Research on system identification method of buffer system based on experimental signal pre-processing

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Abstract: In order to obtain the theoretical model of a gun buffer system, the dynamic vibration characteristics of the buffer system were studied by force hammer experiments, and the vibration acceleration response curves of the buffer system after being excited by the impact load were obtained. The acquired signals were processed by noise reduction, detrending, data smoothing and frequency domain integration, and the second-order free vibration model was used to identify the stiffness, damping and other parameters of the system, and compared with the experimental results. The deviation between the calculated results of the identification model and the experimental results is less than 4%, which compared with the general direct identification method, the error is reduced by about 10%. This method can provide a simple and fast technical way for the accurate identification of vibration system models.

1. Introduction
The buffer system for a gun is subjected to a strong recoil impact load in the actual working process. The stability and reliability of the buffer system are directly related to the service life of the gun's supporting equipment and the safety of the users. System identification is of great significance in this application background. The mathematical model of the system can be obtained through system identification, which can theoretically explore the motion law of the buffer system, evaluate the dynamic characteristics of the buffer system, avoid resonance, predict the buffer results and evaluate the buffer effect, and provide guidance for the optimal design of the buffer system[1-3]. Traditionally, the system identification methods include recursive least squares method[4,5], ordinary differential equation method[6], stochastic approximation algorithm[6,7] and so on. 

For this buffer system, the input is a strong impact load with a very short action time of about a few milliseconds and a very large impact peak of about tens of thousands of Newtons. Therefore, in the process of system identification, the accuracy of the input data is very high, and the data collected directly cannot be directly used for system identification. Therefore, a series of processing is needed for the collected data to meet the conditions of system identification. System identification methods based on the idea of output signal preprocessing are not uncommon[8], and typical ones include deviation compensation-based system modeling methods[9-12], open-loop and closed-loop dynamic system identification[13], and reduced-order modeling[14], etc, which also have a large number of successful applications. In this paper, we propose a simple and fast identification method for the buffer system under strong impact load excitation, through which the second-order vibration system can be identified.
and the accuracy of the identification results can be greatly improved.

2. Methods
The overall composition of the buffer system mainly includes the mounting seat of the buffered device, the buffer guide shaft, detachable spring and polyurethane. According to different working conditions, springs with different wire diameters and polyurethanes with different hardness can be replaced to form different buffer systems. The schematic diagram of the buffer system is shown in Figure 1.

![Figure 1 A buffer system for guns](image)

2.1 Experiment
Through the force hammer experiment, the vibration response of the buffer system under the action of the impact force can be obtained, which can simulate the actual use environment of the buffer system in a more realistic way. Specifically, a force hammer is used to strike the simulated mass of the buffer system to give the mass an initial impact force. The sensor on the force hammer can perform a force test, and the acceleration sensor can measure the curve of the acceleration of the mass with time. The experimental schematic diagram and experimental setup are shown in Figure 2 and Figure 3.

![Figure 2 Experimental schematic](image)

![Figure 3 Experimental setup](image)

In the experiment, the PCB 086D020 force hammer is used as the signal generator, and the black hard hammer head is used for the hammer head. The hard hammer head has the shortest action time, about 1~2ms, which can produce similar impact and force as in the actual use process. The force sensor sensitivity coefficient on the hammer is 0.2029mv/N. The acceleration response of the mass is measured with a PCB 356A02 three-way accelerometer, its range is 500g, and the sensitivity of the measured buffer direction is 9.63mV/g. In the experiment, the NI USB-4431 data acquisition is used to collect signals, which is used for the acquisition of force and accelerometer signals, and combined with the Labview program, the signals can be directly collected to the computer.

Through the hammer experiment, two sets of time-varying data can be measured, which are the force given to the system by the force hammer and the vibration acceleration response of the system after being subjected to the impact effect of the force hammer, both of which are the input and output of the system, respectively.

2.2 Input Data Processing
The force signal obtained by the actual test of the force hammer experiment is not an ideal rectangular impulse, but a signal in the form of a rapidly increasing and then rapidly decreasing approximate impulse, which is processed and equated to an ideal rectangular impulse force signal in order to improve the reliability of the system identification[^15]. Through the trapezoidal integration method, the input force
signal waveform is integrated to obtain the system impulse variation \( \Delta p \). The value of the impulse force is taken to be the maximum value \( F_{max} \) measured during the action of this force, so that the action time of the impulse force can be approximated as

\[
\Delta t = \frac{\Delta p}{F_{max}}
\]  

(1)

The input signal before and after processing of a set of experimental data are shown in Figure 4 and Figure 5, which are the force signal obtained by direct measurement and the equivalent impulse force signal, both of which have equal input impulse \( \Delta p \) to the system. It is worth noting that the duration of the whole force signal is related to the duration of the acceleration signal stabilization to zero, which can be appropriately selected according to the duration of the acceleration signal.

![Figure 4 The force signal obtained from the test](image1)

![Figure 5 Processed force signal](image2)

2.3 Input Data Processing

Due to the existence of system damping, theoretically the collected acceleration signal is the curve of oscillation decaying continuously with time, but because of the existence of errors such as noise interference, the collected acceleration data curve has more clutter, and because the impact response is very fast, the collected signal error is large, so the acceleration data is processed to some extent.

2.3.1 Detrending and smoothing

The data collected by the accelerometer may have the phenomenon of "zero-point drift", and the curve calculated by integration or quadratic integration will have a large cumulative error, resulting in a speed or displacement that increases or decreases oscillation. The trend item directly affects the correctness of the signal. It is necessary to remove the trend item of the acceleration curve, and use the polynomial least squares method to remove the trend item of the acceleration signal curve.

The vibration acceleration signal data obtained from the experiment is \( \{a_k\} (k=1,2,3,...,n) \), and the estimation function of the signal sequence is set as

\[
\hat{a}_k = b_0 + b_1 k + b_2 k^2 + \cdots + b_m k^m
\]  

(2)

In the formula, \( m \) is the order of the equation, and the sum of squares of the errors of \( \hat{a}_k \) and \( a_k \) is

\[
E = \sum_{k=1}^{n} (\hat{x}_k - x_k)^2 = \sum_{k=1}^{n} (\sum_{i=0}^{m} b_i k^i - x_k)^2
\]  

(3)

Then the condition for the existence of extreme value of \( E \) is

\[
\frac{\partial E}{\partial b_j} = 2 \sum_{k=1}^{n} k^j (\sum_{i=0}^{m} b_i k^i - x_k) = 0 \quad (j = 0,1,2,\cdots,m)
\]  

(4)

After the partial derivatives of the above formula are obtained in turn, a linear equation system is obtained. When \( m = 1 \), it is a linear trend term, as follows:

\[
\begin{align*}
\sum_{k=1}^{n} b_0 + \sum_{k=1}^{n} b_1 k - \sum_{k=1}^{n} a_k &= 0 \\
\sum_{k=1}^{n} b_0 k + \sum_{k=1}^{n} b_1 k^2 - \sum_{k=1}^{n} a_k k &= 0
\end{align*}
\]  

(5)

then,
Then the formula for eliminating the linear trend term is

\[ x_k = a_k - \hat{a}_k = a_k - (b_0 - b_1 k) \]  (7)

The test acceleration curve has more "burrs", making the curve very unsmooth. In order to weaken the influence of interference signals, the calculation method using the moving average method (5-point sliding average) \[16\].

The original acceleration response curve collected is shown in Figure 6 and the acceleration curve after detrending and smoothing is shown in Figure 7. It can be seen that the overall smoothness of the curve is greatly improved, which can greatly reduce the influence of noise signals and sampling frequency.

2.3.2 Acceleration frequency domain integration

Theoretically, since the initial velocity and displacement of the buffer device are zero, the displacement signal can be obtained from the secondary integration of the measured acceleration. Converting the time domain signal into frequency domain signal and integrating directly in the frequency domain after certain filtering process can give more accurate results.

According to the formula of Fourier transform, the Fourier component of the acceleration signal at any frequency can be expressed as

\[ a(t) = A e^{j\omega t} \]  (8)

When the initial velocity and initial displacement are both 0, the displacement signal can be obtained by quadratic integration of the Fourier component of the acceleration signal, that is,

\[ x(t) = \int_0^t \int_0^\tau a(\lambda) d\lambda \ d\tau = -\frac{A}{\omega^2} e^{j\omega t} = X e^{j\omega t} \]  (9)

where \(x(t)\) is the Fourier component of the displacement signal at frequency \(\omega\); \(X\) is the coefficient corresponding to \(x(t)\). Thus the relation of the quadratic integral in the frequency domain is given by

\[ X = -\frac{A}{\omega^2} \]  (10)

After the acceleration signal is converted to the frequency domain filtering process by the above method, the displacement results calculated in the frequency domain are compared with the results directly calculated in the time domain. As shown in the Figure 8, it can be seen that the displacement response curve obtained by direct integration in the time domain has an obvious oscillating upward trend, which is not consistent with the actual experimental measurement, while the displacement response curve calculated in the frequency domain avoids this phenomenon and the results obtained are consistent with the actual one, which can be better used for system identification.
2.3.3 System Identification
Assuming that the system is a second-order free vibration system with parameters including mass $m$, stiffness $k$, and damping $c$. When the buffer system is subjected to external force, the differential equation of motion of the system is

$$m\ddot{x} + c\dot{x} + kx = f(t)$$

(11)

After Laplace transform, the transfer function can be obtained as

$$T(s) = \frac{\mathcal{L}[\text{output}]}{\mathcal{L}[\text{input}]} = \frac{x(s)}{F(s)} = \frac{1}{ms^2 + cs + k}$$

(12)

Finally, using MATLAB System Identification toolbox, building the corresponding transfer function model, and the impulse force obtained from the processing of the data above is the input, and the displacement response curve obtained from the frequency domain integration is the output, and the buffer system is identified.

3. Results & Discussion
Different buffer systems were obtained by changing the spring and the polyurethane of the buffer system, and the acceleration response curves and force input curves of the buffer systems under different working conditions were obtained by conducting multiple experiments on 2 different buffer systems respectively using force hammer experiment. Using the above method for identification, the obtained buffer systems parameters identification results are shown in the Table 1.

| number | H/HA | D/mm | $k$/N·m$^{-1}$ | $m$/kg | $c$/N·s·m$^{-1}$ | $\omega_n$/Hz | $c_c$/N·s·m$^{-1}$ | $\zeta$ | $\omega_d$/Hz |
|--------|------|------|---------------|--------|----------------|---------------|------------------|-------|---------------|
| 1      | 60   | 2.5  | 5.51E+5       | 2.54   | 304.31         | 465.90        | 2365.21         | 0.13  | 461.96        |
| 2      | 80   | 2.5  | 8.40E+5       | 2.92   | 346.94         | 536.73        | 3131.26         | 0.11  | 533.36        |

Among them, $H$ is the Shore hardness of the polyurethane of the buffer system, $D$ is the wire diameter of the spring, $k$ is the system stiffness, $m$ is the system mass, $c$ is the system damping, $\omega_n$ is the system natural frequency, $c_c$ is the system critical damping, and $\zeta$ is the system damping ratio, and $\omega_d$ is the damping vibration frequency.

Using the identification results, the maximum acceleration response after buffering under different working conditions is calculated and compared with the results obtained from the hammer experiment. The error bar graph of the two is shown in Figure 9 and Figure 10, and the error range is 4%. The horizontal coordinate in the figure is the input impulse of the system in the experiment, and the vertical coordinate is the maximum response acceleration. It can be seen from the graph that the difference between the results calculated theoretically using the discrimination parameters and the experimental test results is very small, and the maximum error between the two is not more than 4%.
And after using the general method to directly identify the experiment signal, the result obtained by the identification has a large error with the actual one, and the maximum error is even close to 15%, as shown in Figure 11 and Figure 12.

Therefore, it can be concluded that for this buffer system, the proposed method can significantly improve the accuracy and reliability of the identification results by pre-processing the experimental data with detrending, smoothing and frequency domain integration before system identification, which can reduce the error by about 10% compared with direct identification.

4. Conclusions
In this paper, an identification method for the buffer system that can improve the identification accuracy is proposed. The acceleration response curve after buffering is tested through the hammer experiment of the buffer system for a gun. The trend term of the curve is removed by the least square method, and the acceleration response curve is smoothed by the moving average method, and finally, the time domain signal is transformed into the frequency domain, and the frequency domain integration is performed. Finally, the displacement data obtained by the frequency domain integration is used for system identification. The results show that for the buffer system, the method can significantly improve the accuracy of system identification, the deviation from the experimental results is less than 4%, and the error can be reduced by about 10%. This method can provide a feasible technical approach for accurate identification of vibration system models. Further, the identification results can provide more reliable guidance for the design of the system, and ensure the safety and reliability of the buffer system in use.

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