Design and Demonstration of Experiment Bench with Small Vibration Source and Large Load

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Abstract: We design a kind of vertical vibration experiment bench with small vibration source and large load, and demonstrate that the support spring has no influence on the vibration transmissibility. The spring parameters are designed based on the resonance characteristic of the spring, and the transfer plate is simulated and optimized by Workbench and Hyperworks. The experiment bench makes the large load can be vibrated under the excitation of small vibration source, which provides a reference design for the vibration isolator and other equipment that need to carry out the large load vibration experiment.

1. Introduction

With the development of micro-machining and precision detection technology, high precision instruments have increasingly strict requirements for reducing environmental vibration [1]. Vibration isolator is a kind of equipment which can isolate the ground micro-vibration to the high precision instrument. The development of high-performance vibration isolator can effectively isolate ground vibration and improve the machining or detection accuracy of precision instruments [2]. In the performance experiment of the vibration isolator, the vibration isolator is excited after carrying the load. As the load increases, stronger excitation equipment is needed. Ordinary shakers which provide small vibration source cannot drive large load directly. Large load vibration test usually requires large-scale and expensive vibration table [3,4]. Therefore, it is necessary to design an experiment bench to solve the contradiction between small vibration source and large load.

2. Overall design of the experiment bench

2.1. 3D model structure design

This paper designs an experimental bench for vertical vibration test of a prototype of air spring vibration isolator. The vibration isolator is a cylinder with a diameter of 25cm and a height of 27cm. The design index requires that the load can reach 1000kg. The vibration excitation frequency range is 1-200Hz, and the load response amplitude should less than 1mm. The instruments used in the experiment are the SA-JZ050 shaker with the maximum output force of 500N and the FNV-R3D-VD1 Doppler laser vibrometer.
with the measurement accuracy of 10nm. In the vertical vibration experiment, the load is placed on the vibration isolator in advance, then the bottom of the isolator is excited by the shaker, and the vibration signals before and after isolation are measured by the vibrometer. Finally, the collected signals are processed and analyzed to evaluate the performance of the vibration isolator.

According to the above experimental conditions and relevant requirements, the size parameters of the experiment bench and the layout of each equipment are preliminarily determined. A three-dimensional model of the vertical vibration experiment bench with small vibration source and large load is established in the Solidworks software, as shown in Figure 1.

In this experimental bench, the load is made of some steel plates for easy loading and unloading. The vibration isolator supports the load and is installed above a transfer plate, under which there are four support springs and excitation points of the vibration shaker. The support springs are used to bear gravity, and the shaker exerts vibration excitation on the bottom of the transfer plate. The vibration signals before and after isolation by the isolator can be measured by aligning the two laser vibrometers with the bottom of the transfer plate and the load respectively. The support springs, the shaker and the vibrometer are mounted on the same base plate.

The main characteristic of this experiment bench is using support springs to load up most of the load weight, allowing the shaker with relatively small output force for applying vibration excitation. This design solves the contradiction that small vibration source cannot drive the large load, and implements the vibration isolator for vertical vibration experiment testing requirements.

In addition, as the vibration isolator works in the environment of large load, high pressure and vibration in the experiment, in order to prevent the accidental dumping of the load which may cause damage to personnel or equipment, a protective device is designed at the periphery of the experiment bench. The experiment bench with the protection device is shown in Figure 2.

The protection device mainly includes several protective circles and four groups of support rods. The protection circle has windows for observation and operation. In the experiment, the corresponding number of protection circles can be configured according to the loading height of the load plate. The
protection circles are fixed to each other by connecting blocks. The support rods set are arranged around the protection circle, and the triangular stable structure is formed by the hinge circle body and the extension plate of the bottom plate.

2.2. Demonstration: The support spring has no influence on the vibration transmissibility test

The traditional large vibration table can directly vibrate the large load. In the experimental bench designed in this paper, the shaker excites the transfer plate, and then the vibration is transferred to the load through the transfer plate and the vibration isolator. During the vibration test, the four support springs at the bottom of the transfer plate vibrate simultaneously. The vibration amplitude ratio after vibration isolation and before vibration isolation, also known as vibration transmissibility, is a key index to study the performance of vibration isolator [5,6]. The following is a demonstration: the support spring has no influence on the vibration transmissibility test.

This experimental bench can be simplified into a 2-DOF model, as shown in Figure 3. The base plate of the bench is equivalent to the foundation. The support springs and the fixing mechanism are equivalent to \( k_1 \) and \( c_1 \), and the transfer plate and the vibration isolator shell with its solid connection are equivalent to \( m_1 \). The air spring of the isolator is equivalent to \( k_2 \) and \( c_2 \), and the load plate of the isolator and its solid connection on the isolator are equivalent to \( m_2 \). The vibration incentive force exerted by the shaker on the bottom of the transfer plate is \( F \). The vibration response signal of the transfer plate measured by the vibrator is \( X_1 \), and the vibration response signal of the load plate is \( X_2 \).
The dynamic equation of this model can be expressed as follows in matrix form:

\[
\begin{bmatrix}
    m_1 & 0 \\
    0 & m_2
\end{bmatrix}
\begin{bmatrix}
    \ddot{x}_1 \\
    \ddot{x}_2
\end{bmatrix}
+ \begin{bmatrix}
    c_1 + c_2 & -c_2 \\
    -c_2 & c_2
\end{bmatrix}
\begin{bmatrix}
    \dot{x}_1 \\
    \dot{x}_2
\end{bmatrix}
+ \begin{bmatrix}
    k_1 + k_2 & -k_2 \\
    -k_2 & k_2
\end{bmatrix}
\begin{bmatrix}
    x_1 \\
    x_2
\end{bmatrix}
= \begin{bmatrix} F \end{bmatrix}.
\]

When the initial conditions \( x_0 = x_1 = x_2 = 0 \), \( \dot{x}_0 = \dot{x}_1 = \dot{x}_2 = 0 \), using the Laplace transform, we have

\[
\left[ s^2 [M] + s [C] + [K] \right] \left[ X(s) \right] = \left[ F(s) \right],
\]

where, \([M]\) is the mass matrix, \([C]\) is the damping matrix and \([K]\) is the stiffness matrix. Equation (2) can be solved as

\[
\left[ X(s) \right] = \left[ s^2 [M] + s [C] + [K] \right]^{-1} \left[ F(s) \right],
\]

and convert it to

\[
\begin{bmatrix}
    X_1(s) \\
    X_2(s)
\end{bmatrix} = \frac{1}{\Delta} \begin{bmatrix}
    m_2 s^2 + c_2 s + k_2 & c_2 s + k_2 \\
    c_2 s + k_2 & m_2 s^2 + (c_1 + c_2) s + k_1 + k_2
\end{bmatrix} \begin{bmatrix} F \end{bmatrix},
\]

where,

\[
\Delta = (m_2 s^2 + c_2 s + k_2) \left( m_2 s^2 + (c_1 + c_2) s + k_1 + k_2 \right) - (c_2 s + k_2)^2.
\]

According to Equation (4), the ratio of load vibration \( X_2 \) to the vibration \( X_1 \) of the transfer plate can be obtained as

\[
\frac{X_2(s)}{X_1(s)} = \frac{c_2 s + k_2}{m_2 s^2 + c_2 s + k_2}.
\]

We make \( s = i \omega \), then there is

\[
\frac{X_2(i \omega)}{X_1(i \omega)} = \frac{k_2 + i c_2 \omega}{k_2 - m_2 \omega^2 + i c_2 \omega}.
\]

Then the displacement transfer rate of vibration \( X_1 \) after vibration isolation by air spring to vibration \( X_2 \) before vibration isolation can be expressed as

\[
\frac{X_2}{X_1} = \frac{\sqrt{k_2 + (c_2 \omega)^2}}{\sqrt{(k_2 - m_2 \omega^2)^2 + (c_2 \omega)^2}} = \frac{\sqrt{1 + (2 \xi r)^2}}{\sqrt{1 - r^2} + (2 \xi r)^2}.
\]

Among them, \( r = \frac{\omega_0}{\omega_2} \), \( \omega_2 = \frac{k_2}{m_2} \), \( \xi = \frac{c_2}{2 \sqrt{k_2 m_2}} \).

It can be seen from Equation (8) that the displacement transfer rate is only related to \( m_2, k_2 \) and \( c_2 \), excluding \( k_1 \) and \( c_1 \). This indicates that in accordance with the test scheme of this experimental bench, if the vibration displacement of the transfer plate and the isolation shell is taken as the excitation displacement, then the transmission rate of the vibration isolator has nothing to do with the support spring under the transfer plate. In other words, the support spring has no influence on the vibration transmissibility test.

According to Equation (8), the relation curve between the transmissibility and frequency ratio under the condition of small damping is drawn in Matlab, as shown in Figure 4. Then, based on Equation (4), amplitude-frequency responses of load signal \( X_2 \) and transfer plate signal \( X_1 \) on vibration incentive force \( F \) are obtained respectively, and relation curves are drawn as shown in Figure 5 and Figure 6 respectively. By comparing the three curves, it can be seen that based on the simplified model of this experimental bench, the transmissibility curve of the isolator has only one resonance peak, while the load and the transfer plate have two resonance peaks for the amplitude-frequency response curve of the vibration excitation. In other words, the vibration of the load and the transfer plate are related to the support spring, while the displacement transfer rate of the load on the transfer plate is independent of the support spring.
3. Design and optimization of key parts

3.1. Design of support spring
The support springs bear the main weight of the transfer plate, the vibration isolator and its load. In order to facilitate the installation of the shaker and vibrometer, and ensure the stability of the experiment bench, four support springs are arranged on the bottom surface of the transfer plate. The supporting object of the spring is regarded as a whole, the vibration process is equivalent to the equilibrium state, and the force analysis is carried out:

\[ kx_1 + kx_2 + kx_3 + kx_4 + F_0 = mg , \]  

where \( m \) is the total load mass, \( F_0 \) is the output force of the shaker, \( k \) is the stiffness coefficient of the spring, \( x_1-x_4 \) is the compression of the four springs respectively. In the experiment, the mass \( m \) and the stiffness coefficient \( k \) remain unchanged. When the output force of the exciter \( F_0 \) increases, the spring compression value \( x \) decreases and the load rises, and vice versa. Therefore, the spring support can ensure that the large load vibrate under the excitation of the small vibration source.

Resonance occurs when the spring vibrates near its natural frequency. The amplitude-amplification characteristic of spring resonance can be used to further reduce the requirement on the output force of the shaker. The calculation formula of the natural frequency of the spring mass system is
6

\[ f_0 = \frac{1}{2\pi} \sqrt{\frac{k}{m}}, \]  \hspace{1cm} (10)

where \( k \) is the stiffness coefficient of the spring, and the unit is N/m; \( m \) is the total mass of the system, and the unit is kg. It is known that when the load is 100-1000kg, the theoretical natural frequency range of the prototype vibration isolator is 1.33-2.6Hz. The natural frequency of the spring is determined to be 2Hz, and the stiffness coefficient \( k \) of the calculated and rounded spring system is 200N/mm. The four supporting springs in this experiment table are in parallel structure, and the stiffness coefficient \( k \) of a single spring is 50N/mm. The specific size of spring was designed and calculated in Advanced Spring Design software, and the results are shown in Table 1.

| Table 1 | Specific parameters of the spring |
|---------|----------------------------------|
| Wire diameter of spring | 10mm |
| Number of coils | 9.5 |
| Length of spring line | 1938.34mm |
| Spring outer diameter | 74.6mm |
| Spring weight | 1.19675kg |
| Spring inner diameter | 54.6mm |
| Free length | 200mm |
| Stiffness coefficient | 50N/mm |

3.2. Optimization of transfer plate

It is necessary to measure the vibration of the transfer plate in the experiment to replace the vibration of the bottom of the vibration isolator, so the transfer plate should have a large stiffness to reduce the measurement error. When the vibration isolator bears a load of 1000kg, the displacement difference between the bottom surface of the transfer plate in the vertical direction should not exceed 1mm.

The initial shape of the transfer plate is a disk of the same diameter as the load plate. The static simulation analysis of the transfer plate was carried out using Ansys Workbench. The boundary conditions are set as shown in Figure 7. Fix the four circular areas where the bottom surface of the transfer plate is in contact with the spring, and load a uniform force onto the area above the transfer plate facing to the bottom of the vibration isolator.

The area of the transfer plate is kept unchanged, and the thickness of the transfer plate is modified to obtain the maximum deformation under different thicknesses when the load changes, as shown in Table 2. With the increase of load mass, the stiffness of the transfer plate decreases gradually, and increasing the thickness of the transfer plate can increase the stiffness. After calculation, when the load is 1000kg and the thickness gradually increases from 10mm to 15mm, 20mm, 25mm and 30mm, the stiffness increases by 223%, 115%, 80% and 61% in turn. With the uniform increase of the plate thickness, the increase amplitude of stiffness decreases gradually. If the thickness of the transfer plate is too large, its mass will be too large, and the size of the support spring will increase accordingly. Therefore, the thickness of the transfer plate should be selected moderately.
Table 2  Simulation results of transfer plate before optimization

| Load weight (kg) | Thickness (mm) 10 | 15 | 20 | 25 | 30 |
|------------------|------------------|----|----|----|----|
| 100              | 0.03943          |    |    |    |    |
| 200              | 0.07693          |    |    |    |    |
| 500              | 0.18941          |    |    |    |    |
| 1000             | 0.37686          |    |    |    |    |

In order to minimize the quality of the transfer plate without reducing the stiffness of it, we use Hyperworks to optimize the structure of the transfer plate. The static analysis of the transfer plate shows the same result as in Workbench: the maximum displacement when loading 10000N force is 0.016mm. In the pretreatment of structural optimization, set the optimization target as the minimum volume, and the optimization result is shown in Figure 8.

Figure 8  Optimization result of transfer plate

Based on the optimization result, the transfer plate is redesigned in Solidworks to increase the cross reinforcement plate through the load and support points. Using Ansys Workbench to carry out simulation again, and the boundary condition setting is shown in Figure 9.

Figure 9  The boundary conditions of the transfer plate after optimization

The area of the cross rebar plate is kept unchanged, and the thickness of the cross rebar plate is modified to obtain the maximum deformation of the transfer plate with different thickness when the load changes, as shown in Table 3. By comparing table 2, it can be found that the design of cross rebar plate can reduce the weight by more than 50%, while the stiffness is basically unchanged.
Table 3  Simulation results of transfer plate after optimization

| Load weight(kg) | Thickness (mm) |
|-----------------|----------------|
|                 | 10  | 15  | 20  | 25  | 30  |
| 100             | 0.02617 | 0.01084 | 0.00566 | 0.00342 | 0.00228 |
| 200             | 0.05087 | 0.02089 | 0.01082 | 0.00649 | 0.0043  |
| 500             | 0.12496 | 0.05105 | 0.02630 | 0.01569 | 0.01033 |
| 1000            | 0.24845 | 0.10131 | 0.05211 | 0.03103 | 0.0204  |

4. Conclusions
In this paper, a vertical vibration experiment bench with small vibration source and large load is designed according to the vibration test requirements of a prototype vibration isolator. The structural scheme of this bench is demonstrated, and the key parts are designed and simulated. Through analysis, this experimental bench can allow the vibration excitation of the shaker to be applied to the experimental object with a relatively small output force, and the support spring has no influence on the vibration transmissibility of the vibration isolator. The design of the key parts such as the support spring and the transfer plate is reasonable and reliable, which can meet the requirements of use.

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