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Vibration Suppression of Concrete Pump Boom During Pumping by Feedforward Control Method

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In this paper, boom of truck-mounted concrete pump is taken as the research object. Firstly, both mathematical and simulation models of the boom system are constructed, then vibration characteristics of the boom, especially vibration state of pump truck chassis are analyzed. According to the results of this theoretical analysis, a feedforward control method based on least mean square (LMS) algorithm and the finite impulse response (FIR) filtering algorithm is proposed to suppress the vibration of truck-mounted concrete pump boom which is mainly caused by the operation of pump system. After that, proposed feedforward control method simulation model is established; the effect on vibration suppression performance of it is analyzed. According to the simulation results conducted under three typical operation conditions, acceleration at the end of the boom decreases by 39.3% ~ 52.0% after the feedforward control force is applied. Therefore, it can be seen that the adaptive FIR-based feedforward vibration suppression algorithms designed in this paper can effectively suppress the vibration at the end of the boom.

Ключевые слова: *truck-mounted concrete pump boom; vibration suppression; least mean square (LMS) algorithm; finite impulse response (FIR) filtering algorithm; simulation*

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Introduction

When the boom is excited by the external environment to trigger its vibration, it needs to be controlled and intervened. This kind of research can be divided into two categories according to the different research objects for small booms and for large booms, that is, truck-mounted concrete pump booms.

Firstly, in the vibration suppression study of small arm under external excitation, Qiu, Z. used a combination of acceleration feedback and PPF control to suppress the unwanted vibration of a flexible manipulator due to environmental stimuli or caused by the torque of an AC servomotor during the hub slewing motion [1].Bian, Y. proposed a vibration absorbing method based on modal interaction to attenuate the nonlinear vibration of a flexible manipulator. The vibration absorber is made to establish a 1:1 internal resonance state with the flexible arm, so as to transfer the vibration energy from the flexible arm to the absorber to suppress the nonlinear vibration of the flexible arm based on modal interaction [2]; meanwhile, a vibration absorber based on the servomotor is designed with a 2:1 internal resonance relationship with the flexible arm [3]. Alexander, A. designed an active vibration control scheme using the boom position and boom hydraulic cylinder pressure or boom acceleration as feedback signals, which are computed to obtain the control output by a PD controller that modifies the proportional and differential coefficients according to the current operating conditions. The optimal gain coefficient values are obtained using an extreme value search algorithm. Finally, an objective function was determined using a time-domain signal (e.g., pressure or acceleration), and then different control strategies were compared and investigated [4].

Truck-mounted concrete pump boom will be affected by the vibration of the truck chassis due to the operation of the pumping system. Some researchers have studied the vibration suppression of the boom during the pumping operation. Zorn, S. realized the vibration suppression of the truck-mounted concrete pump boom by controlling the hydraulic cylinders of the last two arm sections with a compensating motion, and the compensating motion made the end operation point of the boom could be kept in its original position. The designed controller consists of three basic components: firstly, the measurement of vibration using acceleration sensors, then the calculation of the target force for vibration damping of the corresponding hydraulic cylinders of each boom section based on this information, and finally the force control signal to reach the target force [5]. Wu Zhiyong applies time series analysis based on the historical vibration data of truck-mounted concrete pump boom to predict its vibration attitude, so that the predicted vibration attitude of the boom is as close as possible to the real attitude, and then dynamically compensates for the nonlinear time delay of the boom system to provide a suitable reference trajectory for the active vibration suppression of the boom. Based on this reference trajectory, the hydraulic cylinders of each boom section are controlled to apply opposite forces on the boom to achieve vibration suppression of the boom [6, 7]. Rongsheng Liu developed an active control strategy using constant-position commandless input shaping technique to suppress vibration. Based on a set of independent modal equations obtained by using the modal method, a double-pulse control in the opposite direction is proposed, which not only suppresses vibration but also avoids the change of the equilibrium position of the boom system after the action of active control. In addition, the lag time that exists in the actual system is also taken into account to obtain a better control effect [8, 9]. Li Jiantao studied the modal frequency information of truck-mounted concrete pump boom in three typical attitudes during operation by comparing simulation and experiment, established a corresponding database, calculated the modal characteristic equations of the boom based on the database, and finally designed an active vibration suppression method based on the results of modal analysis [10]. Yi Huang conducted an experimental study on the vibration suppression of the boom system of a truck-mounted concrete pump using an active control strategy based on frequency domain parameter identification. Considering that the dynamic characteristics of the boom system will change with the change of the boom attitude, a global model of the boom system under active control is established and parameter identification is carried out based on the double-normalized least-mean method, and the acceleration signal at the end of the boom is selected as the feedback variable to optimize the active control force of the hydraulic cylinder [11].

1 Vibration Mechanism Analysis of Boom System

Pump truck chassis is the installation base of pump truck boom system, and it is also the most important source of boom vibration. Chassis is directly connected to the vehicle chassis, so the mechanical friction and impact caused by the bursting of diesel fuel in the diesel engine of the automobile chassis and the reciprocating motion of the crank mechanism will lead to a certain degree of vibration. Without connecting the hydraulic pumps of the pumping system and the boom system, the vibration signals of the pump truck's chassis are collected as shown in Fig.1.

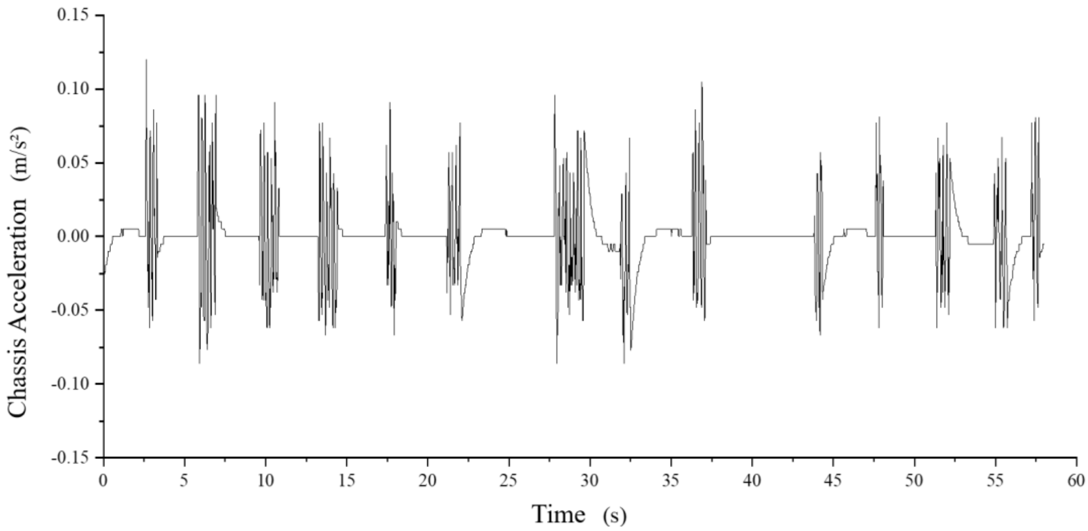


Fig. 1 – Vibration signals of the diesel engine acting on the chassis

It can be found that the vibration of the chassis under the excitation of the diesel engine alone is not significant. Therefore, the pumping system is the main source of vibration of the pump truck chassis. In the pumping system work process, the pumping hydraulic cylinder will carry out periodic reciprocating motion, the hydraulic cylinder pushes the concrete cylinder to realize a rapid continuous change of direction, and then from the hopper to inhale the concrete and to the conveying pipe pushed into the near-continuous concrete. At the end of every half pumping piston movement cycle, the pendulum valve and distribution valve complete the reversal, at this time, the pumping hydraulic circuit rapid reversal, the pressure inside the system reaches an instantaneous peak. During the pumping process, the vibration signal of the pump truck undercarriage near the pumping system is measured as shown in Fig.2.

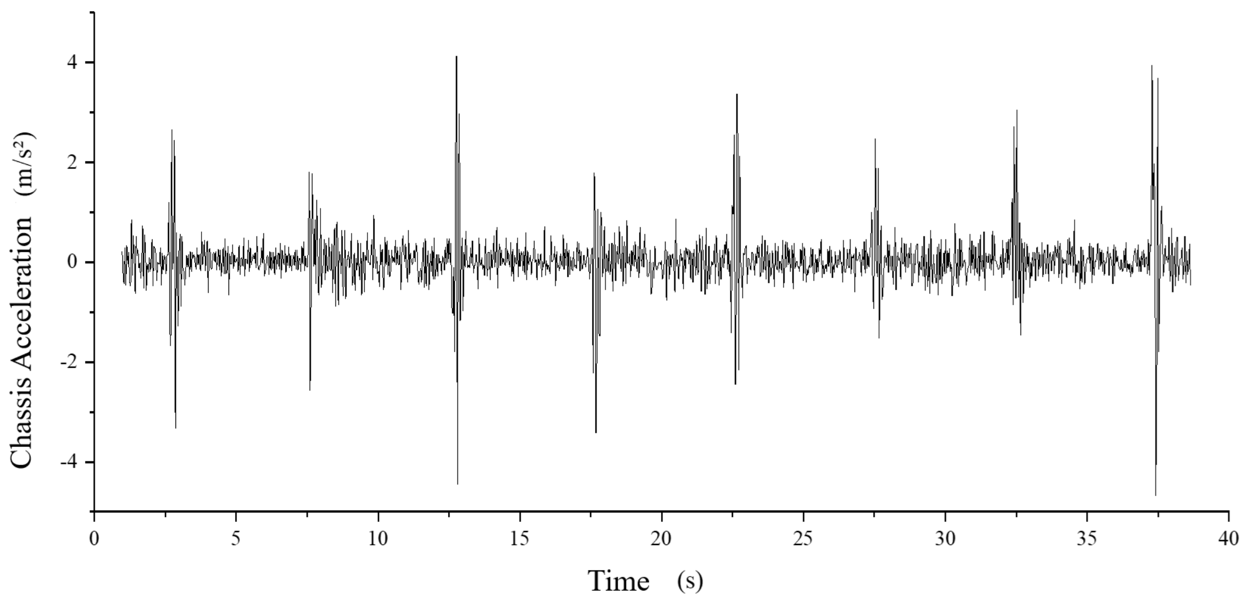


Fig. 2 Vibration excitation signal of pumping system on the pump truck chassis

It can be seen that the pumping excitation signal has an obvious periodicity. In order to get the overall excitation input of the pump truck bottom frame boom system, the vibration signal at the bottom of the boom is measured during the pumping operation, and the result is shown in Fig.3.

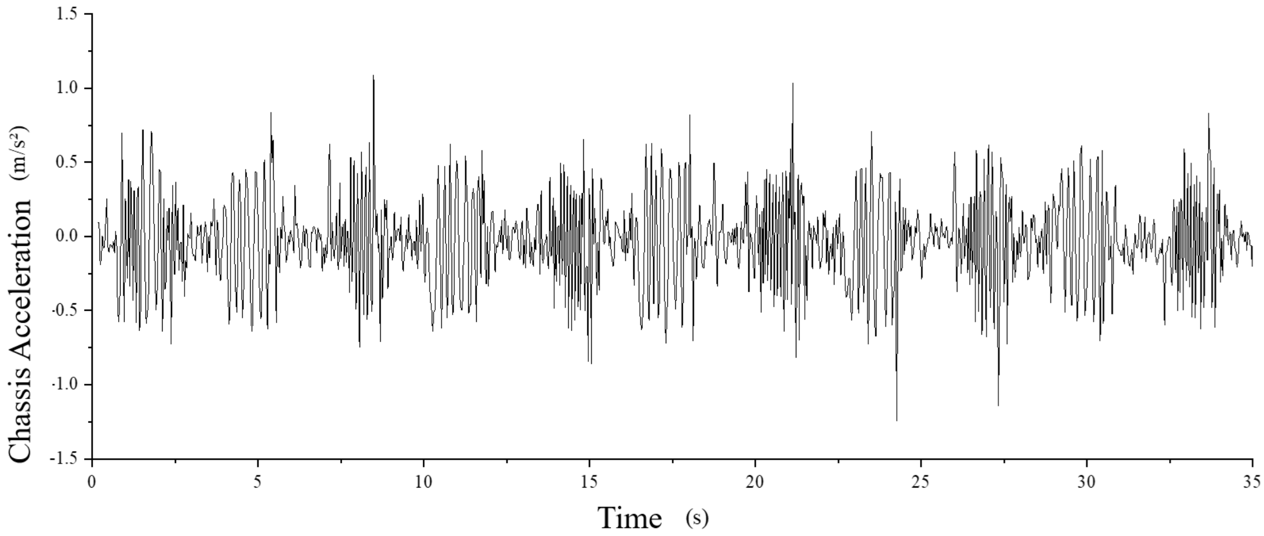


Fig. 3 Vibration signals at the bottom of the boom

The vibration signal in Fig.3 is analyzed by Fourier transform, and its frequency power spectrum is obtained as shown in Fig.4, which shows that the main action frequency of the excitation input signal of the boom system is 0.32Hz, which is same as the working frequency of the pumping system, and therefore it can be proved that the operation of the pumping system is the main excitation source of the bottom of the boom.

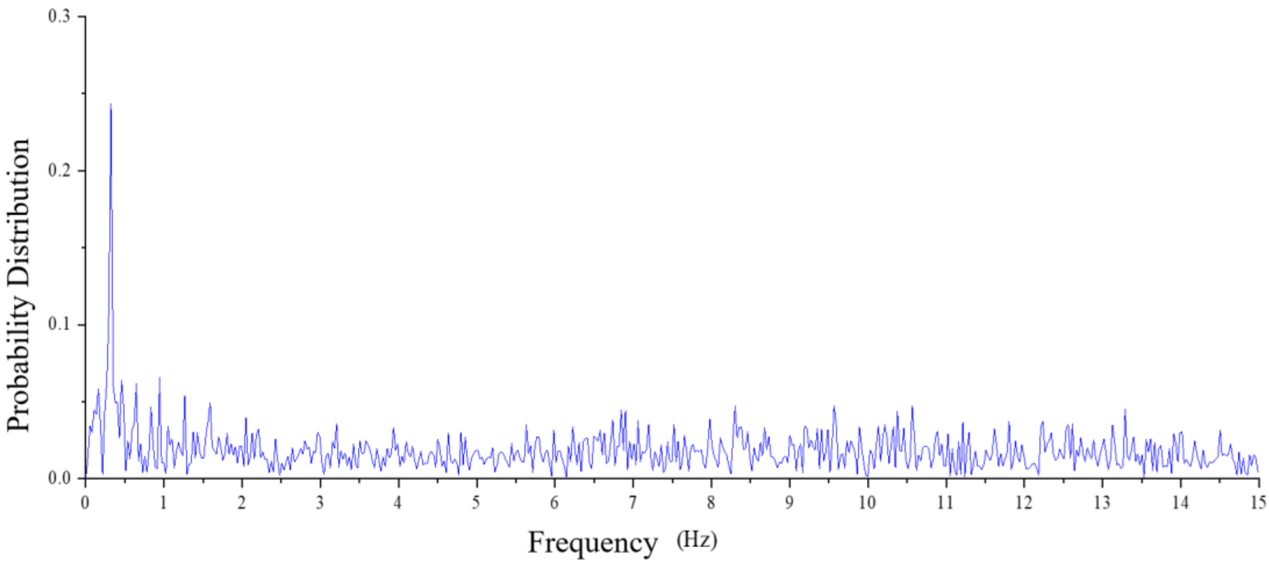


Fig. 4 Frequency spectrum of rotary table vibration signal

As shown in Fig.5, the mechanical structure of the boom system of a truck-mounted concrete pump mainly consists of six parts. These parts are connected through single-degree-of-freedom articulated rotary joints, denoted as θ^i (i=1...6), and are connected around the horizontal Z-axis of the local right-handed Cartesian coordinate system $\sum^i$, which is established based on the rotary table or the preceding boom section. The relative hydraulic cylinder displacement between the cylinder barrel and the connecting mechanism's rod is denoted as s^i (i=1... 6). All joint variables are combined into a vector called the drive variables $\boldsymbol{\theta}\in\mathbb{R}^7$. The X-axis of $\sum^i$ is parallel to the direction of the line connecting the articulation points of the boom sections.

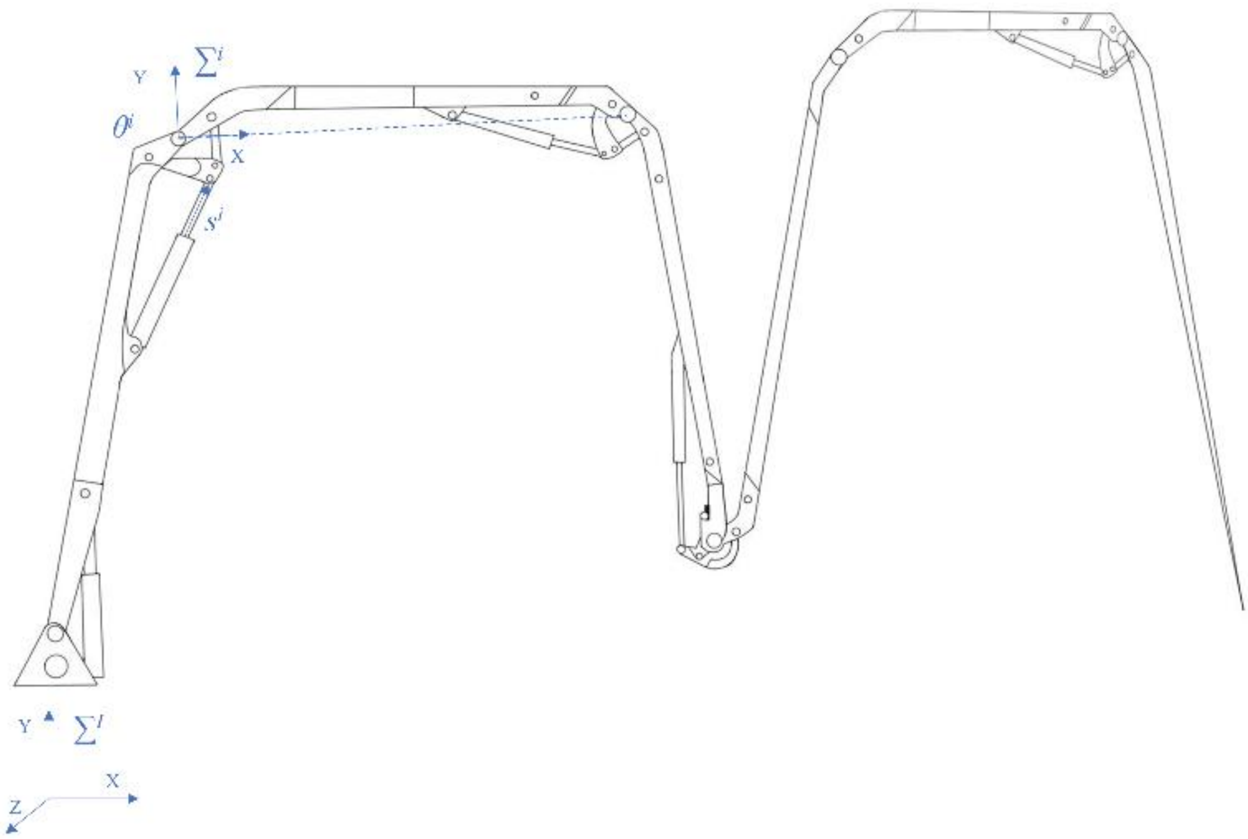


Fig. 5 Concrete pump boom structure schematic diagram

A twist virtual joint is added around the Z-axis of each local coordinate system on each boom section. Each boom section comprises a rigid body part, a virtual joint, and massless spring c_i and damper d_i (not shown in Fig.5). The angle of the virtual joint in the orthogonal coordinate system fixed to its respective boom section i is denoted as γ^i, as shown in Fig.6.

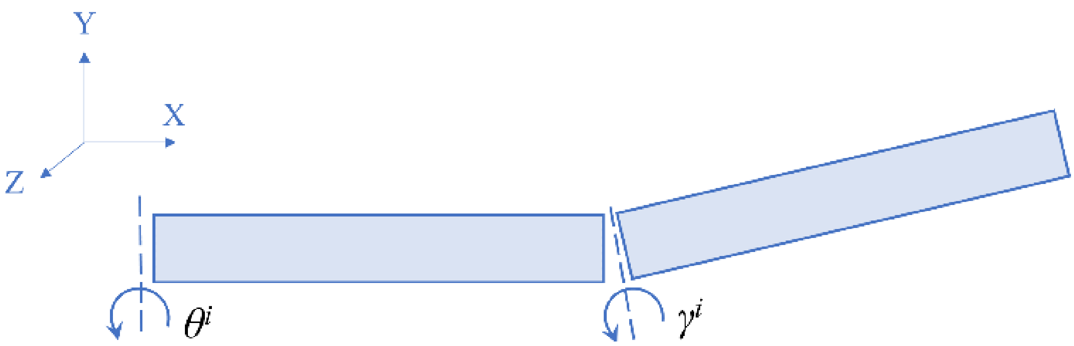


Fig. 6 Boom section rigid-flexible coupling model schematic

The angles of all virtual joints are combined into a vector defined as the elastic variables $\boldsymbol{\gamma}\in\mathbb{R}^6$. The drive variables and elastic variables are combined into generalized variables $\boldsymbol{q}=[\boldsymbol{\theta}^T\ \boldsymbol{\gamma}^T\ ]^T\in\mathbb{R}^{13}$.The forward kinematics formula for the local coordinate system $\sum^i$ is calculated as follows:

$$\begin{aligned} v^i &= J_v^i(q)\dot{q} \\ \omega^i &= J_\omega^i(q)\dot{q}, \quad i=\{0,...,6\} \end{aligned} \tag{1}$$

In this equation, v^i, $\omega^i\in\mathbb{R}^3$ represent the translational and rotational velocities of the local coordinate system $\sum^i$ of the i-th boom section relative to the inertial coordinate system $\sum^I$. The Jacobian matrices $\boldsymbol{J}_v^i$ and $\boldsymbol{J}_\omega^i$ correspond to the translational and rotational motions, respectively. To simplify the analysis, the boom system is assumed to be a rigid structure, which means $\boldsymbol{\gamma}=\mathbf{0}$. Thus, we have:

$$\begin{aligned} v^i &= J_{v,r}^i(\theta)\dot{\theta} \\ \omega^i &= J_{\omega,r}^i(\theta)\dot{\theta} \end{aligned} \tag{2}$$

In equation (2), solving for the velocities in the local coordinate system is simplified, but the structural flexibility of each boom section is neglected. When the flexible deformation components γ are added, the Jacobian matrix can be divided into rigid and elastic parts:

$$\begin{aligned} v^i &= J_{v,\theta}^i(q)\dot{\theta} + J_{v,\gamma}^i(q)\dot{\gamma} \\ \omega^i &= J_{\omega,\theta}^i(q)\dot{\theta} + J_{\omega,\gamma}^i(q)\dot{\gamma} \end{aligned} \tag{3}$$

In this equation, the Jacobian matrices $\boldsymbol{J}_{v,\theta}^i(q)$, $\boldsymbol{J}_{\omega,\theta}^i(q)$, $\boldsymbol{J}_{v,,\gamma}^i(q)$ and $\boldsymbol{J}_{\omega,\gamma}^i(q)$ describe the influence of the drive speed and the elastic speed $\dot{\gamma}$ on v^i and ω^i,, respectively. In this study, the primary focus is on the six-section boom system above the rotary table. Since the base rotation parameter θ^0 does not affect the static deformation in the direction of gravity, θ^0 can be excluded from the vector of drive coordinates $\boldsymbol{\theta}$. Consequently, the dimension of the generalized variables is reduced by one, resulting in $\boldsymbol{q}=[\boldsymbol{\theta}^T\ \boldsymbol{\gamma}^T\ ]^T\in\mathbb{R}^{12}$. Without θ^0, the rotational degree of freedom of the boom system is ignored, simplifying it to a planar structure. Additionally, by disregarding the impact of the connecting mechanism, the planar mechanical structure model of the concrete pump truck boom system can be described by the following formula:

$$M(\theta,\gamma)\begin{bmatrix}\ddot{\theta}\\ \ddot{\gamma}\end{bmatrix}+c\left(\theta,\dot{\theta},\gamma,\dot{\gamma}\right)+\begin{bmatrix}g_\theta(\theta,\gamma)\\ g_\gamma(\theta,\gamma)\end{bmatrix}+\begin{bmatrix}0\\ D\dot{\gamma}+K\gamma\end{bmatrix}=T_u\begin{bmatrix}u\\ 0\end{bmatrix}. \tag{4}$$

In this model, $\boldsymbol{M}\in\mathbb{R}^{12\times12}$ is the mass matrix, $\boldsymbol{c}\in\mathbb{R}^{12}$ is the vector of Coriolis and centrifugal forces, $\boldsymbol{g}\in\mathbb{R}^{12}$ is the gravity vector, which is divided into the drive variable block $\boldsymbol{g}_\theta(\theta,\gamma)\in\mathbb{R}^6$ and the elastic variable block $\boldsymbol{g}_\gamma(\theta,\gamma)\in\mathbb{R}^6$, $\boldsymbol{K}\in\mathbb{R}^{6\times6}$ and $\boldsymbol{D}\in\mathbb{R}^{6\times6}$ are the stiffness and damping matrices of the system, respectively, $\boldsymbol{T}_u^I$ represents the kinematic relationship between the vector of hydraulic cylinder lengths and the drive coordinate vector, $\boldsymbol{u}\in\mathbb{R}^6$ is the input vector of the boom system, composed of the output forces of the six hydraulic cylinders.

2 Feed-Forward Vibration Suppression Method

In this paper, a feed-forward compensated vibration suppression controller is designed and implemented based on digital filtering and adaptive filter theory. According to the working condition characteristics of the truck-mounted concrete pump boom, the adaptive FIR vibration suppression algorithm with easy implementation and significant performance is designed by combining the least mean square (LMS) algorithm and the finite impulse response (FIR) filtering algorithm. Typically, the FIR filter design process assumes the existence of a detection system for generating a signal related to the main disturbance. In this paper, the output signal of the acceleration sensor installed at the bottom of the boom is used as the reference signal related to the excitation of the boom, which can be filtered to generate the necessary control actions to eliminate the influence of the main excitation. A schematic diagram of the feedforward compensated vibration suppression control algorithm is shown in Fig.7.

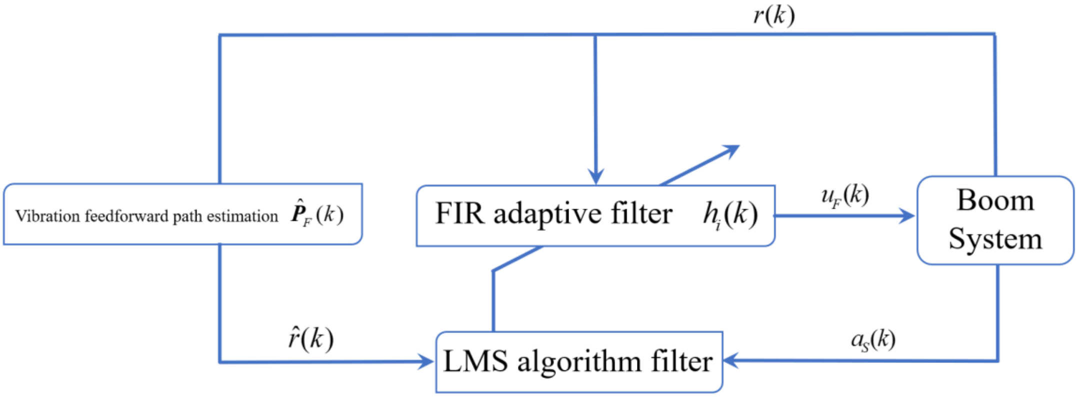


Fig. 7 Schematic diagram of the feedforward compensation vibration suppression control algorithm

The filtering process of the feed-forward compensation algorithm uses an adaptive FIR filter, which defines its i th order coefficients at the k th sampling time as $h_i(k)$, and the output of the filter, $u_F(k)$, is given by:

$$u_F(k) = \sum_{i=0}^{N-1} h_i(k) r(k-i), \tag{5}$$

where N is the order of the filter.

Before the output is measured by the accelerometer, the control signal needs to pass through a portion of the boom's physical system, defining that portion of the system's feedforward path as $\boldsymbol{P}_F(k)$. This path is also referred to as the offset path or error path. At this point, the vibration $a_F(k)$ of the boom system caused only by the feedforward control input can be obtained by the following equation:

$$a_F(k) = \sum_{j=0}^{M-1} g_j \sum_{i=0}^{N-1} h_i(k) r(k-i-j), \tag{6}$$

where g_j is the discrete impulse response of the control input to output path $\boldsymbol{P}_F(k)$, which is assumed to be of order M. At this point, the net output $a_S(k)$ of the system can be expressed as:

$$a_S(k) = a_u(k) + \sum_{j=0}^{M-1} g_j \sum_{i=0}^{N-1} h_i(k) r(k-i-j), \tag{7}$$

where $a_u(k)$ is the end-of-arm acceleration response due to the effect of the main disturbance (vibration at the bottom of the boom). Since the order of the convolutions can be interchanged without changing the calculation, Eq. (7) can be rewritten as:

$$a_S(k) = a_u(k) + \sum_{i=0}^{N-1} h_i(k)\bar{r}(k-i)\,, \tag{8}$$

where:

$$\bar{r}(k-i) = \sum_{j=0}^{M-1} g_j r(k-i-j)\,. \tag{9}$$

By rearranging the convolution, the signal $\bar{r}(k-i)$ is defined, which can be estimated by the filter-based test operation. The true impulse response of the vibration feedforward part of the boom system $\boldsymbol{P}_F(k)$ is estimated by the FIR filter, noting that the estimated value of $\bar{r}(k)$ is $\hat{r}(k)$.

In order to obtain the most appropriate filter coefficients $h_i(k)$, the coefficients need to be adjusted to minimize the cost function $J = E[a_S^2(k)]$, which is quadratically related to the output. For the LMS algorithm, the instantaneous value of $a_S^2(k)$ can be used as an estimate of the expected value of *J*. Therefore, a simple gradient descent algorithm can be used to ensure that the problem of minimizing the cost function converges to its global optimum, and thus the adaptive iterative algorithm for filter coefficients $h_i(k)$ can be written as:

$$h_i(k+1) = h_i(k) - \lambda \frac{\partial J}{\partial h_i(k)}\,, \tag{10}$$

where λ is the convergence factor. According to the definition of the cost function, the differentiation term in Eq. (10) can be changed to:

$$\frac{\partial J}{\partial h_i(k)} = 2a_S(k)\frac{\partial a_S(k)}{\partial h_i(k)}\,. \tag{11}$$

According to Eq. (8), the derivative of $a_S(k)$ with respect to $h_i(k)$ is $\bar{r}(k-i)$ using the estimated value of the filtered reference signal $\hat{r}(k-i)$; instead, the most rapid descent algorithm for the adaptive FIR controller coefficients given by Eq. (10) can be rewritten as:

$$h_i(k+1) = h_i(k) - 2\lambda a_S(k)\hat{r}(k-i)\,, \tag{12}$$

where λ determines the speed and stability of the adaptive process of the controller coefficients.

3 Simulation analysis

Based on the structural parameters of the pump truck boom system studied in this paper, a simplified two-dimensional dynamic model of the boom system was established in AMESim as shown in Fig.8. The model includes the planar model of the boom mechanical structure such as the arm joints, connecting mechanisms, and the hydraulic system model including the hydraulic pump, independent metering valves, hydraulic cylinders, and other auxiliary components, in which the friction at the joints of the boom is established by using the Stribeck friction model; and the damping coefficients are set to be low in order to simulate the flexible characteristics of the joints of the boom and the structure of the boom.

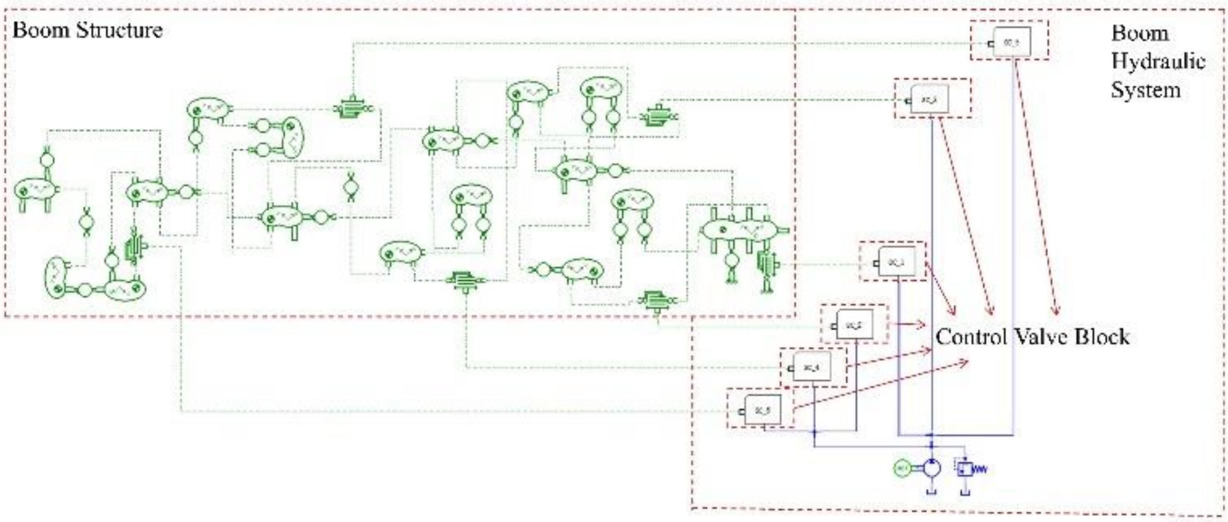
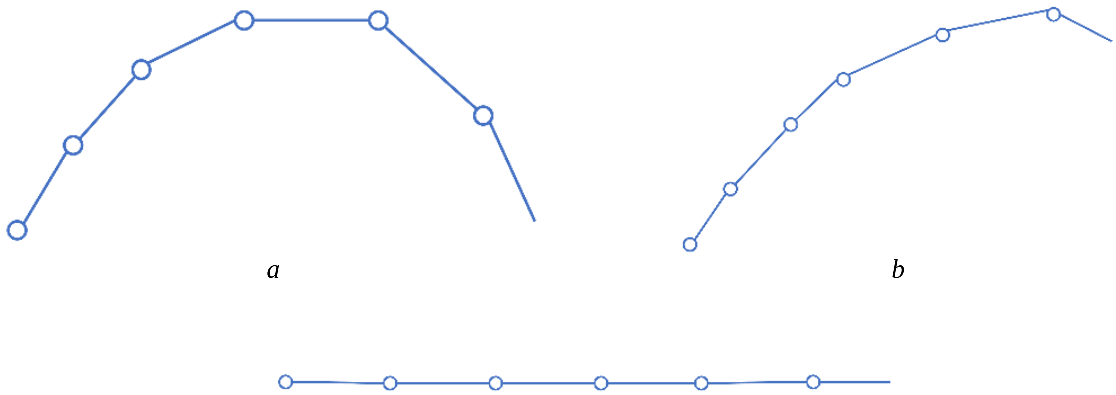


Fig. 8 Boom structure and hydraulic system parameter modeling

The 62-meters-long boom system of the truck-mounted concrete pump studied in this paper has six arm joints, so the boom system has six degrees of freedom in its operating plane, and the sufficiently high degrees of freedom make the end of the boom basically reach any point of the operating plane. In addition, when the operating point of the end of the boom is determined, the boom can be set to a variety of different attitudes. In the construction process, the low-end position arch attitude shown in Fig. 9(a) and the high-end position arch attitude shown in Fig. 9(b), as well as the horizontal linear attitude shown in Fig. 9(c), are usually used. According to the data collected by a company, 95% of truck-mounted concrete pump booms are in the three attitudes as described in Fig.9 during operation [10]. Therefore, in this paper, the end vibration suppression of the boom in these three attitudes during pumping of the pump truck is investigated.



a

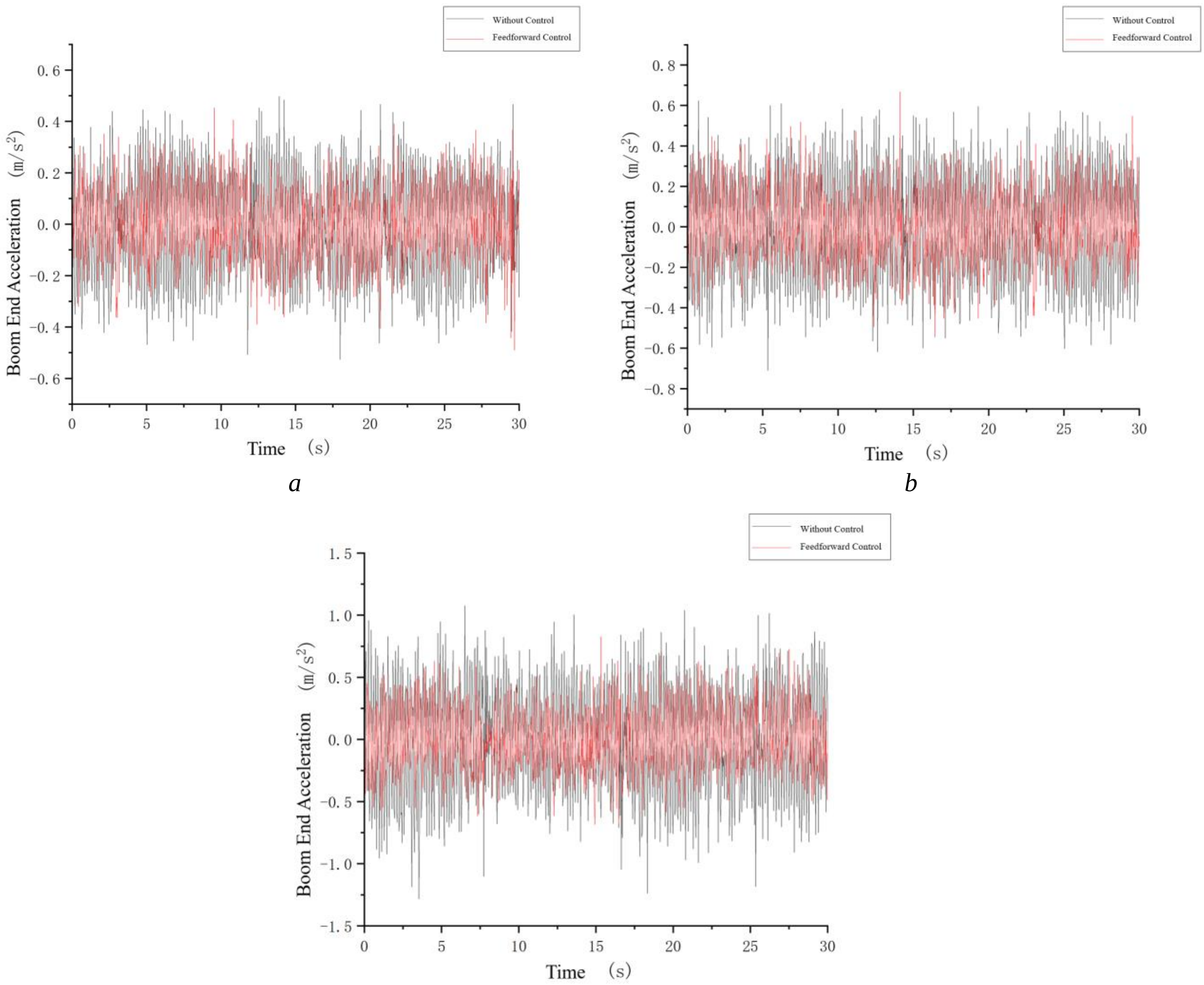
b

c

Fig. 9 The boom positions under consideration:

a – low-end position arch attitude; *b* – high-end position arch attitude; *c* – horizontal linear attitude

In order to validate the designed feedforward vibration suppression algorithm and determine the parameters of the control algorithm, the vibration suppression simulation test was carried out based on the AMESim-Simulink compound simulation model mentioned above. The results of the vibration amplitude comparison simulation test are shown in Fig.10.



c

Fig. 10 Simulation results:
a – low end position arch attitude; *b* – high end position arch attitude; *c* – horizontal linear attitude

At the beginning of the test, the size of the excitation force shown in Fig.3 is added to the bottom of the boom, so that the boom vibrates freely under the action of the external force, and the vibration signal at the end of the boom is recorded for 30s after the vibration of the boom is stabilized. Then the control input is obtained by using the first hydraulic cylinder based on the feed-forward vibration suppression algorithm designed in this paper. After the situation is stabilized, the vibration signal at the end of the boom is also recorded for 30s.

The above figure shows the simulation test results of the vibration amplitude change at the end of the truck-mounted concrete pump boom before and after applying the active feedforward during pumping operation in three attitudes. It can be noticed that the acceleration at the end of the sixth section of the boom decreases by 39.3%, 42.9% and 52.0%, respectively, after the feed-forward control force is applied.

Conclusion

It can be seen that the FIR-based feedforward vibration suppression algorithms designed in this paper can effectively suppress the vibration at the end of the boom. This is because the feed-forward vibration suppression algorithm is designed based on the vibration at the bottom of the boom, which directly corresponds to the external disturbance shock, and relies on the first hydraulic cylinder to apply the reverse vibration compensation to the boom, which can theoretically completely suppress the vibration of the pump truck boom. But due to the complex structure of the boom the vibration suppression effect cannot reach 100%.

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