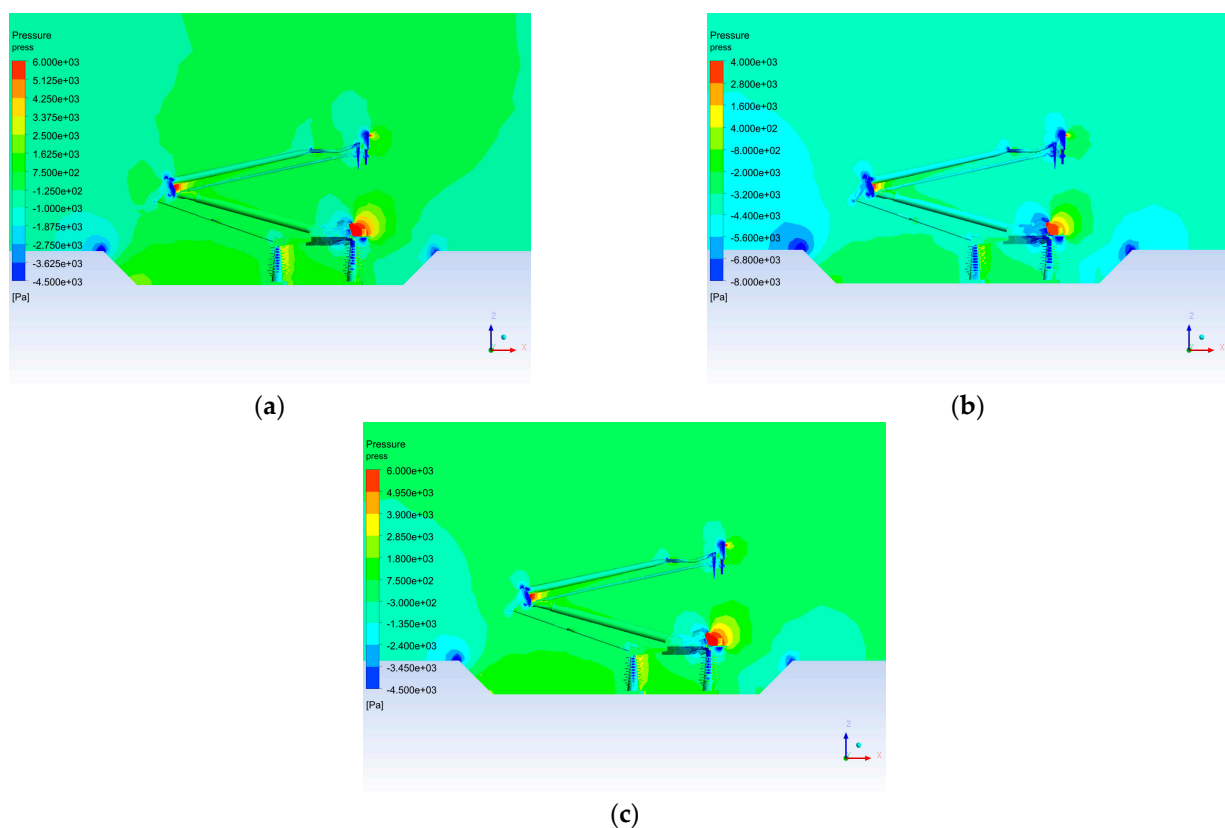


Figure 7. Pressure cloud image of train pantograph section at 400 km/h under the knuckle-downstream operating condition. (a) Before entering the tunnel. (b) In the tunnel. (c) After exiting the tunnel.

Through the computational simulation of the flow field, the average values of drag, lateral force, and lift experienced by the main components of the structure during operation can be obtained, as shown in Table 1. It is observed that the drag forces exerted on all structures are primarily concentrated on the bow head and the balancing beam, with the pull rods experiencing smaller forces due to their smaller windward surface area.

The lateral forces on all components are minor, having negligible impact on the overall displacement of the pantograph structure. Among the components, aside from the bow head, upper arm, and upper pull rod, which experience upward lift forces, other structures are subjected to downward lift forces. The bow head, in particular, experiences significant lift, substantially affecting the vertical displacement of the structure. This indicates that the aerodynamic loads on the pantograph are mainly manifested on the bow head.

Table 1. The resistance, lateral force, and uplift force (average values) of the main components in the knuckle-downstream operating direction at 400 km/h.

Component	Resistance [N]	Lateral Force [N]	Uplift Force [N]
Pantograph head	−710.62	−0.7	202.98
Pantograph head bracket	−640.19	0.85	−55.3
Upper arm	−132.61	−0.72	153.07
Link arm	−17.74	−34.9	45.44
Lower arm	−51.36	8.86	−147.54
Stabilizer	−9.18	0.89	−22.75
Whole	−2422.5	−9.87	878.09

2.4. Comparison of Aerodynamic Loads on Low-Carbon High-Speed Pantographs under the Knuckle-Upstream and Knuckle-Downstream Operating Conditions

This section focuses on comparing the aerodynamic characteristics of the airflow around the pantograph when operating at a speed of 400 km/h in the knuckle-downstream and knuckle-upstream operating conditions, as shown in Figure 8. The analysis aims to highlight the differences in aerodynamic properties of the pantograph under these two operational conditions. Figure 8 presents simulation and analysis diagrams in both the knuckle-downstream and knuckle-upstream operating conditions, along with the coordinate system, where the positive direction of the X-axis is aligned with the direction of the train's movement, and the positive direction of the Z-axis corresponds to the vertical direction of the train.

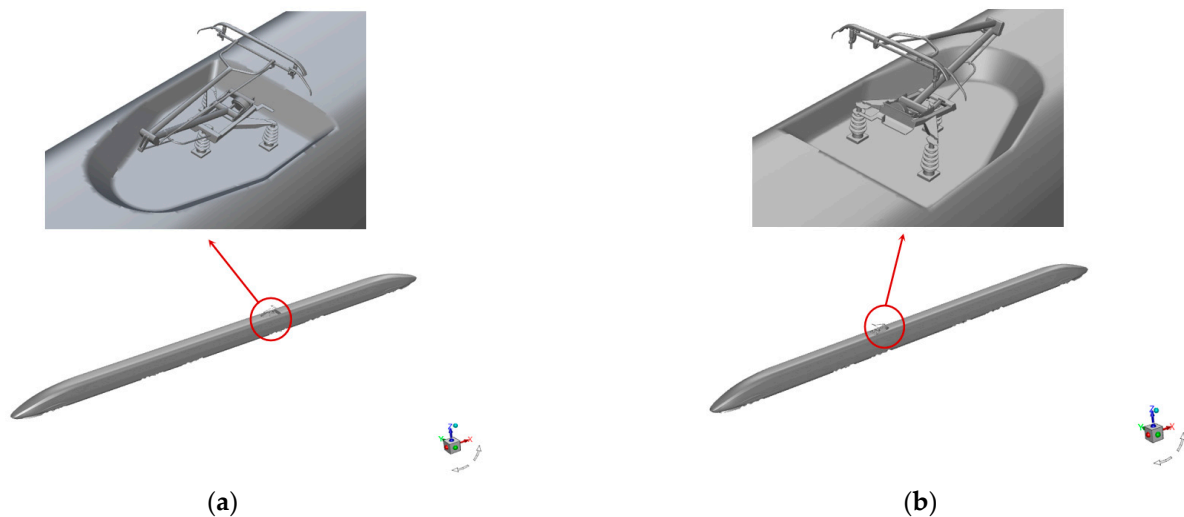


Figure 8. Pantograph knuckle-downstream and knuckle-upstream operation condition simulation and analysis diagram. (a) Knuckle-downstream operating direction. (b) Knuckle-upstream operating direction.

Figure 9 presents the distribution of surface pressure on the windward and leeward sides of the pantograph when operating at a speed of 400 km/h through a tunnel in both knuckle-downstream and knuckle-upstream operating conditions. Surface pressure contour maps are captured at three moments: before entering the tunnel, inside the tunnel, and upon exiting the tunnel. Regardless of whether the knuckle-downstream operating

condition or the knuckle-upstream operating condition, the bow head and the joints where the upper and lower arms connect experience significant negative pressure. The amplitude of surface pressure on the pantograph is similar in both conditions, with higher surface pressure values on the windward side of the base frame before entering and after exiting the tunnel, where the maximum positive pressure is about 4000 Pa, and the maximum negative pressure is approximately -3200 Pa. After the high-speed train enters the tunnel, the amplitude of the pressure experienced on the pantograph's surface significantly increases, with the maximum positive pressure reaching about 8000 Pa and the maximum negative pressure reaching -4500 Pa.

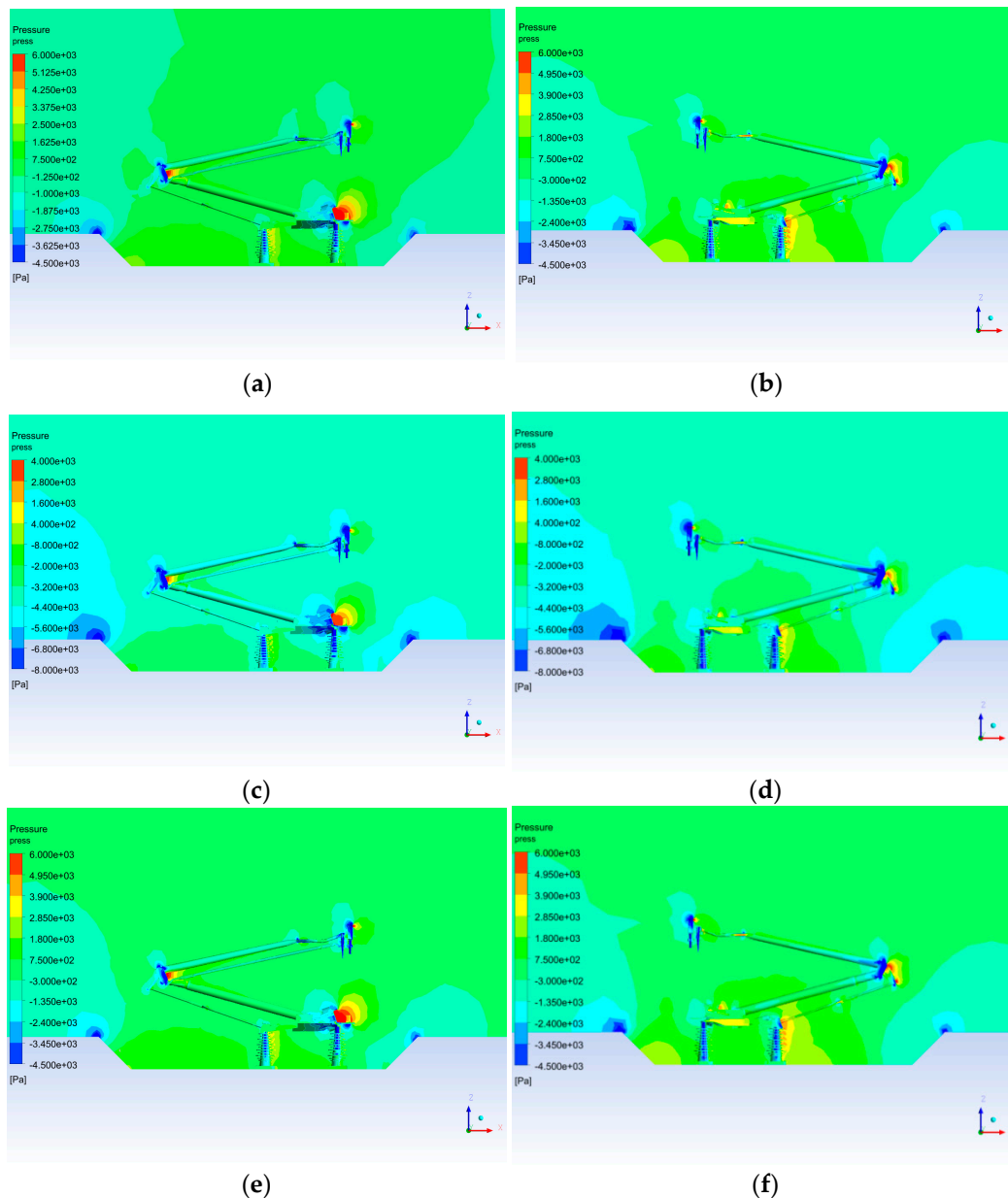


Figure 9. Pantograph knuckle-downstream and knuckle-upstream operation condition simulation and analysis diagram. (a) Pantograph knuckle-downstream operating condition before entering the tunnel. (b) Pantograph knuckle-upstream operating condition before entering the tunnel. (c) Pantograph knuckle-downstream operating condition in the tunnel. (d) Pantograph knuckle-upstream operating condition in the tunnel. (e) Pantograph knuckle-downstream operating condition after exiting the tunnel. (f) Pantograph knuckle-upstream operating condition after exiting the tunnel.

Table 2 presents the average values of drag, lateral force, and lift experienced by the main components of the structure during both knuckle-downstream and knuckle-upstream operating conditions. Through a comparative analysis of the three forces exerted on the entire pantograph, it was found that the drag forces in both knuckle-downstream and knuckle-upstream operating conditions are similar, with the drag in the knuckle-downstream condition slightly exceeding that in the closed condition. However, there are significant differences in the lateral forces experienced, with smaller lateral forces in the knuckle-downstream condition and larger ones in the closed condition. Additionally, in the closed condition, the pantograph experiences an upward lift, while in the knuckle-downstream condition, the lift exerted on the pantograph is downward. An analysis of the aerodynamic loads on the main components of the pantograph reveals differences primarily in the upper and lower arms and the upper and lower pull rods. In both the knuckle-downstream and closed operating modes, the direction of lift on these four rods is opposite, which is a major reason for the variance in lift forces experienced by the pantograph under the two operating conditions.

Table 2. The resistance, lateral force, and uplift force (average values) of the main components under the knuckle-upstream and knuckle-downstream operating conditions at 400 km/h.

Component	Resistance [N]		Lateral Force [N]		Uplift Force [N]	
	Knuckle-Downstream	Knuckle-Upstream	Knuckle-Downstream	Knuckle-Upstream	Knuckle-Downstream	Knuckle-Upstream
Pantograph head	−697.92	−647.63	−18.34	−3.3	212.89	156.89
Pantograph Head bracket	−618.06	−627.06	13.22	5.96	−39.81	58.11
Upper arm	−136.2	−138.72	−0.97	−30.38	154.03	−43.16
Link arm	−21.47	−15.49	−45.01	−26.03	52.93	−39.73
Lower arm	−31.01	−29.94	3.35	−68.5	−86.26	89.45
Stabilizer	−8.01	−53.5	1.59	−0.12	−4.72	40.39
Whole	−2785.67	−2722.62	2.66	−145.58	−386.01	264.73

2.5. Comparison of Aerodynamic Loads on Pantographs at Different Operating Speeds

This section is dedicated to comparing the aerodynamic characteristics of the airflow around the pantograph under the knuckle-upstream operating condition at different speeds of 400 km/h, 450 km/h, 500 km/h, and 600 km/h, analyzing the aerodynamic properties of the pantograph across these four scenarios.

Figure 10 presents the surface pressure distribution on both the windward and leeward sides of the pantograph under the knuckle-upstream operating condition as it passes through a tunnel at speeds of 400 km/h, 450 km/h, 500 km/h, and 600 km/h, capturing the surface pressure contour maps of the pantograph at the same moment in time. It is observed that with the increase in operating speed, the pressure experienced on the surface of the pantograph significantly increases, although the distribution of surface pressure on the pantograph does not undergo substantial changes.

Tables 3–5, along with Figure 11, provide the average values of drag, lateral force, and lift experienced by the main components of the pantograph at the same moment during operation at different speeds. It is observed that as the operating speed increases, the drag force exerted on the structure significantly increases. The lateral forces on the pantograph also increase with speed, but since lateral forces are relatively small, they do not have a substantial impact on the overall displacement of the pantograph structure. However, the lift experienced by the components of the pantograph shows a notable increase with higher operating speeds, which has a considerable effect on the vertical displacement of the structure.

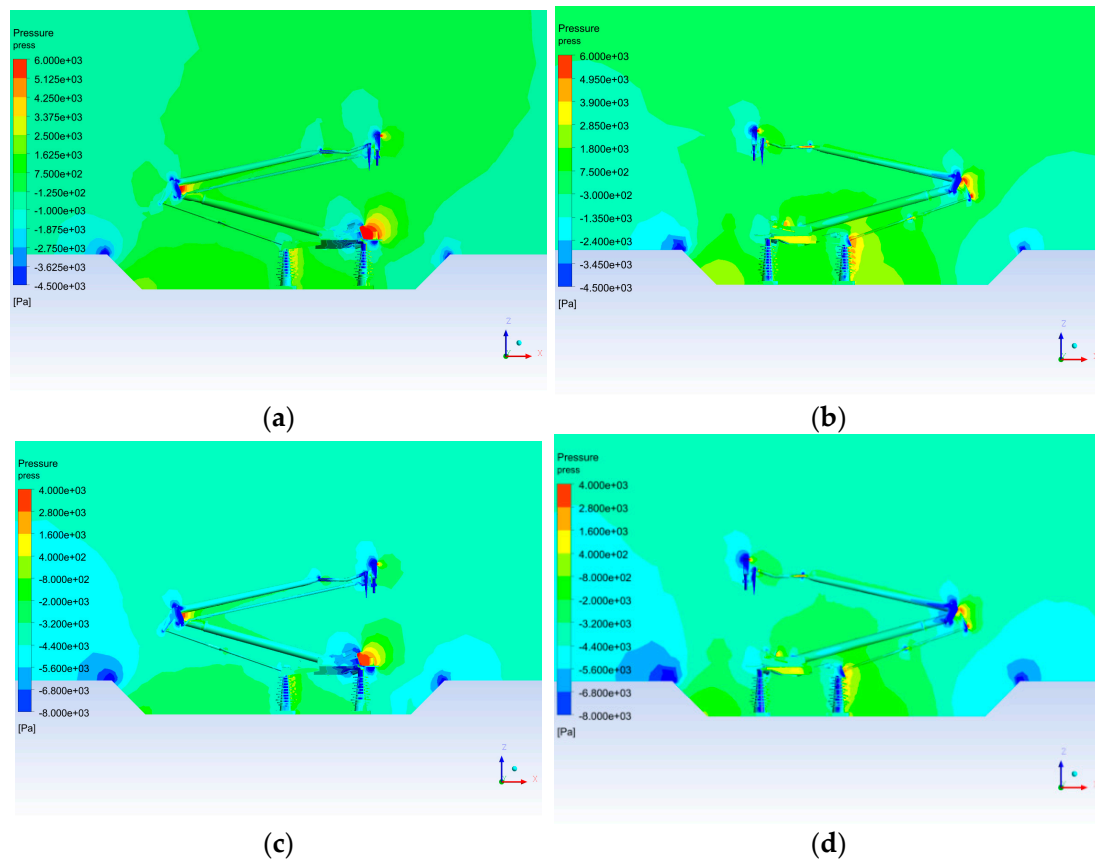


Figure 10. Pressure cloud image of train pantograph section at different operating speeds. (a) 400 km/h. (b) 450 km/h. (c) 500 km/h. (d) 600 km/h.

Table 3. The average resistance of pantograph under the knuckle-upstream operating condition at different running speeds.

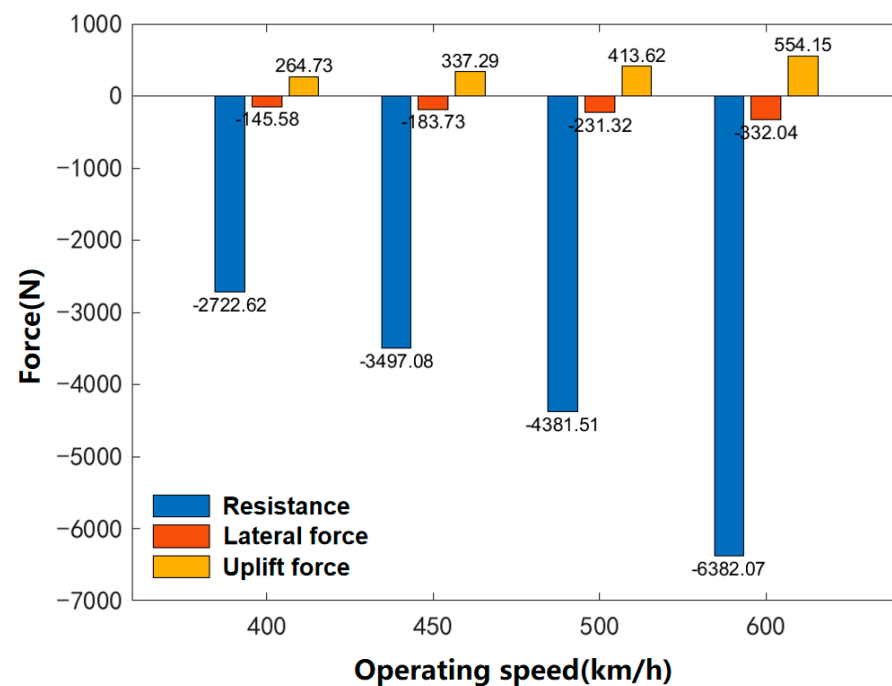
Component	Resistance [N]			
	400 km/h	450 km/h	500 km/h	600 km/h
Pantograph head	−647.63	−834.06	−1048.69	−1537.76
Pantograph head bracket	−627.06	−810.65	−1024.41	−1520.24
Upper arm	−138.72	−179.36	−226.64	−335.09
Link arm	−15.49	−19.55	−24.01	−33.7
Lower arm	−29.94	−37.6	−45.78	−62.73
Stabilizer	−53.5	−68.65	−85.91	−124.74
Whole	−2722.62	−3497.08	−4381.51	−6382.07

Table 4. The average lateral force of pantograph under the knuckle-upstream operating condition at different running speeds.

Component	Lateral Force [N]			
	400 km/h	450 km/h	500 km/h	600 km/h
Pantograph head	−3.3	−4.57	−8.13	−11.92
Pantograph head bracket	5.96	7.59	8.95	11.9
Upper arm	−30.38	−36.55	−46.01	−74.06
Link arm	−26.03	−32.76	−40.01	−55.54
Lower arm	−68.5	−87.57	−107.75	−145.36
Stabilizer	−0.12	−0.18	−0.45	−2.88
Whole	−145.58	−183.73	−231.32	−332.04

Table 5. The average uplift force of pantograph under the knuckle-upstream operating condition at different running speeds.

Component	Uplift Force [N]			
	400 km/h	450 km/h	500 km/h	600 km/h
Pantograph head	156.89	200.56	248.05	345.22
Pantograph head bracket	58.11	75.12	91.63	133.9
Upper arm	−43.16	−55.2	−69.34	−103.92
Link arm	−39.73	−50.82	−63.34	−91.1
Lower arm	89.45	113.89	140.53	196.54
Stabilizer	40.39	51.24	63.26	88.4
Whole	264.73	337.29	413.62	554.15

**Figure 11.** The resistance, lateral force, and uplift force (average values) of the pantograph under the knuckle-upstream operating condition at different operating speeds.

3. Pantograph Structural Modal Simulation Analysis

The modal simulation analysis of the pantograph structure mainly includes two parts: establishing a modal simulation model of the pantograph structure and analyzing the natural frequency and modal simulation of the pantograph. We establish a simulation model of the pantograph using 3D simulation software, and analyze the natural frequency and modal vibration mode of the pantograph using Workbench 2020 R2.

3.1. Pantograph Structural Modal Simulation Model

The high-speed pantograph, serving as the contact interface providing electrical power to high-speed trains, possesses a relatively complex structure. The physical model of the pantograph includes numerous small components such as bolts, washers, and fillets, which have minimal impact on the modal analysis of the structure. Therefore, in modeling, the pantograph can be simplified. The simplified pantograph, as shown in Figure 12, mainly comprises components like the bow head, carbon strip, bow head support, upper arm, balancing rod, lower arm, pull rod, base frame, and transmission mechanism.

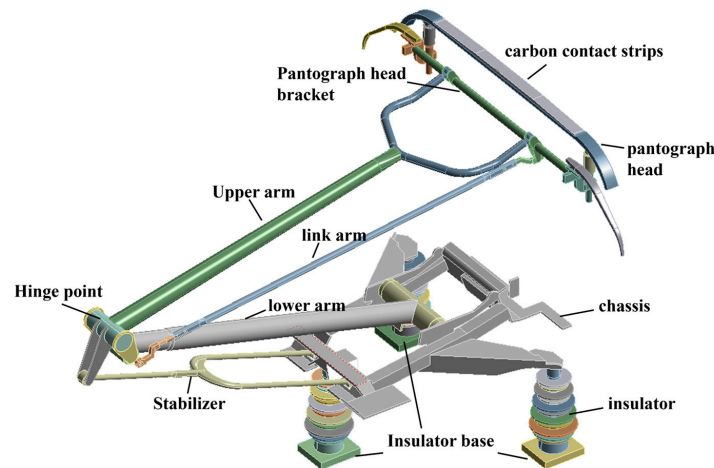


Figure 12. The simplified modal dynamics simulation model of the pantograph.

3.2. Pantograph Modal Simulation Analysis Results

The modal characteristics of the pantograph were analyzed under two different conditions: without prestress and with prestress. Since the pantograph can be considered as fixed to the top of the train, its boundary conditions were set as follows: the three mounting seats of the base frame were fixed constraints, eliminating all six degrees of freedom, as shown in Figure 13 (the planes where the red dots are located represent fixed constraints). For critical hinge locations, revolute constraints contact settings were used (such as the connection between the upper arm and the bow head system, the connection between the ends of the pull rod, the connection between the upper arm and the balancing rod, the connection between the balancing rod and the base frame system, and the connection between the lower arm and the base frame system), as illustrated in Figure 14 (the locations marked with red circles indicate hinge constraints). To replicate the modal test constraint conditions, a vertical spring constraint was added at the midpoint of the carbon strip on the bow head to simulate the catenary contact. Research [38] has shown that with increasing train speeds, for dynamic simulations of the catenary-pantograph interaction at speeds exceeding 120 km/h, using a fixed stiffness spring to model the catenary contact is more reasonable. To simulate the effect of the pantograph's airbag [39], a spring constraint of 3300 N/m was added between the lower arm and the base frame, as shown in Figure 15. For the condition without prestress, the initial catenary contact force was not considered; for the condition with prestress, a lifting torque was applied so that the vertical spring reaction force on the carbon strip of the bow head was 70 N (to describe the static contact force between the catenary and the pantograph), as illustrated in Figure 16.

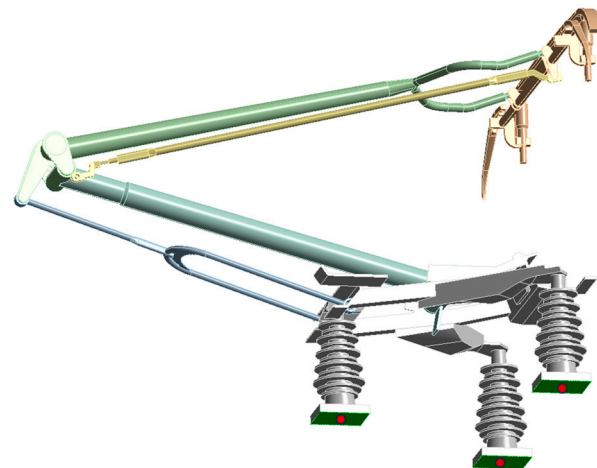


Figure 13. Simulation and analysis diagram of fixed constraint base.

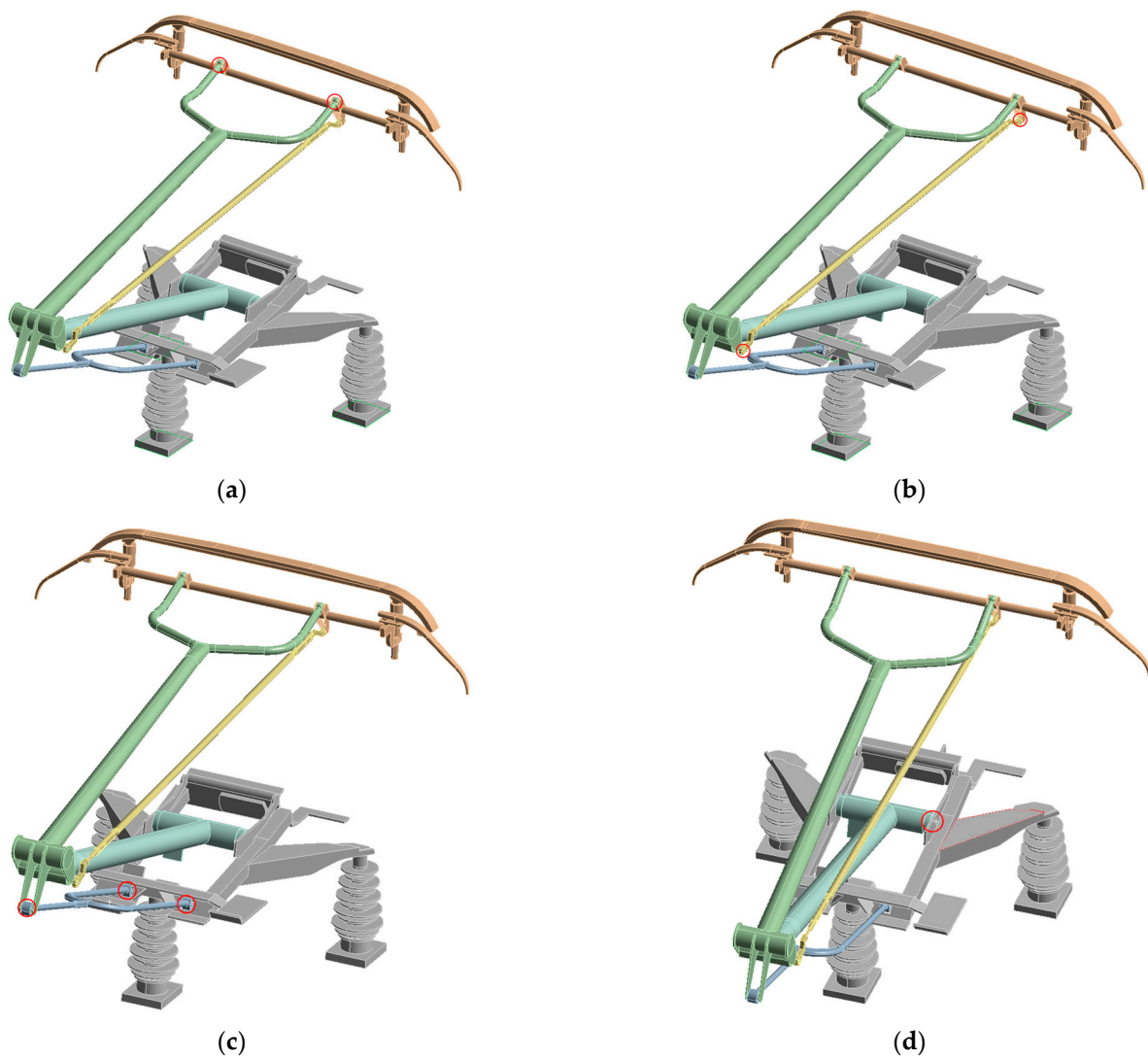


Figure 14. Simulation and analysis diagram of hinge constraints on key connecting parts. (a) the connection between the upper arm and the bow head system. (b) the connection between the ends of the pull rod. (c) the connection between the ends the stabilizer rod. (d) the connection between the lower arm and the base frame system.

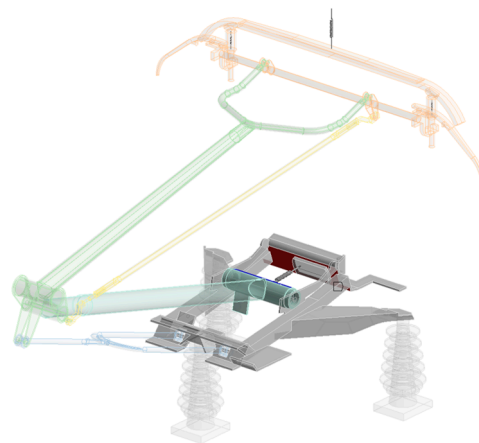


Figure 15. Simulation and analysis diagram of spring constraint.

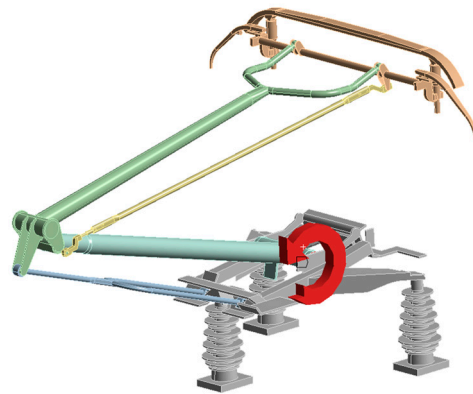


Figure 16. Simulation and analysis diagram of preload.

The material parameters for the main components of the pantograph are set as shown in Table 6.

Table 6. Material parameters of the main components of the pantograph.

Component	Material Type	ρ [kg/m ³]	E [GPa]	N [-]
Pantograph head	Carbon alloy	1000	180	0.3
Pantograph head bracket	Stainless steel	7930	193	0.247
Upper arm	Stainless steel	7930	193	0.247
Link arm	Aluminum alloy	2700	71	0.33
Lower arm	Stainless steel	7930	193	0.247
Stabilizer	Aluminum alloy	2700	71	0.33

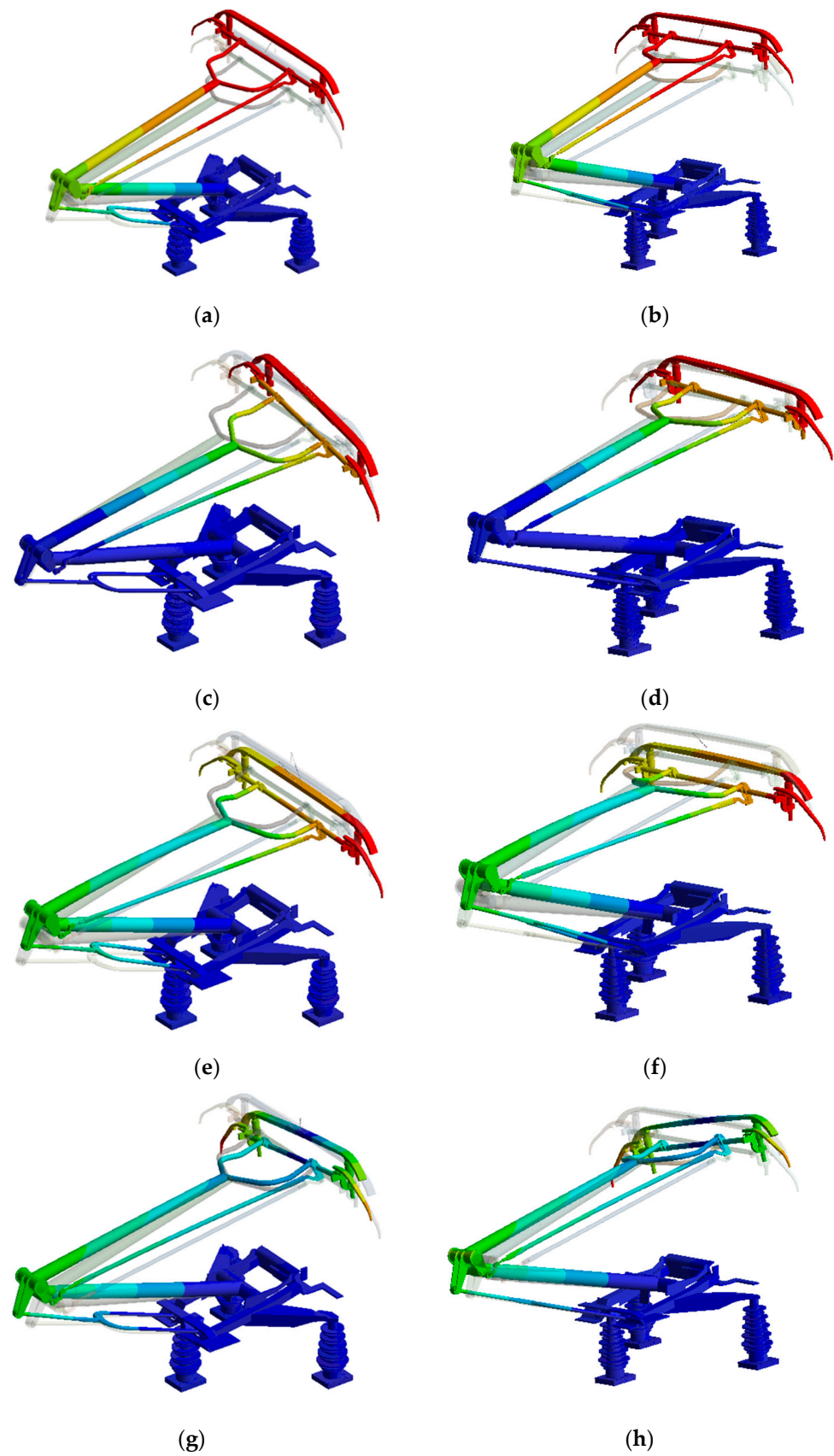
Through finite element software simulation, the first 10 natural frequencies were obtained, as shown in Table 7.

Table 7. The first 10 natural frequencies of the pantograph structure.

Order	Frequency [Hz]		
	With Preload	Without Preload	Error
1	2.29	2.28	0.24%
2	5.10	5.20	1.92%
3	8.79	8.79	0.03%
4	10.48	10.48	0.02%
5	13.41	13.40	0.05%
6	17.22	17.32	0.55%
7	23.13	22.71	1.82%
8	25.51	25.53	0.07%
9	29.31	29.37	0.21%
10	34.84	34.85	0.01%

The results from the modal analysis, which reveal the natural frequencies, show that for both the pantographs without prestress and with prestress, the first 10 modal frequencies are within 40 Hz, and the first five frequencies are all below 12 Hz. Therefore, it can be considered that the pantograph is a typical system with a dense modal distribution and that its frequencies are relatively low. The influence of prestress on the natural frequencies of the pantograph is minor and does not alter the trend of the modal shapes for each order.

The first to fifth modal shape diagrams of the entire pantograph structure, obtained through simulation software, as shown in Figure 17.

**Figure 17.** *Cont.*

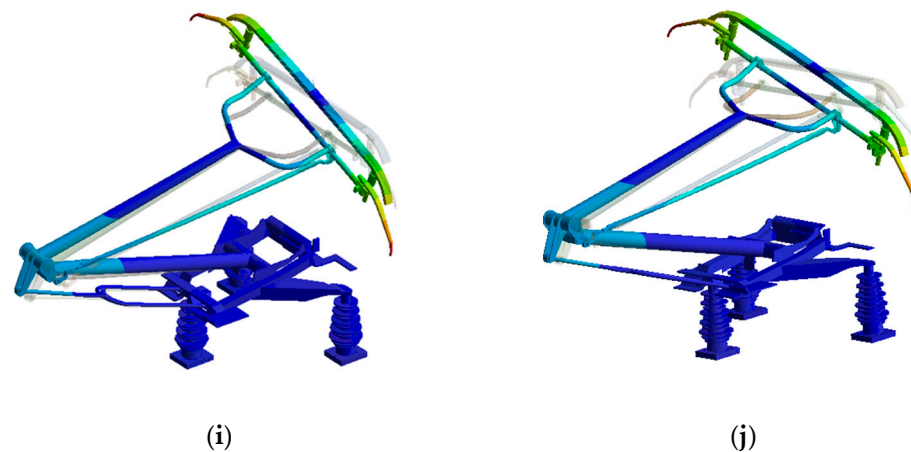


Figure 17. Diagram of the 1st to 5th modal shapes of the overall structure of the pantograph. (a) The first modal shape diagram of the pantograph without preload. (b) The first modal shape diagram of the pantograph with preload. (c) The second modal shape diagram of the pantograph without preload. (d) The second modal shape diagram of the pantograph with preload. (e) The third modal shape diagram of the pantograph without preload. (f) The third modal shape diagram of the pantograph with preload. (g) The fourth modal shape diagram of the pantograph without preload. (h) The fourth modal shape diagram of the pantograph without preload. (i) The fifth modal shape diagram of the pantograph without preload. (j) The fifth modal shape diagram of the pantograph without preload.

From the modal diagrams of the entire pantograph structure, it can be observed that the first modal frequency is characterized by vertical oscillation of the bow head and upper arm, the second mode involves lateral oscillation of the bow head and upper arm, the third mode is characterized by fore-and-aft oscillation of the upper and lower arms, the fourth mode involves lateral torsional oscillation of the bow head, upper arm, and pull rod, and the fifth mode is lateral torsional oscillation of the bow head and pull rod. The first modal frequency is the vertical oscillation of the bow head and upper arm, the second is the lateral oscillation of the bow head and upper arm, the third is the lateral torsional oscillation of the bow head, upper arm, and pull rod, the fourth is the fore-and-aft oscillation of the upper and lower arms, and the fifth is the lateral torsional oscillation of the bow head and pull rod.

In this chapter, finite element software was utilized to construct the modal simulation model of the high-speed pantograph, and modal characteristic analyses were performed on two types of high-speed pantographs. The analysis found that, under the same conditions, the modal vibration trends of the two types of pantographs are essentially consistent; moreover, the first 10 vibration frequencies of both pantographs are concentrated within 40 Hz, and the values of the first five frequencies are all less than 15 Hz. Therefore, the pantograph can be considered a typical low-frequency vibration structure.

4. Establishment of the Pantograph-Induced Vibration Model

The motion differential equations for the pantograph structure can be expressed as follows:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f\} \quad (7)$$

where $[M]$ is the mass matrix, $[C]$ is the damping matrix, and $[K]$ is the stiffness matrix, all of which are square matrices of order n , where n is the degrees of freedom of the system. $\{f\}$ represents the generalized external force vector, indicating the external forces acting on each degree of freedom. $\{x\}$ is the generalized displacement matrix of the system, with each parameter representing the generalized displacement of the corresponding degree of freedom. $\{\dot{x}\}$ and $\{\ddot{x}\}$, respectively, denote the generalized velocity vector and generalized acceleration vector of the system.

A coordinate transformation exists for $\{x\}$:

$$\{x\} = [U] \times \{y\} \quad (8)$$

where $[u]$ is the coordinate transformation matrix. Substituting Equation (8) into Equation (7) transforms the system of motion differential equations into modal coordinates:

$$[M][U]\{\ddot{y}\} + [C][U]\{\dot{y}\} + [K][U]\{y\} = \{f\} \quad (9)$$

Multiplying both sides of the equation by the transpose of the modal matrix $[U]^T$:

$$[U]^T[M][U]\{\ddot{y}\} + [U]^T[C][U]\{\dot{y}\} + [U]^T[K][U]\{y\} = [U]^T\{f\} \quad (10)$$

Let us define the generalized mass matrix, generalized damping matrix, generalized stiffness matrix, and generalized excitation as follows:

$$\begin{aligned} [M_n] &= [U]^T[M][U] \\ [C_n] &= [U]^T[C][U] \\ [K_n] &= [U]^T[K][U] \\ \{f_p\} &= [U]^T\{f\} \end{aligned} \quad (11)$$

The system of motion differential equations can be expressed as below:

$$[M_n]\{\ddot{y}\} + [C_n]\{\dot{y}\} + [K_n]\{y\} = \{f_p\} \quad (12)$$

The system of differential equations obtained after modal coordinate transformation, Equation (12), can be decomposed into n individual differential equations:

$$m_i\ddot{y}_i + c_i\dot{y}_i + k_iy_i = f_{pi}, (i = 1, 2, 3, \dots, n) \quad (13)$$

dividing both sides of the equation by m_i , we have the following:

$$\begin{aligned} k_i &= m_i\omega_{ni}^2 \\ c_i &= 2m_i\zeta_i\omega_{ni} \end{aligned} \quad (14)$$

thus, Equation (13) can be transformed into the following:

$$\ddot{y}_i + 2\zeta_i\omega_{ni}\dot{y}_i + \omega_{ni}^2y_i = \frac{f_{pi}}{m_i}, (i = 1, 2, 3, \dots, n) \quad (15)$$

Equation (15) represents the differential equation of motion for the i mode.

For a multi-degree-of-freedom system, the system of motion differential Equation (15) can be represented in matrix form as follows:

$$\begin{Bmatrix} \dot{Y} \\ \ddot{Y} \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega^2 & -2\delta\omega \end{bmatrix} \begin{Bmatrix} Y \\ \dot{Y} \end{Bmatrix} + \begin{Bmatrix} 0 \\ F \end{Bmatrix} \quad (16)$$

further, it can be expressed as below:

$$\dot{X}(t) = AX(t) + U(t) \quad (17)$$

where

$$\begin{aligned} X(t) &= \begin{Bmatrix} Y \\ \dot{Y} \end{Bmatrix} \\ A &= \begin{bmatrix} 0 & 1 \\ -\omega^2 & -2\delta\omega \end{bmatrix} \\ U(t) &= \begin{Bmatrix} 0 \\ F \end{Bmatrix} \end{aligned} \quad (18)$$

where $X(t)$ represents the state vector of the system, A is the system matrix, and $U(t)$ is the external load vector. Equation (17) is referred to as the system's state equation. Through the aforementioned derivation, the external load vector and the state vector of the system at each time step can be obtained in the matrix form of the motion equation.

For the pantograph structure, the displacement of each measurement point over time can be represented as follows:

$$W(x, y, t) = \sum_{i=1}^N f_i(x, y) q_i(t) \quad (19)$$

where $W(x, y, t)$ represents the displacement variation over time (x, y) at a measurement point with coordinates, $\Phi_i(x, y)$ represents the i mode at coordinates in (x, y) generalized coordinates, and $q_i(t)$ represents the displacement variation over time Nt a point in modal coordinates. N denotes the number of selected modes.

5. Pantograph Flow-Induced Vibration Analysis

During the operation of high-speed trains, the vibration of the pantograph structure has a significant impact on the stable current collection of high-speed trains and is also affected by the air. Flow-induced vibration is also an important factor affecting the stability of pantograph current collection. In this chapter, the aerodynamic load characteristics obtained above are combined with the results of modal analysis. The modal superposition method is used to analyze the time-domain response of the pantograph of a high-speed train under the action of aerodynamic loads. This article maps the fluid forces obtained from pneumatic simulation to the components of a fully flexible pantograph model, and studies the flow induced vibration effect of high-speed trains during operation due to the action of fluid forces.

Selected typical measurement points of the pantograph are analyzed for the characteristics of flow-induced vibration response at different operating speeds. The selected measurement points include: the middle part of the pantograph head, the middle part of the head support bracket, the middle part of the upper arm rod, the middle part of the pull rod, the connection part of the upper and lower arm rods, the middle part of the lower arm rod, and the middle part of the balance rod, as illustrated in Figure 18.

The modal superposition method is utilized to analyze the time-domain displacement response of multiple measurement points on the pantograph under external aerodynamic loads. During the simulation process, the contact wire is equivalently represented as a spring. The time-domain response of the contact force between the pantograph and the contact wire under different conditions can be considered as the product of the spring stiffness and the displacement response of the pantograph head. We analyze the maximum, minimum, average values, and standard deviation of the contact force, as well as assess the offline rate of the pantograph during train operation.

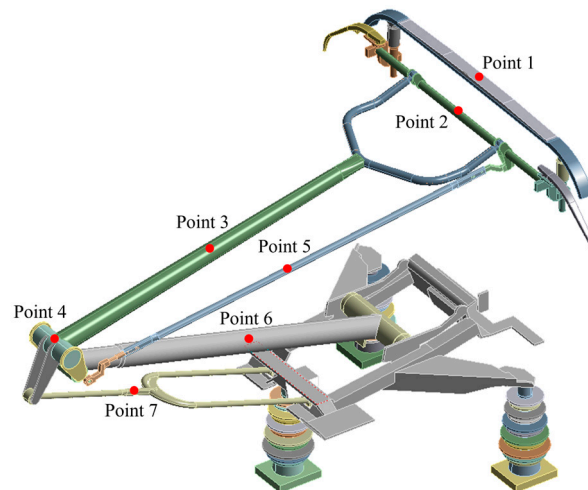


Figure 18. Simulation and analysis diagram of displacement response measurement point selection (Points 1–7 represent respectively carbon contact strips, pantograph head bracket, upper arm, hinge point, link arm, lower arm and stabilizer).

5.1. Comparison of Displacement Results of Flow-Induced Vibration Simulation at Different Speeds under Knuckle-Downstream and Knuckle-Upstream Operating Condition

The displacement of the pantograph represents the contact state between the pantograph head and the overhead contact system. The stability of the bow displacement is crucial for the normal operation of the pantograph. If the displacement is unstable, it may lead to poor contact or even circuit disconnection. Stable displacement helps improve the efficiency of electrical energy transmission. Abnormal displacement may cause poor contact between the pantograph and the overhead contact line, thereby affecting the safety of train operation. Therefore, the simulation of pantograph displacement is of great significance.

The displacement responses of the pantograph under knuckle-downstream operating condition at speeds of 400 km/h, 500 km/h, and 600 km/h are depicted in Figures 19–21.

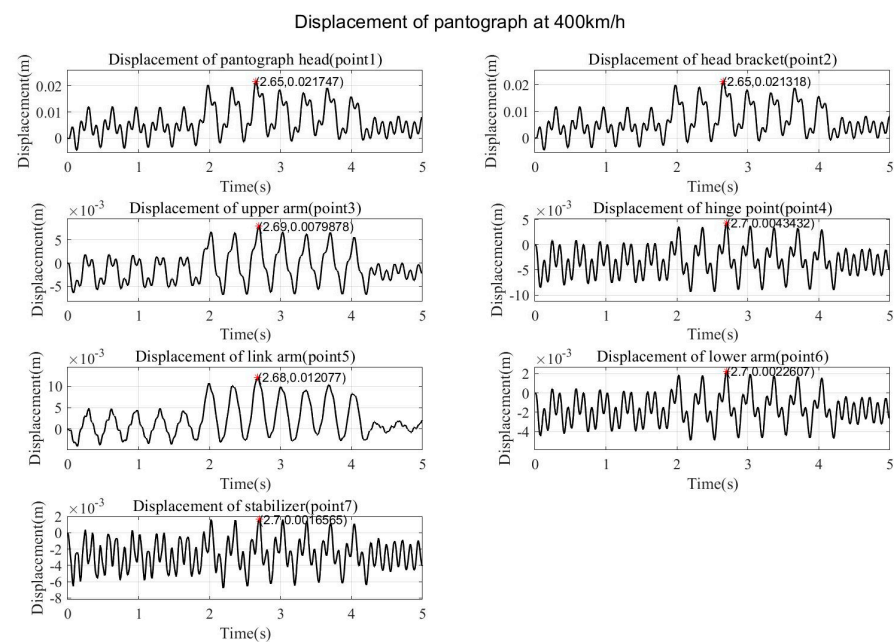


Figure 19. Simulation and analysis diagram of displacement response of pantograph measuring points at 400 km/h (knuckle-downstream operating condition).

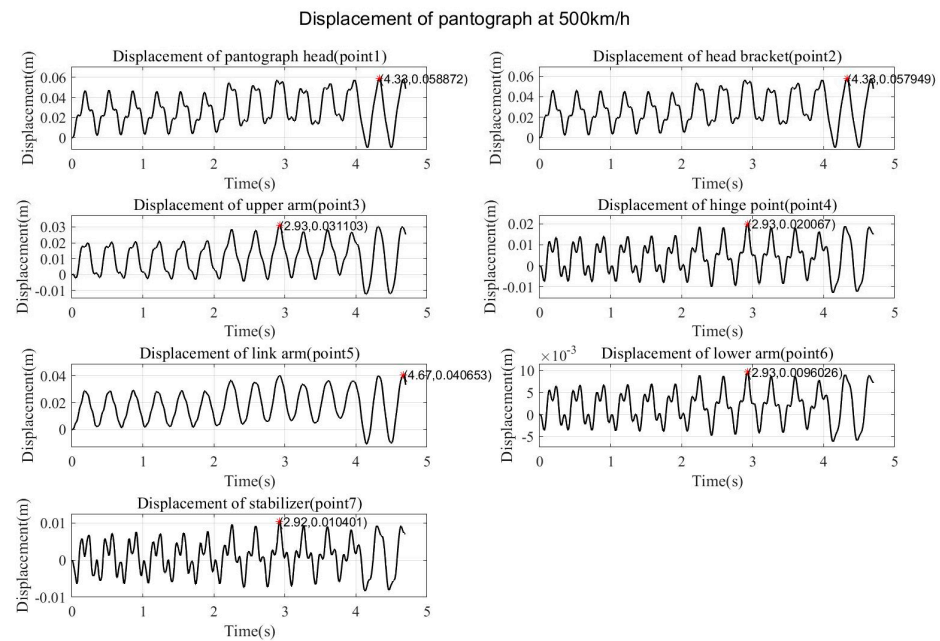


Figure 20. Simulation and analysis diagram of displacement response of pantograph measuring points at 500 km/h (knuckle-downstream operating condition).

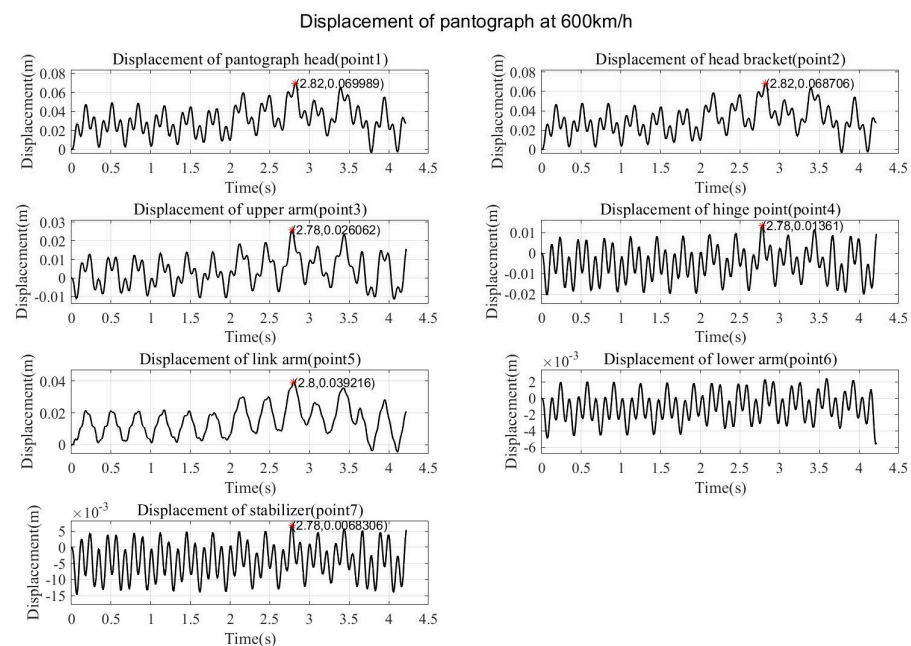


Figure 21. Simulation and analysis diagram of displacement response of pantograph measuring points at 600 km/h (knuckle-downstream operating condition).

It can be observed that with the increase in train operating speed, the displacement gradually increases under knuckle-downstream operating condition of the pantograph. The maximum values reach 2.17 cm, 3.06 cm, 5.89 cm, and 7.00 cm, respectively, which align with the aerodynamic simulation results, indicating that higher operating speeds result in greater aerodynamic lift forces acting on the pantograph. Additionally, prior to entering the tunnel (approximately 2 s), the displacement amplitudes of each component are noticeably smaller compared to after entering the tunnel, consistent with the phenomenon of a sharp increase in aerodynamic loads experienced by the pantograph after entering the tunnel.

The displacement responses of the pantograph under the knuckle-upstream operating condition at speeds of 400 km/h, 500 km/h, and 600 km/h are depicted in Figures 22–24.

It can be observed that with the increase in train operating speed, the displacement gradually increases under the closed-state condition of the pantograph. The maximum values reach 3.97 cm, 4.89 cm, 5.19 cm, and 7.33 cm, respectively, which align with the aerodynamic simulation results indicating that higher operating speeds result in greater aerodynamic lift forces acting on the pantograph. Additionally, prior to entering the tunnel (approximately 2 s), the displacement amplitudes of each component are noticeably smaller compared to after entering the tunnel, consistent with the phenomenon of a sharp increase in aerodynamic loads experienced by the pantograph after entering the tunnel.

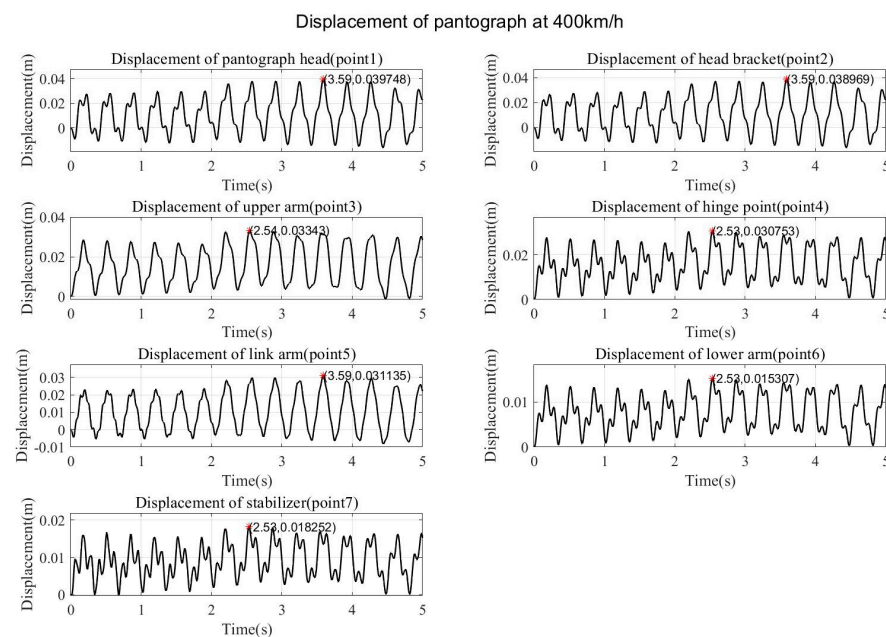


Figure 22. Simulation and analysis diagram of displacement response of pantograph measuring points at 400 km/h (knuckle-upstream operating condition).

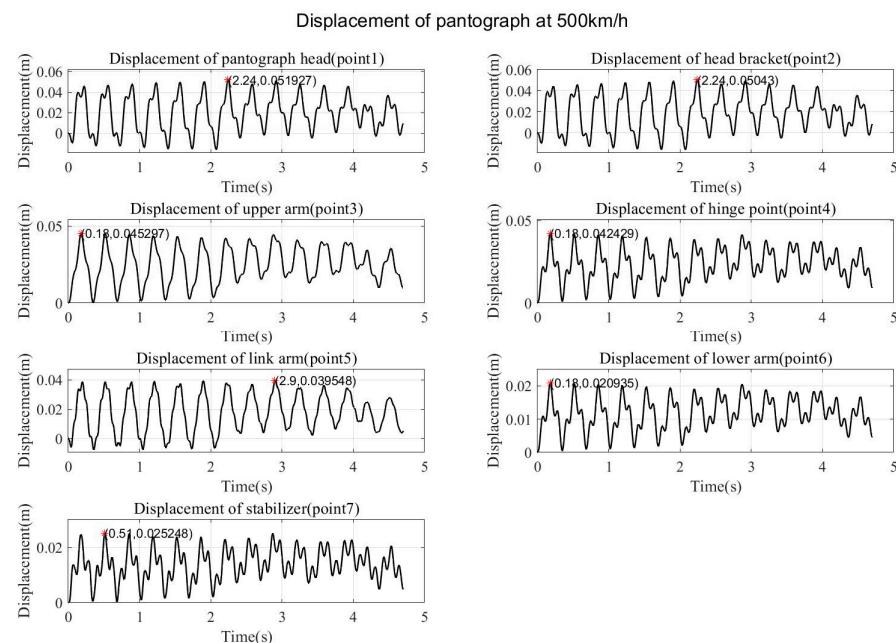


Figure 23. Simulation and analysis diagram of displacement response of pantograph measuring points at 500 km/h (knuckle-upstream operating condition).

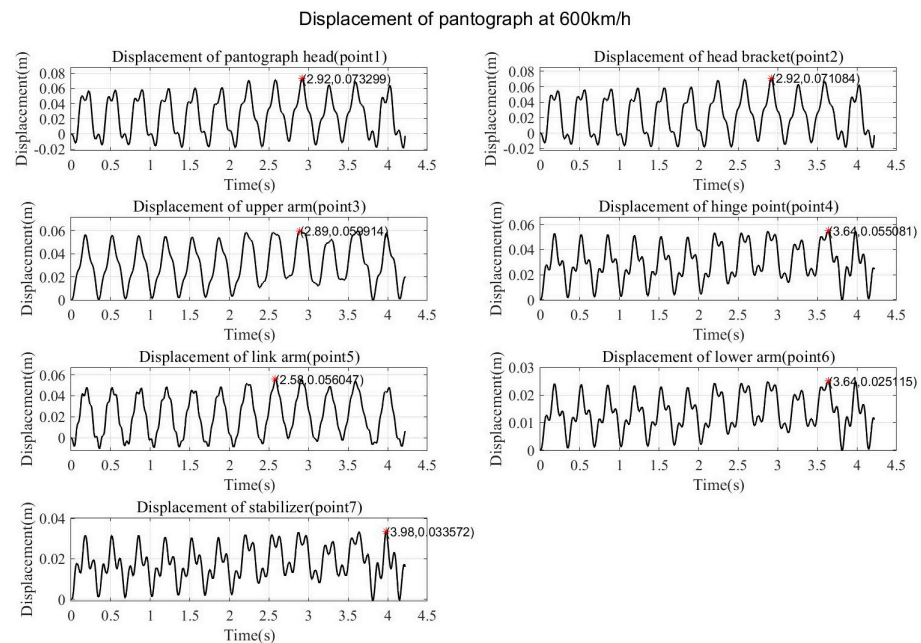


Figure 24. Simulation and analysis diagram of displacement response of pantograph measuring points at 600 km/h (knuckle-upstream operating condition).

Compared to the knuckle-downstream operating condition, when the pantograph operates in the knuckle-upstream operating condition, the displacement of the pantograph head slightly increases. The increase in bow head displacement may have adverse effects on the stability of train operation, energy transmission efficiency, and bow net lifespan. Excessive displacement may cause instability and a decrease in the electrical energy transmission efficiency of trains during high-speed operation.

5.2. Comparison of Contact Force Results of Flow-Induced Vibration Simulation at Different Speeds under Knuckle-Downstream and Knuckle-Upstream Operating Conditions

The time-domain response of the contact force between the pantograph and the contact wire in the knuckle-downstream operating condition under operating speeds of 400 km/h, 500 km/h, and 600 km/h obtained from simulations is illustrated in Figures 25–27.

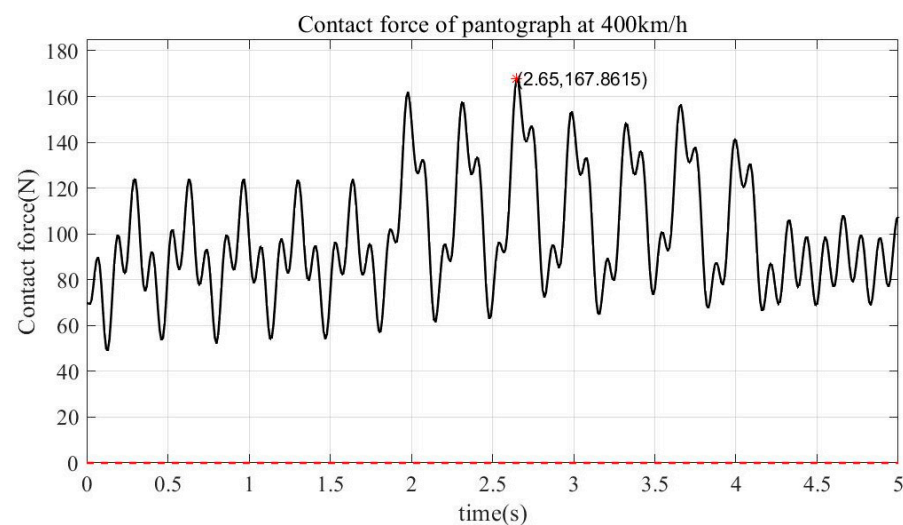


Figure 25. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 400 km/h (knuckle-downstream operating condition).

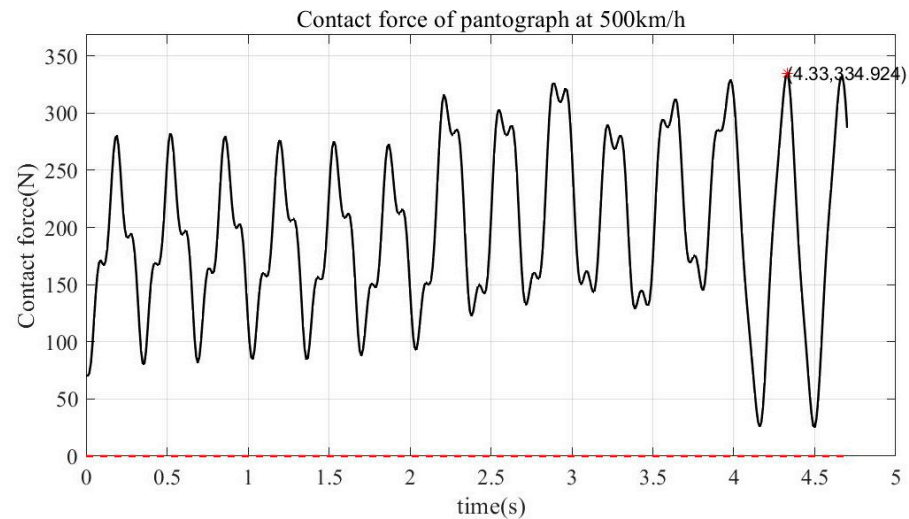


Figure 26. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 500 km/h (knuckle-downstream operating condition).

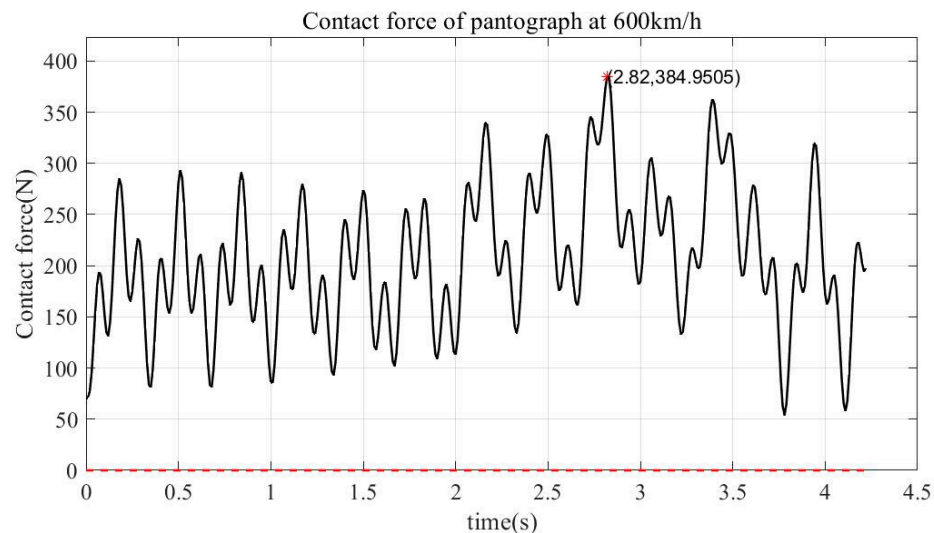


Figure 27. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 600 km/h (knuckle-downstream operating condition).

Under various speed conditions in the knuckle-downstream operating condition, the contact forces between the pantograph and the contact wire are presented in Table 8. The average contact forces of the contact wire are all less than $0.00097v^2 + F_0$ (where F_0 represents the static contact force of the contact wire), which complies with the requirements outlined in the referenced literature and standards [40].

Table 8. Contact force and offline rate of pantographs under different speed conditions when the pantograph is in the knuckle-downstream operating condition.

	Maximum Contact Force [N]	Minimum Contact Force [N]	Average Contact Force [N]	Standard Deviation of Contact Force [N]	Offline Rate [%]
400 km/h	167.86	49.04	97.98	24.90	0
450 km/h	207.57	50.79	129.20	29.59	0
500 km/h	334.92	25.68	281.71	71.86	0
600 km/h	384.95	54.56	205.73	65.43	0

Analysis reveals that the contact force between the pantograph and the contact wire significantly increases after entering the tunnel compared to before (approximately 2 s prior). With the increase in operating speed, the contact force of the pantograph with the contact wire (maximum contact force, average contact force, and standard deviation of contact force) increases. Additionally, significant mechanical wear occurs between the pantograph carbon slide and the contact wire.

The time-domain responses of the contact force between the pantograph and the contact wire in the knuckle-downstream operating condition under operating speeds of 400 km/h, 500 km/h, and 600 km/h obtained from simulations are illustrated in Figures 28–30.

In the knuckle-upstream operating condition, the contact forces between the pantograph and the contact wire are presented in Table 9. The average contact forces of the contact wire are all less than $0.00097v^2 + F_0$ (where F_0 represents the static contact force of the contact wire).

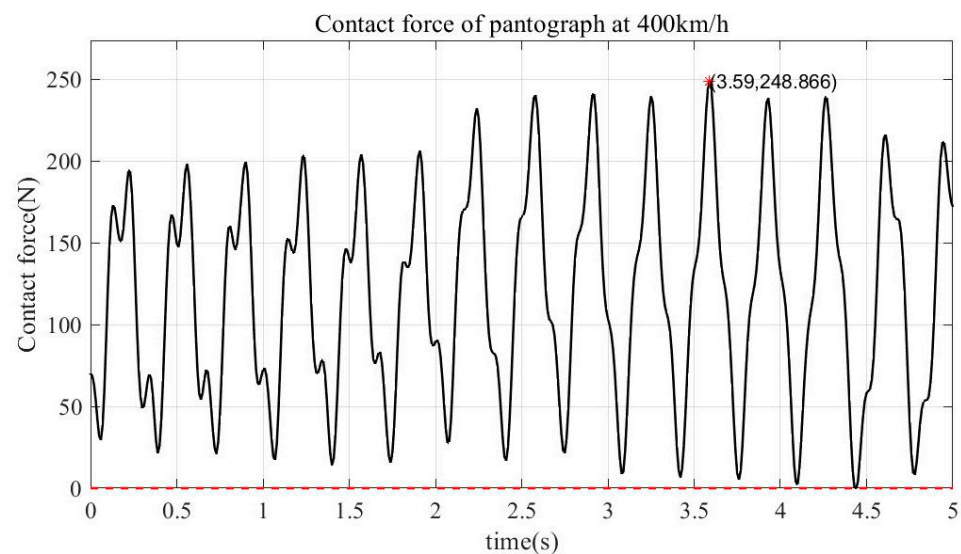


Figure 28. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 400 km/h (knuckle-upstream operating condition).

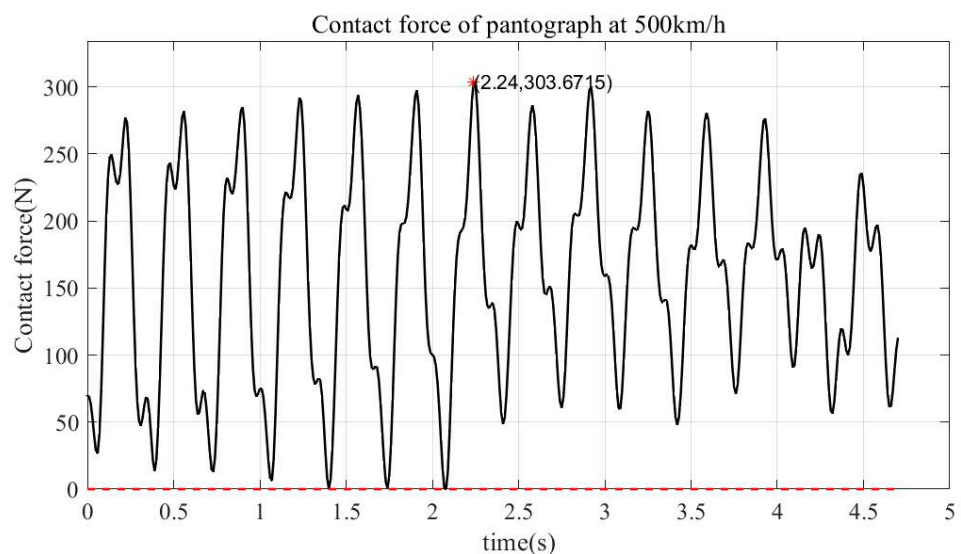


Figure 29. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 500 km/h (knuckle-upstream operating condition).

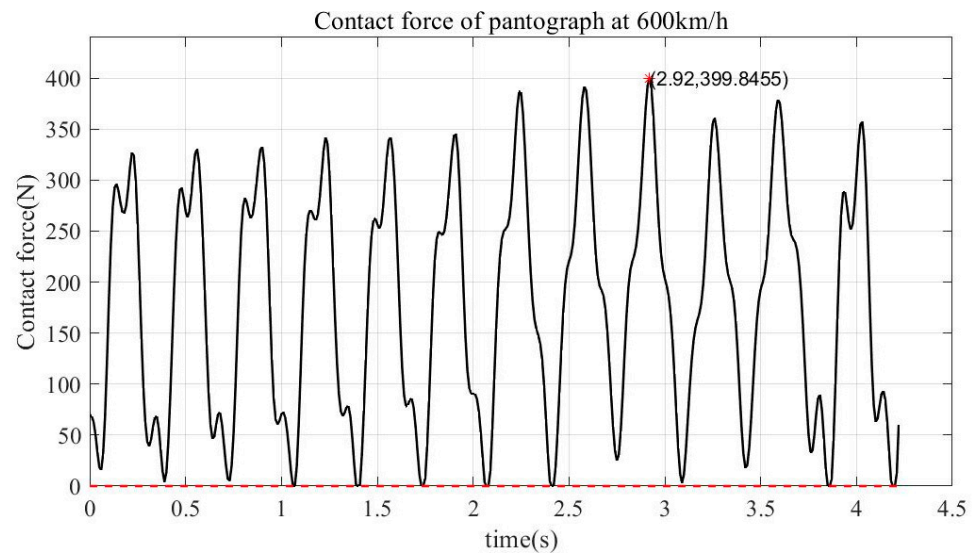


Figure 30. Simulation and analysis diagram of time-domain response of pantograph-catenary contact force at 600 km/h (knuckle-upstream operating condition).

Table 9. Contact force and offline rate of pantographs under different speed conditions when the pantograph is in the knuckle-upstream operating condition.

	Maximum Contact Force [N]	Minimum Contact Force [N]	Average Contact Force [N]	Standard Deviation of Contact Force [N]	Offline Rate [%]
400 km/h	248.87	0	115.85	63.85	0.4
450 km/h	290.12	0	133.44	77.53	1.45
500 km/h	303.67	0	153.38	76.61	0.64
600 km/h	399.85	0	171.63	112.59	3.55

Analysis reveals that the contact force between the pantograph and the contact wire significantly increases after entering the tunnel compared to before (approximately 2 s prior). With the increase in operating speed, the contact force of the pantograph with the contact wire (maximum contact force, average contact force, and contact force standard deviation) increases. Additionally, significant mechanical wear occurs between the pantograph carbon slide and the contact wire. Compared to the knuckle-upstream operating condition, the pantograph experiences upward aerodynamic forces, leading to an increase in the maximum contact force, average contact force, and contact force standard deviation of the contact wire in the knuckle-upstream operating condition. This results in an increase in the offline rate, thereby affecting the stability of the current collection.

Analysis using simulation software reveals that adjusting the stiffness of the equivalent spring of the pneumatic cushion and the mass of the pantograph head can reduce the contact force between the pantograph and the contact wire during high-speed operation. The larger the stiffness of the equivalent spring of the pneumatic cushion and the mass of the pantograph head, the smaller is the contact force between the pantograph and the contact wire, as shown in Figure 31 (where the X-axis units 1 represent the pantograph head density and spring stiffness in the knuckle-downstream operating condition with a speed of 400 km/h).

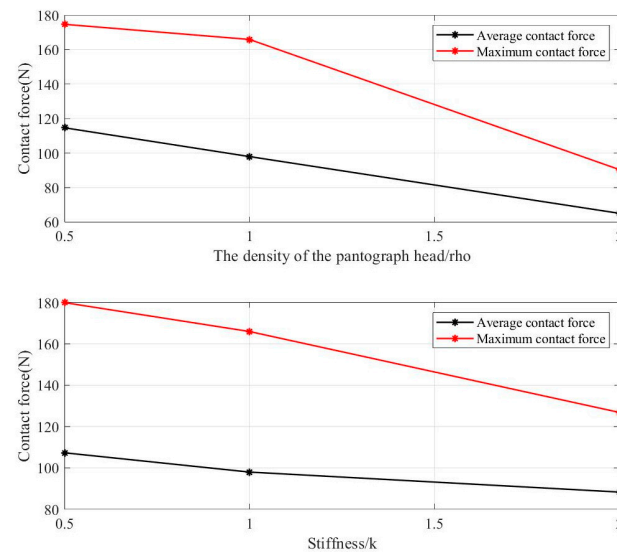


Figure 31. Response diagram of pantograph head density/stiffness to pantograph mesh contact force.

6. Conclusions

This article presents a novel method for establishing a full-flexible high-speed pantograph flow-induced vibration model using external aerodynamic loads and modal simulation data of the pantograph as inputs, combined with the modal superposition method. This method was utilized to analyze the dynamic response of high-speed pantographs under different speeds and knuckle-downstream and knuckle-upstream operating conditions. The following findings were obtained: Compared to before entering the tunnel (approximately 2 s prior), the displacement and contact force of the pantograph significantly increase after entering the tunnel. As the operating speed increases, the displacement and contact force of the pantograph also significantly increase. Compared to the knuckle-upstream operating condition, when running in the knuckle-downstream operating condition, the pantograph experiences upward aerodynamic forces, leading to an increase in the displacement of the pantograph and the maximum contact force, average contact force, and contact force standard deviation of the contact wire. This results in an increase in the offline rate, thereby affecting the stability of the current collection.

Analysis of flow-induced vibration of high-speed pantograph structures under the condition of 400 km/h operating speed reveals significant vibration reduction potential in the stiffness of the pneumatic cushion equivalent spring and the density of the pantograph head. Effective reduction of the contact force between the pantograph and the contact wire during high-speed operation can be achieved by adjusting the stiffness of the pneumatic cushion equivalent spring and the density of the pantograph head. These findings provide support for the design optimization of new pantograph structures.

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