Longitudinal Coupled Vibration of Parallel Hoisting System with Tension Balance Devices

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Keywords: longitudinal coupled vibration, parallel hoisting system, tension balance device, dynamic behaviors.

Abstract. In long travel lifting systems, multiple parallel ropes are connected to conveyance through a set of termination devices to achieve the lifting process. Due to external excitations coming from different wear levels and manufacturing error of rope grooves on the friction pulley, or rope slipping, each rope length is different and accordingly results in different rope tension. Therefore, a tension balance device is applied as the termination device to reduce the tension differences. In order to describe the dynamic behavior, rope sockets of the tension balance device are simplified into a lumped mass at the end of each rope, and a longitudinal vibration model of a parallel hoisting system with tension balance device is built. Both the normal and unsteady working conditions of the tension balance device are considered. To deal with the complex constraints between ropes and hoisting conveyance, Lagrange multipliers are used, and the equations are numerically solved. The characteristics of longitudinal vibration frequency are depicted and the dynamic behaviors are revealed, which are essential for optimizing the parallel hoisting system with the tension balance device. That will minimize the effects of external excitations and improve the security of the system.

1 INTRODUCTION

Hoisting system is an efficient vertical transportation to lift human and payload to different levels [1, 2]. Hoisting ropes are axially translating media with time-varying length [3, 4]. Hoisting ropes, due to their flexibility, loading conditions, and internal damping characteristics, mainly determine the resonances of longitudinal and lateral vibration of the system [5, 6]. This may lead to large vibration amplitude and mechanical breakdowns of the equipment especially in high-rise systems. Therefore, parallel hoisting system is now used to improve system safety [7]. Kumaniecka and Nizioł[8] analyzed the longitudinal-transverse vibration with a varying-length rope and focused on the parametric resonances of the rope. Huang [9] studied the dynamic stability of a moving string in three dimensions. The partial differential equations of motion are derived by using Hamilton's principle and later simplified as ordinary differential equations by Galerkin's method. Huang [10] also investigated qualitative aspects of parametric excitation due to a non-constant traveling velocity of a viscoelastic string. Sandilo and Van Horssen studied autoresonance phenomena of one rope in a space–time-varying mechanical system [11].

However, the longitudinal vibration of a hoisting rope with tension balance device is less studied in parallel hoisting system. In this paper, the flexible hoisting system is regarded as a flexible multi-body structure. The governing equations of the flexible hoisting system are developed by using the Lagrange equations technique. The characteristics of longitudinal vibration frequency are depicted and the dynamic behaviors are revealed, which are essential for optimizing parallel hoisting system with the tension balance device. This work will minimize the impact of external excitations and improve the security of the system.

2 DESCRIPTION OF THE MODEL OF A LIFTING SYSTEM

Long travel lifting system mainly includes the following parts by reference to Fig. 1: Sheaves, multiple hoisting ropes, tension balance devices, hoisting conveyance and rigid guide. The end of the rope is fixed on rope sockets (3-a) of the tension balance device. Tension balance devices used to adjust the tension distance between the hydraulic cylinder (3-b) and the base (3-c) according to the tension [12].Multiple parallel ropes are connected to conveyance through a set of termination devices to achieve the lifting process. Eccentric pulleys and sheaves and systematic resonance in the electronic control system are typical causes of longitudinal vibrations of a lifting system. The model is based on the following assumptions: (a) Only longitudinal vibration is considered. (b) Neglecting the influence of the compensation rope.

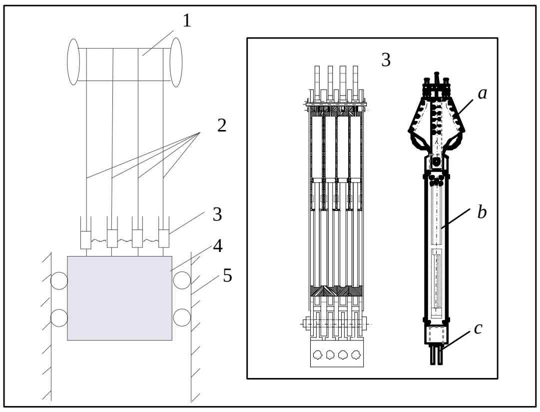

Figure1: Lifting system

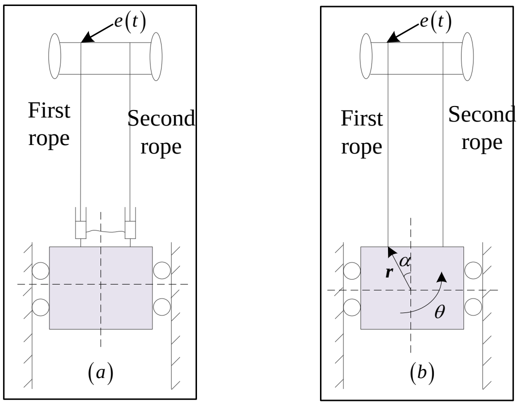

Figure 2: Two hoisting ropes with tension balance (a) and without tension balance (b)

3 VIBRATION MODEL

In order to derive the differential equations of motion, the dynamic model considers the case of n hoisting ropes. The kinetic energy is:

$$T=\frac{1}{2}\sum_{i=1}^{n}\rho\int_{0}^{l_i(t)}\left(u_{i,t}+vu_{i,x}+v_i\right)^2dx+\frac{1}{2}m_i\left(u_{i,t}\left(l_i,t\right)+vu_{i,x}\left(l_i,t\right)+v_i\right)^2+\frac{1}{2}m\dot{u}_c^{\ 2}+\frac{1}{2}J\dot{\theta}^2 \tag{1}$$

where ρ, m_i, m and J are rope mass per unit length, rope mass of sockets on tension balance device, hoisting conveyance mass and inertia moment, respectively. u_i is longitudinal vibration displacement. u_c is the displacement of conveyance. Let $\rho_s = \rho + m_i\,\delta\,(x - l_i)$ for which the kinetic energy can be written as:

$$T=\frac{1}{2}\sum_{i=1}^{n}\rho_s\int_{0}^{l_i(t)}\left(u_{i,t}+vu_{i,x}+v_i\right)^2dx+\frac{1}{2}m\dot{u}_c^{\ 2}+\frac{1}{2}J\dot{\theta}^2 \tag{2}$$

The potential energy and dissipation energy is a function of the vibration displacement:

$$E_e=\sum_{i=1}^{n}\left(\int_{0}^{l_i(t)}T_{i,s}u_{i,x}dx+\int_{0}^{l_i(t)}EA\frac{1}{2}u_{i,x}^{\ 2}dx\right)+\frac{1}{2}\sum_{i=1}^{4}G_i\theta^2 \tag{3}$$

$$E_g=-\rho_s\sum_{i=1}^{n}\int_{0}^{l_i(t)}u_i g\mathrm{d}x-mgu_c \tag{4}$$

$$D=\sum_{i=1}^{n}\left(\frac{1}{2}\mu_i\int_{0}^{l_i(t)}u_{i,t}^{\ 2}dx\right) \tag{5}$$

where G_i is guide stiffness, $T_{i,,s}$ is the rope tension. g is the constant acceleration of gravity. And the Lagrangian will be

$$L=T-\left(E_e+E_g\right) \tag{6}$$

The solutions of longitudinal displacement are assumed in the form:

$$u_i = \sum_{j=1}^{R} \psi_j q_{i,j} \tag{7}$$

where $q_{i,j}$ is the generalized coordinate, $\psi_j = \sin(i-1/2)\pi\xi$ is the mode shape functions. This Lagrangian leads to the following set of simultaneous differential equations of motion

$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial L}{\partial \dot{q}_i} - \frac{\partial L}{\partial q_i} + \frac{\partial D}{\partial \dot{q}_i} = F_i + \sum_{k=1}^{N} \lambda_k \frac{\partial g_k}{\partial q_i} \tag{8}$$

where, $\boldsymbol{q} = \left[\boldsymbol{q}_1^{\mathrm{T}} \quad \cdots \quad \boldsymbol{q}_n^{\mathrm{T}} \quad u_c \quad \theta\right]^{\mathrm{T}}$, $\boldsymbol{g} \in R^{N\times 1}$ is the boundary conditions vector and its definition are denoted in Section 4, λ_k is the Lagrange multipliers. This is a linear system of homogeneous equations, which can be considered as the $n+1$ components of the block matrix equation

$$[\boldsymbol{M}]\ddot{\boldsymbol{q}} + [\boldsymbol{C}]\dot{\boldsymbol{q}} + [\boldsymbol{K}]\boldsymbol{q} = [\boldsymbol{F}] + \boldsymbol{\lambda}^{\mathrm{T}}\boldsymbol{Q} \tag{9}$$

where the matrices are defined by:

$$\begin{aligned}
\boldsymbol{M}^i &= \rho l_i \boldsymbol{M}_1 + m_i \boldsymbol{M}_0 \\
\boldsymbol{C}^i &= \rho v_i \left(\boldsymbol{C}_1 + \boldsymbol{C}_2 - \boldsymbol{C}_2^{\mathrm{T}}\right) + \mu_i l_i \boldsymbol{C}_1 \\
\boldsymbol{K}^i &= \rho a_i \boldsymbol{K}_1 - \frac{\rho v^2}{l_i}\boldsymbol{K}_2 + \frac{EA}{l_i}\boldsymbol{K}_3 - \mu_i v_i \boldsymbol{K}_4 \\
\boldsymbol{F}^i &= \left(\rho l\left(g - a_i\right) - \rho v_i^2\right)\boldsymbol{F}_1 - m_i g \boldsymbol{F}_2 - \left(\rho g l_i - \rho v_i^2\right)\boldsymbol{F}_3 - m_i a_i \boldsymbol{F}_4
\end{aligned}$$

$$\boldsymbol{M}^{n+1} = \begin{bmatrix} m & 0 \\ 0 & J \end{bmatrix}, \boldsymbol{C}^{n+1} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}, \boldsymbol{K}^{n+1} = \begin{bmatrix} 0 & 0 \\ 0 & \sum_{i=1}^{4} G_i \end{bmatrix}, \boldsymbol{F}^{n+1} = \begin{bmatrix} mg \\ 0 \end{bmatrix}$$

$\boldsymbol{M}_i, \boldsymbol{C}_i, \boldsymbol{K}_i, \boldsymbol{F}_i$ are mode matrixes with boundary conditions and $\boldsymbol{Q}$ is Jacobi matrix related to the constraint equations g_k. After elimination of constraints, we can obtain the complete solution of the problem.

4 CASE STUDY

In this section, simulation results of the two ropes in this paper are discussed.

Case (a):

When guide stiffness G_i is soft as Fig. 2(b). The constraint relationship between the hoisting rope and conveyance without a tension balance device should be fulfilled as followed:

$$g_1 = u_c + r\theta\sin\alpha - u_1(1,t), \mathrm{g}_2 = u_c - r\theta\sin\alpha - u_2(1,t) \tag{10}$$

in which, r is the distance to the connecting point as Fig. 2(b), and a rotation around the axis of angle θ (in the positive trigonometric sense).

Case (b):

When tension balance devices are applied in the system as shown in Fig. 2(a), constraint condition is:

$$g_1 = \frac{1}{2}\left(u_1(1,t) + u_2(1,t)\right) - u_c \tag{11}$$

5 NUMERICAL CALCULATION

In this simulation, two hoisting ropes one with a tension balance device the other without are selected for comparison. The fundamental parameters of the system are listed in Table 1.

Table 1 Fundamental parameters of the system

Parameter	Value	Unit	Parameter	Value	Unit
m	35000	kg	ρ	8.5	kg/m
m_i	400	kg	G_i	2×107	Nm
EA	1.138×105	N	J	2000	kg·m^2
μ_i	0.4125	kg/s	l_0	2100	m

By numerical calculations, the results demonstrate the nonlinear dynamic interaction between the components of the lifting system, during the hoisting process and when the system is non-stationary. The roots determine the natural frequencies of the system. The natural frequencies are determined from the roots of the frequency equation associated with Eq. (9) by calculating the eigenvalue.

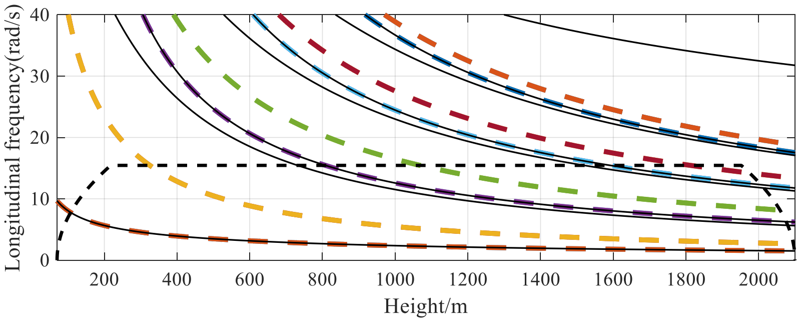

Figure 3: Frequency characteristic diagram (without tension balance device in solid line ω, with tension balance device in dotted line $\bar{\omega}$. Excitation frequency in black dotted line)

In both cases, the first order is longitudinal low frequency and they are exactly the same. As shown in Fig. 3, odd orders of longitudinal frequency are exactly consistent ($\omega_{2i-1} = \bar{\omega}_{2i-1}$), while the even orders are higher than that with tension balance device ($\omega_{2i} > \bar{\omega}_{2i}$). The second order is the rotational frequency of hoisting conveyance.

The numerical output demonstrates the rotational vibration resonates at 700m in Fig. 4 (The frequency diagram also shows the intersection of the excitation frequency and the rotation frequency). Longitudinal vibration of the hoisting conveyance is basically similar between two models. However, the resonance peak in other parts of hoisting rope is much higher than that with tension balance device (Fig. 5). The results show that the tension device fulfills its function and achieves the rope tension balances.

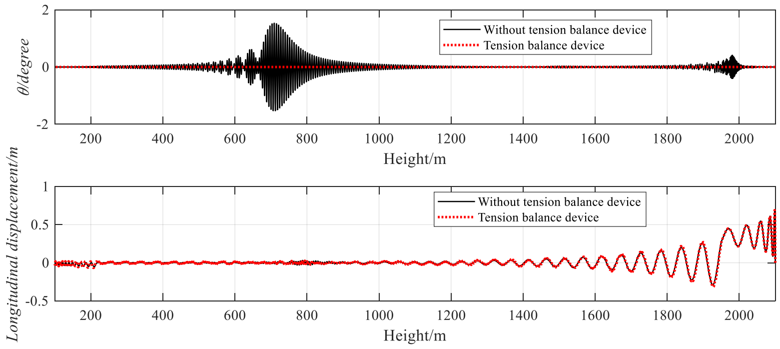

Figure 4: Terminal response when the excitation is applied to the first rope

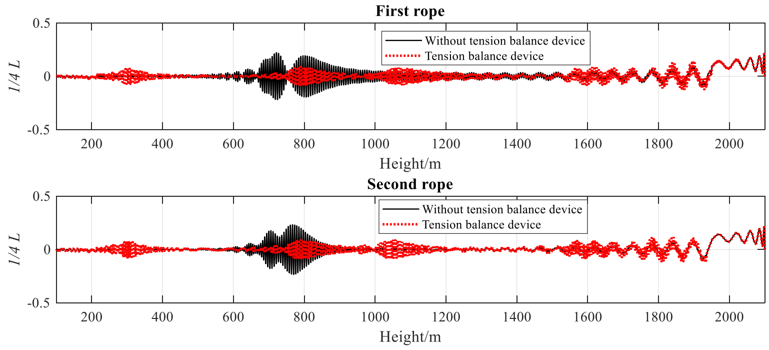

Figure 5: Vibration responses: $u_i(1/4l_i, t)$ with tension balance device or not.

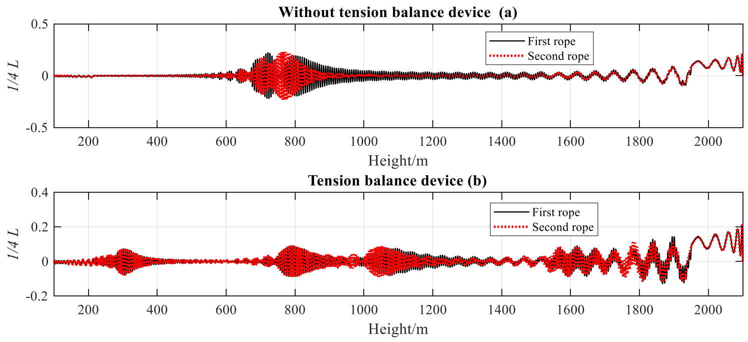

Figure 6: Displacements of two ropes at quarter length with respect to height.

The external excitation is applied at the top end of the first rope and the bottom constraint force acts on the second rope. From Fig. 6(a), the excitation applied to the first rope is indirectly transmitted to another rope through the hoisting conveyance. Since the ends of two ropes are connected to tension balance device directly, their responses are basically the same (Fig. 6(b)).

6 CONCLUSIONS

Vibration phenomena in long-travel lifting systems lead to poor ride quality. Eccentric pulleys and sheaves, systematic resonance in the electronic control system generated vibrations are typical causes of longitudinal vibrations of the lifting system. By modeling longitudinal coupled vibration of parallel hoisting systems with or without tension balance devices, it can be observed that the hoisting conveyance is subjected to rotational vibration while is excited by an eccentric pulley. The tension balance device can be adapted to mitigate the effects of vibrations and rope tension difference. It can avoid the system's rotational resonance and reduce the resonance amplitude reduced to about half. Other passive and active vibration suppression techniques can be utilized to control vibration phenomena.

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