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Full Text

Preamble

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Abstract

In this paper, an effective active vibration control method was investigated to further improve the positioning accuracy of the Five-hundred-meter Aperture Spherical radio Telescope (FAST) feed cabin. Actual operation data from FAST were collected to analyze the vibration characteristics of the feed cabin in multiple directions. A simplified model of the cabin-cable system was established to evaluate the effects of a mass damper on different vibration frequencies and modes. On this basis, an active mass damper system and control system were designed for the cabin with multiple degrees of freedom and modal variation characteristics. Theoretical calculation and simulation proved that the system has a significant effect on improving the damping of the cabin-cable system and suppressing vibration of the FAST feed cabin.

Key words: telescopes – methods: data analysis – instrumentation: miscellaneous – techniques: miscellaneous – Astronomical Instrumentation – Methods and Techniques

1. Introduction

The Five-hundred-meter Aperture Spherical radio Telescope (FAST) is the largest and most sensitive single-aperture radio telescope [Figure 1: see original paper]. Since its opening in 2020, FAST has maintained stable and efficient operation, achieving numerous important detections across scientific targets including pulsars, radio bursts, and neutral hydrogen. Meanwhile, the engineering team has been exploring technical enhancements to improve the telescope's competitiveness in frontier astronomical research, such as developing new receivers to increase survey efficiency and expand observation frequency ranges, and applying new measurement and control technologies to improve observing sensitivity and pointing accuracy.

The FAST feed cabin [Figure 2: see original paper] is suspended by six long-span parallel cables. This cable system exhibits complex dynamics and is susceptible to pulsating wind disturbances. Precise positioning of the feed inside the cabin is achieved through three-stage series control. First, the feed cabin is towed by six cables, controlling the cabin's position and attitude accuracy within 48 mm and 1°, respectively. Second, the AB-axis mechanism, which can rotate around two orthogonal axes inside the cabin, compensates for deviation between the feed's attitude under optimized cable forces and the expected observation attitude. Finally, the Stewart platform inside the cabin compensates for residual error in real time, maintaining feed positioning accuracy better than 10 mm RMS (Nan 2006; Tang et al. 2011; Jiang et al. 2019, 2020). The current feed control scheme fully considers the dynamic characteristics of adjustment mechanisms at all levels and effectively avoids resonance risk in multi-stage control of flexible structures by restricting control bandwidth, enabling the telescope to achieve expected observational sensitivity and pointing accuracy in the L/S band (Qian et al. 2020).

Analysis of FAST's actual operation data reveals that residual positioning error of the feed primarily originates from feed cabin vibration caused by the long-span cables. Therefore, suppressing cabin vibration represents a new approach to further improve feed positioning accuracy. Under the existing feed support scheme, restraining vibration by adjusting structural parameters alone is difficult, necessitating consideration of additional vibration control equipment. By increasing damping in the cabin-cable system, vibration at each modal frequency can be directly reduced, while the response bandwidth of the original closed-loop controller can be increased to further enhance positioning accuracy.

Mass dampers are widely used in engineering and are generally categorized as tuned mass damper (TMD), active mass damper (AMD), and semi-active tuned mass damper (SATMD). The mass damper connects as a substructure to the main structure where vibration suppression is required. TMD [FIGURE:3(a)] relies on mechanical springs and dampers to suppress specific vibration modes. AMD [FIGURE:3(b)] generates force through a driving mechanism according to actual vibration of the main structure, enabling suppression of a wide range of vibration modes. SATMD [FIGURE:3(c)] is primarily passive, but the substructure can be adjusted within a limited range to produce different forces and adapt to variation in vibration frequency.

Specific to FAST feed cabin application, the damping and stiffness characteristics present unique challenges: the cabin exhibits six degrees of freedom in space. The workspace is a spherical cap located about 140 m above the reflector bottom, with an aperture exceeding 200 m [Figure 4: see original paper]. As the feed cabin position changes, cable network length and shape vary, leading to constant changes in vibration modes and frequencies. The natural frequencies of the cabin-cable system are very low, primarily concentrated below 1 Hz, with the first-order resonance frequency even below 0.2 Hz. The vibration modes are complex and frequencies constantly changing, making TMD effectiveness very slight during most operating time. Due to these low vibration frequencies, any mass damper with spring elements would struggle to maintain the mass block's equilibrium position. Therefore, AMD is the optimal choice due to its active actuation capability to adapt to changes and maintain mass block equilibrium.

Lefteris Koutsoloukas and Nikolaos Nikitas (Koutsoloukas et al. 2022) summarized 208 application cases of mass dampers on large buildings or facilities, with AMDs accounting for 4%. AMD application difficulties lie in requiring external energy to generate driving force, and achieving good vibration suppression requires maintaining stability and reducing control system time lag, which increases application cost. However, under perfect application conditions, AMD demonstrates strong robustness and flexibility across various disturbances and responses, and can achieve optimal control effects by selecting appropriate control algorithms for different control objectives.

In vibration suppression studies of the FAST feed support system, Su and Duan designed an electrorheological fluid damper for fluctuating wind vibration of the cables (Su et al. 2003). The damper, installed on a long section of suspension cable near the cabin, was controlled by a wind speed measuring device to counteract wind disturbance. The damper could reduce cable vibration in specific modes, thereby indirectly reducing cabin vibration. However, the configuration and dynamic changes of the cabin-cable system were not considered, preventing analysis and verification of damper effectiveness in complex and changing operating environments.

To suppress feed cabin vibration around first-order resonant frequencies, Dong and Duan designed a set of multi-tuned mass dampers installed on the feed cabin (Dong et al. 2002). The main purpose was to suppress pulsating wind interference. Each mass damper only suppressed horizontal vibration in a specific mode and could not cope with frequency changes during operation. Furthermore, installing more TMDs would inevitably increase cabin mass, affecting observation quality.

In this paper, vibration directions that significantly affect observations are first determined by collecting and analyzing actual FAST operating data. On this basis, an AMD system suitable for the multi-degree-of-freedom and frequency variation characteristics of the feed cabin is designed. Finally, its effect on improving cabin-cable system damping and suppressing feed cabin vibration is verified through theoretical calculation and simulation analysis.

2. Vibration Data Analysis

The feed cabin follows different trajectories and positions across observation cases, but vibration characteristics remain similar. Therefore, a typical operating data set is selected, with vibration data extracted and transformed from the telescope's global coordinates to the cabin's local coordinates. On this basis, vibration can be decomposed along the primary optical axis direction to determine influence on astronomical observation.

This paper analyzes a FAST observation record in tracking mode on 2022 September 22. The tracking lasted approximately 1500 s. The data set contains complete time-series motion information of the feed cabin , including planned and actual positions in the telescope's global coordinate system, as well as planned and actual attitude angles rotating along axes in the feed cabin's local coordinate system.

As shown in [Figure 5: see original paper], FAST's global coordinate system is defined with the active reflector's spherical center as origin, north direction as X-axis, east direction as Y-axis, and zenith direction as Z-axis. Two local coordinate systems exist in the feed cabin [Figure 6: see original paper]. The first is fixed on the feed cabin, with the cabin's mass center as origin and Z-axis pointing to the cabin top; X and Y-axis directions follow the right-hand rule. Rotation angles along the X, Y, and Z axes of this local coordinate system are defined as Roll, Pitch, and Yaw angles, respectively. The second is fixed on the feed's phase center and is parallel to the first in its initial state.

During observations, two important directions are defined. As shown in [Figure 5: see original paper], the primary optical axis is defined as a straight line through the feed's phase center and the active reflector's spherical center, indicating the telescope's pointing direction. The focal plane is perpendicular to the primary optical axis and passes through the feed's phase center. Therefore, feed cabin vibration can be decomposed from global coordinates to this axial-plane combination to analyze influence on telescope pointing.

As shown in [Figure 7: see original paper], translation error of the feed cabin is derived by calculating the difference between theoretical and actual coordinates in the global coordinate system. The feed's phase center is not at the feed cabin's centroid; attitude error of the feed cabin transfers to the receiver feed to produce position deviation. When the cabin's attitude angle changes, rotation error around the Z-axis in the local coordinate system is less than 10^{-6} rad, which does not affect observation and is ignored in analysis. In rotation error analysis, only error caused by rotation of Pitch and Roll angles is considered.

[Figure 8: see original paper] shows angle error of the feed cabin. The rotation error is obtained from the difference between actual and planned attitude angles as shown in the figure. As shown in [Figure 7: see original paper] and [Figure 8: see original paper], during the initial 600 s of operation, the system transitions from high-speed source changing to tracking observation. The large inertia of the cabin-cable system and integral controller caused oscillation requiring more time for suppression. Therefore, data after 1000 s is selected when the cabin is in smooth tracking status to analyze vibration caused by cabin-cable dynamics.

2.1. Translational Error

The feed cabin can be treated as a rigid body relative to the long-span cables. Its translation directly transfers to the feed inside the cabin, representing one of the main sources of feed positioning errors. To further analyze translational error influence on FAST observation accuracy, translational error should be decomposed from the global coordinate system to the focal plane and primary optical axis in the local coordinate system.

As shown in [Figure 9: see original paper], the spatial error vector of the feed's phase center (a) extends from theoretical to actual position, and the primary optical axis vector (b) extends from the global coordinate system origin to the theoretical position. The angle between the spatial error vector and primary optical axis can be obtained using the vector angle formula. The length of the spatial error vector can be calculated. The component along the primary optical axis is L cos and the component in the focal plane is L sin . Therefore, translation error is decomposed on the primary optical axis and focal plane as shown in [Figure 10: see original paper].

2.2. Rotational Error

Attitude error of the feed cabin also transmits to the feed, introducing additional displacement. Therefore, analyzing rotational error influence on observation accuracy and range is necessary. Due to small angles, the receiver platform rotation can be approximated as linear displacement, also called angular displacement. Following the previous attitude angle definition, total angular displacement error L_1 caused by angle error around X and Y axes can be calculated through the formula:

$$L_1 = l\sqrt{(\Delta\mathrm{Pitch})^2 + (\Delta\mathrm{Roll})^2}$$

In this equation, l is the distance between the two local coordinate system origins (from cabin mass center to feed phase center), with a value of 1.75 m. ΔPitch and ΔRoll represent differences between planned and actual attitude angles of the feed cabin. The angular displacement error caused by rotation around each axis and the total angular displacement error synthesized by vector are shown in [Figure 11: see original paper].

The steady-state error of total angular displacement error is first eliminated by Fourier transform. Then the total angular displacement error is decomposed toward the primary optical axis and focal plane as shown in [Figure 12: see original paper]. The cabin mass center (O_C) connects to the feed phase center by a virtual connecting rod of length l = 1.75 m. F_1 represents the feed's planned position and F_2 represents actual position. L_1 is the distance between F_1 and F_2, given by the equation above. The angular displacement is decomposed to the primary optical axis and focal plane, with angle c between the vector from actual to theoretical position and the focal plane. From the diagram relationship, angle c is half the angle between F_1 and F_2: arcsin(L_1/(2l)). Furthermore, the component l_{z1} of angular displacement on the primary optical axis can be obtained as l_{z1} = L_1 sin c, and the component l_{j1} on the focal plane is l_{j1} = L_1 cos c.

2.3. Frequency Domain Analysis

To compare translational error and angular displacement error influence on main vibration frequency, errors decomposed on the primary optical axis and focal plane are transformed to the frequency domain. After removing steady-state error, the results are shown in [Figure 14: see original paper], with maximum amplitudes listed in .

As shown in , the maximum feed positioning error values in two directions are 1.473 mm and 1.305 mm, both caused by translation error. These two directions have comparable values and represent the main error sources. Angular displacement error also affects feed positioning error, primarily in the focal plane direction.

3. Mass Damper Scheme

Based on error analysis from the preceding sections, the following conclusions can be synthesized: (1) Both translational error and angular displacement error influence feed cabin observation accuracy, but the feed is more affected by translational vibration error. AMD effectiveness depends critically on suppression of translational error. (2) A single mass damper with multiple degrees of freedom is highly complex and difficult to achieve. Therefore, the multi-directional nature of vibration determines that AMD number and position should not be singular. (3) Due to space and weight constraints in the cabin, AMD cannot be installed along both focal plane and primary optical axis directions.

After weighing advantages and disadvantages, the optimum solution distributes three AMD sets spaced 120° apart at the maximum diameter of the cabin's outer edge. As shown in [Figure 15: see original paper], masses move along guides parallel to the primary optical axis. Mass movement is controlled by a linear drive mechanism, eliminating need for mechanical springs and additional dampers. This not only simplifies structure and saves space but also improves stability. This design can directly and effectively suppress translation errors in the primary optical axis direction and minimize angular displacement errors caused by cabin rotation. Since the cabin exhibits compound motion, vibration in the focal plane will also be suppressed to a certain range after the mass damper increases system damping.

4. Effect Analysis of Mass Damper in Feed Cabin

To analyze damper vibration-suppressing capability and preliminarily determine reasonable damper parameters, a simplified multi-DoF model of the cabin-cable system is established with an AMD installed. The overall model is shown in [Figure 16: see original paper], where m_1 is 15 t representing long-span cable mass, m_2 is 30 t representing cabin mass, and m is damper mass at 1/10 of cabin mass (3 t). The first-order natural frequency of the FAST feed cabin support system is 0.18 Hz and the damping ratio is 0.002. System stiffness and damping factor are 50,000 N m^{-1} and 500, respectively.

Based on force analysis of three mass blocks, the three displacements are merged into a vector. The dynamic equation of the system without AMD is established as:

$$M\ddot{x} + C\dot{x} + Kx = f$$

where f is external force applied on each mass. The first-order and second-order undamped natural frequencies of the model can be calculated by solving the characteristic equations, yielding 0.18 Hz and 0.42 Hz respectively. The first-order damping ratio is 0.002 and second-order damping ratio is 0.005, equivalent to the FAST prototype.

To verify AMD effect with velocity feedback, the system dynamics equation after adding AMD is established:

$$M\ddot{x} + C\dot{x} + Kx = F_{\text{AMD}} + f$$

where $F_{\text{AMD}} = -G_v\dot{x}$. To analyze system response, the equation is transformed into state space form:

$$\dot{z} = Az + Bu$$

where G_v is velocity feedback gain, k_a is elastic coefficient, and c_a is damping coefficient of AMD. In this paper, AMD design depends only on linear drive mechanism to control mass block motion, so c_a and k_a values are 0, and velocity feedback gain $G_v = 500$.

Comparison of system response between original system and system equipped with velocity feedback AMD is shown in [Figure 17: see original paper]. When velocity feedback is used, the first-order damping ratio becomes 0.0451 and second-order damping ratio becomes 0.0121. The first-order damping ratio is approximately 19 times higher, and the second-order damping ratio about 2 times higher than the original system. These results demonstrate that AMD with velocity feedback is effective, improving first-order and second-order damping ratios of the feed cabin to help the cabin reach steady-state more quickly.

5. Design and Simulation of the Vibration Platform for Feed Cabin

To further verify vibration-suppressing effect of the mass damper system on multiple vibration directions of the feed cabin, an indoor simplified vibration platform is designed [Figure 18: see original paper]. The AMD design aims to restrain feed cabin translation in the primary optical axis direction and rotation around X and Y axes. To simplify feed cabin vibration modes, a rigid bracing mechanism with three degrees of freedom replaces the prototype cable-cabin structure. The support mechanism uses a Hooke's hinge to constrain feed cabin rotation along the horizontal axis. The cabin mass center is adjusted slightly below the Hooke's hinge, allowing the cabin to vibrate freely as a single pendulum. The Hooke's hinge connects to the support structure with a spring, enabling vertical vibration. Main parameters of the vibration platform are shown in .

Vertical translation frequency can be adjusted by spring stiffness, while rotation frequency along the horizontal axis can be adjusted by distance between the Hooke's hinge center and cabin mass center.

5.1. Simulation Model of Vibration Platform for Feed Cabin

To investigate required parameters and vibration suppression effect of AMDs on the feed cabin vibration platform, simulation was performed before physical modeling experiments. The SimMechanics toolkit in MATLAB/SIMULINK environment was used to build the whole system simulation model. Its operation flow chart is shown in [Figure 19: see original paper]. The whole simulation model includes the feed cabin platform without mass damper and the cabin platform with AMD. In the second part, the cabin platform is attached to an AMD module, plus two measurement modules and AMD feedback controller.

Parameters of the two feed cabins are identical to compare vibration before and after AMD installation. The mass damper model consists of three mass blocks connected to the cabin platform through linear drivers. Two measurement modules provide feedback signals to the damper controller: a velocity measurement module outputs platform vibration speed, and a displacement measurement module outputs damper mass block displacement relative to the cabin platform.

As shown in [Figure 20: see original paper], the feedback controller calculates driving force applied to mass blocks according to feedback signals. Feedback gain coefficient is determined based on maximum vibration amplitude and maximum mass moving range. With maximum mass guide length of 300 mm in the simulation model, travel feedback gain restricts mass block displacement within this limit. While satisfying mass block displacement constraints, larger velocity feedback gain enables faster return to steady-state. Here displacement feedback gain is -0.05 and speed feedback gain is 5.

Through the disturbance generator, excitation is created to make the vibration platform model equivalent to actual cabin amplitude. With and without AMD, cabin angular error and translation error in the primary optical axis direction are output to visually verify AMD effectiveness.

5.2. Simulation Results

First, an impulse disturbance of identical amplitude is input to the model group. Simulation results of feed cabin rotational error and translation error in the primary optical axis direction, with and without AMD, are shown in [Figure 21: see original paper] and [Figure 22: see original paper].

Next, actual disturbance forces from four different operating modes ("Tracking," "SwiftCalibration," "SnapShotdec," and "OnTheFlyMapping") up to 150 s long are solved by inverse dynamics from observation record data and input to the disturbance actuator to simulate environmental pulsating wind interference. Force components include cabin force in the primary optical axis direction and force rotating the cabin around X and Y axes. Simulation results of the cabin's two error types in the primary optical axis direction, with and without AMD, are shown in [Figure 23: see original paper] and [Figure 24: see original paper].

Calculating RMS values of error signals with and without AMD on the platform yields the results shown in .

[Figure 21: see original paper] and [Figure 22: see original paper] show that under impulse interference, vibration decay time of the AMD-equipped cabin model is significantly reduced. While the original system requires about 400 s to reduce vibration below 10^{-4} m relying on its lower damping ratio, the AMD-equipped system takes less than 90% of that time to achieve the same reduction. Combining four different operation modes, under disturbing forces solved by inverse dynamics, the AMD-equipped system reduces translational error by 55% and rotational error by 60% compared to the original system. Angular displacement error can be obtained from rotational error, and overall error can be obtained by combining translational and rotational errors in orthogonal directions. The AMD-equipped system reduces overall error to 41% of the original system.

Simulation results demonstrate that AMD significantly reduces translational and rotational errors on the primary optical axis, verifying AMD effectiveness.

6. Conclusions

This paper analyzed FAST feed cabin operating data. Through quantitative comparison of translational and rotational errors in the feed cabin's local coordinate system, we determined that the cabin is more affected by translational error. Combined with the feed cabin working environment, a mass damper system including three AMDs was designed, spaced 120° apart at the maximum diameter of the cabin's outer edge.

A simplified cabin-cable system model with three degrees of freedom was established with an AMD installed to analyze damper vibration suppression capability. Analysis results show that velocity feedback can significantly restrain low-frequency cabin vibration.

A simplified simulation model of the feed cabin vibration platform was established to analyze mass damper system effectiveness on multiple vibration directions. Results show AMD has significant suppression effect on vibration in the primary optical axis direction and rotation error around X and Y axes caused by pulsating wind. After AMD installation, time to restore stability is reduced to about 10% of the original system. When facing continuous interference forces, AMD can reduce primary optical axis translational error to 45% of the original system and X/Y axis rotational error to 40% of the original system. Overall error was reduced to 41% of the original system. Simulation experiments successfully verified AMD effectiveness.

This paper provides reference for FAST 2nd generation feed cabin design, which will include AMD installation space with weight reduction. Future research will continue designing AMD with better suppression effects. Based on the existing design, an active variable gain mechanism for the AMD control system

(Nagashima 1997) will be added to resolve the contradiction between mass block stroke limitation and vibration suppression effectiveness.

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