Design and analysis of an active-controlled hydraulic low-frequency vibration isolator

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Abstract: This study proposes an active-controlled hydraulic low-frequency vibration isolator. The vibration isolator can achieve large equivalent mass and quasi-zero dynamic stiffness by combining the fluid inerter with the hydraulic active compensation. Thus, good vibration isolation performance at ultra-low frequency can be realised. The model for the dynamics of the device is set up. The principle of low-frequency vibration isolation is explicated. The natural frequency and its influencing factors are analysed. Finally, preliminary experimental research works are carried out. The results indicate that low-frequency vibration can be effectively isolated by the device proposed in this study.

1 Introduction

In ship industry, the floating vibration isolation technology has been able to isolate the high-frequency vibration very effectively for a long time, but there is no good isolation method for low-frequency vibration. The isolation of low-frequency vibration is still a challenge for the development of submarines for a lower underwater acoustic emission. In addition, the low-frequency vibration on the car seat can increase the probability of drivers suffering from back pain, which causes physical and mental damage [1]. However, the isolation of low-frequency vibrations is difficult, because it requires reducing the system’s dynamic stiffness to lower its natural frequency while increasing the static stiffness to support the equivalent mass of the system.

Traditional passive vibration isolation devices have a satisfactory isolation performance on small-amplitude and high-frequency vibration, but it is hard to suppress large-amplitude and low-frequency vibration. The low-frequency vibration isolator needs to retain a sufficiently low dynamic stiffness. By incorporating quasi-zero stiffness systems, one has a chance to set up a high-static, low-dynamic stiffness system to balance the requirement of low stiffness at high frequencies and high stiffness at low frequencies [2]. The first method is the parallel connection of components that have negative and positive stiffness [3]. For example, in 2013, Le and Ahn [4] designed a car seat isolator with the principle of connecting positive and negative stiffness in parallel; he also added a pneumatic unit to achieve active vibration control. The experimental results showed that the device can achieve satisfactory low-frequency vibration isolation. The second method is to design a non-linear structure or using non-linear materials [5, 6] to obtain a dynamic zero stiffness of the system. For example, in 2007, Virgin et al. [7] designed a vibration isolator based on a non-linear structure. The isolator was non-linear and can be approximated as a non-linear spring. The force–displacement curve has a section with zero slope, which can be used as the zero stiffness characteristic required by engineering applications. When the parameter selection is appropriate, a good low-frequency vibration isolation effect can be obtained.

However, the mechanical structure of positive and negative stiffness in parallel is prone to being unstable for heavy objects. There are other types of device configurations through non-linear structures, although the structure is very simple yet poorly repeatable. In order to achieve large static stiffness and low dynamic stiffness, the hydraulic active vibration isolation technology can be used. The active vibration isolation technology can not only effectively attenuate low-frequency vibrations but also adapt to changes in external disturbance frequency. Hydraulic pressure can also be used to support heavy loads and provide large static stiffness.

In this paper, a fluid inerter and hydraulic active-compensation are combined to operate as an active-controlled hydraulic low-frequency vibration isolator. The model is set up. The natural frequency and its influencing factors are analysed. Preliminary experimental verification is carried out. This paper presents a promising contribution to the applications of low-frequency vibration isolation.

2 System design and modelling

2.1 Principle of isolation

Fig. 1 shows the structural diagram of the device designed in this paper. There is a load on the platform. The piston rod drives under the action of force. Meanwhile, the fluid in the piston cylinder reciprocates in the external helix, which increases equivalent mass of the system and works as a fluid inerter. A spring is installed in the piston cylinder to provide elastic force. A damping orifice is arranged on the piston to increase the damping.

The bottom of the piston rod extends into the pressure cavity. As the piston vibrates, the pressure in the pressure cavity changes accordingly. At the same time, the flow rate of the oil in the pressure cavity is precisely adjusted through the hydraulic circuit to keep the output force of the system unchanged. Thus, the effect of zero stiffness is realised.

The specific action of the hydraulic circuit is as follows: when the piston vibrates upwards, the oil pressure of the pressure cavity is reduced. The controller comprehensively considers the speed sensor and the oil pressure sensor and controls the rotation of the motor to drive the valve spool upwards through the worm gear screw. Then, the pumped oil flows into the pressure cavity through the oil pipe 2 and oil pipe 1 to increase the oil pressure. When the piston vibrates downwards, the valve spool drives downwards correspondingly. Then, the oil flows into the tank through the oil pipe 1 and oil pipe 3 to achieve a pressure drop.

The flow rate is proportional to the area of the valve port. When the piston speeds up, the signals of the speed sensor and oil pressure sensor become larger. Then, the controller controls the rotation of the motor to increase the area of the valve port, which increases the flow rate correspondingly, and vice versa [8, 9]. Thus, a precise control of the flow rate and oil pressure is achieved.
Fig. 1  Structural diagram of the active-controlled low-frequency vibration isolator

Fig. 2  Simplified model of the vibration isolator

2.2 Modelling of isolator dynamics

When the system is at a standstill, the accumulator 1 regulates the oil pressure in the pressure cavity to balance the load's weight, namely,

\[ m_{h} \text{g} = P_{t}A. \]

As shown in Fig. 2, when there is an external stimulation vibration \( F' \), the piston vibrates up and down, and the governing equations of the system dynamic can be obtained as [10]

\[
\begin{align*}
(m_{h} + b)x + C_{a} \ddot{x} + K_{a} x &= F' + mg - f_{d}, \\
f_{d} &= P_{t}A, \quad F = F' + mg,
\end{align*}
\]

where \( b \) is the inertia, \( x \) is the displacement of the piston rod, \( C_{a} \) is the damping value, \( K_{a} \) is the spring stiffness, \( f_{d} \) is the force acting on the piston, \( A \) is the cross-sectional area of the piston rod and \( P_{t} \) is the oil pressure inside the pressure cavity.

The pressure in the pressure cavity changes as [11]

\[ \frac{V_{t}}{\beta_{e}} P_{t} = A \dot{x} - Q. \]

where \( V_{t} \) is the total volume of the pressure cavity, \( \beta_{e} \) is the volume elastic modulus of the hydraulic oil and \( Q \) is the inlet and outlet flow of the oil in the pressure cavity regulated by the hydraulic circuit.

On the basis of hydraulic principles [12], the flow changes as

\[ Q = K_{v} \dot{x}_{v} - K_{c} P_{t}, \]

where \( K_{v} \) is the flow gain of the valve, \( \dot{x}_{v} \) is the displacement of the valve core and \( K_{c} \) is the flow-pressure coefficient of the valve. The three-way spool is zero-open, \( K_{c} = 0. \)

Since the value of \( \dot{x}_{v} \) is determined by the signal fed back from the speed sensor and oil pressure sensor, set \( x_{v} / \dot{x} = k. \) Combining (1)–(3), the system dynamic balance equation can be obtained as

\[ (m_{h} + b)\ddot{x} + C_{a} \ddot{x} + \left( K_{a} + \frac{\beta_{e} A^{2}}{V_{t}} - \frac{\beta_{e} AK_{v}}{V_{t}} \right) x = F. \]  

(4)

After the dimensionless treatment, (4) is recast as

\[ x + \frac{C_{a}}{m_{h} + b} \ddot{x} + \frac{K_{a} + (\beta_{e} V_{t}) A^{2} - (\beta_{e} AK_{v} / V_{t}) k}{m_{h} + b} x = \frac{F}{m_{h} + b}. \]  

(5)

The system's natural frequency is independent of external inputs. In order to analyse the system's natural frequency, the differential equation of simple harmonic motion is obtained as

\[ \ddot{x} + 2\xi \omega_{n} \dot{x} + \omega_{n}^{2} x = 0, \]

(6)

where \( \xi \) is the damping ratio of the system and \( \omega_{n} \) is the system's natural frequency, which are calculated as [13]

\[
\begin{align*}
\xi &= \frac{C_{a}}{\sqrt{4(K_{a} + (\beta_{e} V_{t}) A^{2} - (\beta_{e} AK_{v} / V_{t}) k)(m_{h} + b)}}, \quad (7a) \\
\omega_{n} &= \sqrt{\frac{K_{a} + (\beta_{e} V_{t}) A^{2} - (\beta_{e} AK_{v} / V_{t}) k}{m_{h} + b}}. \quad (7b)
\end{align*}
\]

The linear system's natural frequency is

\[ \omega_{n} = \sqrt{\frac{k}{m}}. \]  

(8)

By comparing (7b) with (8), it is easy to find that the equivalent mass and equivalent stiffness of the system are no longer the original \( m_{h} \) and \( K_{a} \), but \( m' \) and \( k' \), given as follows:

\[ m' = m_{h} + b \]  

(9a)

\[ k' = K_{a} + \frac{\beta_{e} A^{2}}{V_{t}} - \frac{\beta_{e} AK_{v}}{V_{t}} k. \]  

(9b)

In order to obtain a very low natural frequency, increasing the equivalent mass and obtaining quasi-zero stiffness have become two important methods. It can be seen from (9a) that the existence of the fluid inerter increases the system's equivalent mass. At the same time, the value of \( k \) is adjusted by the motor and controller, so that the equivalent stiffness can also be adjusted to be very low, even near zero. Therefore, low-frequency vibration can be effectively isolated by this isolator.

The performance of the system can be determined by the relationship between the ratio of input force and the force that is transmitted to the base and the input frequency, that is, the transfer function of the system. The force applied on the base is

\[ F_{T} = C_{a} \dot{x} + K_{a} x + f_{d}. \]  

(10)
Equations (4) and (10) can be rewritten in frequency domain, and the system's transfer function can be obtained as

\[ G(s) = \frac{F_T(s)}{F(s)} = \frac{C_\alpha s + K_\alpha + \left(\beta e/V_t\right) A^2 - \left(\beta e A K_\nu/V_t\right) k}{(m_h + b)s^2 + C_\alpha s + \left(\beta e/V_t\right) A^2 - \left(\beta e A K_\nu/V_t\right) k}. \]  

(11)

### Simulation

#### 3.1 Resonant frequency of the system

Table 1 lists the parameter values in the simulation. When \( k' = 30 \) is selected, the Bode diagram of the system can be obtained and plotted in Fig. 3. It can be seen from the program that system's resonant frequency has dropped to 0.07 Hz. The results show that system's bandwidth for vibration isolation can be increased by the combination of active compensation principle and inerter. It can truly achieve low-frequency vibration isolation.

#### 3.2 Influence of parameters

The fluid inerter of the system can have different inertances by changing the turns and radius of the helix [14, 15]. Table 2 lists four different inertances, and the simulate responses of the system with these inertances are plotted in Fig. 4. As shown in the figure, the larger the inerter, the better the vibration isolation performance. However, the excessively high inerter will increase the external load requirement. Thus, the selection of the inerter should be reasonable.

Table 3 lists three different \( k' \) values, and the simulated Bode diagram under different \( k' \) values is plotted in Fig. 5. It can be seen that the smaller the \( k' \), the smaller the system's equivalent dynamic stiffness, and the smaller the resonant frequency. It can be easily explained according to physics. When \( k' \) is smaller, the flow rate in and out is controlled more precisely, so that the change of force \( F_T \) transmitted to the foundation is smaller, and the vibration isolation effect is better.

### Experimental verification

#### 4.1 Control principle and experimental device

The device diagram of the test system is shown in Fig. 6. The controller comprehensively considers the signals of speed sensor and oil pressure sensor to control the rotation of the motor. Then, the worm gear screw drives the valve spool to move up and down accurately, so that the area of the valve port can be accurately controlled to achieve precise oil filling and oil discharge for pressure cavity.

#### 4.2 Preliminary test results

In order to measure the vibration isolation performance, we conducted a test on the whole device. Force sensors were installed under the load and the pressure cavity to measure the input force \( F \) and output force \( F_T \). The voltage signals of the sensors were collected by the peripheral component interconnect (PCI) data acquisition card, and the voltage signals \( V \) and \( V_T \) corresponded to

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**Table 1** List of parameters of the isolator

| \( V_t \), m³ | \( A \), m² | \( C_\nu \), Ns/m | \( \beta e \), N/m² |
|----------------|-------------|-----------------|------------------|
| \( 1 \times 10^{-2} \) | \( 3 \times 10^{-4} \) | 0.1 | \( 1 \times 10^9 \) |

**Table 2** List of four different inertances (kg)

| \( b_1 \) | \( b_2 \) | \( b_3 \) | \( b_4 \) |
|-----------|-----------|-----------|-----------|
| 50        | 100       | 150       | 200       |

**Table 3** List of three different \( k' \) values

| \( k'_1 \) | \( k'_2 \) | \( k'_3 \) |
|-----------|-----------|-----------|
| 15        | 30        | 60        |

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*Fig. 3* Bode diagram of the system

*Fig. 4* Bode diagram under different inertances

*Fig. 5* Bode diagram under different \( k' \) values
the device’s input force $F$ and output force $F_T$, respectively. The vibration isolation effect of the system can be obtained by comparing $V$ and $V_T$.

A preliminary test was carried out, and the data comparison curve is plotted in Fig. 7. As shown in Fig. 7, the inclination after vibration isolation is significantly reduced, and approximately horizontal. The amplitude of force change caused by vibration is also significantly reduced. In summary, the vibration isolation performance is obvious.

5 Conclusion

In this paper, an active-controlled hydraulic low-frequency vibration isolator has been proposed, which aims to effectively isolate low-frequency vibration and improve the working environment of large machinery, such as warships and submarines. The external helix, the damping orifice and the spring in the device provide equivalent mass, damping and elastic force, respectively. The motor is controlled to rotate worm wheel, and the screw matched with the worm wheel is used to drive the valve core up and down accurately in order to achieve precise control of the area of the valve port, so as to achieve accurate compensation of the oil pressure in the pressure cavity. Thus, the change of force transmitted to the foundation is almost zero. Through the analysis, the system’s dynamic equation has been obtained. The system’s resonant frequency has been calculated, and its influencing factors have been discussed. Additionally, a preliminary test has been carried out. The data showed that low-frequency vibration can be obviously isolated by this isolator.

6 References

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