Research Article

Container Quay Crane Structural Response under Trolley Traveling

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Received 20 July 2022; Accepted 1 September 2022; Published 20 September 2022

Academic Editor: Ivan Giorgio

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Structural response of quay crane should take more attention with the growth of worldwide logistics demand and quay crane size. This paper analyzes the response mechanism of the metallic structure of the quay crane induced by trolley traveling. An on-site test has been conducted. The time domain of crane girder acceleration signal shows that the main influence of the trolley traveling on crane structure is the high-frequency impact passing through the hinge point. The spectrum shows the main frequency of structure vertical vibration and impact vibration. Single girder model and theoretical derivation reveal that the vertical displacement response is quasistatic. A refined numerical model of the whole crane structure is established. The interaction between the trolley and the girder is realized by S-M contact method. The calculated results are basically consistent with the measured signal. The acceleration result indicates the same information as on-site test such that the structural acceleration response of in-service crane is affected by trolley in high-frequency band. The displacement results in three directions of 10 points on the girder show that the trolley traveling has significant effect on the vertical displacement. The lateral displacement shows the eccentric loading condition of girder. The displacement spectrum indicates that the trolley has little dynamic effect on displacement response of crane structure. The response can be treated as quasistatic.

1. Introduction

With the continuous growth of international trade, the logistics industry is developing rapidly. The quayside container crane, referred to as the quay crane, as the frontline logistics equipment of the port, is an important part of maintaining the container throughput of the port. With the growth of port logistics throughput, the quay crane structure continues to increase in size. Trolley with hoisting load is the only moving load in crane structure. The Port Equipment Manufacturers Association pointed out that the alternating stress on the bridge crane structure mainly comes from the high-frequency moving load in the service state. The effect of trolley traveling on safety of quay crane structure must be studied [1].

Studies at home and abroad have been done on beam structure response caused by car running. Huang Guoping [2] analyzed the longitudinal dynamic response of the bridge under the traffic flow based on the Euler beam theory and found that the bridge displacement caused by the traffic flow would cause fatigue wear on the bridge parts. Chen et al. [3] used beam elements to simulate bridges, dealt with the wheel-rail contact relationship with the close-fitting hypothesis and creep theory, and established a train-bridge-vehicle coupling model to achieve the coupled vibration of the vehicle and the bridge. Multiworking condition calculations are carried out for vehicle velocity, route, track spectrum, damping ratio, and other factors.

Based on the plate and shell theory, Prakash and Barik [4] analyzed the dynamic response of the bridge deck when the vehicle moved at a constant velocity and obtained the center deflection of the bridge deck before and after the reinforcement with stiffening and found that the I-shaped reinforcement contributed more to the strength of the bridge deck (Cheng et al. [5]).

The three models of moving load, moving mass, and spring damping mass were compared when analyzing the vehicle-bridge system, and it was believed that the moving
load model was the most stable. The improved moving load method is used to numerically model the vehicle-bridge system and the response of the bridge midspan under different mass and spring stiffness is analyzed. The simplified model has better calculation results except for the first resonance velocity.

At home and abroad, the research on the dynamic response of the trolley beam system mainly takes the bridge structure and high-velocity railway as the research object, and the high-velocity railway is the majority. Few take port crane trolley and the girder as the research object, and most of them take the single beam structure as the research object, lacking the structural dynamics analysis of the whole machine.

At present, there are three main research methods for the car beam system [6, 7]. The car beam system is simplified to the dynamic equation of the loaded beam [8–11]. Modeling and analysis of the actual car beam 3D model are based on multibody dynamics [6, 12–15] and transient dynamic analysis of vehicle beam using finite element modeling [16–18]. The analysis methods based on dynamic equations are mostly used to analyze the dynamic characteristics of vehicles and simple track structures such as high-velocity trains, but they have certain limitations in the noncentral load analysis of complex beam structures [19–21]. Multibody dynamics is suitable for dynamic analysis of complex structures, and it can establish refined models of beams and trolleys and realize vehicle-rail coupling. Based on the transient dynamic analysis of finite element vehicle-beam coupling, complex metallic structures and vehicle-beam coupling models can be established after simplification.

The main methods of vehicle-beam coupling model based on finite element theory include coupling method, birth-death element method, and contact method. All three simplify the vehicle load into a mass element. The difference is that the coupling method couples the node where the mass element is located with the node on the beam element and couples the mass element with the next node of the beam element at each time step to realize vehicle movement. The birth-and-death element method arranges a mass element on each node of the beam element to simulate the vehicle running on the beam in the form of a birth-and-death element; the contact method establishes a contact element between the mass element and the beam element to realize their interaction [22, 23]. Due to the refined modeling of the finite element method being relatively complex, the beam elements are used mostly as the main analysis tool for vehicle-beam coupling modeling domestically and abroad. The above three finite element analysis methods of vehicle beam are limited to the coupling modeling between mass element and beam element. The research object of this paper is the quay crane, which is a kind of port logistics equipment with a relatively large structure. The girder structure of the large quay crane is a thin-walled box girder shaped like an irregular trapezoid. The front and rear girders are connected by hinge points, and there is a gap between the trolley tracks at the hinge points. The trolley runs on the track beam in side of the girder, so the girder is under no-central force during the trolley running. The complexity of the structure at the girders and hinge points of the quay crane should be considered when analyzing the interaction between the vehicle and the girder. According to the particularity and load characteristics of the quay crane structure, the finite element plate and shell element are used in this paper to establish the refined structure model of the quay crane, and the mass-shell contact method is used for the first time to realize the modeling analysis of the trolley running on the refined girder. The trolley is simplified as mass elements, the girder structure is refined using shell elements, and the interaction between them is simulated by contact elements.

It should be pointed out that because of the chain rule, the amplitude will be reduced in acceleration of the frequency below 1/(2π) Hz. While the amplitude of acceleration of the frequency above 1/(2π) Hz will be enlarged, this makes the signal above 1/(2π) Hz more significant in acceleration and the signal below 1/(2π) Hz more significant in displacement.

The content of this paper is arranged as follows. Firstly, on-site test data were used to study the acceleration signal characteristics of the quay crane under normal working conditions. Secondly, based on the Euler beam theory, the single girder model with moving load is established. The displacement result under different working conditions is analyzed. Dynamic effect of trolley is derived. Then, a refined model of the quay crane was established. The S-M contact method is used to realize the interaction of the trolley mass and girder flange. The acceleration response is used to analyze the high-frequency component induced by trolley traveling. The displacement responses of ten measuring points on the girder of the quay crane model are analyzed. Time domain signal is used for displacement characteristic analysis. Frequency domain signal is used for low-frequency component analysis and proves the theoretical derivation of quasistatic status of response. The main influence of the trolley on the displacement of the quay crane is summarized.

2. Metallic Structure of Quay Crane

As shown in Figure 1, the main structure of the quay crane is the water and land side gantry. The water and land side girder are suspended from the gantry. The tension rod connects the ladder frame on the top of the gantry and the
front and rear girder to form a balanced structure. The struts maintain stability of the sea side and land side structures. When loading and unloading goods on the quay crane, the trolley runs along the track on the girder, and the container spreader is lowered to the ship deck or the ground for loading and unloading through the pulley and wire rope winding system on the trolley. The metallic structure of the quay crane consists of slender thin-walled girders. Among them, the girder is divided into multisection beams by suspension and tie rod support to increase the overall stability. Therefore, the quay crane girder is composed of several simply supported beams in the mechanical model.

3. On-Site Test

For long-term monitoring, a structural health monitoring (SHM) system has been established on the investigated quay crane. The vertical acceleration of girder hinge region for the last five years is shown in Figure 2. The acceleration signals of critical sites are converted to effective value. The energy of vertical vibration increases yearly. This is because of long-time structure abrasion. The signal in red circle is significantly lower than before, which is the time after the girder was maintained.

The effective value can reduce the data size for SHM purpose. However, it can only represent the tendency of the quantity and remove the extreme value. To obtain the real time signal of the structure response, an on-site test of quay crane girder under normal operating conditions is carried out. The total length of the quay crane girder is 127 m, the nominal lifting capacity is 60 t, and the full load including the trolley and spreader is 80 t. The nominal velocity of the trolley is 4 m/s. The acceleration sensor is installed in the hinge point area of the front and rear girder, and the model of the sensor is IMI 608A11. The test acceleration direction of the sensor is perpendicular to the ground, and the installation position and direction are shown in Figure 3. The sampling frequency is 2500 Hz, and the sampling time is 60 s.

Figure 4 shows the measured acceleration signal of the hinge points. When the trolley is running, the acceleration value of crane structure gradually increases and reaches maximum value in the middle. This is because the position of the measured site happens to be in the middle of the entire girder structure. The trolley rail gap which induces impact load takes place in the hinge region. When the trolley is getting closer to the hinge point, the vibration of girder system has a trend of increasing gradually. The trolley track has unwelded gaps at the hinge point, and the vibration signals surge at the hinge point compared with adjacent incremental signals, indicating that the vibration of the...
trolley is sharply enhanced here. The impact caused by the vibration will accelerate the wear of the parts of track system and influence the safety and efficiency of the quay crane.

The spectrum of the measured vertical acceleration data is shown in Figure 5. The signal of 50 Hz and frequency doubling signal come from the electromagnetic interference of the sensor power line. The main frequency of vertical acceleration takes place within the frequency range of 0–200 Hz. The frequency of maximum amplitude is 1.25 Hz in the low-frequency band and about 100 Hz in high-frequency band. The quay crane is a large metallic structure. The main frequency of the free vibration acceleration is in low-frequency band located in 0–5 Hz. The high frequency of the measured results is mainly due to the dynamic load impact of the trolley passing through rail gap.

For the purpose of distinguishing analysis on normal signal and impact signal, the 20 s–30 s signal of time domain is extracted from the vertical acceleration test data of crane girder structure. During this time segment, the trolley was passing through hinge point. The 0–20 s and 30 s–60 s signals are extracted for the analysis of normal operation of the quay crane without hinge point impact induced by trolley.

The spectrum of the extracted signal is shown in Figure 6. When the trolley is running on the landside and waterside girder, the main frequency band of the vertical acceleration of the quay crane structure is 0–5 Hz. But from landside to waterside, the amplitude increases in high-frequency band of 10–200 Hz significantly. The main reason for this phenomenon is that the waterside girder is supported by hinge and tie rods, which form simple support beam, while the landside girder is supported by gantry and diagonal brace, which is more rigid. The section inertia of waterside girder is smaller than that of the landside girder. There is no large static load such as machine room on landside girder. So the overall stiffness is smaller. The impact induced by trolley passing through hinge point increases the amplitude of high-frequency band following the acceleration response.

When the trolley is passing through the hinge point, the frequency of acceleration is mainly concentrated around 100 Hz. The duration of each impact is short, but the accumulated energy in the frequency domain is high. The ultralow frequency under 1/(2π) Hz is not obvious, while the high-frequency signal is evident in the acceleration signal, which is the frequency band that the trolley mainly influences on crane structural response.

The high-frequency dynamic impact caused by trolley passing through rail gap results in the wear of the rail pad as well as some bolt failures as shown in Figure 7. The failure mechanism analysis is not the purpose of this paper, but SHM on logistics equipment including structural response is necessary to avoid the productivity reduction and safety of the quay crane.

4. Model of Quay Crane

4.1. Single Girder Model of Quay Crane. In this section, single segment of girder with simple support is established based on the Euler-Bernoulli theory. It is assumed that the trolley is not separated from the girder during the running process, and the dynamic equation of the girder segment under the action of the trolley is derived. The displacement solution of the model is obtained by the Newmark method.

The front and rear girder hinge points of the quay crane and the girder segment between the front girder and middle tie rod hinge points are taken as the research objects. The girder segment is simplified as a simply supported beam, and the trolley is a moving mass acting on the simply supported beam.

After simplification, the dynamic balance equation of the beam segment can be expressed as

\[
EI \left( \frac{d^4 y(x,t)}{dt^4} \right) + m \left( \frac{d^2 y(x,t)}{dt^2} \right) + c ( \dot{y} x(t)) \right\} \delta(x - x_i),
\]

where \( EI \) is the stiffness of the girder, \( m \) and \( m_i \) are the mass per unit length of the girder and the mass of the trolley respectively, \( c \) is the damping of the girder structure, \( x \) is the longitudinal coordinate value along the girder, and \( y \) is the vertical displacement value of the girder. \( \gamma(x,t) = \varphi(x) \eta(t); \gamma_i(x) = \sin(i\pi x/L) \), \( i = 5 \) in this paper; \( \delta(x-xt) \) is the Dirac function. When the value in the parentheses is 0, the function value is 1.

The right side of the equal sign is the external load, and the second-order partial derivatives in square brackets represent the vertical acceleration of the girder vibration at the location of the trolley, the vertical acceleration caused by the vertical velocity change of the girder caused by the trolley moving, and the centrifugal acceleration caused by the trolley moving on the vertical curve due to the curvature of the girder in the process of the vibration. The matrix expression of the dynamic balance equation of the girder is

\[
[M] \{ \ddot{\eta}(t) \} + [C] \{ \dot{\eta}(t) \} + [K] \{ \eta(t) \} = \{ F(t) \},
\]

where
\[
\eta(t) = \begin{bmatrix}
\eta_1(t) \\
\eta_2(t) \\
\eta_3(t) \\
\eta_4(t) \\
\eta_5(t)
\end{bmatrix},
\]
\[
M = \begin{bmatrix}
I + \gamma D_{11} & \gamma D_{12} & \cdots & \gamma D_{15} \\
\gamma D_{21} & I + \gamma D_{22} & \cdots & \gamma D_{25} \\
\vdots & \vdots & \ddots & \vdots \\
\gamma D_{51} & \gamma D_{52} & \cdots & I + \gamma D_{55}
\end{bmatrix},
\]
\[
F = \begin{bmatrix}
\gamma g D_1 \\
\gamma g D_2 \\
\vdots \\
\gamma g D_5
\end{bmatrix},
\]
\[
C = \begin{bmatrix}
2(\xi_1 \omega_1 + \gamma \Omega D_1 E_1) & 2\gamma \Omega D_1 E_1 & \cdots & 2\gamma \Omega D_1 E_5 \\
2\gamma \Omega D_2 E_1 & 2(\xi_2 \omega_2 + \gamma \Omega D_2 E_2) & \cdots & 2\gamma \Omega D_2 E_5 \\
\vdots & \vdots & \ddots & \vdots \\
2\gamma \Omega D_5 E_1 & 2\gamma \Omega D_5 E_2 & \cdots & 2(\xi_5 \omega_5 + \gamma \Omega D_5 E_5)
\end{bmatrix},
\]
\[
K = \begin{bmatrix}
\omega_1^2 - \gamma \Omega^2 D_1^2 G_1 & -\gamma \Omega^2 D_1^2 G_2 & \cdots & -\gamma \Omega^2 D_1^2 G_5 \\
-\gamma \Omega^2 D_2^2 G_1 & \omega_2^2 - \gamma \Omega^2 D_2^2 G_2 & \cdots & -\gamma \Omega^2 D_2^2 G_5 \\
\vdots & \vdots & \ddots & \vdots \\
-\gamma \Omega^2 D_5^2 G_1 & -\gamma \Omega^2 D_5^2 G_2 & \cdots & \omega_5^2 - \gamma \Omega^2 D_5^2 G_5
\end{bmatrix}.
\]

**Figure 6**: Spectrum of measured vertical acceleration at hinge point: (a) trolley passing through landside girder, (b) trolley passing through waterside girder, and (c) trolley passing by rail gap.
where \( D_i = \sin i\Omega t \); \( D_{ij} = \sin i\Omega t \sin j\Omega t \); \( E_i = \cos i\Omega t \); \( G_i = i2\sin i\Omega t \); \( \gamma = 2\, m^2/(m\, L) \); \( \Omega = \pi v/L \). \( L \) is beam length; modal damping ratio \( \xi_n = 0 \); natural vibration circle frequency is

\[
\omega_n = \left( \frac{n\pi}{L} \right)^2 \sqrt{\frac{EI}{m}} \tag{4}
\]

Equation (2) is a second-order differential equation with variable coefficients. The mass matrix and external force vector are only related to the mass of the trolley, the mass of the girder, and the order of the mode shape. The centrifugal acceleration term in the dynamic balance equation (1) is omitted from the calculation in this section, and its influence on the results is calculated and compared in Section 4.3.

4.2. Result. The actual working conditions of the quay crane and the load of the girder include four parts: the trolley, the spreader, the spreader hanger, and the hoisting weight. In this section, vertical response analysis at midspan of the single girder under simple working condition is conducted. The working conditions studied are as follows: the trolley runs at a constant velocity of 240 m/min, 350 m/min, and 480 m/min without lifting weight of 20 t, with a spreader of 40 t, with lifting load of 60 t, and a full load of 80 t. Among them, 240 m/min is the nominal velocity of the trolley, 350 m/min is the maximum velocity of the trolley, and 480 m/min is the ultra high velocity of trolley for quay crane nowadays. The parameters of the model are shown in Table 1.

In this section, the Newmark method [7] is used to solve the problem, and the midspan vertical displacement of the girder segment calculated from the parameters in Table 1 is shown in Figure 8.

It can be seen from Figure 8(a) that the deflection of the vertical displacement time-history curve at midspan increases with the growth of the moving load. The maximum dynamic deflection of the trolley is less than 5 mm when it is

**Table 1: Parameters of dynamic equilibrium equations model.**

| Parameter                                      | Numerical value (t) |
|-----------------------------------------------|---------------------|
| Beam length (m)                               | 20                  |
| Moment of inertia of beam cross section (m^4) | 0.038               |
| Loading mass (t)                              | 20, 40, 60, 80      |
| Trolley velocity (m/min)                      | 240, 350, 480       |
Figure 8: Midspan vertical displacement of girder under different load or velocity conditions: (a) the trolley velocity $v_t = 240$ m/min, and (b) the load mass $m_t = 80$ t.

Table 2: Dynamic coefficient of vertical displacement.

| Load (t) | Static displacement (m) | Velocity | Dynamic displacement (m) | Dynamic factor |
|----------|-------------------------|----------|--------------------------|----------------|
| 20       | $-4.177 \times 10^{-4}$ | 240 m/min| $-4.300 \times 10^{-4}$  | 1.029          |
|          |                         | 480 m/min| $-4.312 \times 10^{-4}$  | 1.032          |
| 80       | $-1.671 \times 10^{-2}$ | 240 m/min| $-1.720 \times 10^{-2}$  | 1.029          |
|          |                         | 480 m/min| $-1.731 \times 10^{-2}$  | 1.035          |

Figure 9: Displacement result considering the centrifugal term.
not loaded and about 15 mm when it is fully loaded. When the trolley moves at a constant velocity of 240 m/min, there is a small fluctuation in the displacement time-history curve, which is related to the inertial impact on the structure during the movement of the load. As the dynamic load value of the girder increases, the fluctuation value caused by the vertical inertia impact of the load increases.

The time for the trolley to pass through the single beam is inversely proportional to the velocity, which means total time span shortens as the trolley velocity increases. Under the same moving mass, the maximum displacement value is proportional to the velocity. The static load conditions of the trolley in the span of the girder under no-load and full-load conditions are calculated. As shown in Table 2, the displacement

| Operation condition | Centrifugal term | No centrifugal term | Relative error |
|---------------------|------------------|---------------------|---------------|
| 20 t 240 m/min      | $-4.313e-4$      | 240 m/min           | 0.301%        |
| 20 t 480 m/min      | $-4.329e-4$      | 480 m/min           | 0.393%        |
| 80 t 240 m/min      | $-1.743e-2$      | 240 m/min           | 1.3%          |
| 80 t 480 m/min      | $-1.750e-2$      | 480 m/min           | 1.9%          |

Table 3: Numerical results of vertical displacement of girder span.

![Refined model of the quay crane](image)
4.3. Influence of Centrifugal Acceleration on Midspan Vertical Displacement. Considering the influence of centrifugal acceleration, the midspan displacement response girder under each working condition in 4.2 is calculated. The results are shown in Figure 9. The comparison of the maximum displacement values in the midspan is shown in Table 3. The centrifugal acceleration increases the displacement amplitude. When the overloaded trolley is running at a velocity exceeding the nominal velocity, the effect of centrifugal items on the displacement result is not more than 2%.

The response of single girder under trolley traveling is easy to realize with Euler beam theory, while the actual condition of quay crane girder is more complex because of the multihanging structure of girder.

With the conclusion of 4.2, the trolley mass in (1) is treated as constant load. The equation becomes

$$EI \frac{d^4 y(x,t)}{dx^4} + m \frac{d^2 y(x,t)}{dt^2} + c \frac{dy(x,t)}{dt} = m_1 g \delta(x-x_i). \quad (5)$$

The dynamic displacement of the girder $y$ can be expressed as

$$y(x,t) = \sum_{i=1}^{\infty} \phi_i(x) \eta_i(t), \quad (6)$$

where the $n$th mode of vibration is

$$\phi_n(x) = \sin \frac{n \pi x}{L}. \quad (7)$$

$x$ is the arbitrary longitudinal position of the girder, and $t$ is time. If the damping term is ignored, the single girder model with moving constant load can be solved as

$$y(x,t) = \frac{2m_1 g L^3}{mL} \sum_{n=1}^{N} \frac{1}{\omega_n^2 - \Omega_n^2} \left( \sin \Omega_n t - \frac{\Omega_n}{\omega_n} \sin \omega_n t \right) \phi_n(x), \quad (8)$$

where $\Omega_n = (n \pi v/L)$ is the driving frequency of the trolley.

When the velocity of the trolley is small, the driving frequency of the trolley is correspondingly small, which is much less than $\omega_n$. The response of the quay crane structure is quasistatic under these circumstances.

When $n=1$, the vertical displacement of the girder is

$$y(x) = \frac{m_1 L^3 g}{48EI} \frac{\pi v t}{L} = F_i \sin \pi f t, \quad (9)$$

### Table 4: The vibration characteristic of quay crane.

| Degree | Frequency (Hz) | Mode of vibration |
|--------|----------------|-------------------|
| 1      | 0.307          | Lateral swing of waterside girder |
| 2      | 0.451          | Lateral swing of landside girder |
| 3      | 0.747          | The waterside and landside girders swing in the same direction |
| 4      | 0.809          | Longitudinal vibration of the upper structure (deformation of portal leg on waterside) |
| 5      | 0.820          | Longitudinal vibration of the upper structure (deformation of portal leg on landside) |
| 6      | 1.084          | Vertical swing at the end of the girder (the same direction as the gantry) |
| 7      | 1.423          | Vertical swing at the end of the girder (the reverse of the gantry) |
where $F_t$ is the static displacement of girder when the trolley is at the worst position and $f$ is the frequency of the structure under trolley traveling.

5. Overall Structure Model of Quay Crane

The refined modeling analysis of the whole metallic structure is carried out for the measured quay crane. The model of the overall crane structure and the details of the hinge points between landside and water side girder are shown in Figure 10.

The quay crane studied is double-girder type. The main structure is a box-section girder. The material of main structure is Q235 (equal to ASTM A306GR55). The trolley is traction type, and the total weight of the trolley and the hoisting load is about 80 t when fully loaded. In this section, the finite element commercial software ANSYS is used to carry out the structural modeling and response analysis of the whole machine. The SHELL181 element is used to model the girders to better simulate the dynamic response caused by the girder structure and the load of the trolley. The metallic structures such as gantry and diagonal braces are built with BEAM44 and PIPE16 elements. LINK10 element is used to model the backstay and forestay to avoid multi-degree-of-freedom analysis caused by hinge point coupling. The MASS21 element is used for trolley load modeling. Traditional key-point-line contact cannot realize the interaction between mass and shell elements. In this paper, the $S$-$M$ contact method is applied to the dynamic analysis of the quay crane for the first time. By establishing the target element on the flange of trolley girder and the contact element on the trolley mass element, the inertial force of the trolley motion and its mass are applied to the thin-walled shell element of the trolley girder. The DOF of trolley mass has
been constrained in other directions except the motion direction along the track beam. Displacement commands were applied to mass element in the moving direction to realize the interaction between mass and shell surface.

5.1. Modal Analysis. Modal analysis is used to obtain the nature frequency of the quay crane structure. The first seven modal shapes of the whole machine are shown in Table 4. The first three orders are lateral vibrations, the fourth and fifth orders are longitudinal vibrations, and the rest are vertical vibrations. The first seven orders are located in 0–2 Hz band. Most of them are lower than 1 Hz, which is difficult to be detected for the available equipment.

5.2. Vertical Acceleration of Crane Structure. ANSYS is used to analyze the transient dynamics of measured quay crane under the influence of trolley traveling. The velocity of the trolley is 4 m/s, which is the nominal velocity. The traveling length of the trolley is 88 m. The influence of the track irregularity and the dead weight of the structure are ignored. The integration time step is 0.0004 s. Figure 11 shows the results of vertical dynamic load at the same hinge point as the measured position. The spectrum of the calculated acceleration signal is shown in Figure 12. Similar to the measured acceleration signal spectrum, the main energy of the hinge point acceleration signal is concentrated within 200 Hz. Since the fitting of the rail and the beam has been
simplified in FE model, the rail, the rail pad, the rail bolt, and the fitting of track and the track beam in the actual quay crane structure have been ignored. So the energy of the calculated impact signal is smaller than the measured one, especially in high-frequency band caused by the omitted component mentioned above. The impact segment of calculated acceleration signal at the rail discontinuity points is extracted and compared with the impact segment of the measured signal.

The comparison of time domain signal is shown in Figure 13. The shape of the signal and the peak size of the shock are similar. The error analysis is shown in Table 5. In this section, three indexes related to shock signals are selected for calculation and comparison. Among them, the amplitude refers to the maximum value of the absolute value of the difference between adjacent peaks and valleys. The relative error is the quotient of the absolute error and the test value. The main frequency of the impact signal obtained by calculation and test is 80.65 Hz, and the amplitude is about 290 m/s². It can be seen that the simulation result is consistent with the test one. It can be used as a reference for dynamic analysis of quay crane structures.

5.3. Displacement of Crane Structure. The structural response of the quay crane caused by the trolley is further analyzed. Figure 14 shows the positions of the ten measuring points of the quay crane girders and the specific descriptions of vertical, longitudinal, and transverse.

The vertical direction is perpendicular to the horizontal plane. The longitudinal direction is parallel to the girder, that is, the running direction of the trolley. The lateral direction is the horizontal direction perpendicular to the girder. The results of displacement are shown in Figures 15–17. With the movement of the trolley, the measuring point at the rear end of the girder firstly produces displacement. The whole machine’s girder span, the front girder midspan, and the front end of the front girder produce displacement responses one after another. As the quay crane girder structure is suspended by four gantry points and supported by six front and rear tie rods to form a multisection simply supported beam, therefore the displacement response of the previous section of the beam will produce reverse fluctuate when the trolley moves to the next section of the beam.

The magnitude and fluctuation direction of the same measuring point on the left and right sides of the vertical displacement are almost the same. Among all the measuring points, the largest displacement fluctuation is located on girder end and boom head, followed by the four measuring points of the waterside girder. The displacement of girder-left, girder-right, mid-girder-left, and mid-girder-right is smaller than those of other measuring points. The measuring points at both ends of the girder are supported by tie rods with low rigidity. The four measuring points named boom-left-1, boom-left-2, boom-right-1, and boom-right-2 are far from the rigid support of the gantry. So the vertical displacement results are larger than those closer to the gantry, such as the measuring points of mid-girder and girder-left and girder-right. When the trolley moves to the next beam
segment, the reverse fluctuation of the vertical displacement response is more significant than the displacement responses in the other two directions.

The longitudinal displacement is caused by the inertial load of the trolley moving forward, and the longitudinal displacement curves of all measuring points show a roughly upward trend with time. The five measuring points on the land side first generate longitudinal displacement, and the amplitude is greater than that of the five measuring points on the water side, which is also related to the working condition of the trolley moving from the land side to the water side. If the trolley moves in the opposite direction, the above results will also have the opposite trend.

The lateral displacement of the same measuring point on right and left girders has the opposite direction, which is related to the eccentric loading condition of the rail girder. When the trolley was passing through the measuring point, the left girder twists to the right and produces the right lateral displacement and vice versa. In the lateral displacement results, the displacement fluctuations of measuring points on the water side and land side are smaller than those of other side points, as shown in Figure 17, and the trend is gentle. According to the above analysis, the lateral displacement of girder under trolley traveling is mainly due to the torsional lateral deformation caused by the trolley load bias relative to the girder, as in Figure 10. The measure points at both ends of the girders are almost unaffected by the torsion of the girder, so the results are the least obvious. In the displacement results of the three directions, the maximum vertical displacement is 120.6 mm, and the minimum longitudinal displacement is 11.47 mm, all of which are at the measurement point of the last end of the girder.

The vertical displacement spectrum of each measuring point is shown in Figure 18. The result shows that the low-frequency vibration information is retained more completely in the displacement signal. The first frequency of the displacement of each measuring point is \( v/L = 4/88 = 0.454 \text{ Hz} \), which is the disturbance frequency of the structure under trolley traveling proposed in 4.3. This means the nominal velocity of the crane trolley is slow enough and has little dynamic effect on quay crane structure response. The velocity of the trolley exceeding resonant velocity will induce large dynamic effect, which is the length of the girder times the natural frequency of the quay crane. But this condition can never happen. Because the maximum trolley velocity for quay crane is less than 10 m/s, while the minimum resonant velocity of the studied crane is 27 m/s. This condition can be used in bridge with large span or the structure with low velocity moving load compared to the length of the structure.

6. Conclusion

1. The acceleration response at the girder hinge point under the working condition of the quay crane in service is measured on site. The trolley passed from landside to waterside girder. The time series of acceleration shows the high-frequency impact induced by trolley passing through hinge point. The spectrum shows that the main frequency band of the structure acceleration is 0–200 Hz. The separation analysis shows that the impact induced by trolley passing through will affect the following acceleration response on the high-frequency band of about 100 Hz. The acceleration shows weakness in ultralow-frequency analysis.

2. According to the actual working conditions of the quay crane, the dynamic responses of the single girder under different working conditions are analyzed. When the trolley runs at full load at a velocity exceeding the nominal velocity of 480 m/min, the dynamic effect of the vertical displacement response of the girder is still not obvious. From the Euler equation, it is deduced that the inertia term of the equation can be ignored when the trolley velocity is low. The displacement of girder is quasistatic.

3. The advantages of plate and shell elements are utilized to model complex structures, a refined model of the quay crane including the geometric features of the actual measuring points is established, and the S-M contact method is applied to the dynamic analysis of the quay crane structure for the first time. The acceleration response of the measured position is obtained. The analysis shows that under the working conditions, the vertical acceleration response of the quay crane is mainly affected by the impact of the trolley passing the hinge point in high-frequency band 10–200 Hz, which confirms the inference of the measured results.

4. The displacement responses of ten measuring points of the girder and the trolley in the whole machine model are extracted. The time domain of the displacement shows the trolley traveling effects mainly in vertical direction. Lateral displacement reveals the eccentric loading condition of girder. The frequency domain of displacement response reveals that under trolley traveling with nominal velocity, the displacement response of the structure is quasistatic. The dynamic load effect of the trolley on the quay crane structure is small.

5. The influence of trolley traveling on crane structure response is mainly in high frequency, which is shown in acceleration response. The displacement response of crane structure reveals that the trolley has little dynamic effect on it. The displacement response can be treated as quasistatic status. The trolley velocity should exceed the first resonant velocity, which is 27 m/s in this case far beyond the nominal velocity of the trolley, to have significant dynamic effect on the crane structure response. This deduction can be used in bridges with large span or structures with low velocity moving load compared to the length of the structure.

Data Availability

The data used to support the findings of this study can be obtained from the corresponding author upon request.
Conflicts of Interest

The authors declare no conflicts of interest in this article.

Acknowledgments

This research received specific grant from National Natural Science Foundation (62073213).

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