Research on multi-parameter correction of vibration system based on genetic Algorithm

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Abstract. In order to solve the problem of fast solving the vibration system under complex excitation, a co-simulation method based on the mathematical simulation model and the multi-parameter correction of the vibration system is proposed. According to the dynamics theory of multi-body systems, the dynamic equations of the system are derived. Runge-Kutta method and MATLAB software are used to solve the dynamic equations. Compared with ADAMS multi-body simulation, the accuracy and efficiency of the dynamic equations are proved. The vibration test is carried out to obtain the acceleration data required to construct the multi-objective function. Optimal solution of modified parameters is obtained by multi-objective genetic algorithm. Finally, the vibration test bench system was used to verify the feasibility of the multi-parameter correction method. The results show that the error of the modified objective function of the constructed vibration test bench system is greatly reduced, which verifies the effectiveness of the multi-parameter correction method.

1. Introduction
Multi-body dynamics modelling and simulation is one of the main methods for studying the dynamic performance of mechanical vibration systems. For the dynamics under the large number of working conditions and complex transient excitation types to solve vibration mechanics problems in real time, the calculation time of the simulation system constructed by commercial software is far from meeting the requirements. For example, when studying the real-time correction of the posture of a self-propelled anti-aircraft gun in the shooting process, it is necessary to obtain its dynamic response in each shooting state in real time, so as to provide the posture information for the fire control system. Simplified models should be simplified for specific problems, and the dynamic equations of small scale and fast-solving mathematical models derived from this can meet the requirements of real-time prediction of system dynamic response and improve system performance. Therefore, the mathematical model dynamic equation derived by using the simplified model has obvious advantages in solving the vibration mechanics problems under multiple operating conditions and transient excitation.

In the process of constructing the dynamic model equation, there are some problems such as structural simplification, idealization of contact relationship, parameter measurement error, etc. The dynamic model equation is bound to be quite different from the actual vibration system. Therefore, how to accurately determine the dynamic parameters in the model equation has become an urgent problem to be solved in the dynamic simulation of the vibration system [1-3].
Qingguo Fei [4] took the nonlinear beam structure as an example, and used the radial basis function neural network to modify the three parameters of the nonlinear elastic coefficient. Through comparison with the experimental results, the effectiveness of the parameter modification was verified. Qinlong Wang [5] and others carried out a multi-rigid body dynamics simulation of a tracked vehicle, with the power spectrum density of different roads as the goal, and screened out three main influencing parameters. An approximate model between the objective function and the parameters to be corrected is constructed using the optimized super-Latin experimental design and the Kriging proxy model, and the corrected parameters are obtained through the pattern search method.

The parameter correction work based on the commercial software simulation model requires a lot of iterative calculations. The engineers proposed a method of constructing a proxy model to make up for the shortcomings of the method such as long calculation time and non-adjustable parameters [6]. But at the same time, the proxy model brings a contradiction between the number of parameters and the accuracy of the model [7]. The proxy model constructed by the fitting method has a limited number of parameters to be corrected, and its correlation with kinetic parameters will also decrease as the number of parameters increases.

The establishment of a mathematical model can not only guarantee the calculation time and accuracy at the same time when solving the vibration mechanics problems under multiple working conditions, but also can maximize the scale and accuracy of the modified dynamic parameters. This paper takes the vibration test bench as an example, establishes the dynamic model of the system by Newton Euler method, and uses the Runge-Kutta method to calculate the system response. In addition, a virtual prototype simulation was performed in Adams software. The comparison of the two simulation results verifies the credibility and efficiency of the mathematical model. In this paper, the multi-objective genetic algorithm is used to modify the multi-dynamic parameters in the vibration system to improve the simulation accuracy, and the validity of the parameter modification method is verified by comparing with the experimental results.

2. Multi-parameter correction method of vibration system
The prerequisite for the development of kinetic parameter correction is the construction of the objective function. The correctness of the parameter correction results is related to the effectiveness of the objective function construction. The test result is the most effective data reflecting the response of the vibration system. Therefore, the time and frequency domain information of the test data should be analyzed, and the objective function of parameter correction should be established according to them.

The multi-objective parameter correction takes the degree of agreement between the measured data and the simulation data as the evaluation index, and uses an optimization algorithm to search in the parameter design space to screen out the optimal solution, thereby correcting the kinetic parameters [8]. NSGA-II is a non-dominated sorting genetic algorithm with elite strategy [9]. Compared with other multi-objective genetic algorithms, its fast and non-dominated sorting method and the introduction of crowding operator [10] greatly improve its search performance.

The multi-parameter correction technology flow chart of the vibration system proposed in this paper is shown in Figure 1.

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Figure 1. Flow chart of vibration system multi-parameter correction technology
This technical route analyzes the topological relationship and simplifies the model for the research object, and verifies the credibility of the deduced model by comparing the calculation of the model derived by dynamic theory and the simulation of commercial software. Secondly, vibration tests are carried out under typical working conditions to construct multi-objective functions. Finally, the parameters to be corrected are determined through vibration mechanics analysis, and the multi-objective genetic algorithm [11] is used to correct multiple correction parameters in the dynamic model to obtain the Pareto front and select the optimal solution.

3. Vibration test bench system modelling

3.1. Vibration test system model

Figure 2 is a three-dimensional model of the vibration test system. The vibration isolator is fixed on the bench surface of the electric vibration table through the threaded connection of the base. The vibration isolator contains tower springs, three-jaw damping discs and other components, and its mechanical properties can be simplified to springs and viscous dampers that are perpendicular to each other in three directions. The core shaft of the vibration isolator is connected with the aluminium vibration body by bolts, and the counterweight is placed in a specific position and then connected with the vibration body by five studs.

![Figure 2. 3D modal of vibration test system consists of an electric vibration bench, four vibration isolators, an aluminium vibration body, and several clump weight.](image)

3.2. Newton Euler Method Modelling

The coordinate system definition of the kinematic analysis of the vibration bench is shown in Figure 3. The origin O of the inertial coordinate system G and the origin O1 of the vibration body coordinate system V are located at the initial centre of mass position of the vibration body and the counterweight; the vibration bench coordinate system T is the following coordinate system of the vibration bench, and the origin O2 is located on the upper surface of the vibration bench. The geometric centre position.

The force generated by the vibration isolator is provided by springs, viscous dampers, etc. When working near the rated load, the displacement of the vibration isolator core shaft works in a specific smaller range, so its elastic coefficient and viscous damping coefficient can be approximated as constants.

The vibration body has six rigid body degrees of freedom. The position of the Centroid of the vibration body is described by three rectangular coordinates \((x, y, z)\) in the inertial coordinate system G; the azimuth angle change of the vibration body in the inertial coordinate system G is described by three Euler angle coordinates \((\psi, \Theta, \phi)\). The horizontal stiffness of the vibration bench is much greater than the vertical stiffness, Therefore, assuming that it has only three degrees of freedom, its displacement is \((z_T, \Theta_T, \phi_T)\). The displacement of the vibration bench is the excitation of the vibration system, which is measured by the sensor. When subjected to external excitation, the spring deformation of the vibration isolator is calculated from the displacement of the vibration body and the vibration bench as:
\[
\begin{bmatrix}
  p_{x,i} \\
p_{y,i} \\
p_{z,i}
\end{bmatrix} = \begin{bmatrix}
  x \\
y \\
z
\end{bmatrix} + \begin{bmatrix}
  \psi \\
\theta \\
\phi
\end{bmatrix} \times \begin{bmatrix}
  x_i \\
y_i \\
z_i
\end{bmatrix} \begin{bmatrix}
  0 \\
0 \\
0
\end{bmatrix} + \begin{bmatrix}
  \psi_t \\
\theta_t \\
0
\end{bmatrix} \times \begin{bmatrix}
  H_{x_i} \\
H_{y_i} \\
H_{z_i}
\end{bmatrix}
\]
i = 1, 2, 3, 4
(1)

\( (p_{x,i}, p_{y,i}, p_{z,i}) \) is the deformation of the three vertical springs of the first vibration isolator, \( (L_{x_i}, L_{y_i}, L_{z_i}) \) is the space coordinate of the first vibration isolator in the vibration body coordinate system V, and \( (H_{x_i}, H_{y_i}, H_{z_i}) \) is the space coordinate of the first vibration isolator in the vibration bench coordinate system T. The force analysis of the vibration body is shown in Figure 4.

Figure 4. Schematic diagram of force analysis of vibrating body

The force action on the vibration isolator on the vibration body is simplified to \( A_i \sim \bar{A}_i \), and all the force actions are based on the direction of the vibration body coordinate system V. According to Newton's Euler equation, the force analysis is performed with the center of mass of the vibration body and the counterweight as the origin. The dynamic equations of the vibration body are:

\[
m\ddot{x} = \sum_{i=1}^{4} F_{x,i}
\]
\[
m\ddot{y} = \sum_{i=1}^{4} F_{y,i}
\]
\[
m\ddot{z} = \sum_{i=1}^{4} F_{z,i} - mg
\]
\[
\begin{bmatrix}
  I_{XX} & -I_{XY} & -I_{XZ} \\
-I_{XY} & I_{YY} & -I_{YZ} \\
-I_{XZ} & -I_{YZ} & I_{ZZ}
\end{bmatrix} \begin{bmatrix}
  \ddot{x} \\
\ddot{y} \\
\ddot{z}
\end{bmatrix} + \begin{bmatrix}
  \psi \\
\theta \\
\phi
\end{bmatrix} \times \begin{bmatrix}
  I_{XX} & -I_{XY} & -I_{XZ} \\
-I_{XY} & I_{YY} & -I_{YZ} \\
-I_{XZ} & -I_{YZ} & I_{ZZ}
\end{bmatrix} \begin{bmatrix}
  \dot{\psi} \\
\dot{\theta} \\
\dot{\phi}
\end{bmatrix} + \sum_{i=1}^{4} \begin{bmatrix}
  F_{x_i} \\
F_{y_i} \\
F_{z_i}
\end{bmatrix} \begin{bmatrix}
  L_{x_i} \\
L_{y_i} \\
L_{z_i}
\end{bmatrix}
\]

The force of the vibration isolator on the vibration body in the equations is as follows:

\[
\begin{bmatrix}
  F_{x,i} \\
F_{y,i} \\
F_{z,i}
\end{bmatrix} = \begin{bmatrix}
  k_{x,i} p_{x,i} \\
k_{y,i} p_{y,i} \\
k_{z,i} p_{z,i}
\end{bmatrix} \begin{bmatrix}
  c_{x,i} \dot{p}_{x,i} \\
c_{y,i} \dot{p}_{y,i} \\
c_{z,i} \dot{p}_{z,i}
\end{bmatrix} \quad i = 1, 2, 3, 4
\]

where \( m \) is the total mass of the vibration body and the counterweight; \( I_{XX}, I_{YY}, I_{ZZ} \) are the moments of inertia of the vibration body and the counterweight around the OX1, OY1, OZ1 axis; \( I_{XY}, I_{XZ}, I_{YZ} \) are the product of inertia of the block in the XY, XZ, and YZ planes, \( k_{Xi}, k_{Yi} \) and \( k_{Zi} \) are the elastic coefficients of the ith isolator tower spring in the front, back, left, and right directions; \( c_{Xi}, c_{Yi}, c_{Zi} \) are the viscous damping coefficient of the vibration in the front and rear, left and right, up and down directions; \( FC_i \) is the Coulomb friction of the ith vibration isolator in the up and down direction.

3.3. Virtual prototype simulation

The multi-body dynamics model of the vibration bench system is established in Adams software, and the elastic force, Coulomb friction force and viscous damping force of the vibration isolator are
simulated by writing functions. The degree of freedom of the component is restricted by the motion relationship of the fixed pair. In addition, the excitation of the vibration bench is defined by writing a function, and simulation is performed to obtain the response of the vibration body under the corresponding excitation. The system takes the displacement of the three degrees of freedom of the vibration bench as input excitation, and controls the displacement of the vibration bench as:

\[ z(t) = 6.2 \times 10^{-4} \sin(40\pi t) \text{mm}; \quad \psi_\text{a} = 1.14 \times 10^{-4} \sin(40\pi t) \text{rad}; \quad \theta_\text{a} = 5.7 \times 10^{-4} \sin(40\pi t) \text{rad}. \]

The mathematical model and Runge-Kutta method formula are programmed in MATLAB and VC++6.0 software respectively. The excitation setting of the mathematical model is consistent with the dynamic simulation, the simulation time is 10s, and the calculation step is 1ms. Figures 5 and 6 show the comparison of partial pose response curves of the mathematical model and the simulation model of the vibration bench under uniform working conditions. Table 1 compares the simulation time of various software.

![Figure 5. Comparison of displacement simulation results between Matlab and Adams](image1)

![Figure 6. Comparison of angle displacement simulation results between Matlab and Adams](image2)

| Software type | Adams | MATLAB | VC++6.0 |
|---------------|-------|--------|---------|
| Resolving time | 74.7s | 0.875s | 0.034s |

Figures 5 and 6 show that the mathematical model simulation has the same accuracy as commercial software. Table 1 shows that the calculation time of self-compiled code to solve the dynamic equation is greatly reduced compared with Adams software. In summary, this method has high accuracy and fast calculation, which can significantly improve the scale and speed of parameter correction.

4. Vibration response test

The vibration test of the shaker system is carried out in the laboratory. The test signal is the vertical acceleration at the four corners of the shaker and the upper surface of the vibration body. The data acquisition system consists of a piezoelectric three-directional acceleration sensor, an Elsys408 voltage data acquisition system, and a tranAX data acquisition system. Software composition, sampling frequency is 1KHz. The layout of the test equipment and sensors is shown in Figure 7.
During the test, due to the gap and collision between the components of the vibration system, the acceleration signal will appear in the middle and high frequency components. Therefore, the vertical acceleration signal of each data collection point was zero-averaged and low-pass filtered. Use Fourier transform and inverse Fourier transform to perform low-pass filtering of test data, with a cut-off frequency of 25HZ.

According to the vertical acceleration signals of the four measuring points and the distance between the measuring points, the four-dot matrix method is used to obtain the acceleration component of the vibration body. The result is shown in Figure 8.

5. Dynamic model multi-parameter correction

5.1. Objective function construction

The premise of parameter modification is to establish a modified objective function to quantify the similarity between test results and simulation results. Acceleration data are used to construct the target function of parameter correction. The dimensionless mean square error of the three acceleration component test data and simulation data are used to describe the degree of agreement of the acceleration curve.

\[
f = \frac{1}{n} \sum_{i=1}^{n} \left( \frac{|a_i - a'_i|}{\max(|a_i|,|a'_i|)} \right)^2
\]

where \( f \) is the dimensionless mean square error between the simulation data and the test data, \( a_i \) is the simulation acceleration value at time \( t \), \( a'_i \) is the test acceleration value at time \( t \), the value range of \( f \) is 0–4, the closer the function value is to 0, then two The better the curve fits the degree, that is, the closer the simulation result and the test result are.

5.2. Modified kinetic parameter determination

The frequency of the experimental results under constant frequency excitation is basically the same as that of the simulation calculation results, but there is still a certain difference in amplitude. The quality parameters and elastic coefficients have been tested experimentally. Therefore, the viscous damping coefficient of the vibration isolator is the key factor in the difference of simulation results.

The overall damping value of the vibration isolator is related to the magnitude of the acceleration. It is precisely because of the coupling of the damping value of the vibration isolator to the dynamic response that the single-objective optimization idea cannot be used to complete the correction of the dynamic parameters. Therefore, this paper uses a genetic algorithm based on multi-objective optimization to simultaneously correct the viscous damping parameters of the four vibration isolators.
5.3. Modified mathematical model

The mathematical model describing the correction problem of this dynamic model is as follows:

\[
\begin{align*}
\min & \quad f_\psi, \\
& \quad f_\theta, \\
& \quad f_z; \\
\text{s.t.} & \quad c_{31}^L \leq c_{31} \leq c_{31}^U, \\
& \quad c_{32}^L \leq c_{32} \leq c_{32}^U, \\
& \quad c_{33}^L \leq c_{33} \leq c_{33}^U, \\
& \quad c_{34}^L \leq c_{34} \leq c_{34}^U.
\end{align*}
\] (5)

Where \( f_\psi, f_\theta \) and \( f_z \) are the modified objective functions in the OX, OY and OZ axis directions. The superscripts L and U of the correction function respectively represent the upper and lower limits of the value interval of the viscous damping coefficient. It is stipulated that the lower limit of kinetic parameters is half the initial value, and the upper limit is twice the initial value.

5.4. Result analysis

The parameters of NSGA-II are set as: the initial population individual number is 100; the maximum evolutionary generation number is 200; the optimal individual coefficient is 0.3; the fitness function deviation is set to 1e-100. Figure 9 shows the calculated Pareto frontier, and Table 2 shows some of the multi-objective modified optimal solutions and their objective function change rates.

| Serial number | \( f_\psi \) | Rate of change /% | \( f_\theta \) | Rate of change /% | \( f_z \) | Rate of change /% |
|---------------|-------------|-------------------|-------------|-------------------|-------------|-------------------|
| 1             | 0.2439      | 13.9              | 0.1163      | 37.9              | 0.0267      | 86.8              |
| 2             | 0.2492      | 12.1              | 0.1015      | 45.8              | 0.0274      | 86.4              |
| 3             | 0.2804      | 11.1              | 0.0906      | 51.7              | 0.0290      | 85.6              |
| 4             | 0.2401      | 15.3              | 0.0834      | 55.5              | 0.0457      | 77.4              |
| 5             | 0.2371      | 16.4              | 0.1139      | 39.3              | 0.1046      | 48.2              |
| 6             | 0.2368      | 16.5              | 0.1318      | 29.7              | 0.1267      | 37.2              |
| 7             | 0.2365      | 16.6              | 0.1412      | 24.7              | 0.1376      | 31.8              |
| 8             | 0.2362      | 16.7              | 0.1538      | 17.9              | 0.1530      | 24.2              |
| 9             | 0.2359      | 16.8              | 0.1679      | 10.4              | 0.1689      | 16.3              |
| 10            | 0.2357      | 16.9              | 0.1856      | 1.0               | 0.1764      | 12.6              |
| Before correction | 0.2836 | 0 | 0.1875 | 0 | 0.2019 | 0 |
It is not difficult to see from Figure 9 that the Pareto front basically conforms to a spatial curve distribution, the angular acceleration correction objective function solution in the OX axis direction is mostly concentrated around 0.24, the angular acceleration correction objective function around the OY axis is consistent with the optimization direction of the acceleration correction objective function along the OZ axis.

It can also be seen from Table 2 that in the design space of the correction parameters, when the value of the correction objective function of the OX axis is around 0.24, the change law of the correction objective function of the OY axis and the OZ axis is the same. Therefore, under comprehensive consideration, the second solution is considered to be the most effective Good, the final solution to the multi-objective correction problem. The parameter group corresponding to the second solution is substituted into the equation, and the dynamic response and the calculated result before correction are compared with the experimental test result. The comparison result is shown in Figure 10.

It can be seen from Figure 10 that after dynamic parameter correction, the angular acceleration amplitude error in the OX axis direction is reduced from 24.06% to 6.56%; the angular acceleration amplitude error in the OY axis direction is reduced from 38.8% to 24.07%; the acceleration in the OZ axis direction The amplitude error is reduced from 104.89% to 20.77%. The objective function and acceleration amplitude in the three directions have been reduced to varying degrees, indicating that the accuracy of dynamic modelling has been improved, and it also proves that the method of multi-objective genetic algorithm is used in the correction of dynamic parameters. Is reasonable and effective.

6. Summary
A method to modify the dynamic parameters of the vibration system based on the dynamic equation is presented in this paper. Taking the vibration bench system as the research object, the accuracy and efficiency of the dynamic model are verified by the comparison of dynamic equation solving and multi-body dynamic simulation. The vibration bench system test was carried out. The test data was processed through zero-averaging and low-pass filtering. The acceleration component of the vibration body was obtained by the four-dot matrix method, and the objective function to correct the problem was established. The non-dominated sorting genetic algorithm NSGA-II with elite strategy is used to solve the correction problem, and the Pareto front with multi-objective parameter correction is obtained, and the optimal solution is obtained after screening. Through the analysis of the results before and after the correction, the accuracy of the kinetic model has been significantly improved, and the effectiveness of the parameter correction based on the multi-objective genetic algorithm has been verified.

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