Experimental Validation on Active Vibration Reduction System for 2.4m×2.4m Transonic Wind Tunnel Model

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Abstract. An active vibration reduction system has been developed to solve pitch vibration problems of the force test models in 2.4m×2.4m transonic wind tunnel. The twin flexure hinges structure types and PD control algorithm are used in the system. The constitution of active vibration reduction system is presented. The results of Ground Tests and wind tunnel tests are given. The influence of the system on the static signal of the balance and its performance are discussed. The ground tests show that the system can increase the damping of the sting more than tenfold. It is shown that the vibration amplitude decreases by more than 70% in wind tunnel tests. The wind tunnel tests also reveal that the active vibration reduction system has relatively strong performance on the suppression of the model vibration.

1. Introduction
The tail support is a force-measuring model support usually used in 2.4m Transonic Wind tunnel. It has the characteristics of low support frequency and small structural damping. Under the condition of air separation on the surface of the model, vibration of the model and the support system will inevitably occur [1], especially the vibration of the balance and the support rod combination. It will seriously affect the quality of test data and test safety. Effective vibration reduction measures will help to improve the quality of wind tunnel test data and broaden the range of test attack angle.

It is well known that the amplitude of a model under unsteady aerodynamic excitation is inversely proportional to the product of stiffness of the support and structural damping. Increasing the bracing stiffness helps to reduce the vibration amplitude of the model, but it is difficult to increase the bracing stiffness substantially due to the restriction of bracing material, diameter and bracing interference requirements. Increasing support damping, especially support rod damping, is an effective method to reduce model vibration. Literature [2] also emphasizes that the key to solve the problem of model vibration of high dynamic pressure transonic wind tunnel is to greatly increase the support rod damping.

The passive damping principle is described in detail in Freymann’s research [3]. Installation of passive tuned mass damper TMD [4] at the head of the model or viscoelastic material [5] around the support rod is limited by the installation space and damping material characteristics, and its vibration reduction effect is not ideal.

With the emergence of small size, large output force and fast response stacked piezoelectric ceramic actuator, it is possible to greatly increase the damping of support rod by using active control technology. The basic principle is to generate reverse dynamic bending moment under voltage excitation by...
installing a piezoelectric ceramic actuator at a local position of support rod and utilizing its inverse piezoelectric effect. So as to restrain the vibration of the model [6]. The European Transonic Wind Tunnel (ETW) has 14 piezoceramic actuators looped between the balance and the strut to reduce all degrees of freedom except model roll. Because the actuator is close to the center of the model and the driving force is small, a small-size low-voltage piezoelectric ceramic actuator can achieve good vibration reduction effect [7]. The piezoelectric actuators of Sting-Tip Damper and Sting-Top Damper developed by the National Transonic Wind Tunnel (NTF) in the United States have the same arrangement and operating principle as that of ETW. Wind tunnel tests verify their vibration suppression capability under most test conditions. In this paper, the composition of active vibration reduction system for 2.4m Transonic Wind tunnel is briefly introduced, ground test and wind tunnel test results are given, and the effect of active vibration reduction system on calm state signal and its performance are discussed.

2. Active Damping System

Active vibration damper system consists of mechanical structure and control system. The mechanical structure consists of a controlled model/balance/strut system and an active joint of an embedded piezoceramic actuator (piezoelectric element). Active connector is divided into front connector and rear connector. As shown in Figure 1., piezoelectric elements are fully arranged in the lower part of the connector by pre-tensioned bolting. The upper and lower elastic hinges are used to amplify the micro-displacement axial movement of the piezoelectric element into a larger pitching motion of the model. Eight piezoelectric elements are evenly arranged above and below the lower hinge to ensure almost equal moment on both sides of the hinge and to prevent the piezoelectric elements from bearing large shear and bending loads.

![Active adapter schematic](image)

1. rear adapter; 2. front adapter; 3. Piezo elements; 4. spring hinge;

**Figure 1.** Active adapter schematic

The control system hardware consists of piezoelectric power amplifier, DSP core controller, acquisition and output card, DC-link of DC high-voltage power supply box, uninterruptible power supply UPS, signal transfer box, etc., as shown in Figure 2.
The active vibration damper system firstly uses the transfer function characteristics between piezoelectric excitation and balance response to obtain active control parameters through test calibration. Then the controller output is calculated based on the balance real-time vibration signal. The output signal is amplified by power and drives the piezoelectric element to move. Through the double elastic hinge structure of active damping joint, the bending vibration of the support rod is converted into bending vibration. If the bending vibration is just opposite to the pitching direction of the model, the pitching vibration of the model is suppressed.

Modal decoupling is the key to control algorithm and helps eliminate the interference between support mode and balance mode. Balance Longitudinal Element U1 and U2. The dynamic signal mainly consists of the first two modes. After ignoring the high-order response, it can be expressed as follows:

\[ U_1 = q_1 \Phi_{11} + q_2 \Phi_{21} \]  
\[ U_2 = q_1 \Phi_{12} + q_2 \Phi_{22} \]

When single frequency excitation occurs at the first mode natural frequency respectively.

\[ U_{1,f1} = q_1 \Phi_{11} \]  
\[ U_{2,f1} = q_2 \Phi_{12} \]

Similarly, when single-frequency excitation occurs at the natural frequency of the second-order mode.

\[ U_{1,f2} = q_1 \Phi_{21} \]  
\[ U_{2,f2} = q_2 \Phi_{22} \]

Thus, the decoupling matrix can be obtained
Based on the mode frequency identified by the mode, the structure is excited by a narrow bandwidth random signal, the structure response is measured, the frequency response function is calculated, and the open loop transfer function of the structure is fitted.

Using PD control algorithm, the relationship between closed-loop transfer function and open-loop transfer function

\[ H_\alpha(s) = H_\beta(s) \frac{H_\alpha(s)(sf + g)^T H_\alpha(s)}{1 + (sf + g)^T H_\alpha(s)} \]  

(6)

The eigenvalue of a closed-loop system is determined by the following equation.

\[ 1 + (sf + g)^T H_\beta(s) = 0 \]  

(7)

The eigenvalue of the closed-loop system can be solved by any set of control parameters \( f \) and \( g \). Hence, the appropriate control parameters can be found, or the control parameters can be designed directly using the Simulink design function of MATLAB.

3. Test results and discussion

Figure 3 shows the balance response signals when the model is hammered when the active damper system is under control or uncontrolled. It can be seen from the diagram that when the system is uncontrolled, the vibration of the model has not been fully attenuated after 20 seconds, while approximately 2 seconds after active control, the model has been basically attenuated, which shows that the active damping system developed has the ability of fast vibration reduction.

![Figure 3. Balance responses during impulse tests](image)

Table 1 shows the vibration frequency and damping of the model/balance/strut system with different control objectives. The results show that, in general, the control effect increases with the increase of control objectives. Damping of the system is greatly increased after active control. For example, when the first-order control target of pitch is 95%, the system damping increases to 14.6 times that of uncontrolled control, and when the second-order control target is 88%, the system damping increases to 14.8 times that of uncontrolled control. The active damping system has the ability to increase the system damping by more than 10 times, thus realizing the purpose of greatly increasing the system damping to suppress the model vibration.
Table 1. System Vibration Frequency and Damping for Different Control Objectives

| Control goal | 1st (%) | 2nd (%) | $f_1$ (Hz) | $\gamma_1$ (%) | $f_2$ (Hz) | $\gamma_2$ (%) |
|--------------|---------|---------|------------|---------------|------------|---------------|
| 0.0          | 0.0     | 8.90    | 0.44       | 16.90         | 0.26       |
| 90           | 88      | 8.89    | 2.37       | 16.80         | 3.84       |
| 92           | 90      | 8.89    | 3.34       | 16.81         | 4.83       |
| 95           | 95      | 9.63    | 6.43       | 16.52         | 17.40      |

Figure 4 shows the vibration reduction effect of a typical test state model. When uncontrolled, test angle of attack is 8°. The maximum amplitude of model vibration is about 3.83V for U1 peak difference and 3.15V for U2 peak difference. After control, the vibration amplitude decreased by about 73.0% and 76.7%, respectively, indicating that the system has a strong ability to suppress model vibration. From test angle of attack to 8°, from the U1 amplitude spectrum, the pitch mode amplitude of the support is reduced by 74.6% and that of the model/balance by 90.2% after control. The results of the wind tunnel test further demonstrate that the active vibration reduction system can significantly reduce the vibration of the support and model/balance pitch modes at the same time.

![Figure 4. Vibration reduction Results](attachment:image1.png)

Whether the active vibration reduction system has an impact on the balance force measurement data is a key issue to be considered in the development of the active vibration reduction system. Table 2 shows the average value of each component signal of the balance when the high voltage is not on, the high voltage is on, the bias voltage is on and the piezoelectric element is checked. From the table, it can be seen that the static value of the 6 elements of the balance does not change significantly at different stages, which indicates that the active damping system has little impact on the static force measurement data of the balance when it works.
Table 2. Average static state signals

| stage | Off   | On    | Offset | Check |
|-------|-------|-------|--------|-------|
| U1(mv)| 1859.3| 1859.1| 1859.2 | 1859.1|
| U2(mv)| 185.39| 185.35| 185.25 | 185.29|
| U3(mv)| -1057.4| -1057.7| -1056.8| 1056.6|
| U4(mv)| -634.93| -634.98| 635.59 | -635.56|
| U5(mv)| -620.28| -619.89| -620.18| -620.15|
| U6(mv)| -1.8358| -1.8368| -1.8358| -1.8369|

The test results show that for every 80V increase in bias voltage, the angle of attack of the model increases by 0.01°, at 320V bias, the model angle of attack increases by only 0.04°, it has little effect on angle of attack and can be corrected appropriately when processing data.

The performance of active vibration damper system is our key concern. From the results of many wind tunnel tests, power amplifiers mostly use only half the capacity, even though they have sufficient control capability for larger model vibration active damping systems.

In addition, since the active damping system can only suppress pitch vibration of the model, it is necessary to develop a new active joint and corresponding control system for yaw vibration of some models. One is to develop an active damping system which can simultaneously suppress the pitch and yaw vibration of the model, the other is to retain the existing active damping system to suppress the pitch vibration of the model and redevelop an active damping system to suppress the polarized vibration, in which the active joint is arranged at the front of the strut.

4. Conclusion

The model active damping system of 2.4m Transonic Wind tunnel has been verified by model test and its damping effect and capability have been verified.

Active vibration damper system can suppress both strut pitch and model/balance pitch mode vibration, greatly increasing the damping of model/balance strut system by more than 10 times.

During wind tunnel tests, the active damping system can significantly reduce the vibration amplitude of the model by more than 70%, and has sufficient capacity to restrain the pitching vibration of the model.

Active vibration damper system has little effect on calm state signal and does not affect static force measurement data.

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