Analysis Research on Operation Condition of Mining Crane

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Abstract. Mining cranes play an important role in the transportation and installation of underground equipment. Operating swing angles will seriously affect the operations safety. To ensure the operation safety of mining cranes, this paper firstly established a mathematical model of mining crane hoisting swing, and discussed affecting factors of the oscillation. Then, the orthogonal test method has been carried out, and the optimal and serious working conditions were obtained. Finally, the accuracy of the numerical results have been cross-verified on the ADAMS virtual platform. Results show that the swing trend increases with the increase of pitch acceleration, rotary acceleration and rope length, especially the rotary angular acceleration. The vibration variation trends of Matlab numerical results and ADAMS simulation results are consistent, which cross-verifies the two extreme operation conditions are valid and reliable. It provides a useful reference for the future research.

1. Introduction

In recent years, with the rapid development of mining technology, the development and application of mining machinery have played an important role in the transfer, assembly and disassembly of underground equipment. However, due to the movement characteristics of the mining crane, the heavy objects will inevitably swing during pitching and slewing movements. If the swing angle is too large, it will reduce the efficiency and safety of underground production.

Therefore many researchers have tackled the wobble problem of operation. Huili REN deduced the control equation that coupled the dynamic response of suspended working platform and load swing, and determined the key parameters significance and revealed how to adjust the system design to avoid critical conditions [1]. Y. Sakawa studied the motion equations of crane and load swing with new calculation methods to control crane swing [2-3]. Others optimize the control system and controller. For example, Feng Liu and M.I. Solihin, et al. researched the anti-swing control strategy of portal crane with experiments. The crane working efficiency could be greatly improved in the field test by optimizing the control coefficient [4-5]. B.H. Lin designed a PID controller to control the slewing and amplitude movement of the crane, and adjusted the controller parameters by trial and error method [6]. Maghsoudi, et al. controlled fast and accurate trolley positioning and minimum payload swing [7]. Sung Kun et al. used fuzzy logic control to suppress load swing. The control system is controlled by position servo control and fuzzy logic control are composed of two parts [8]. Takagi K et al.
introduced a design scheme of centralized control system that coupled the lifting (boom luffing) direction and the rotation direction, and achieved two independent controls successfully [9]. Henry RJ and Yu Zhu et al. proposed a control system with delayed position feedback, and simulated the controlled system by fully nonlinear motion equations [10-11]. Zheng Tian et al. designed and analysed a partially enhanced coupled nonlinear controller with the Lyapunov Technology and LaSalle’s invariance theorem [12]. Smoczek J et al. proposed a data-driven time-to-failure prediction model based on the fuzzy logic and an adaptive crane control system designing evolutionary algorithm [13]. Qinghua Chen et al. analysed of the working conditions of the luffing mechanism, and the dynamic simulation analysis was carried out based on the equivalent element method and ADAMS simulation software [14].

Although the above researches had a certain effect on the control effect of lifting swing, there is still a certain distance from the actual operation control. Based on the actual operating conditions, this paper establishes the dynamic model of the mining crane and discusses swing influencing factor, and then performs an orthogonal test. Finally, the result accuracy was cross verified by the ADAMS.

2. Mathematical Model of Crane Hoisting Swing

In order to facilitate the analysis of the characteristics of the hoisting swing of the mining crane, the following simplifications are made: Considering hoisting weight as mass block \( m \). The mass of the hook and rope is negligible relative to the hoisting weight. The elastic deformation of the wire rope and the boom is ignored. The crane structure is shown in Figure 1. These symbols represent heavy object, wire rope, lifting arm, rotating table, lower cover and luffing hydraulic in ascending order. The physical model of the mining crane is shown in Figure 2. The crane parameters are shown in Table 1.

![Figure 1. Mining crane structure model](image1)

![Figure 2. Physical model of the mining crane](image2)

| Symbol | Meaning | Symbol | Meaning | Symbol | Meaning |
|--------|---------|--------|---------|--------|---------|
| \( \theta \) (rad) | Swing angle | \( \alpha \) (rad) | Rotation angle | \( L_b \) (m) | Boom length |
| \( \theta_1 \) (rad) | Swing angle in the ZOQ plane | \( \beta \) (rad) | Pitch angle | \( m \) (kg) | Mass block |
| \( \theta_2 \) (rad) | Swing angle outside the ZOQ plane | \( L \) (m) | Wire rope length |

According to the Lagrangian dynamics equation, a dynamic model of the mass block \( m \) is established for the angle \( \theta_1 \) and \( \theta_2 \). The kinetic energy of the mass block \( m \) is:

\[
T = \frac{1}{2} m \left( x_m^2 + y_m^2 + z_m^2 \right) \tag{1}
\]

\( x_m, y_m, z_m \) are mass block coordinates in the Oxyz coordinate system.

The potential energy of the mass block \( m \) is:

\[
V = mg(L_0 \sin \beta - L \cos \theta_1 \cos \theta_2) \tag{2}
\]

In addition, the system Lagrangian function is shown in formula (3), where \( L \) is the Lagrangian operator with \( L=T-V \):

\[
\begin{align*}
\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{\theta}_1} \right) - \frac{\partial L}{\partial \theta_1} &= 0 \\
\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{\theta}_2} \right) - \frac{\partial L}{\partial \theta_2} &= 0
\end{align*}
\tag{3}
\]
The dynamic equation of the hoisting system can be obtained by bringing in the various parameters:

\[ \ddot{\theta}_1 = \left( \ddot{\alpha}^2 + \frac{L_1 \beta (\cos \beta + \sin \beta)}{L} \right) - \frac{g}{L} \theta_1 - \frac{2L_1}{L} \dot{\theta}_1 \dot{\theta}_2 + \frac{L_1 \sin \beta (\dot{\alpha}^2 + \dot{\beta}^2)}{L} \right) \frac{L}{L} \beta \cos \beta - \mu_1 \theta_1 \]  

(4)

\[ \ddot{\theta}_2 = \left( \ddot{\alpha}^2 + \frac{L_2 \beta (\cos \beta + \sin \beta)}{L} \right) - \frac{g}{L} \theta_1 - \frac{2L_2}{L} \dot{\theta}_1 \dot{\theta}_2 - \frac{L_2 \dot{\alpha}^2 \sin \beta}{L} - \mu_2 \theta_2 \]  

(5)

As shown in equations (4) and (5), when the hoisting machine hoists heavy objects, not only will there be friction between the steel cable and the pulley, but there will also be air damping, and the damping item is added \( \mu_1 \cdot \dot{\theta}_1 \) and \( \mu_2 \cdot \dot{\theta}_2 \), \( \mu_1 \) and \( \mu_2 \) are the comprehensive damping coefficients.

3. Factor simulation of Crane Hoisting Swing

To achieve the influencing factors, the single pitch motion and single rotary motion under different working conditions and the pitch and rotary motion with different rope lengths are simulated.

3.1. Single pitch motion

Firstly, the single pitch process is acceleration-uniform velocity–deceleration motion. The rope length of \( L = 1.3 \text{m} \), boom length of \( L_B = 3.5 \text{m} \) and pitching angle of \( \beta = 0 \text{–} 0.785 \text{rad} \). The compilation function was input into Matlab to compare and analyse the swing characteristics. The input parameters of the first and second working conditions were shown in Table 2.

| Movement stage       | Pitch angular acceleration of \( \ddot{\theta} \) (rad/s²) | Rotation angular acceleration of \( \ddot{\alpha} \) (rad/s²) |
|----------------------|----------------------------------------------------------|----------------------------------------------------------|
| Time \( t(s) \)      | First condition                                          | Second condition                                         |
| Acceleration motion  | 0–4                                                      | 0.013                                                    | 0.017                                                    |
| Uniform motion       | 4–22.5                                                   | 0                                                       | 0                                                        |
| Deceleration motion  | 22.5–26.5                                                | -0.013                                                  | -0.017                                                  |
| Free oscillation     | 26.5–60                                                  | 0                                                       | 0                                                        |

The results of the first and second conditions are shown in Figure 3 and Figure 4. In the single pitch motion, the angle \( \theta_1 \) will be generated. When the pitch motion is over, the lifting system will gradually converge with reciprocating motion due to damping. There is no effect on the angle \( \theta_2 \). The greater the pitch angular velocity, the greater the angle \( \theta_1 \), that is, the more severe swing.

![Figure 3. The angle \( \theta_1 \) results of two conditions](image1)

![Figure 4. The angle \( \theta_2 \) results of two conditions](image2)
3.2. Single rotary motion

Then, the single rotation process is also acceleration-uniform velocity-deceleration motion. The rope length of $L=1.3$ m, boom length of $L_B=3.5$ m, rotation angle of $\alpha=0$–$3.489$ rad. The input parameters of the third and fourth working conditions were shown in Table 2.

The results of the third and fourth operation conditions are shown in Figure 5 and Figure 6. The swing angle $\theta_1$ and $\theta_2$ will be generated simultaneously, and the lifting weight makes sinusoidal motion with amplitude attenuation both inside and outside the lifting arm plane. The faster the rotary speed, the greater the swing amplitude. Obviously, the amplitude of $\theta_2$ is greater than that of the $\theta_1$.

3.3. Pitch and rotary compound motion with different rope lengths

The system carries out pitching and rotary motion continuously, and sets the arm length of $L_B=3.5$ m, pitch angle of $\beta=0$–$0.785$ rad and return angle of $\alpha=0$–$3.489$ rad. The lifting motion is analysed under three different rope length $L$. The rope length $L$ of working condition of the 5th, 6th and 7th is 0.95 m, 1.3 m and 1.9 m respectively in Table 3.

| Movement stage          | Time (s) | Pitch angular acceleration $\beta$ (rad/s²) | Rotation angular acceleration $\alpha$ (rad/s²) |
|-------------------------|----------|---------------------------------------------|-----------------------------------------------|
| Pitch acceleration motion | 0–4      | 0.0087                                      | 0                                             |
| Pitch uniform motion     | 4–22.5   | 0                                           | 0                                             |
| Pitch deceleration motion | 22.5–26.5 | -0.0087                                     | 0                                             |
| Rotary acceleration motion | 26.5–29.5 | 0                                           | 0.087                                         |
| Rotary uniform motion    | 29.5–39.5 | 0                                           | 0                                             |
| Rotary deceleration motion | 39.5–42.5 | 0                                           | -0.087                                        |
| Free oscillation motion  | 42.5–60  | 0                                           | 0                                             |
Simulation results are shown in Figure 7 and Figure 8. In pitch and rotary motion, the swing angle $\theta_1$ and $\theta_2$ increase with rope length. After the rotation motion, the lifting system makes sinusoidal motion with amplitude attenuation, and the whole system presents a cone pendulum motion. Under the same working conditions, the amplitude of $\theta_2$ is greater than that of $\theta_1$.

The analysis shows that the three important factors affecting the swing of the lifting weight are the pitch angle acceleration of $\dot{\beta}$, rotation angle acceleration of $\dot{\alpha}$ and rope length of $L$. Traditional analysis methods need to analyse lots of working conditions so that the workload is heavy. To find the optimal and serious conditions quickly, the orthogonal experiment method can solve this problem effectively.

4. Orthogonal test analysis

Orthogonal experimental design is an experimental design method that uses a set of standardized orthogonal tables to study multiple factors and multiple levels. The experimental plan can be designed according to the design principles and the experimental results can be analysed. The minimum number of experiments can be carried out. At the same time, the multi-factors are investigated and studied, and then the most reasonable and optimal test results can be obtained. Therefore, it is possible to carry out orthogonal experimental design for mining cranes to optimize the design of working conditions. By analysing the amplitude of $\theta_1$ and $\theta_2$, the optimal and serious operating parameters of the mining crane can be obtained.

4.1. Orthogonal design

Before carrying out the orthogonal test design of mining crane, it is necessary to determine the number of factors and level of this orthogonal test. The three factors of pitch angular acceleration, rotational angular acceleration and hoisting rope length greatly affect the swing angle during the operation of the crane, so these three factors are selected as test factors. And according to the actual operating parameters of the mining crane, three levels are selected for research. The specific values are shown in Table 4 below.

| Table 4. Crane operation swing factor level table |
|-----------------------------------------------|
| Level | Factor A | Factor B | Factor C |
|       | Angular speed of pitch (rad/s) | Angular speed of rotation (rad/s) | Lifting rope length (m) |
| 1     | 0.017    | 0.105    | 1.5      |
| 2     | 0.026    | 0.140    | 2.0      |
| 3     | 0.035    | 0.175    | 2.5      |

4.2. Numerical results

This test is a three-factor and three-level test, so the orthogonal table $L_9$ is used to design the test. By processing the dynamic model of the hoisting system, the mathematical form of the model and each...
set of test parameters are written into a form that can be read by the Matlab/Simulink simulation platform, so that the simulation model established by the Matlab/Simulink platform to test each group of experiments. The simulation solution is carried out, and the amplitudes of the swing angle $\theta_1$ and $\theta_2$ of the hoisting system can be obtained. The simulation results are shown in Table 5. It can be directly seen that the minimum values of angle $\theta_1$ and angle $\theta_2$ are 0.011 rad and 0.058 rad whose the amplitude $A_1$ and $A_2$ are 27.532 mm and 146.132 mm. Meanwhile, the maximum values of angle $\theta_1$ and angle $\theta_2$ are 0.026 rad and 0.111 rad, respectively, and the amplitude $A_1$ and $A_2$ are 39.684 mm and 165.408 mm.

| Test number | Factor composition ABC | $\theta_1$ (rad) | $\theta_2$ (rad) | Amplitude $A_1$(mm) | Amplitude $A_2$(mm) |
|-------------|------------------------|-----------------|-----------------|--------------------|--------------------|
| 1           | A1B1C1                 | 0.012           | 0.068           | 17.828             | 101.396            |
| 2           | A1B2C2                 | 0.017           | 0.094           | 34.716             | 187.383            |
| 3           | A1B3C3                 | 0.024           | 0.098           | 60.252             | 244.478            |
| 4           | A2B1C2                 | 0.012           | 0.060           | 23.108             | 120.634            |
| 5           | A2B2C3                 | 0.018           | 0.094           | 44.198             | 234.489            |
| 6           | A2B3C1                 | 0.026           | 0.111           | 39.684             | 165.408            |
| 7           | A3B1C3                 | 0.011           | 0.058           | 27.532             | 146.132            |
| 8           | A3B2C1                 | 0.019           | 0.111           | 28.77              | 166.631            |
| 9           | A3B3C2                 | 0.026           | 0.101           | 51.97              | 200.793            |

5. Result analysis and verification

5.1. Analysis results

The test result analysis is shown in Table 6. In the test range, if the range of a certain column is the largest, the factor will mostly change the result of the objective function. Therefore, the effect order for angle $\theta_1$ is rotation angular velocity, pitch angular velocity and hoisting rope length. Similarly, that of $\theta_2$ is rotation angular velocity, hoisting rope length and pitch angular velocity.

| The sum of the level of each factor 1 | Factor A | Factor B | Factor C | Factor A | Factor B | Factor C |
|---------------------------------------|----------|----------|----------|----------|----------|----------|
| The sum of the level of each factor 2 | 0.055    | 0.034    | 0.058    | 0.259    | 0.186    | 0.289    |
| The sum of the level of each factor 3 | 0.056    | 0.077    | 0.053    | 0.270    | 0.309    | 0.250    |
| Average value of each factor 1 level | 0.018    | 0.011    | 0.019    | 0.086    | 0.062    | 0.096    |
| Average value of each factor 2 level | 0.019    | 0.018    | 0.018    | 0.089    | 0.100    | 0.085    |
| Average value of each factor 3 level | 0.019    | 0.026    | 0.018    | 0.090    | 0.103    | 0.083    |
| Range R                               | 0.001    | 0.014    | 0.001    | 0.004    | 0.041    | 0.013    |

Factor primary → secondary B C A

To ensure the accuracy of the swing orthogonal test of the mining crane and make the mining hydraulic crane run more safely and reducing the impact of vibration swing. Two extreme working conditions of A3B1C3 and A2B3C1 have been cross verified with ADAMS, respectively. The specific parameters and results are shown in Table 7.

| Working condition | Angular speed of pitch (rad/s) | Angular speed of rotation (rad/s) | Lifting rope length (m) | 01 (rad) | 02 (rad) |
|-------------------|--------------------------------|----------------------------------|-------------------------|----------|----------|
| A3B1C3            | 0.035                          | 0.105                            | 2.5                     | 0.011    | 0.058    |
| A2B3C1            | 0.026                          | 0.175                            | 1.5                     | 0.026    | 0.111    |

The A3B1C3 and A2B3C1 condition is that pitch angular velocity of 0.035 rad/s and 0.026 rad/s, rotation angular velocity of 0.105 rad/s and 0.175 rad/s and hoisting rope length of 2.5 m and 1.5 m. This test will provide reference for the anti-pendulum control in the future.
5.2. Cross verification
For verifying the correctness of Matlab numerical results obtained by orthogonal experimental, it is necessary to do comparison cross verification. Specifically, the spatial motion trajectory of crane boom and pitching and rotary angles of each rotary joint were programmed by the Step function in ADAMS. The multi-body kinematics simulation analysis was carried out. The results are shown in Figure 9 and Figure 10. The red square line is angle $\theta_1$ and the black asterisk line is angle $\theta_2$. The variation trend of the curves is consistent with the calculated results of Matlab.

![Figure 9. Results of the A3B1C3 condition](image1)

![Figure 10. Results of the A2B3C1 condition](image2)

The results of optimal operation condition A3B1C3 are shown in Table 8. The angle range of $\theta_1$ and $\theta_2$ are 0-0.010rad and 0-0.057rad, respectively. For the serious operation condition A2B3C1, that of are 0-0.019rad and 0-0.113rad, respectively. The error of $\theta_1$ and $\theta_2$ are 3.64% and 1.72% under the optimal condition A3B1C3 and that of are 3.84% and 1.80% under the serious condition A2B3C1. It's a perfectly acceptable error.

| Working condition | $\theta_1$(rad) | Simulation result | Error (%) | Numerical result | $\theta_2$(rad) | Simulation result | Error (%) |
|-------------------|----------------|------------------|-----------|-----------------|----------------|------------------|-----------|
| A3B1C3            | 0.011          | 0.0106           | 3.64      | 0.058           | 0.057          | 1.72             |
| A2B3C1            | 0.026          | 0.025            | 3.84      | 0.111           | 0.113          | 1.80             |

6. Conclusion
Through establishing the mathematical model of crane hoisting system, discussing three factors affecting the oscillation. While the operating conditions of mining crane were optimized by orthogonal experimental design method, and the optimal and serious working conditions were obtained. The accuracy of the Matlab numerical results were cross-verified by simulation results of ADAMS.

The conclusions are as follows: (1) Three factors of pitch angular acceleration, rotational angular acceleration and hoisting rope length affect the oscillation. (2) The swing trend increases with the increase of the three factors, especially the rotary angular acceleration. (3) The optimal operating parameters of the crane are that the pitch angular velocity is 0.035 rad/s, the rotation angular velocity is 0.105 rad/s, and the hoisting rope is 2.5 m. (4) The serious operating parameters of the crane are that the pitch angular velocity is 0.026 rad/s, the rotation angular velocity is 0.175 rad/s, and the hoisting rope is 1.5 m. (5) By comparing the numerical results and simulation results, the vibration variation trends are consistent, which verifies the accuracy of the results. The optimization design has certain guiding significance to improve the safety and efficiency of underground operation.
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