Experimental Investigation on Active Longitudinal Vibration Suppression of the Thrust Bearing

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Abstract. The thrust bearing in a marine propulsion shafting system is the main transmission path of vibration induced by the longitudinal fluctuating forces of the propeller, and has significant influence on the coupled vibration and acoustic radiation of the shafting-hull system. An active control method for suppressing the longitudinal vibration of the thrust bearing is presented, which is able to reduce the coupled vibration of the shafting-hull system. The active method was experimented on a shafting system, which comprised two electromagnetic actuators symmetrically mounted on the thrust bearing base. The two actuators can response to an adaptive control strategy and theoretically suppress the dynamic forces between the thrust bearing and its base and resulting reduce vibrations in the hull structure. Experimental results demonstrated that the active control method was able to reduce vibration of the thrust bearing effectively, and the power spectra were decreased by about 90%.

1. Introduction
The coupled vibration of the shafting-hull system caused by the propeller thrust ripple is the main reason for the low frequency acoustic radiation, and it is difficult to control, which has become one of the important factors restricting the ship acoustic stealth performance [1]. The thrust bearing base as the key link on propeller shafting-hull system, its acoustic characteristics have an important influence on the vibration and acoustic radiation of the shafting-hull coupled system [2]. Theoretically, improving the ship’s line shape to make the flow field of the stern better and improving the propeller design to reduce the pulsating force, are the fundamental ways to increase the stealth performance of ships. However, the problem of wake field and propeller hydrodynamic are very complex. Even with the use of seven blade propeller, the longitudinal pulsating load is too large. It is mainly improved by experience and experiment presently [3]. On the other hand, there are also many practical difficulties in trying to avoid the hull resonance by improving the structure design of the hull [4]. The propulsion shafting is the main way to propagate the vibration of the propeller [5]. Therefore, it is a practical and feasible way to solve the problem from the vibration transfer path.

Shafting vibration control can reduce the vibration transmission to the hull and avoid the acoustic radiation. But it is quite difficult. On the one hand, we must guarantee the basic functions of the shafting, so that the vibration control of shafting can’t cause excessive thrust loss and damage the stern tube sealing device. On the other hand, we must ensure the impact resistance of the propulsion shafting. The propulsion shafting is mainly designed based on its functional, which has little room for dynamic modification of each component. The theoretical and applied research on propulsion shafting vibration control is less by optimization. The key point is still to add additional control devices on the propulsion shafting. Therefore, many scholars have put forward some control methods, such as
resonant converter, passive vibration absorption, active auxiliary thrust bearing, distributed active vibration absorption, and so on [6~11]. Although theoretical studies and simple system tests show that these methods are feasible, there is still a lack of verisimilitude experimental validation, and their actual effect and practicality are not quite clear.

In this paper, for the problem of shafting vibration control, a new shafting longitudinal vibration control scheme is proposed based on the summary of advantages and disadvantages of the existing vibration reduction methods. The active method was experimented on a shafting system, which comprised two electromagnetic actuators symmetrically mounted on the thrust bearing base.

2. Experiment Model
The shafting test bench structure mainly consists of the propulsion motor, elastic coupling, intermediate bearing, thrust bearing, tail bearing, propeller bearing, propeller weight and shafting section etc, as shown in Figure 1. The stern of bench installs the propeller thrust simulation device, which is used to simulate the propeller load force. Two active actuators are mounted symmetrically on the base of the thrust bearing.

![Figure 1. The vibration suppression experiment model on the thrust bearing](image)

The active actuator symmetry is installed on the thrust bearing base, and a high sensitivity accelerometer sensor is arranged on the thrust bearing base for the vibration transmission control of the shafting system. The control loop is shown in Figure 1. The vibration acceleration signal of thrust bearing base is integral to the vibration velocity signal by a charge amplifier. Corresponding to the vibration velocity signal, the controller generates the control voltage. And according to the voltage and the control algorithm, the power amplifier pushes the active actuators to generate the electromagnetic force and reduce the vibration of the thrust bearing base. In this experiment, the static thrust is generated by the air spring. The signal source is the synthesis of the random signals with 30Hz, 60Hz, 90Hz and the periodic signal with 20Hz ~100Hz. And the exciter emits the same interference force, acting on the propeller weight and causing the shafting vibration. So the vibration response of the thrust bearing base is the synthesis of periodic and random responses.

3. Adaptive Feedback Control Algorithm
As the dynamic parameters of the shafting bearing depend on the revolving speed, there are some different for the steady system on vibration control method [12]. The current model-free control algorithm requires precise frequency estimation, narrowband filter and fast online optimization to control the periodic interference by frequency selection. The stability, convergence and performance of the algorithm will need further study. Through the online identification of the control channel model can realize the broadband control, it is usually disturbed by the vibration signal of the shafting frequency modulation for the operation of shafting. So it may reduce the control channel modeling accuracy, or even false results. The adaptive model online identification control method can solve this problem and the common methods are based on LMS (Least Mean Squared) algorithm presently.

The control principle frame based on model online identification is shown in Figure 2, where $\hat{H}(z)$ is the identified control channel model, $d(n)$ is the disturbance signal, $y(n)$ the is control channel output, $e(n)$ is the control error, $e(n)=d(n)-y(n)$. $\tilde{d}(n)$ is the observed disturbance signal, $W(z)$ is the
controller and limit the output amplitude [10]. The coefficient of $W(z)$ is

$$\theta = \left\{ \theta_0, \theta_1, \ldots, \theta_{\alpha-1} \right\}^T$$

($\alpha$ is positive integer), and its adjusting equation is:

$$\theta(n+1) = S_u \left( \theta(n) + \mu \frac{e(n)\rho(n)}{\gamma + \|\rho(n)\|^2} \right) \tag{1}$$

Where $\rho(n) = \left\{ z(n), z(n-1), \ldots, z(n+1-\alpha) \right\}^T$, $S_u$ is the first order of the Sigmoid function, $0<\mu<1$, $\gamma>0$. The control signal $u(n) = S \left( \theta^T(n) r(n) \right)$, where $r(n) = \left\{ \tilde{d}(n), \tilde{d}(n-1), \ldots, \tilde{d}(n+1-\alpha) \right\}^T$. In Figure 2, the observed disturbance signal $\tilde{d}(n)$ is regarded as the reference signal. $\tilde{d}(z) = d(z) - \left( \hat{H}(z) - H(z) \right) u(z)$, and $\tilde{d}(n) = d(n)$ when $\hat{H}(z) = H(z)$.

![Figure 2. Block diagram of the adaptive control of periodical disturbances](image)

The frequency response of the control channel may need to be re-identified for the case of a significant change in speed. The modeling method is divided into two stages. In the first stage, we first obtain the FIR (Finite Impulse Response) model including the control channel of the periodic disturbances, and use the LMS recursive algorithm to get the finite impulse response sequence of the control channel. The FIR sequence contains the false feature information for the periodic disturbances in the response. It is because the LMS method regards the periodic vibration as the small damping mode of the system. Therefore, the real system finite impact response must be restored by the initially identified FIR sequence. In the second stage of modeling, the subspace filtering method is used to identify and remove small damping characteristics and unstable characteristics from the FIR sequence, and get the finite impulse response that reflects the real characteristics of the system.

### 4. Experimental Results

The external characteristic tests of actuators use the excitation of the white noise signal source. These tests can obtain the vibration acceleration and driving voltage for the inertial body of actuators, and calculate the frequency response function. When the control effect is tested with actuators, the stern exciter is used as the signal source (white noises), and the changes in the response of the shafting and the bearing seat are obtained under different excitations. So it can quantitatively evaluate the effect of the control method under disturbances.

In practice, two longitudinal vibration controllers need synchronous operation, so synchronicity needs to be tested. The test results are shown in Figure 3, Point 9 and Point 10 represent vibration acceleration of the two inertial bodies respectively, and Point 11 represents input voltage. As shown in Figure 3, the synchronization performance meets the control requirements when the two longitudinal vibration controllers operate in series. The resonant frequency of actuators is near 1.5Hz, which produces large vibration at the frequency and is bad for the fatigue strength of the spring. So, this
In order to enhance the damping of longitudinal actuators, the active damping can be introduced to suppress the resonance peak, as shown in Figure 4. We can see that after inducting active damping, the natural vibration of longitudinal vibration actuators is not obvious in the frequency response characteristics, and there is no change in the performance of the working section. In Figure 4, the peak value of 50Hz is power frequency interference.

Taking the test speed 90r/min as an example, the control frequency band is 10Hz~100Hz. The corresponding control result is shown in Figure 5, where red indicates uncontrolled and green indicates controlled. Seeing the Figure 5, we find that the longitudinal vibration of the thrust bearing base is greatly attenuated. The amplitude of the power spectrum is decreasing by 90% in 5Hz~200Hz, but the high frequency vibration is decreasing slightly with the different rotate speed. The test data also shows that the longitudinal vibration of thrust bearing base is stronger than vertical vibration, and vertical vibration is stronger than horizontal vibration. Because vertical vibration is mainly caused by longitudinal vibration, it can be reduced directly by controlling longitudinal vibration. In addition, there is almost no change in horizontal vibration, which indicates that longitudinal control has little effect on horizontal vibration. Due to the symmetry of bearing base, longitudinal excitation and control force are not easy to produce the eccentric load in horizontal direction.
According to the above test results, we can conclude that:

1) The frequency response characteristics of the longitudinal vibration actuators conform to the design requirements. For actuators, the natural frequency is 1.5Hz, the damping ratio is less than 5%.

Figure 5. Vibration response of the thrust bearing base before and after control (red corresponding to before and green to after)
the feedback working frequency covers 5Hz to 250Hz, and the 0.5V excitation voltage is corresponding to the output force of 80N~100N, which meet the requirements of vibration control.

2) Taking the longitudinal vibration of the thrust bearing base as the control target, the actuators can suppress the longitudinal vibration of thrust bearing. After controlling, the power spectrum peak is decreased about 90% (about 10dB), and the vertical vibration of thrust bearing can also be suppressed like the longitudinal vibration control.

5. Conclusion
In this paper, a new active control method for longitudinal vibration of shafting is presented. The vibration of shafting-shell coupled is studied and the control device is verified through model testing. The device can directly control the vibration of the shafting and observably reduce the shell vibration caused by the pulsating thrust through the driving shafting theoretically. The experimental verification was carried out in shafting test model, and test results show that: while two longitudinal vibration controllers are symmetrically arranged on the thrust bearing base and tandem operate, their synchronization is good, which can satisfy the control requirements; taking the longitudinal vibration of the thrust bearing base as the control target, the control device can suppress the longitudinal vibration of thrust bearing effectively. The power spectrum peak is decreased about 90% after controlling.

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