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Study on longitudinal vibration characteristics of lifting coupling system of quayside container crane with variable length wire rope

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Abstract

Aiming at the quayside container crane, to research longitudinal vibration of lifting coupled vibration system with variable length wire rope, the differential equations of the longitudinal vibration of the system are developed employing the Hamilton's principle, is solved by discreting using finite difference method. The results show that the coupling consistency exists in the vibration of the hoisting system, the vibration response changes abruptly when the acceleration changes, the maximum amplitude occurs in the acceleration stage of hoisting, and the residual vibration exists when the velocity of the load drops to zero. The vibration instability of the girder and the wire rope has a great influence on the efficiency and safety of quayside container crane operation. The governing equations derived can provide reference to further study on quayside container crane.

Keywords

Quayside container crane, variable length wire rope, longitudinal vibration, finite difference method, vibration instability.

1. Introduction

Dynamic load impact will occur on the structure during the lifting process, when a quayside container crane lifts heavy objects, due to the sudden departure from the ground or the change of acceleration and speed of the lifting load, which will aggravate the fatigue of the structure, thus reducing the service life of the structure and affecting the stability and efficiency of the operation. Therefore, it is necessary to study the vibration characteristics of hoisting system. Many scholars have studied the vibration of variable length wire rope in mine, elevator and other lifting equipment. Kaczmarczyk S et al. [1] considered the influence of drum and transmission pulley, established the differential equation of longitudinal vibration of deep well winding hoisting wire rope by Hamilton principle using moving coordinate system method, solved the frequency, vibration displacement and dynamic tensions of the wire rope at each concentrated mass of the system by multi-scale method. Zhu et al. [2] analyzed the stability of variable length beam and rope under different boundary conditions from the energy point of view, and gave the exact expression of chord energy variation. Zhang et al. [3] established the longitudinal vibration control equations of flexible transmission system with arbitrary variable length by Newton method and Hamilton principle respectively, and solved them by Rayleigh-Ritz method. Cao et al. [4] considered the action of head sheave and string, derived the solution method and approximate analytic expression of longitudinal and torsional coupling vibration displacement, tension and torsion of hoisting wire rope in the process of entering and leaving cage in mine based on step function. Zhang et al. [5] established the longitudinal vibration model of variable length wire rope, solved and analyzed the vibration response and dynamic tension of wire rope under

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two different working conditions of elevator up and down using Galerkin method, in the condition of the ideal operation curve. Wang et al. [6] considered the lateral excitation at both ends of the rope, established the governing equation of the forced coupling vibration of the variable length traction rope of the elevator suspension system, and solved the transverse vibration displacement and energy of the elevator in the process of ascending by the finite difference method. These models focused on the coupling vibration of variable length wire rope, however, the coupling vibration of wire rope and upper flexible structure was not involved. In this paper, establish a model of variable length wire rope hoisting coupling system of quayside container crane, study and analyse the longitudinal vibration characteristics of girder and wire rope considering this factor.

2. Model of the lifting coupled vibration system

Lifting coupled vibration system of the quayside container crane can be simplified as an axially moving string with time-varying length, with a driving pulley fixed on the girder connecting to the upper end and a hoisting load at its lower end, ignoring the specific structure of the spreader, as shown in Fig.1. In this model, the mass of the moving car is m_t, the mass of the driving pulley is m_p, the mass of the superstructure of the wire rope is m_e, the mass of the substructure of the wire rope is M. The girder is simplified as a spring with equivalent stiffness k_b and mass m_b. The flexible wire rope has Young's modulus E, cross section area S, and the density per unit length ρ. The origin of the coordinate is set at the centroid of the pulley when the system is statically balanced. The length of the wire rope is $l(t)$, the longitudinal vibration displacement of the rope at $x(t)$ is $u(x,t)$, the vibration deflection of the girder is γ, and the initial wire rope length is L_0 in the process of hoisting. The instantaneous translational velocity and acceleration of the wire rope are $\dot{x}(t)=v(t)$ and $\dot{v}(t)=a(t)$ respectively, the instantaneous velocity of lifting load off the ground is v_0, where the overdot denotes time differentiation.

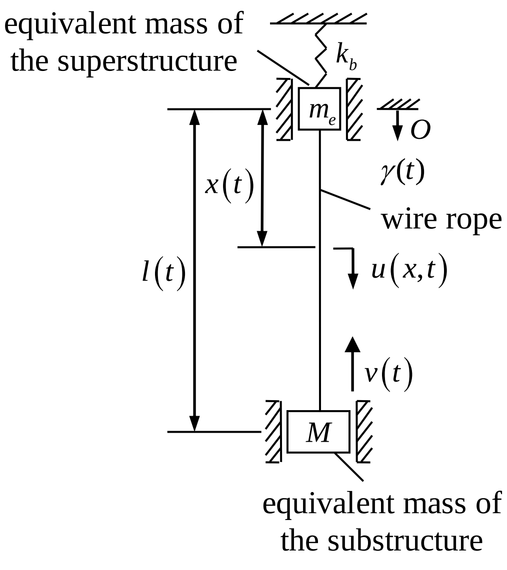

Fig. 1 Model of lifting coupled vibration system of the quayside container crane

The model base on the following assumptions: 1. the influence of longitudinal vibration of the girder is ignored; 2. the influence of coupling vibration between the surrounding structures and the wire rope among the drum and the driving pulley on the girder are ignored. 3. Young's modulus E, cross section area S, and density of the rope ρ are always constants; 4. the bending stiffness of rope, all the damp and friction, and the influence of air current are ignored; 5. only longitudinal vibration is considered here. The elastic distortion of rope arousing from the longitudinal vibration is much less

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than the length of the rope; and 6. the slip of pulley and the influence of pulley and wire rope at meshing section on the system are ignored.

The kinetic energy of the lifting coupled vibration system is given by

$$E_k = \frac{1}{2} m_e \dot{\gamma}^2 + \frac{1}{2}\rho\int_0^{l(t)} (v+\dot{\gamma}+\frac{Du}{Dt})^2 dx + \frac{1}{2} M\,(v+\dot{\gamma}+\frac{Du}{Dt})^2\Big|_{x=l(t)} \tag{1}$$

where the first term on the right of Eq.(1) represents the kinetic energy of the upper structure of the wire rope, the second term represents the kinetic energy of wire rope and the third term represents the kinetic energy of lifting weight. And m_e represents the equivalent mass of substructure can be expressed as

$$m_e = m_b + m_p + m_s \tag{2}$$

the differential operator is expressed as

$$\frac{D}{Dt} = \frac{\partial}{\partial t} + v\frac{\partial}{\partial x} \tag{3}$$

The elastic strain energy of the system is

$$E_e = T_b^i \gamma + \frac{1}{2} k_b \gamma^2 + \int_0^{l(t)} (T^i u_x + \frac{1}{2} EAu_{xx}) dx \tag{4}$$

where $(\ )_x$ denotes partial differentiation with respect to displacement x, T_b^i and $T^i(x)$ represent the initial static tension of the simplified spring and the wire rope at $x(t)$ are expressed as

$$T_b^i = m_e g + \rho g l(t) + Mg \tag{5}$$

$$T^i(x) = Mg + \rho[l(t) - x]g \tag{6}$$

The gravitational potential energy of the system is

$$E_g = E_g^i - m_e g\gamma - \int_0^{l(t)} \rho g(\gamma+u)dx - Mg\,(\gamma+u)\big|_{x=l(t)} \tag{7}$$

Where E_g^i is the strain energy in the initial state, and the second term on the right of Eq.(7) represents the gravitational potential energy of the upper structure of wire rope, the third term represents the gravitational potential energy of the wire rope, the fourth term represents the gravitational potential energy of the load.

Substituting Eqs.(1), (4) and (7) into Hamilton's Principle

$$\int_{t_1}^{t_2} (\delta E_k - \delta E_e - \delta E_g) dt = 0 \tag{8}$$

and applying the time condition and boundary condition

$$\gamma(0)=0, \dot{\gamma}(0)=0, u(0,t)=0, u_t(0,t)=v_0 \tag{9}$$

yields the following system of equations for the deflection:

$$\begin{gathered} m_e\ddot{\gamma} + \rho\int_0^{l(t)} (\dot{v}+\ddot{\gamma}+u_{tt}+\dot{v}u_x+2vu_{xt}+v^2u_{xx})dx + M\,(\dot{v}+\ddot{\gamma}+u_{tt}+\dot{v}u_x+2vu_{xt}+v^2u_{xx})\big|_{x=l(t)} \\ +T_b^i + k_b\gamma - m_e g - \rho g l(t) - Mg = 0 \end{gathered} \tag{10}$$

$$\rho(\dot{v}+\ddot{\gamma}+u_{tt}+\dot{v}u_x+2vu_{xt}+v^2u_{xx}) - T_x^i - \rho g - ESu_{xx} = 0, 0 < x < l(t) \tag{11}$$

$$M\,(\dot{v}+\ddot{\gamma}+u_{tt}+\dot{v}u_x+2vu_{xt}+v^2u_{xx}) + T^i - Mg + ESu_{xx} = 0, x = l(t) \tag{12}$$

where $(\ )_t$ denotes partial differentiation with respect to time t. In this system equations (11) and (10) describe the dynamics of the wire rope and the superstructure respectively, equation (12) defines

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motion of the end mass. When the vibration displacement of the girder is not considered, equating to $\gamma=0$, the equations (11) and (12) are in agreement with those obtained in reference [5].

3. Discretization of finite difference method

A new variable $\xi = x/l(t)$ is introduced to normalize the original variable, where $\xi\in[0,1]$, and longitudinal vibration displacement of wire rope is represented as $\hat{u}(\xi,t)=u(x,t)$, spatial variable ξ is discreted into $\xi_0=0$, ξ_1, ξ_2, $\cdots$, $\xi_{n-1}=1$, where $\xi_i=\xi_0+ih$, $h=1/n$. The displacement $\hat{u}(\xi_i,t)$ is defined as $U_i(t)=\hat{u}(\xi_i,t)$, directing central difference for partial differential of relative dimensionless space variable in equation of motion, and introducing the following difference terms as shown in Eq.(13) and Eq.(14), where $i=1, 2, \cdots, n-1$

$$\left.\frac{\partial\hat{u}}{\partial\xi}\right|_{\xi_i}=\frac{U_{i+1}-U_{i-1}}{2h}+O(h^2) \tag{13}$$

$$\left.\frac{\partial^2\hat{u}}{\partial\xi^2}\right|_{\xi_i}=\frac{U_{i+1}-2U_i+U_{i-1}}{h^2}+O(h^2) \tag{14}$$

and substituting Eq.(13) and Eq.(14) into equations (10-12), then yields the ordinary differential equation as shown in equation (35)

$$\boldsymbol{M}\begin{bmatrix}\ddot{\gamma}\\ \ddot{\boldsymbol{U}}\end{bmatrix}+\boldsymbol{C}\begin{bmatrix}\dot{\gamma}\\ \dot{\boldsymbol{U}}\end{bmatrix}+\boldsymbol{K}\begin{bmatrix}\gamma\\ \boldsymbol{U}\end{bmatrix}=\boldsymbol{F} \tag{15}$$

where $\boldsymbol{U}(t)=[U_1(t),U_2(t),\cdots,U_n(t)]^T$ is the displacement vector of longitudinal vibration, $\boldsymbol{M}(t)$, $\boldsymbol{C}(t)$, $\boldsymbol{K}(t)$ and $\boldsymbol{F}(t)$ are matrixes of mass, damp, stiffness and generalized force respect to $\boldsymbol{U}(t)$ and $\gamma(t)$, respectively. The real-time values of the vibration deflection of the girder $\gamma(t)$ and the longitudinal vibration response of the wire rope $u(x,t)$ may be yielden by solving the ordinary differential equations (15) with numerical methods.

4. Simulation results and analysis

The system vibration response can be calculated and analyzed through Eq.(15) by selecting the parameters of 65t-55m quayside container crane in a port. For the quayside container crane, rated lifting speed, lifting acceleration and starting and braking time are 1m/s, 0.5m/s^2, and 2s, respectively. The length parameters of each section of the girder are $L_{AB}=26.0\text{m}$, $L_{BC}=26.0\text{m}$, $L_{CH}=15.1\text{m}$, $L_G=56.0\text{m}$, respectively. The parameters of the coupled system are $m_{b0}=135021.16\times10^3\text{kg}$, $m_p=4\cdot144\text{kg}$, $m_s=18258\text{kg}$, $I=7.5648\times10^{10}\text{mm}^4$, $\rho=7.8\times10^3\text{kg/m}^3$, $E=8\cdot2.1\times10^5\text{MPa}$, $d=32\text{mm}$, $M=82.5\text{t}$, $l(0)=41\text{m}$, where m_{b0} represents total mass of the single front girder, $l(0)$ represents initial length of the wire rope, the total numbers of the front girder and the wire ropes are 2 and 8 respectively.

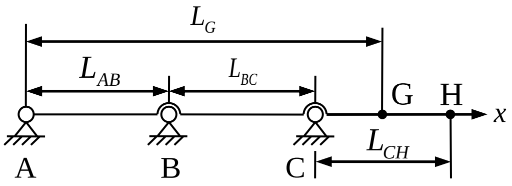

Fig. 2 Schematic of the front girder of the quayside container crane

Figure 2 shows the structural sketch of the front girder of the quayside bridge, in which AB section is the simply supported girder, BCH section is the extended simply supported girder, point A is the hinge joint of the front girder and rear girder, point G is the maximum forward extension location of front girder. And the sea side direction is taken as the positive direction of coordinate axis.

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According to reference [7], $k_b = 9.9288 \times 10^7$ N/m and $m_b = 92171$kg at the position of L_G, be located at the maximum forward elongation position of the front girder of the trolley during loading and unloading, denotes the equivalent stiffness and the equivalent mass of the front girder, respectively.

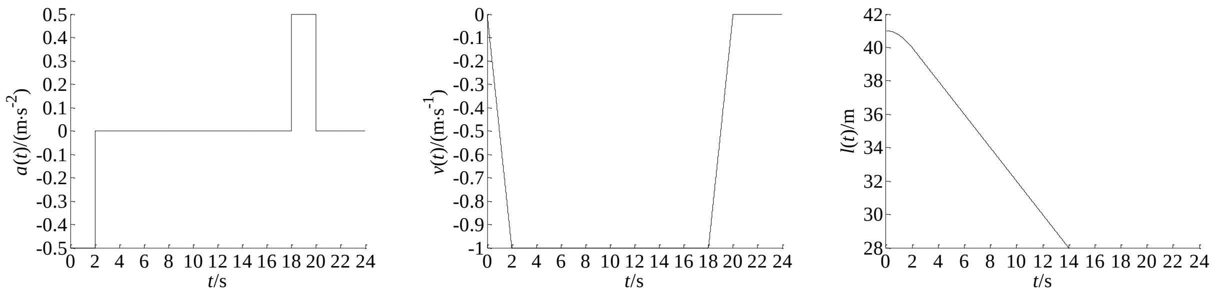

Fig. 3 Upward movement profile curves of the lifting load

Fig. 3 displays the assumed acceleration, velocity, displacement, curves of the lifting coupled vibration system of the quayside container crane, where the processes of acceleration, deceleration and uniform speed upwards are included. Utilizing the curves as the input of equation (15) may obtain the longitudinal vibration responses of front girder and lifting load as seen in Fig. 4 and Fig. 5 respectively, and may yield the dynamic tension curves of wire rope at upper pulley and lower pulley of quayside container crane hoisting system as shown in Fig. 6. The whole calculation process starts from the third stage of the hoisting process [8], and the initial hoisting velocity is 0.05m/s.

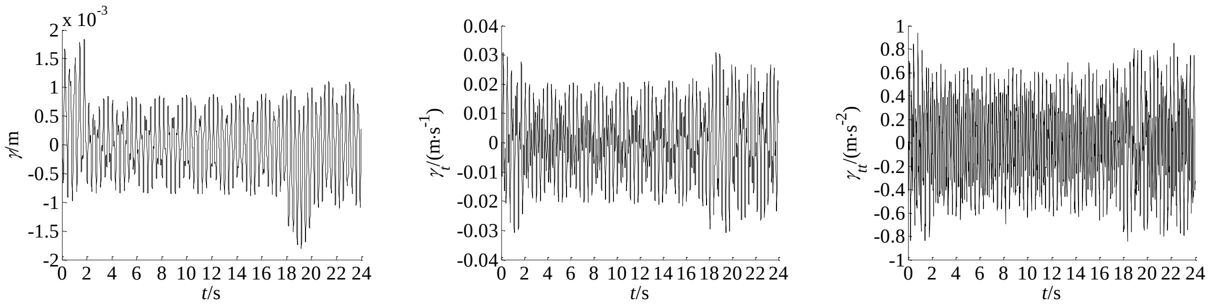

Fig. 4 Vibration response curves of the girder

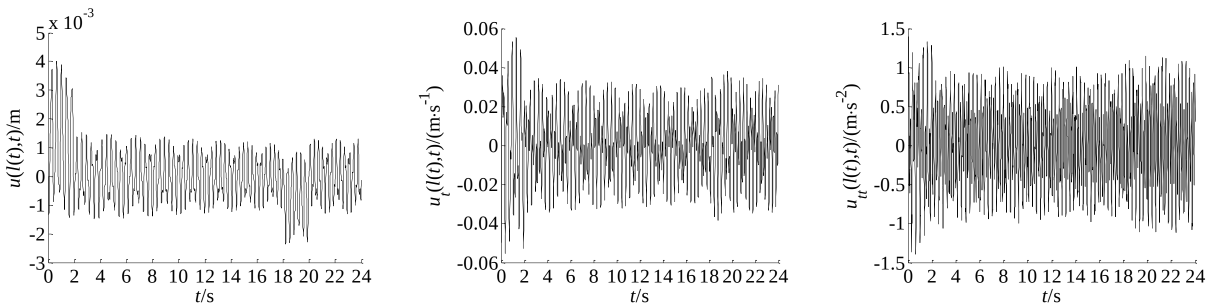

Fig. 5 Vibration response curves of the lifting load

The dynamic tension of wire rope at upper pulley is

$$T(0,t) = T^i(0) + ES\,u_x\big|_{x=0} = \rho g l(t) + Mg + m_e\ddot{\gamma} + k_b\gamma \tag{16}$$

and the dynamic tension of wire rope at lower pulley is

$$T(l(t),t) = T^i(l) + ES\,u_x\big|_{x=l(t)} = Mg + ES\,u_x\big|_{x=l(t)} \tag{17}$$

From Fig. 5 and Fig. 6, it is observed that there is obvious coupling vibration between the girder and the lifting load during the lifting process, both of which vibrate up and down around the equilibrium position. The maximum amplitude occurs in the acceleration stage, which has the greatest impact on the dynamic load of the structure, and residual vibration exists when the speed drops to zero. In the course of hoisting, the vibration responses of the girder and the lifting load have a sudden change

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when the acceleration changes, and there is a small range of up-and-down vibration near each amplitude, and the small range vibration disappears when the speed drops to zero. The sudden change and vibration of vibration responses of the girder and the lifting load will cause discomfort of drivers and affect the accuracy of operation. Figure 7 displays the dynamic tensions of the wire rope at the upper and lower pulleys are consistent with the vibration of the girder and the lifting load. The dynamic tension at the upper pulley as well as the impact is the largest. The sudden change of the dynamic tension and the reciprocating vibration will cause the fatigue of the wire rope and reduce the service life of the wire rope. The dynamic load responses and dynamic tensions mentioned above are most intuitively reflected in the dynamic lifting coefficient [8-9]. These phenomena have already been observed in practice.

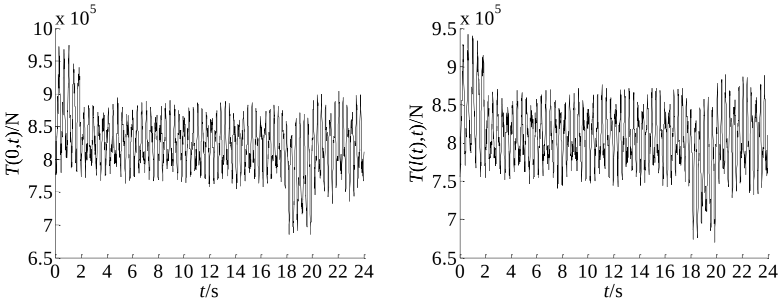

Fig. 6 Dynamic tension curves of the wire rope at the upper and lower pulleys

5. Conclusion

(1) Aiming at the quayside container crane, establishes a model of lifting coupling vibration system including variable length wire rope and girder, transmission pulley, trolley and lifting load. The partial differential equation of longitudinal vibration of the system is established by Hamilton principle, the finite difference method is used to discretize the solution, and the real-time value of the vibration of the variable length wire rope and the girder during hoisting are obtained, respectively.

(2) Based on the model, given the motion process of the lifting system, the vibration response of the girder and the lifting load, and the dynamic tensions of the wire rope at the upper and lower pulleys are obviously coupled. The maximum amplitude occurs in the acceleration stage, and there is a sudden change when the acceleration changes, and there is residual vibration after the motor closes. Measures should be taken to reduce the fatigue caused by vibration and tension vibration of wire rope.

(3) The research in this paper can provide theoretical reference for the overall design of cranes, and the model can lay a foundation for further study of the coupling vibration of wire rope and structure in different directions between load and drum of quayside container crane under different working conditions.

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