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Ground-based Air-bearing Equivalent Verification for Active Vibration Isolation for Space Microgravity Utilization

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Abstract:

Ground-based verification of space active vibration isolation systems faces fundamental challenges because traditional gravity compensation methods - such as quasi-zero stiffness mechanisms, suspension systems, and magnetic levitation - cannot simultaneously realize an ideal microgravity state across multiple degrees of freedom (DOFs) due to mechanical coupling. Horizontal air-bearing systems can emulate the microgravity state in three DOFs simultaneously, but whether the air-bearing environment can serve as a faithful equivalent to true microgravity, and whether the system exhibits sufficient long-term stability, remain open questions. In this paper, we establish a dynamic model of the air-bearing system and theoretically analyze the equivalence of multi-DOF testing. We then design a verification environment comprising an air supply circuit, air-bearing structures, and a high-precision measurement and control system. Based on this platform, we conduct static control, wide-band excitation, and long-term stability tests on the active vibration isolation system, and evaluate the influence of the air-bearing environment on isolation performance. Both theoretical analysis and experimental results demonstrate that the air-bearing simulation method can effectively substitute for the space microgravity environment in verification, offering a viable technical pathway for performance testing of active vibration isolation systems.

Key words: Microgravity simulation; Floating platform; Active vibration isolation; Ground verification;

Introduction

Active vibration isolation for spacecraft has become a critical research topic in space engineering (Ye and Wang, 2026; Herranz et al. 2013). As space science experiments and precision optical payloads impose ever-stringent requirements on the microgravity environment, the performance targets for micro-vibration isolation have risen progressively from about 1e-6g to 1e-7g, and even approaching 1e-8g in some cases (Liu et al. 2025). To ensure reliable long-term operation of high-precision active isolators on orbit, ground-based performance validation is typically required prior to launch (Gong et al. 2020). Nevertheless, it has been recognized that the 1g environment on Earth differs fundamentally from the space microgravity environment in terms of mechanical behavior, which hinders the realistic emulation of the isolator's equivalent flexible support and near-zero stiffness dynamics under microgravity conditions (Wu et al. 2019).

Current ground simulation technologies primarily encompass approaches such as suspension systems, air bearings, quasi-zero stiffness mechanisms, magnetic suspension, and multi-stage gravity offloading structures.

Suspension systems, owing to their simple configuration and high load capacity, are widely adopted for ground-based gravity offloading experiments of spacecraft and space mechanisms (Yan et al. 2025). They operate by counterbalancing the gravitational force of

the payload using flexible cables or hanging fixtures, thereby approximating microgravity motion along selected directions (Wei et al. 2023). In essence, however, such a system behaves like a pendulum: only when the pendulum length is large and the motion amplitude is small can it produce a near-microgravity environment within a limited region orthogonal to the suspension direction (Gao et al. 2021).

Quasi-zero stiffness (QZS) mechanisms have demonstrated effective vibration isolation in ground-based active isolation systems (Li et al. 2024). Their performance, however, hinges on nonlinear mechanical configurations for stiffness cancellation, and is therefore highly susceptible to parameter inaccuracies, assembly tolerances, and variations in the operating point. Any departure from the designed equilibrium position causes the equivalent stiffness to vary rapidly (Ma et al. 2022). In addition, QZS mechanisms are prone to coupling between translational and rotational modes in multi-DOF setups an effect that becomes particularly pronounced near the natural frequency which complicates the task of achieving satisfactory isolation simultaneously for both translational and yaw degrees of freedom (Chen et al. 2019).

Magnetic levitation vibration isolation is attractive for its contactless operation, active controllability, and low friction (Wu et al. 2019). Yet under 1g ground conditions, the system must exert a sustained static electromagnetic force to counteract the gravity of the payload, incurring extra power consumption, nonlinear magnetic effects, control noise, and electromagnetic coupling stiffness (Gong et al. 2020). For massive

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payloads, the required strong magnetic fields and large currents can further lead to thermal drift, compromised attitude stability, and restricted control bandwidth (Yan and Wang. 2025). Despite its high control flexibility, maglev technology therefore still poses challenges for long-duration, high-mass, six-DOF precision levitation validation (Ye and Wang. 2025; Ma et al. 2022).

Beyond the ground-based support methods discussed above, microgravity can also be simulated using drop towers (Zhang et al. 2021) and parabolic aircraft flights (Zhang et al. 2017). While these techniques provide multi-DOF microgravity conditions, the available test time is brief, making them unsuitable for validating active vibration isolation systems that require long-duration or ultra-low-frequency operation.

Air bearings are widely employed in ground-based testing of satellite control systems (Rybus et al. 2016), with available configurations including three-DOF (Wu et al. 2018) and five-DOF air-bearing setups (Wu et al. 2023). Their key advantage is that the air bearings generate a stable air film, enabling nearly frictionless support in the horizontal plane (Rybus et al. 2024) and full decoupling in two horizontal translational axes and one rotational axis.

In this work, we employ a DOF-decoupled equivalence method that breaks down the six-DOF space problem into several independent directional channels for verification, enabling equivalent ground-based testing. The present system is designed for active vibration isolation of a space optical clock, and must therefore demonstrate a combined six-DOF performance of $2e-7g$. On the basis of the air-bearing system's performance tests, we need to address the following three key issues in depth:

(1) Equivalence of the horizontal air-bearing environment

In space, the microgravity environment essentially provides a six-DOF near-free-floating condition. Air-bearing platforms, however, generally offer motion capability in only three horizontal degrees of freedom. Hence, based on the system dynamics model and the coupling between DOFs, we must examine the dynamical equivalence between the air-bearing environment and actual space microgravity, and confirm the viability of a direction-wise test method for representing the full six-DOF dynamic characteristics.

(2) Multi-DOF disturbance effects in the air-bearing system

Despite the favorable low-friction behavior of the air-bearing platform, disturbances can still propagate among degrees of freedom due to air film fluctuations, supply pressure noise, and structural vibrations, potentially compromising the ultra-high precision micro-vibration control. Hence, the disturbance characteristics and coupling mechanisms present in the air-bearing environment, and their impact on overall system performance, require thorough investigation.

(3) Long-term operational stability

Over extended operation, the air-bearing system is susceptible to air supply variations, environmental vibrations, and thermal drift, all of which can alter its dynamic behavior. It is therefore essential to assess the long-term stability of the air-bearing environment and to

characterize how such stability affects the active isolation performance over time.

In response to the aforementioned issues, we develop a dynamic model for the air-bearing active vibration isolation system and analyze the dynamic equivalence between the DOF-decoupled test approach and the actual six-DOF space configuration. The disturbance behavior of the air-bearing system is examined, and the impact of the air-bearing platform on the performance of the active isolator is quantitatively assessed via long-term stability tests. This work thus furnishes both a theoretical foundation and experimental support for ground-based validation of active vibration isolation systems intended for space microgravity applications.

Composition of the Active Vibration Isolation

Air-Bearing Test System

Overall Scheme

To enable ground-based performance verification of an active vibration isolation system intended for space microgravity applications, this work develops a low-disturbance ground simulation testbed based on an air-bearing platform. The overall system comprises five main subsystems: an air-bearing microgravity simulation platform, a stator/floater assembly, an electromagnetic actuation system, a measurement and control system, and a host computer control system. The system layout is illustrated in Fig. 1, and a photograph of the actual hardware is provided in Fig. 2.

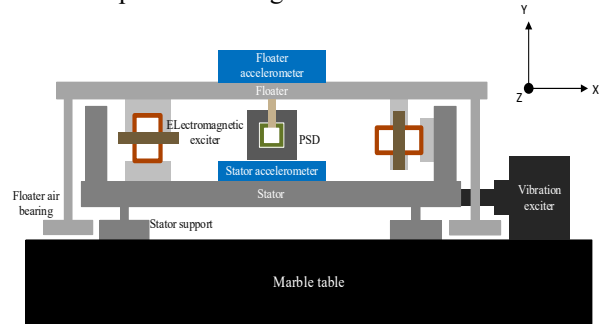


Fig. 1 Structure diagram of the active vibration isolation system

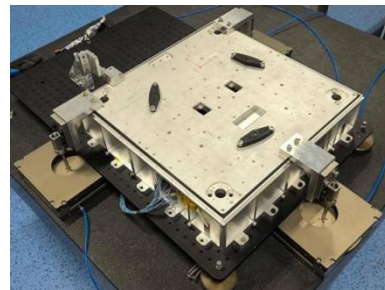


Fig. 2 Diagram of the actual device

In contrast to traditional ground-based gravity offloading approaches, this work exploits the nearly frictionless motion of the air-bearing platform in the horizontal plane to establish a microgravity simulation environment with low coupling and low disturbance. In

the test setup, the stator is coupled to external disturbance sources, whereas the floater is the payload to be isolated. An umbilical cable connects the stator and the floater, and sensors continuously monitor their relative motion. Based on these measurements, electromagnetic actuators generate the required control forces to realize active vibration isolation.

Air-bearing system

The pneumatic subsystem of the air-bearing system comprises an air compressor, air filters, pressure regulators, and the air bearings themselves, arranged in a multi-stage filtration configuration with branched supply lines. The overall system design is illustrated in Fig. 3.

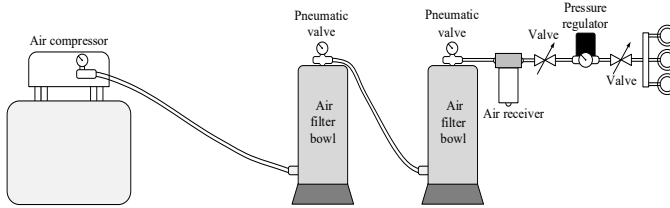


Fig. 3 Air-bearing circuit system structure diagram

For the air-bearing system, the following aspects deserve particular attention:

(1) Air-bearing cleanliness: A three-stage filtration architecture is employed in the pneumatic system to effectively eliminate particulate contaminants from the supply air, preventing debris from entering the air film gap and inducing localized contact or abrupt stiffness changes.

(2) Air film behavior: The air film thickness is highly dependent on the supply pressure. Through pressure regulators and stabilizers, the supply pressure is adjusted and buffered according to the actual counterweight, ensuring that the air film maintains consistent physical characteristics and the air bearing remains in proper equilibrium.

(3) Air pad balancing: The floater air pads are supplied from a single pressure regulator with a common supply line, eliminating pressure discrepancies that could otherwise cause tilting or translational misalignment of the floater platform.

Measurement and Control and Electronics System

The electronics subsystem of the active vibration isolator comprises several modules: the stator main controller, stator measurement and control unit, floater measurement and control unit, motion sensors, electromagnetic actuators, communication module, and a host computer. All modules interact through electrical interfaces and a shared communication bus. The stator main controller includes the main board and motor drives; both the stator and floater measurement/control units feature signal conditioning, A/D conversion, and digital filtering. The sensor suite provides three position-sensitive detectors (PSDs) for relative pose measurement and two triaxial accelerometers, while the actuation is handled by eight voice coil motors. To preserve the fidelity of ground tests relative to the flight system, the

electronics architecture including control logic, sampling rates, communication protocols, and actuator placement is kept identical to that of the on-orbit system. This design greatly improves the engineering value of the ground test results. The overall structure is depicted in Fig. 4.

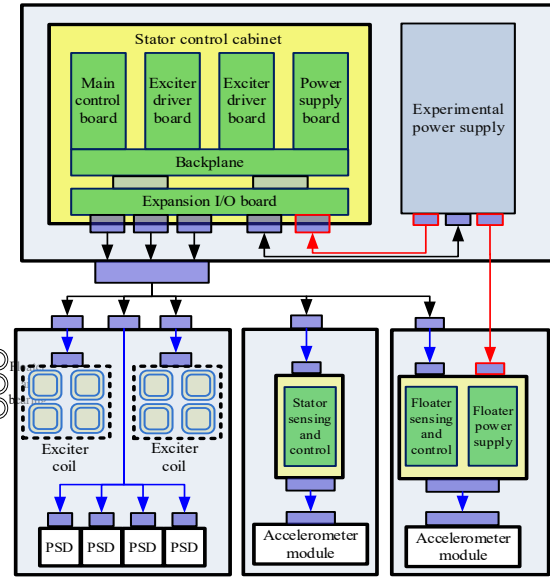


Fig. 4 Active Vibration Isolation System Electronics

Active Vibration Isolation Control System

The system uses a dual-loop control architecture combining pulse-averaged acceleration feedback and displacement feedback. Acceleration and relative displacement/attitude are measured to drive eight electromagnetic actuators, generating counter-forces for active vibration suppression, while the displacement loop ensures collision-free operation. Hardware includes four 2D PSDs, eight single-axis actuators, and triaxial accelerometers on both stator and floater for six-DOF sensing. Two control modes (displacement-only and dual-loop) are available, as shown in Fig. 5.

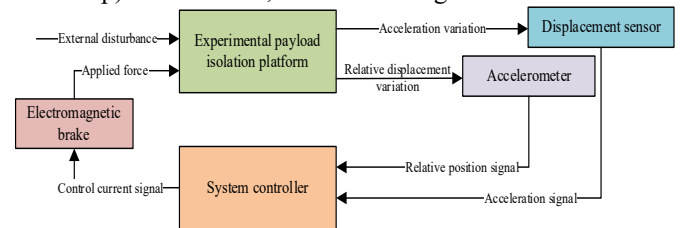


Fig. 5 Active Vibration Isolation System Control Flow Chart

Dynamic model and equivalence analysis of the air-bearing vibration isolation system

Six-degree-of-freedom dynamics and control system model

The model of the air-bearing platform is shown in Fig. 6. In the initial stable state, the stator and the float

adopt the same global coordinate system $\Sigma O - XYZ$.

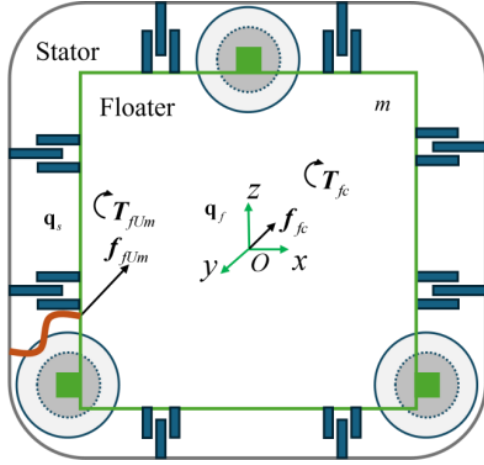


Fig. 6 Diagram of the dynamic coordinate system

The coordinate of the float can be expressed as:

$$\mathbf{q}_f = [\mathbf{r}_f \quad \boldsymbol{\theta}_f]^T$$

The stator coordinate can be expressed as:

$$\mathbf{q}_s = [\mathbf{r}_s \quad \boldsymbol{\theta}_s]^T$$

Where $\mathbf{r} \in \mathbb{R}^3$ denotes the translational motion with three degrees of freedom $[x \ y \ z]^T$, $\boldsymbol{\theta} \in \mathbb{R}^3$ denotes the rotational motion with three degrees of freedom $[\theta_x \ \theta_y \ \theta_z]^T$.

Based on rigid body dynamics, under the condition that the stator is much heavier than the floater, the dynamic equation for the floater of the air-bearing platform is given by:

$$\begin{bmatrix} m & 0 \\ 0 & J_f \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{r}}_f \\ \ddot{\boldsymbol{\theta}}_f \end{bmatrix} = \begin{bmatrix} \mathbf{f}_{fUm} + \mathbf{f}_{fc} \\ -\dot{\boldsymbol{\theta}}_f \times J_f \boldsymbol{\theta}_f + \mathbf{T}_{fUm} + \mathbf{T}_{fc} \end{bmatrix} \quad (1)$$

where $m \in \mathbb{R}$ denotes the mass of the isolation payload, $J_f \in \mathbb{R}^3$ the floater inertia matrix, $\mathbf{f}_{fUm} \in \mathbb{R}^3$ and $\mathbf{f}_{fc} \in \mathbb{R}^3$ the forces on the floater from the umbilical cable and control, respectively, and $\mathbf{T}_{fUm} \in \mathbb{R}^3$ and $\mathbf{T}_{fc} \in \mathbb{R}^3$ the corresponding torques. The floater is assumed to be free of direct disturbances by default.

$$\begin{bmatrix} M_{11} & M_{12} \\ M_{21} & M_{22} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}}_c \\ \ddot{\mathbf{q}}_z \end{bmatrix} + \begin{bmatrix} C_{11} & C_{12} \\ C_{21} & C_{22} \end{bmatrix} \begin{bmatrix} \Delta \dot{\mathbf{q}}_c \\ \Delta \dot{\mathbf{q}}_z \end{bmatrix} + \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} \begin{bmatrix} \Delta \mathbf{q}_c \\ \Delta \mathbf{q}_z \end{bmatrix} + \begin{bmatrix} \mathbf{Q}_c \\ \mathbf{Q}_z \end{bmatrix} = \begin{bmatrix} \mathbf{F}_c \\ \mathbf{F}_z \end{bmatrix} \quad (4)$$

In the ground-based air-bearing environment, the constrained degrees of freedom are limited by the mechanical structure, such that the offset of the floater's center of mass relative to that of the stator tends to zero, i.e., $\Delta \mathbf{q}_z \rightarrow 0$ and $\Delta \dot{\mathbf{q}}_z \rightarrow 0$. For the attitude variation of the floater under small-range motion, the rotational

$$\ddot{\mathbf{q}}_z = -M_{22}^{-1} M_{21} \ddot{\mathbf{q}}_c - M_{22}^{-1} C_{21} \Delta \dot{\mathbf{q}}_c - M_{22}^{-1} K_{21} \Delta \mathbf{q}_c \quad (5)$$

$$(M_{11} - M_{12} M_{22}^{-1} M_{21}) \ddot{\mathbf{q}}_c + (C_{11} - M_{12} M_{22}^{-1} C_{21}) \Delta \dot{\mathbf{q}}_c + (K_{11} - M_{12} M_{22}^{-1} K_{21}) \Delta \mathbf{q}_c = \mathbf{F}_c \quad (6)$$

Equivalence between the three-DOF model and the six-DOF constrained model

The focus is placed on the difference of Eq. (6) between the three-DOF and six-DOF cases. Under the

The force and torque contributed by the umbilical cable are expressed as:

$$\begin{bmatrix} \mathbf{f}_{fUm} \\ \mathbf{T}_{fUm} \end{bmatrix} = -K \begin{bmatrix} \mathbf{r}_f - \mathbf{r}_s \\ \boldsymbol{\theta}_f - \boldsymbol{\theta}_s \end{bmatrix} - C \begin{bmatrix} \dot{\mathbf{r}}_f - \dot{\mathbf{r}}_s \\ \dot{\boldsymbol{\theta}}_f - \dot{\boldsymbol{\theta}}_s \end{bmatrix} \quad (2)$$

The active control feedback force (exemplified by a PID controller) is expressed as:

$$\begin{bmatrix} \mathbf{f}_{fc} \\ \mathbf{T}_{fc} \end{bmatrix} = -K_p \begin{bmatrix} \mathbf{r}_f - \mathbf{r}_s \\ \boldsymbol{\theta}_f - \boldsymbol{\theta}_s \end{bmatrix} - K_d \begin{bmatrix} \dot{\mathbf{r}}_f - \dot{\mathbf{r}}_s \\ \dot{\boldsymbol{\theta}}_f - \dot{\boldsymbol{\theta}}_s \end{bmatrix} - K_a \begin{bmatrix} \ddot{\mathbf{r}}_f \\ \ddot{\boldsymbol{\theta}}_f \end{bmatrix} \quad (3)$$

Here, $K \in \mathbb{R}^{6 \times 6}$ denotes the stiffness matrix, $C \in \mathbb{R}^{6 \times 6}$ the damping matrix, $K_p \in \mathbb{R}^{6 \times 6}$ and $K_d \in \mathbb{R}^{6 \times 6}$ the proportional and derivative gains of the PID controller for relative displacement feedback, and $K_a \in \mathbb{R}^{6 \times 6}$ the absolute acceleration feedback gain. (Note: for single-degree-of-freedom decoupled control, K_p and K_d are diagonal; when coupling is taken into account, they may be set as non-diagonal. The acceleration gain K_a is applied only to the three translational degrees of freedom.)

Three-degree-of-freedom dynamics and control system model

On the ground, only two translational directions and the rotation about the perpendicular axis are controlled at the same time. Hence, the six DOFs are partitioned into controllable and constrained DOFs, with zero control forces and torques applied to the constrained ones.

Taking a plane in the $[x \ y \ \theta_z]$ coordinate system as an example, the vectors are reordered into constrained and unconstrained ones.

$$\mathbf{q}_c = [x \ y \ \theta_z]^T, \quad \mathbf{q}_e = [\theta_x \ \theta_y \ z]^T,$$

$$\mathbf{F}_c = [F_x \ F_y \ T_z]^T, \quad \mathbf{F}_z = 0$$

The six-DOF dynamics are expressed in a block form after rearrangement:

variables are higher-order small quantities, and their influence on the system's dynamic response can be neglected; hence, $\mathbf{Q}_c \rightarrow 0$ and $\mathbf{Q}_z \rightarrow 0$. When $\mathbf{F}_z = 0$, meaning that the output forces on the constrained degrees of freedom are zero, substituting the above conditions into the dynamic equations yields:

constraints, $M_{22}^{-1} \rightarrow 0$, and the dynamic equation for the controlled degrees of freedom can be written as:

$$M_{11} \ddot{\mathbf{q}}_c + C_{11} \Delta \dot{\mathbf{q}}_c + K_{11} \Delta \mathbf{q}_c = \mathbf{F}_c \quad (7)$$

However, if $M_{22}^{-1} \neq 0$, it can be written as

$M_{11}' = M_{11} - M_{12}M_{22}^{-1}M_{21}$, and similarly for C_{11}' and K_{11}' . The dynamic equation for the controlled degrees of freedom can then be written as:

$$M_{11}'\ddot{\mathbf{q}}_c + C_{11}'\Delta\dot{\mathbf{q}}_c + K_{11}'\Delta\mathbf{q}_c = \mathbf{F}_c \quad (8)$$

Following the rearrangement in Eq. (4), the mass-inertia, stiffness, and damping matrices are given by:

$$\begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & J_{\theta_z\theta_z} & J_{\theta_z\theta_x} & J_{\theta_z\theta_y} & 0 \\ 0 & 0 & J_{\theta_x\theta_z} & J_{\theta_x\theta_x} & J_{\theta_x\theta_y} & 0 \\ 0 & 0 & J_{\theta_y\theta_z} & J_{\theta_y\theta_x} & J_{\theta_y\theta_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & m \end{bmatrix},$$

$$\begin{bmatrix} K_{xx} & K_{xy} & 0 & 0 & 0 & K_{xz} \\ K_{yx} & K_{yy} & 0 & 0 & 0 & K_{yz} \\ 0 & 0 & K_{\theta_z\theta_z} & K_{\theta_z\theta_x} & K_{\theta_z\theta_y} & 0 \\ 0 & 0 & K_{\theta_x\theta_z} & K_{\theta_x\theta_x} & K_{\theta_x\theta_y} & 0 \\ 0 & 0 & K_{\theta_y\theta_z} & K_{\theta_y\theta_x} & K_{\theta_y\theta_y} & 0 \\ K_{zx} & K_{zy} & 0 & 0 & 0 & K_{zz} \end{bmatrix},$$

$$\begin{bmatrix} C_{xx} & C_{xy} & 0 & 0 & 0 & C_{xz} \\ C_{yx} & C_{yy} & 0 & 0 & 0 & C_{yz} \\ 0 & 0 & C_{\theta_z\theta_z} & C_{\theta_z\theta_x} & C_{\theta_z\theta_y} & 0 \\ 0 & 0 & C_{\theta_x\theta_z} & C_{\theta_x\theta_x} & C_{\theta_x\theta_y} & 0 \\ 0 & 0 & C_{\theta_y\theta_z} & C_{\theta_y\theta_x} & C_{\theta_y\theta_y} & 0 \\ C_{zx} & C_{zy} & 0 & 0 & 0 & C_{zz} \end{bmatrix}$$

Note: It is assumed by default that the coupling between the rotational and translational degrees of freedom in the stiffness and damping matrices of the umbilical cable is essentially zero or negligibly small.

Then, through matrix operations, the following can be derived:

$$M_{11} - M_{12}M_{22}^{-1}M_{21} = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & J_{\theta_z\theta_z} - [J_{\theta_z\theta_x} \ J_{\theta_z\theta_y}] \begin{bmatrix} J_{\theta_x\theta_x} & J_{\theta_x\theta_y} \\ J_{\theta_y\theta_x} & J_{\theta_y\theta_y} \end{bmatrix}^{-1} \begin{bmatrix} J_{\theta_x\theta_z} \\ J_{\theta_y\theta_z} \end{bmatrix} \end{bmatrix} = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & J'_{\theta_z\theta_z} \end{bmatrix}$$

$$K_{11} - M_{12}M_{22}^{-1}K_{21} = \begin{bmatrix} K_{xx} & K_{xy} & 0 \\ K_{yx} & K_{yy} & 0 \\ 0 & 0 & K_{\theta_z\theta_z} - [J_{\theta_z\theta_x} \ J_{\theta_z\theta_y}] \begin{bmatrix} J_{\theta_x\theta_x} & J_{\theta_x\theta_y} \\ J_{\theta_y\theta_x} & J_{\theta_y\theta_y} \end{bmatrix}^{-1} \begin{bmatrix} K_{\theta_x\theta_z} \\ K_{\theta_y\theta_z} \end{bmatrix} \end{bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} & 0 \\ K_{yx} & K_{yy} & 0 \\ 0 & 0 & K'_{\theta_z\theta_z} \end{bmatrix}$$

$$C_{11} - M_{12}M_{22}^{-1}C_{21} = \begin{bmatrix} C_{xx} & C_{xy} & 0 \\ C_{yx} & C_{yy} & 0 \\ 0 & 0 & C_{\theta_z\theta_z} - [J_{\theta_z\theta_x} \ J_{\theta_z\theta_y}] \begin{bmatrix} J_{\theta_x\theta_x} & J_{\theta_x\theta_y} \\ J_{\theta_y\theta_x} & J_{\theta_y\theta_y} \end{bmatrix}^{-1} \begin{bmatrix} C_{\theta_x\theta_z} \\ C_{\theta_y\theta_z} \end{bmatrix} \end{bmatrix} = \begin{bmatrix} C_{xx} & C_{xy} & 0 \\ C_{yx} & C_{yy} & 0 \\ 0 & 0 & C'_{\theta_z\theta_z} \end{bmatrix}$$

When the off-diagonal coupling terms of the inertia matrix are much smaller than the principal moments of inertia, it can be shown that:

$$\left| M_{11}' - M_{11} \right| \ll M_{11}, \quad \left| C_{11}' - C_{11} \right| \ll C_{11},$$

$$\left| K_{11}' - K_{11} \right| \ll K_{11}$$

On this basis, the zero-pole distributions of Eqs. (7) and (8) can be regarded as essentially equivalent. Hence, by using the same control parameters (e.g., PID) for the $\mathbf{F}_c$ control force as in the original case, the dynamic performance and stability of the 3-DOF model are virtually identical to those of the 6-DOF model.

Verification of the Six-DOF Equivalence via the Three-DOF Model

Assuming that the 3-DOF control system has been verified separately in each of the three states, $[x \ y \ \theta_z]$, $[x \ z \ \theta_y]$, and $[y \ z \ \theta_x]$, can it be inferred that the six-DOF model will be stable and that its performance will meet the requirements?

From Eq. (8), the 3-DOF dynamics for each of the three axes can be assembled into nine dynamic equations. By removing the redundant three axes, Eq. (4) can be rewritten as:

$$\begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & J_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & 0 & J_{\theta_y\theta_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & J_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{r}}_f \\ \ddot{\boldsymbol{\theta}}_f \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} & C_{xz} & 0 & 0 & 0 \\ C_{yx} & C_{yy} & C_{yz} & 0 & 0 & 0 \\ C_{zx} & C_{zy} & C_{zz} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{\theta_y\theta_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \dot{\mathbf{r}}_f - \dot{\mathbf{r}}_s \\ \dot{\boldsymbol{\theta}}_f - \dot{\boldsymbol{\theta}}_s \end{bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} & K_{xz} & 0 & 0 & 0 \\ K_{yx} & K_{yy} & K_{yz} & 0 & 0 & 0 \\ K_{zx} & K_{zy} & K_{zz} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & 0 & K_{\theta_y\theta_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & K_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \mathbf{r}_f - \mathbf{r}_s \\ \boldsymbol{\theta}_f - \boldsymbol{\theta}_s \end{bmatrix} = \begin{bmatrix} \mathbf{F}_x \\ \mathbf{F}_y \\ \mathbf{F}_z \\ \mathbf{T}_x \\ \mathbf{T}_y \\ \mathbf{T}_z \end{bmatrix}$$

Comparing with Eq. (1):

$$\begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & J_{\theta_x\theta_x} & J_{\theta_x\theta_y} & J_{\theta_x\theta_z} \\ 0 & 0 & 0 & J_{\theta_y\theta_x} & J_{\theta_y\theta_y} & J_{\theta_y\theta_z} \\ 0 & 0 & 0 & J_{\theta_z\theta_x} & J_{\theta_z\theta_y} & J_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{r}}_f \\ \ddot{\boldsymbol{\theta}}_f \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} & C_{xz} & 0 & 0 & 0 \\ C_{yx} & C_{yy} & C_{yz} & 0 & 0 & 0 \\ C_{zx} & C_{zy} & C_{zz} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{\theta_x\theta_x} & C_{\theta_x\theta_y} & C_{\theta_x\theta_z} \\ 0 & 0 & 0 & C_{\theta_y\theta_x} & C_{\theta_y\theta_y} & C_{\theta_y\theta_z} \\ 0 & 0 & 0 & C_{\theta_z\theta_x} & C_{\theta_z\theta_y} & C_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \dot{\mathbf{r}}_f - \dot{\mathbf{r}}_s \\ \dot{\boldsymbol{\theta}}_f - \dot{\boldsymbol{\theta}}_s \end{bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} & K_{xz} & 0 & 0 & 0 \\ K_{yx} & K_{yy} & K_{yz} & 0 & 0 & 0 \\ K_{zx} & K_{zy} & K_{zz} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{\theta_x\theta_x} & K_{\theta_x\theta_y} & K_{\theta_x\theta_z} \\ 0 & 0 & 0 & K_{\theta_y\theta_x} & K_{\theta_y\theta_y} & K_{\theta_y\theta_z} \\ 0 & 0 & 0 & K_{\theta_z\theta_x} & K_{\theta_z\theta_y} & K_{\theta_z\theta_z} \end{bmatrix} \begin{bmatrix} \mathbf{r}_f - \mathbf{r}_s \\ \boldsymbol{\theta}_f - \boldsymbol{\theta}_s \end{bmatrix} = \begin{bmatrix} \mathbf{F}_x \\ \mathbf{F}_y \\ \mathbf{F}_z \\ \mathbf{T}_x \\ \mathbf{T}_y \\ \mathbf{T}_z \end{bmatrix}$$

Evidently, the only difference between Eq. (4) and Eq. (1) lies in the cross-coupling matrix associated with the rotational terms. Since the acceleration control terms for active isolation act primarily along the translational degrees of freedom, Eq. (4) is able to completely mimic Eq. (1) in the translational directions.

A rigorous mathematical proof of the equivalence, however, rests on the following premises:

(1) The off-diagonal coupling terms in the stiffness and damping matrices between translational and rotational DOFs are negligible compared with the diagonal (translational-only and rotational-only) terms. This is supported by our previous experimental results and by the literature (Wang et al. 2026).

(2) The angular velocity of the stator is very small, so that the centrifugal and Coriolis forces can be neglected, thus $\mathbf{Q} = 0$. This condition is satisfied in practice, since the angular rate of high-precision pointing satellites is typically on the order of arcseconds per second (Wang et al. 2026).

(3) In air-bearing tests, the added mass alters the actual mass-inertia matrix relative to the original; however, this discrepancy can be eliminated by adding counterweights.

In summary, the proposed method is valid for static-base, high-precision pointing missions where the attitude angular velocity is negligibly small (approaching zero). However, it is not suitable for scenarios involving large angular-rate maneuvers. Given that this condition is satisfied, we can reasonably conclude that, from an engineering experimental standpoint, the three-axis three-DOF air-bearing tests adequately emulate the dynamics and control performance of the full six-DOF system in space.

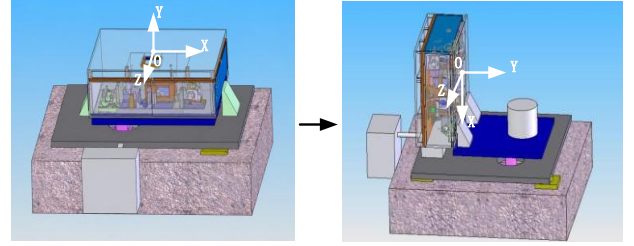


Fig. 7 Schematic of rotation directions

Experimental Verification of the Ground-Based Air-Bearing System

To address the three key scientific questions raised in Section 1, we design the following experiments: a static control test for equivalence validation, an excitation test for evaluating the system's disturbance rejection capability, and a long-term test for analyzing system stability.

Static Active Vibration Isolation Test

The main goal of the multi-axis tests is to assess the effect of vibrations from the ground and the air-bearing system itself on the static performance of the active vibration isolator. The setup is initially placed horizontally for $[x \ z \ \theta_y]$ direction isolation testing; after the criteria are satisfied, the platform is rotated to the $[x \ y \ \theta_z]$ and $[y \ z \ \theta_x]$ orientations for integrated six-DOF evaluation, as shown in Fig. 7. Two control approaches are compared: displacement-based control and a dual-loop pulse-averaging strategy that incorporates both acceleration and displacement feedback. The results are presented in figures for direct comparison.

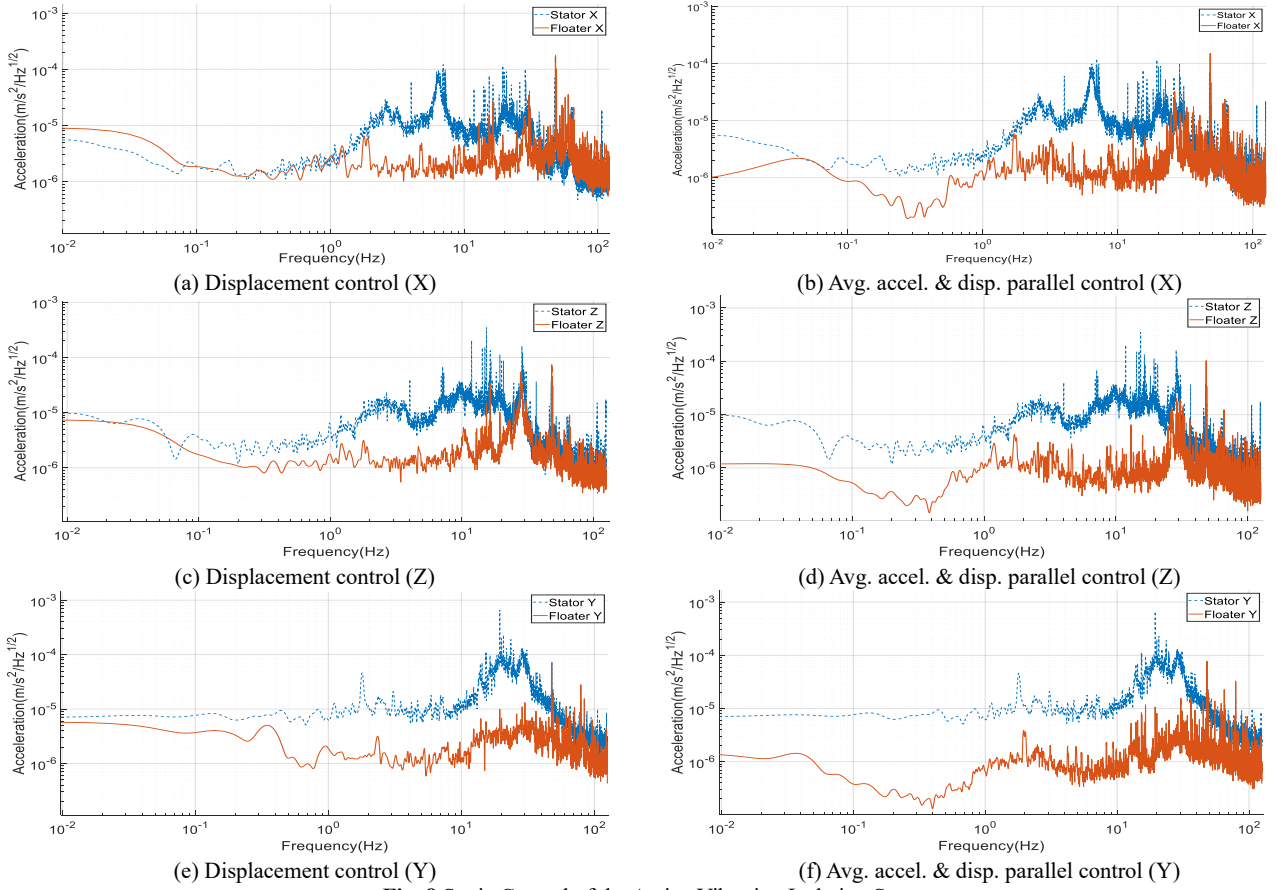


Fig. 8 Static Control of the Active Vibration Isolation System

Fig. 8(a), (c), and (e) show the results of single-loop displacement feedback control, whereas Fig. 8(b), (d), and (f) present those of the dual-loop strategy that combines acceleration feedback with displacement feedback via pulse averaging. In the low-frequency range (0.01 - 1Hz), the stator's vibration noise is approximately $2 \times 10^{-6} m/s^2 / \sqrt{Hz}$; in the mid-frequency band (1 - 20Hz), the dominant suppression band, the noise level rises above $1 \times 10^{-5} m/s^2 / \sqrt{Hz}$. When the platform is rotated to the $[x \ y \ \theta_z]$ and $[y \ z \ \theta_x]$ orientations, however, the larger vertical projection area increases its sensitivity to ambient airflow disturbances, pushing the noise level above $1 \times 10^{-5} m/s^2 / \sqrt{Hz}$ across the entire 0.01-20 Hz band-although this can be mitigated by the active isolation controller.

Single-loop displacement control

In the single-loop displacement feedback control scheme, the system relies primarily on displacement sensor signals to regulate its isolation response. Within the 0.1-20Hz band, the vibration amplitudes of the floater in all directions are reduced to between $1 \times 10^{-6} m/s^2 / \sqrt{Hz}$ and $3 \times 10^{-6} m/s^2 / \sqrt{Hz}$. The closed-loop positioning accuracy is sufficient only in the relatively low frequency range of 0.01-1Hz. However, mechanical coupling from the umbilical cable introduces additional displacement disturbances, which significantly degrade system performance in the low-to-mid frequencies. Consequently, displacement control alone is insufficient to meet high-performance isolation criteria across the full spectrum.

Dual-loop average-acceleration and

displacement parallel control

To overcome this limitation, an additional loop with pulse-averaged acceleration feedback is incorporated into the original displacement controller, as illustrated in Fig. 8(b), (d), and (f). With a band-pass filter of 0.1-20 Hz, the floater vibrations in all directions are reduced to $2 \times 10^{-7} m/s^2 / \sqrt{Hz}$ in the 0.1-1 Hz band and below $1 \times 10^{-6} m/s^2 / \sqrt{Hz}$ in the mid-frequency range. This scheme delivers accurate acceleration outputs across the effective frequency band, thereby not only improving the rejection of low-frequency disturbances caused by the umbilical cable, but also compensating for the shortcomings of displacement control in the low-to-mid frequencies. Nevertheless, the active isolation performance degrades at high frequencies due to the increasing dynamic stiffness of the air bearing. The successful experimental results confirm that, by validating each translational direction separately, the system effectively emulates three-axis microgravity conditions and achieves a high precision of $2e-7g$ in a multi-DOF ground environment.

Active Vibration Isolation Test Under Disturbance

In subsequent experiments, a modal shaker was used to excite the stator with white-noise excitation spanning 0.1-250 Hz, raising the vibration level above $1 \times 10^{-4} m/s^2 / \sqrt{Hz}$ for frequencies above 10 Hz to simulate realistic disturbance conditions. The suppression ratio between the stator and the floater was then measured. These tests allowed us to verify the isolation performance of the dual-loop pulse-averaging control scheme (combining acceleration and displacement feedback) in a

practical environment, with particular emphasis on its broadband response characteristics.

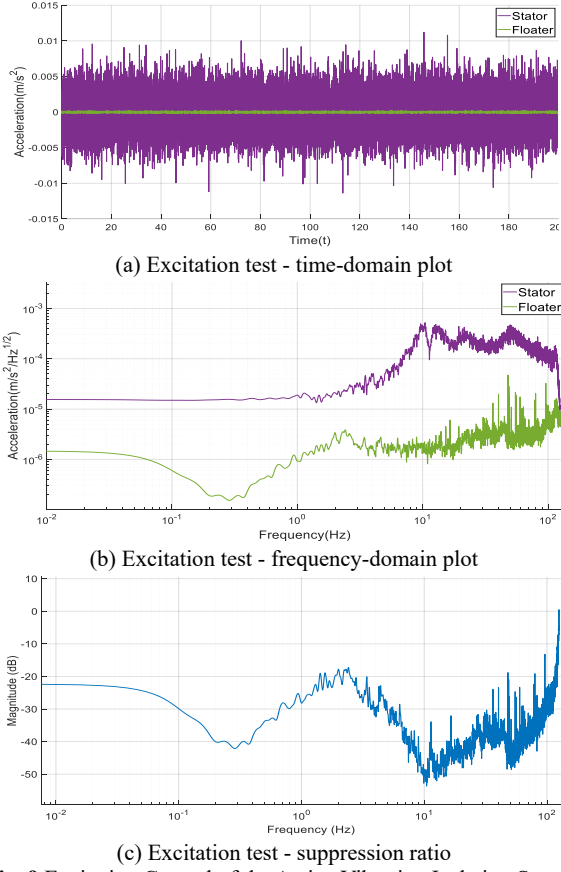
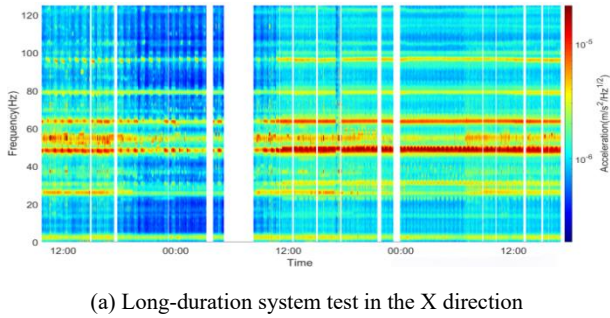


Fig. 9 Excitation Control of the Active Vibration Isolation System



The overall vibration suppression ratios of the isolation system in the three translational directions under white-noise excitation are presented in Fig. 9. Evidently, even with large-amplitude stator vibrations, the floater remains stable, and the suppression ratio stays below -20 dB across the entire frequency range. At high frequencies, passive isolation effectively attenuates the vibrations; in the 0.1-1 Hz low-to-mid band, the active controller further reduces the noise to $2 \times 10^{-7} \text{ m/s}^2 / \sqrt{\text{Hz}}$, corresponding to a 100-fold attenuation. The most pronounced suppression occurs in the 5-20 Hz mid-frequency region, where the suppression ratio reaches as high as -50 dB. These results demonstrate that the system provides excellent full-spectrum vibration isolation and maintains reliable operation even under strong external disturbances. The system exhibits outstanding stability and precision in the presence of dynamic vibration disturbances, and offers a marked improvement over conventional vibration isolation devices in terms of suppression performance.

Long-Term System Stability Test

To verify the long-term stability and reliability of the air-bearing active vibration isolator for future space microgravity missions, a 55-hour endurance test was carried out under the $[x \ z \ \theta_y]$ axis configuration, and the results are presented in Fig. 10. The vertical axis denotes frequency (Hz), the horizontal axis denotes time (days/hours), and the color scale represents the acceleration power spectral density (PSD, in $\text{m/s}^2 / \text{Hz}^{1/2}$), with red indicating strong vibrations and blue indicating weak ones. The overall spectrogram shows a stable background frequency pattern, along with distinct energy concentrations at several discrete frequencies. Although some data gaps occurred due to software-related acquisition issues (visible as vertical white bands in the figure), these did not compromise the normal functioning of the isolator or the assessment of its vibration suppression performance.

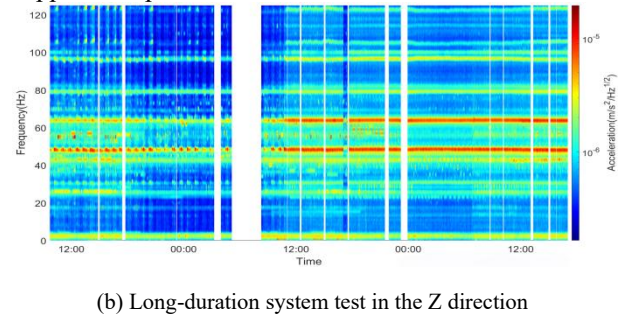


Fig. 10 Active Isolator Long-Duration Test

Throughout the entire frequency spectrum, the active vibration isolator shows clear frequency-dependent behavior and environmental sensitivity. In the low-frequency region (<20 Hz), where ambient disturbances are most prominent, the measured energy levels remain relatively low, confirming that the device effectively attenuates low-frequency vibration transmission—though intermittent energy bands still appear, indicating that such disturbances are reduced but not entirely suppressed. In the mid-frequency band (20-60 Hz), the spectrogram reveals stable striated features, with a pronounced energy concentration near 40 Hz that points to coupling between

the system and its surroundings. A comparison of the two test phases also shows lower overall energy in the earlier phase, suggesting better isolation performance under weaker ambient disturbances. At high frequencies (>60 Hz), the energy distribution becomes sparse, with mostly blue-green tones, implying limited external influence and a natural isolation advantage of the device. Consequently, optimization efforts should be directed primarily toward the low- and mid-frequency bands.

Environmental factors also exert a notable influence on system performance. The long-duration test reveals periodic vibration fluctuations with a cycle of about 40

minutes, which are traced back to transient excitations caused by the air pump's intermittent replenishment of the air-bearing support system—disturbances that propagate through the building floor and affect the system's vibration environment. Furthermore, the continuous 3-day-and-2-night monitoring clearly demonstrates a day-night pattern: isolation performance during nighttime hours (10 PM - 7 AM) is consistently better than during the daytime, primarily owing to the lower ambient vibration background and the marked reduction in human activity - related disturbances.

In conclusion, the active vibration isolator demonstrates excellent stability and reliability over extended operation, effectively attenuating ambient disturbances in both the low- and mid-frequency ranges. Moreover, the diurnal variation and periodic disturbance characteristics have been clearly revealed, providing a solid basis for future test campaigns.

Conclusion

This paper addresses the challenge of ground-based validation for active vibration isolation systems intended for space microgravity applications. A microgravity emulation testbed supported by air bearings is developed, along with a 6-DOF dynamic model that includes mass, inertia, stiffness, and damping coupling terms. The coupling behavior of the 6-DOF air-bearing system and the effects of external disturbances on simulation fidelity are thoroughly analyzed. Through examination of the constrained DOF responses, it is shown that the coupling from non-target DOFs to the target translational DOFs is sufficiently small under air-floating conditions, allowing 3-DOF equivalent simulation of space microgravity on the ground. By reorienting the air-bearing platform, the dynamic characteristics along different axes can be evaluated independently. An experimental setup is then built, and a series of tests are carried out to confirm the 6-DOF isolation capability of the system and its full-spectrum disturbance rejection performance under external vibration inputs. Long-duration tests are also conducted to assess system reliability and to characterize the ambient disturbance patterns in the test environment. The findings establish a solid experimental basis for subsequent ground performance verification and control strategy studies for space-bound active vibration isolation systems.

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Data Availability Not applicable.

Declarations

Competing Interests The authors declare no competing interests.

Ethics Approval Not applicable.

Consent to Participate Not applicable.

Conflicts of Interest The authors declare that they have no conflict of interest.

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