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ADAPTIVE NON-STATIONARY VIBRATION CONTROL FOR MAGNETIC BEARING SYSTEM FROM START-UP TO OPERATIONAL SPEED

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ABSTRACT

In this paper, firstly, we study the asymptotic stability for adaptive nonlinear algorithm in non-stationary condition and we also study the relationship between control level and frequency change rate and disturbance amplitude change rate in detail. We obtain the conclusion that the control result has a great influence on amplitude change rate, and has not influence on frequency change rate while that frequency change rate is known. After that, we apply the adaptive nonlinear algorithm for magnetic bearing system to control the vibration due to unbalance when the rotor would be accelerated from start-up to operational speed beyond its critical speed. The effectiveness of our proposed adaptive nonlinear algorithm is verified by the simulations and experiments.

INTRODUCTION

The forced vibration of the rotor caused by the unbalance excitation force is the fateful problem in a rotating machine for high-speed including the rotor-magnetic bearing system. If the vibration becomes large enough, the rotor cannot increase the rotational speed beyond its the critical speed and the operation will become impossible. It is an important subject to that such suppression of the unbalance vibration, especially in the amplitude of the resonance zone, because it may give a great influence to the rotor system, which the excitation force due to the centrifugal force is received in.

Frequency estimation has been reported as for the case as well that a periodic disturbance when the frequency of disturbance were unknown, and furthermore the adaptive disturbance rejection with the multiple frequencies estimation and a frequency tracking function for the periodic disturbance were proposed by authors[1]~[4] .And then, the control experiment result of the unbalance vibration show that the technique proposed was validity when the rotor is rotating at high speed. On the other hand, generally, it will take fixed time from the start-up to operational speed for the rotor with the rotation machine such as the magnetic bearing. In this case, the rotor will be beyond the critical speed of the rigid mode. Therefore, it is necessary to control the vibration fully as an occasion. If the unbalance vibration become large enough, the control performance gets extremely poor due to that the power amplifier is saturated. In this paper, we study the method about adaptive control in the non-stationary condition in the case of such a frequency of the periodic disturbance was change, and examined the relations between the suppression effect and the change rates of the amplitude and the frequencies. Furthermore, the validity of the adaptive control nonlinear algorithm in the nonstationary condition was confirmed by the simulation and the verification experiment.

ADAPTIVE REJECTION APPROACH FOR NON-STATIONARY PERIODIC DISTURBANCE

In this section, we develop an adaptive algorithm for the control of the non-stationary periodic disturbance. The block diagram of a non-stationary periodic disturbance active adaptive rejection using this method is shown in **FIGURE 1**. Furthermore, the details of the adaptive algorithm are shown in **FIGURE 2.**

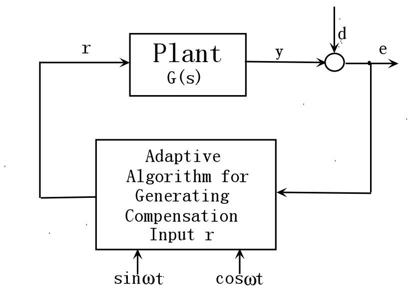


FIGURE 1: Block diagram of the control system rejecting periodic disturbance

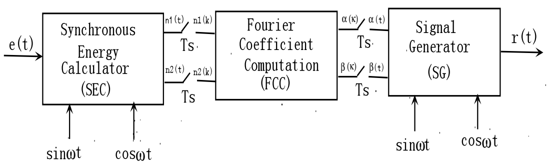


FIGURE 2: Block diagram of adaptive algorithm for internally generating reference signal

In **FIGURE 2** , the left block shows the calculation of the harmonic as sine and cosine function, the middle block shows the modification of Fourier coefficients $\alpha(t), \beta(t)$ of the input signal to plant adaptability, and the right block shows the control input generated for the plant from the Fourier coeffitients. The purpose of this algorithm is such that the generated pseudo-feedforward control input signal to the plant in **FIGURE 2** convolutes in the presence of a disturbance as shown in **FIGURE 1** and cancels the error, thus, the error signal become smaller.

The adaptive algorithm without observation noise was proposed[1][2]. And the adaptation algorithm with the frequency tracking method was examined to control stationary periodic disturbance[3]. In this paper, the adaptation rejection algorithm is examined to control the non-stationary periodic disturbance .

Firstly, the non-stationary periodic disturbance which has single frequency is given as follows:

$$d(t)=\alpha_d(t)sin(\omega_0 t+\Delta\omega t)+\beta_d(t)cos(\omega_0+\Delta\omega t) \quad (1)$$

Here, ω_0 is the frequency of the disturbance at the control start, $\alpha_d(t)$, $\beta_d(t)$ are the amplitude of the disturbance, and $\Delta\omega$ are the frequency change rate of moment. We assume that the transfer function of the plant is $G(s) = A(\omega)e^{j\theta}$. And the control input $r(t)$ to the plant is defined here as follows:

$$r(t)=\alpha(t)sin(\omega_0+\Delta\omega t)t+\beta(t)cos(\omega_0+\Delta\omega t)t \quad (2)$$

Here, $\alpha(t), \beta(t)$ are the Fourier coefficients modified by the adaptive control algorithm.

In this case, the output $y(t)$ of the plant is shown as follows:

$$\begin{aligned} y(t) &= r(t)G(j\omega) \\ &= A(\omega)[\alpha(t)sin[(\omega_0+\Delta\omega t)t+\theta(\omega)] \\ &+ \beta(t)cos[(\omega_0+\Delta\omega t)t+\theta(\omega)] \end{aligned} \quad (3)$$

However, $A(\omega)$ and $\theta(\omega)$ are the gain and the phase of the plant at frequency ω. Then the error signal $e(t)$ of the system becomes:

$$\begin{aligned} e(t) &= y(t)+d(t) \\ &= A(\omega)\{\alpha(t)sin[(\omega_0+\Delta\omega t)t+\theta(\omega_0)] \\ &+ \beta(t)cos[(\omega_0+\Delta\omega t)t+\theta(\omega_0)\} \\ &+ \alpha_d(t)sin(\omega_0+\Delta\omega t)t+\beta_d(t)cos(\omega_0+\Delta\omega t)t \end{aligned} \quad (4)$$

We consider the error signal $e(t)$ in the signal processing, as shown in **FIGURE 3**.

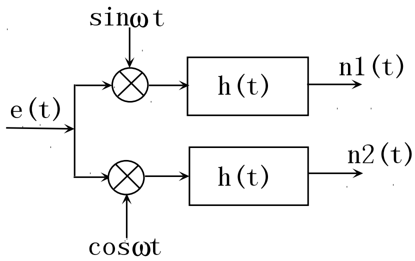


FIGURE 3: Details of the block SEC of **FIGURE 2**

$h(t)$ is a low pass filter and its outputs was written approximatly as follows:

$$\begin{cases} n_1(t) = e(t)sin(\omega_0+\Delta\omega t)t \\ n_2(t) = e(t)cos(\omega_0+\Delta\omega t)t \end{cases} \quad (5)$$

Firstly, we consider $n_1(t)$.

$$\begin{aligned} n_1(t) &= A(\omega)\{\alpha(t)sin[\omega_0+\Delta\omega t)t+\theta(\omega_0)] \\ &+ \beta(t)cos[(\omega_0+\Delta\omega t)t+\theta(\omega_0)]\}sin(\omega_0+\Delta\omega t)t \\ &+ [\alpha_d(t)sin(\omega_0+\Delta\omega t) \\ &+ \beta_d(t)cos(\omega_0+\Delta\omega t)t]sin(\omega_0+\Delta\omega t)t \\ &= 0.5A(\omega)\{\alpha(t)[cos\theta(\omega)-cos(2(\omega_0+\Delta\omega t)t+\theta(\omega))] \\ &+ \beta(t)[sin(2(\omega_0+\Delta\omega t)t+\theta(\omega))+sin\theta(\omega)]\} \\ &+ 0.5\{\alpha_d(t)[1-cos2(\omega_\Delta \omega t)t] \\ &+ \beta_d(t)sin2(\omega_0+\Delta\omega t)t\} \end{aligned} \quad (6)$$

Here, we define the cutoff frequency ω_B of the low pass filter as $\omega_B \ll 2\omega_0$. As result, we can neglect the relatively small gain at high-frequency components to determine the true value of the disturbance frequency ω_0.

In this state, the output $n_1(t)$ of the low pass filter can be written as follows approximately.

$$\begin{aligned} n_1(t) &= 0.5A(\omega)\{\alpha(t)cos\theta(\omega)+\beta(t)sin\theta(\omega)\} \\ &+ 0.5\alpha_d(t) \end{aligned} \quad (7)$$

We can also obtain the output $n_2(t)$ as follows:

$$n_2(t) = 0.5A(\omega)\{\alpha(t)sin\theta(\omega) + \beta(t)cos\theta(\omega)\} + 0.5\beta_d(t) \quad (8)$$

Firstly, let us supposes that disturbance frequency can be identified. In this case, an adaptive law does not depend on phase θ and is defined as follows[2]:

$$\begin{cases} \alpha(k+1) = \alpha(k) - \mu_1(k+1)n_1(k) \\ \beta(k+1) = \beta(k) - \mu_2(k+1)n_2(k) \end{cases} \quad (9)$$

Here, μ_1, μ_2 are the step sizes. Forthermore,

$$\begin{aligned} n_1(k+1) &= 0.5A(\omega)[\alpha(k+1)cos\theta(\omega) \\ &+ \beta(k+1)sin\theta(\omega)] + 0.5\alpha_d(k+1) \\ &= n_1(k) - 0.5A(\omega)\{\mu_1(k+1)n_1(k)cos\theta(\omega) \\ &+ \mu_2(k+1)n_2(k)sin\theta(\omega)\} + 0.5\Delta\alpha_d(k+1)\Delta t \end{aligned} \quad (10)$$

$$\begin{aligned} n_2(k+1) &= n_2(k) - 0.5A(\omega)\{\mu_1(k+1)n_1(k)sin\theta(\omega) \\ &+ \mu_2(k+1)n_2(k)cos\theta(\omega)\} + 0.5\Delta\beta_d(k+1)\Delta t \end{aligned} \quad (11)$$

$\Delta\alpha(k+1), \Delta\beta(k+1)$ is the change rate of the disturbance coefficient in the moment $k \sim k+1$, and Δt are sampling time.
However,

$$\begin{cases} \mu_1(k+1) = \mu_1(k)sgn(n_1(k-1)^2 - n_1(k)^2) \\ \mu_2(k+1) = \mu_2(k)sgn(n_2(k-1)^2 - n_2(k)^2) \end{cases} \quad (12)$$

The above equation become a nonlinear adaptive law.
It shows that the output signal of the filter has great influence on amplitude change rate, and has no relations influence on the frequency change rate of the disturbance from Eqs.(10) and (11). The asymptotic stability of the modification law of this nonlinear system can be guaranteed if the initial value of μ_i is set such fixed condition as $|\mu(0)| < 2\sqrt{2}/A(\omega)$ for the different phase θ from the result[3], because Δt is small than 1/1000 seconds, $\Delta\alpha_d(k+1)\Delta t$ and $\Delta\beta_d(k+1)\Delta t$ is far smaller than 1. It is verified by the theoretical consideration using Lyapunov asympototic stability theorem and the simulation. Therefore, it is shown that asympototic stability of the Eqs.(10) and (11) is guaranteed.

SIMULATION

In this section, we examine the adaptive vibration control to verify the validity of the above algorithm at the resonance zone of the vibration system due to the centrifugal force. The vibration equation of the system is shown as follows here:

$$M\ddot{x} + C\dot{x} + Kx = m_0 r_0 \omega^2 sin\omega t \quad (13)$$

In this case, the response of the system in the forced oscillation state becomes:

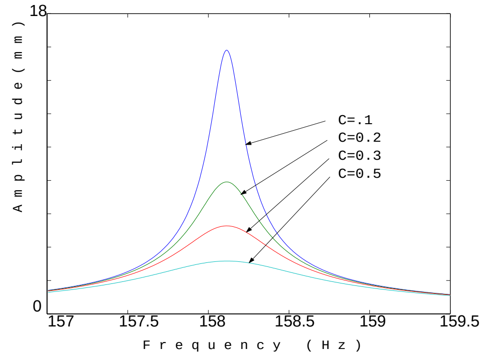
$$x(t) = \frac{m_0 r}{\sqrt{(1-\omega^2/\omega_n^2)^2 + (2\xi\omega/\omega_n)^2}} \quad (14)$$

However, $\xi = \frac{C}{M}$.

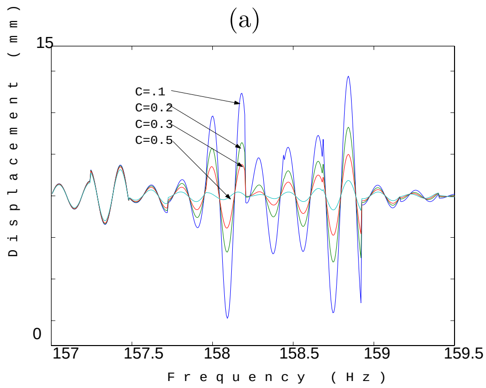
Here, we establish as a parameter of the system with $M = 1Kg, K = 25N/mm, m_0 = 0.006Kg, r = 0.02mm$. In this case, the natural frequency of the vibration system is $\omega_n = 158.11Hz$.

A response curves in the resonance zone of the vibration system for the various attenuation coefficients $C = 0.1, 0.2, 0.3, 0.5$ are shown in **FIGURE 4**(a).

As for the amplitude of the resonance zone, the response magnification varies according to the attenuation coefficient as shown in **FIGURE 4**(a).



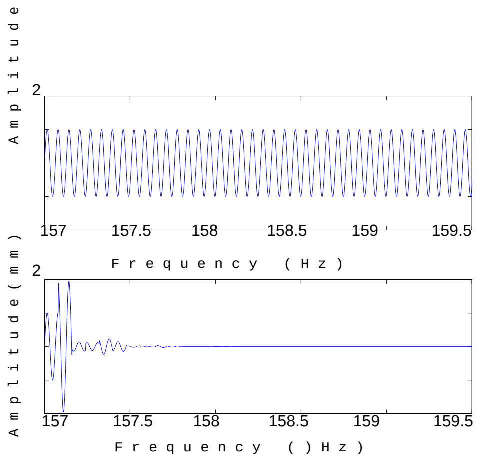
(a)



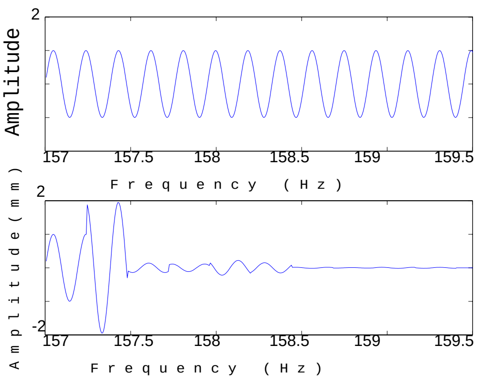
(b)

FIGURE 4: Amplitudes of vibration system and displacement attenuation results

In the case of the frequency change rate which it is the same, the effect of the adaptive suppression is different too from that the response magnification of the amplitude of the resonance zone due to the attenuation coefficient being different. Furthermore, when the amplitude of the disturbance is constant and only the change rate of the frequency is changed, an adaptive suppression result becomes **FIGURE 5**. We can obtain the about same suppression result even if a frequency change rate changes as shown in **FIGURE 5**.



$(a)\Delta\omega = 10Hz/s^2$



$(b)\Delta\omega = 30Hz/s^2$

FIGURE 5: Displacement attenuation results for different frequency change rate.

The above simulation shows that an adaptive suppression effect of the non-stationary without taking an influence in the change rate of the frequency, but was taken in the change rate of the amplitude of the system.
A 10KWh class high temperature superconductivity flywheel system for the electric power storage is examined as an example.

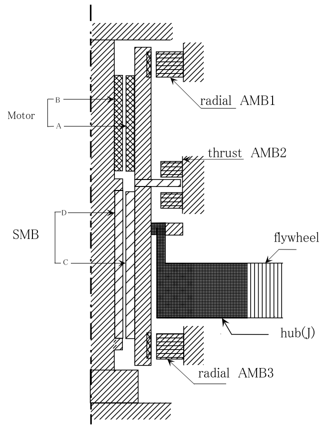
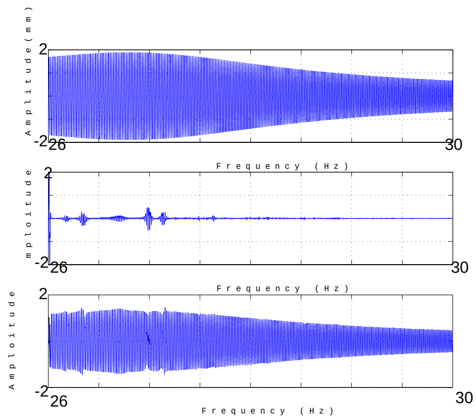


FIGURE 6: Cross-sectional view of outer rotor type flywheel

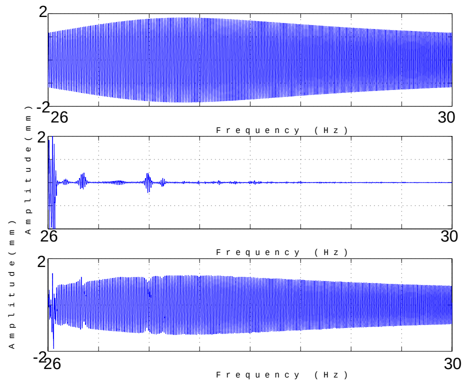
The total weight of this flywheel system was 104 tons, the weight of the rotor was 45.7 tons and its operational speed was 6000rpm. The superconducting magnet is used to lift off the flywheel, and it is applied to both of the axial direction and the radial direction. The natural frequency of the flywheel system about the first bending mode was 32.03Hz by the analysis using the vibration analytic software ANSYS. As for the stabilization control using AMB, we applied a sliding mode control technique and the new natural frequency of the close loop control system was 27.00Hz.

Here, for the case in which the disturbance frequency changes, the response is examined. Furthermore, we applied non-stationary adaptive algorithm to control the vibration because the excitation force due to the centrifugal force in the resonance zone (frequency range in $26Hz \sim 30Hz$) of the rotor system.

In this case, the result of the system response is shown in **FIGURE 7**. Angular accelerations are set as $0.5Hz/s^2$, $2Hz/s^2$, $4Hz/s^2$ respectively. In each figure, the upper figure is the responses in the position of AMB1 and AMB3 using the sliding mode controller for the stability, and the figure in the

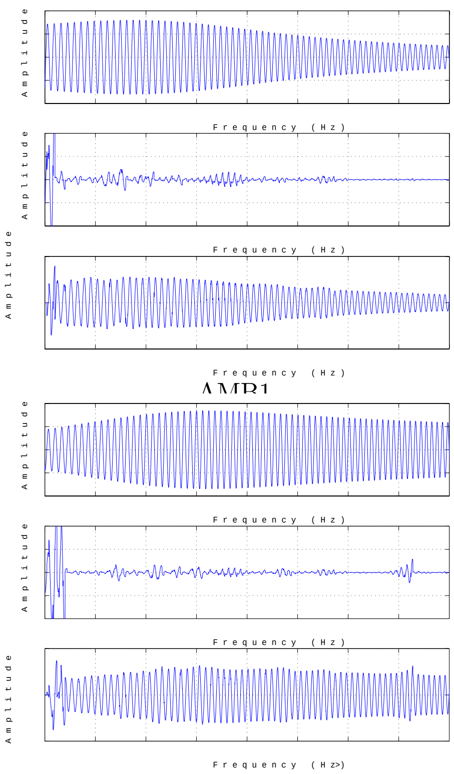


AMB1

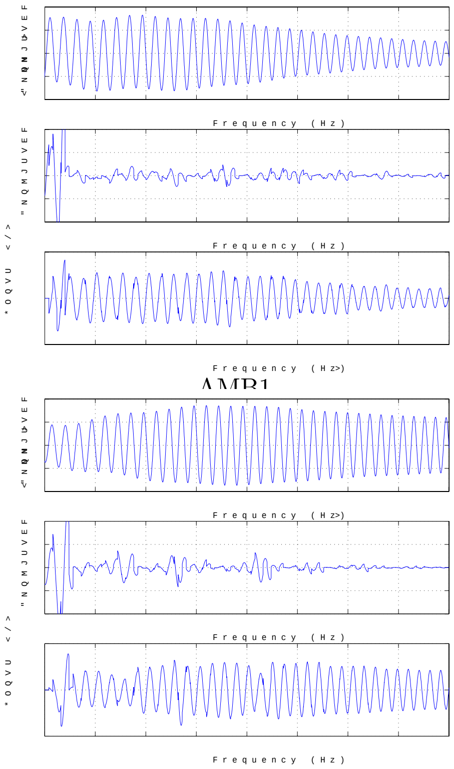


AMB3

(a) $\Delta\omega = 0.5Hz/s^2$



AMB3
(b) $\Delta\omega = 2Hz/s^2$



AMB3
(c) $\Delta\omega = 4Hz/s^2$

FIGURE 7: System response using sliding mode controller and adaptive attenution results

middle are the suppression results where non-stationary adaptive rejection algorithm was applied and the figure of the bottom are the input signals.

The validity of the non-stationary algorithm was confirmed using the flywheel as shown in **FIGURE 7**. It is shown that the suppression result depends on the frequency change rate, because the amplitude change rate depends on the angular acceleration.

VERIFICATION EXPERIMENT

In order to confirm the results of simulation, we carried out verification experiments using the five-axes controlled magnetic bearing. First, we use the analog PID controller to lift off the rotor stably. Then, input the excitation signal to the magnetic bearing through the amplifier, and make it excite. Here, the frequency change rate of the excitation signal (periodic disturbance) was set as $4Hz/s^2$.

Furthermore, we applied the nonlinear adaptive rejection algorithm to conduct the non-stationary vibration suppression experiment. The block diagram of the control system is shown in **FIGURE 8**. The responses of system with and without the nonlinear adaptive algorithm are shown in **FIGURE 9**.

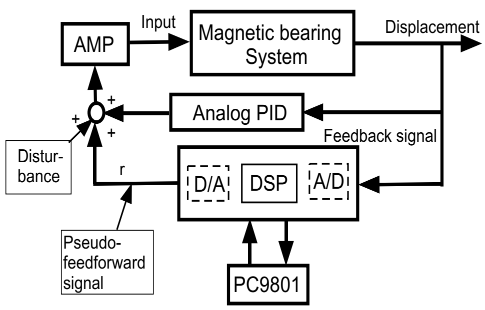


FIGURE 8: Block diagram for control system

The results show the control process converges in about two seconds after executed the controller and this shows that the convergence of this algorithm is fast. The validity of the proposed non-stationary nonlinear adaptive algorithm is verified by the fact that a disturbance in the resonance zone whose the amplitude change rate is big can be controlled fully.

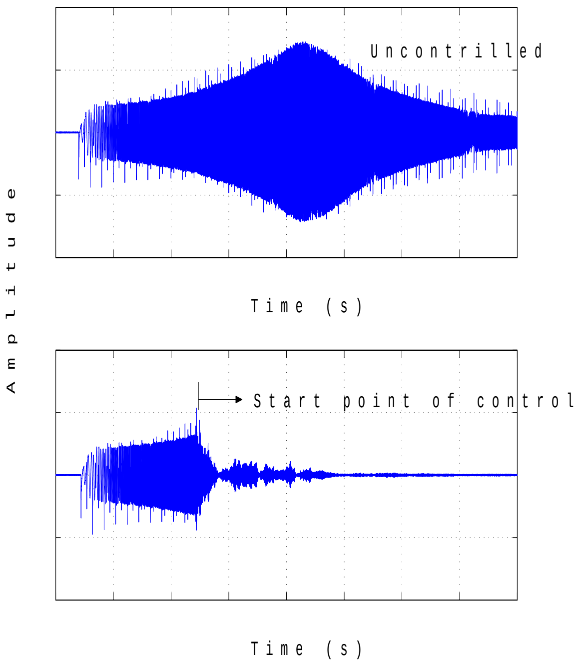


FIGURE 9: Result of verification experiment

CONCLUTIONS

In this paper, we examined the nonlinear adaptive rejection technique about the non-stationary periodic disturbance, and the conclusions are summarized as following:

(1) In the non-steady state, the nonlinear adaptive suppression result is no relations in the frequency change rate, but depends on the amplitude change rate of the signal.

(2) In the case of the rotor system vibration suppression , specially, in the case of the adaptive suppression of the resonance zone that amplitude changes violently was examined.

From the case of the rotor vibration suppression, specially the adaptive suppression of the resonance zone that amplitude changes violently was examined, the result is shown that it could obtain a good adaptive suppression result by the results of the theory analysis, the simulation and the verification experiment in the case of the amplitude change rate was small due to that if the frequency change rate was slow.

For future research, we plan to verify the adaptive suppression algorithm of nonstationary with experiments.

References

[1] Kenzo Nonami, Qifu Fan, and Hirochika Ueyama., Unbalance Vibration Control of Magnetic Bearing Systems Using Adaptive Algorithm with Disturbance Frequency Estimation, JSME International Journal, Series C, 41(2),220-226, 1998.

[2] Zi-he Liu, Kenzo Nonami., Adaptive Vibration Control Using Frequency Estimation for Multiple Periodic Disturbances with Noise, Proceedings of Pioneering International Symposium on Motion and Vibration Control in Mechatronics, pp.199-204, (1999.4).Tokyo.

[3] Kenzo Nonami,Zi-he Liu., Adaptive Unbalance Vibration Control of Magnetic Bearing System Using Frequency Estimation for Multiple Periodic Disturbances with Noise, Proceedings of the IEEE International Conf. on Control Applications, pp.576-581, (1999.8).Hawaii

[4] Tsao T.C.and Qian Y.X.,An Adaptive repetive control scheme for tracking period signals with unknown period,Proc.Amer.contr.conf., (1993),1736-1740,San Francisco,CA

[5] Yajun Zhang,Kenzo Nonami, Hiromasa Higasa.,Feasibility Study of Modeling and Control for 10MWh Class Energy Storage Flywheel System Using Superconduting Magnetic Bearing, Proceedings of the 12th International Symposium on Supercondutivity, pp.161-163,(1999.10) Morioka

[6] Tsao T.C. and Nemani M.,Asympotic rejection of periodic disturbance with uncertain period. proc. Amer contr. conf.(1992),2696-2699.

[7] Kuo S.and,Ji M. and M,J., Passband Disturbance Reduction in Periodic Active Niose Control System, IEEE, Trans. Speech Audio Processing, 14-2,(1996),96-103.

[8] Hillerstrom and J.Sternby, Rejection of periodic disturbance with unknown priod. . . A frequency domain approach. Proc.Amer.contr.conf., (1994), 1626-1631, Baltomore, MD.