Novel Robust and Adaptive Vibration Control for Active Magnetic Bearing System

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Abstract. In this paper, a novel adaptive vibration control algorithm is proposed by introducing a parameter K. The asymptotic stable condition of this algorithm is mathematically derived. In the case of stationary disturbance, the algorithm is an unbalance vibration control for K=0, while for K=1, it becomes an adaptive vibration control with consumed current control. With a non-zero K, the present control algorithm is suitable for non-stationary disturbance. To validate the applicability of the present control algorithm, a series of experiments are carried out under stationary and non-stationary disturbance conditions with K ranging from 0 to 1. Our results indicate that it is possible to find an appropriate value for K to realize a relatively optimal control of both vibration and consumed current simultaneously.

Index Terms **– Magnetic Bearing, Unbalance Adaptive Vibration Control, Non-Stationary Vibration Control, Consumption Current Suppression, Zero Bias Control**

1. Introduction

One person of the authors proposes adaptive vibration control of magnetic bearing system[1] ,furthermore applying this method with the multiple periodic disturbances, the effectiveness was verified[2],[3]. In this paper we propose the control algorithm of the non-stationary adaptive vibration control in the case that the rotational frequency changes, verify the performance through the experiment, in particular, in this research by introducing a new parameter K in non-stationary adaptive vibration control, the convergence conditions of the algorithm how changes, and a relation with the size of K and the error, it is considered.

In order to improve markedly the efficiency of the flywheel system as an electric power storage system, the total consumption energy should be small. If the consumption current of AMB becomes small, the eddy current will be reduced, and also the conquer problems, such as generation of heat, and all effects should turn into a synergistic effect. The consumption current was reduced by applying the proposed non-stationary adaptive vibration control algorithm, and choosing the suitable parameter K paying attention to the error and control input of vibration control. Moreover, the suitable parameter K to which vibration becomes the smallest, and it examined how the suitable parameter K to which both consumption current and an error become to some extent small would be chosen.

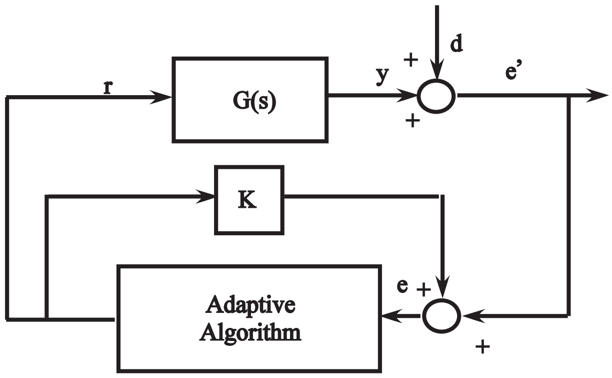

Fig.1 Proposed concept of new non-stationary adaptive vibration control system block diagram

In this research, a consumption current became the smallest at the time of K= 1, and the closed loop system became so much stable even though the rotational speed was accelerated and decelerated. Moreover, it has been considered why K= 1 is a suitable.

2. The non-stationary adaptive vibration control with disturbance frequency change

In this chapter, the non-stationary adaptive vibration control algorithm is expressed. Especially, the parameter K was introduced as the feedback loop system in this paper. The adaptive vibration control algorithm is considered in the cases of acceleration and deceleration for the non-stationary vibration.

.2.1 Non-stationary adaptive vibration control

In this section, the non-stationary adaptive vibration control algorithm with periodic disturbance is expressed. The block diagram of new concept and the non-stationary adaptive vibration control with periodic disturbance is shown in Fig.1. The extreme feature does not need the mathematical model of the plant in Fig.1, and, the parameter K is introduced, this parameter K is used for the adjustment of vibration control performance and consumption current.

Here, variable the frequency depending on time is defined as Ω(t). d(t) is the single periodic frequency disturbance of the non-stationary vibration, the next expression is given as follows.

$$d(t) = \alpha_d(t)\sin(\Omega(t)t) + \beta_d(t)\cos(\Omega(t)t) \quad (1)$$

Because the sensor noise always exists, the low pass filter was used for the reduction of such noise, here not to handle. The transfer function of the plant is defined as G(s)=Aejθ. Here, r(t) is the control input to reduce the unbalance vibration, it is defined as follows;

$$r(t) = \alpha(t)\sin(\Omega(t)t) + \beta(t)\cos(\Omega(t)t) \quad (2)$$

Here, α(t) and β(t) are the fourier coefficients which are corrected step by with the adaptive control algorithm. Y(t) is the system output value that is defined as follows;

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

However, A is the gain of the plant depending on the frequency, and θ is the phase. Furthermore, the signal after the summation is given by e'(t)=y(t)+d(t),and the input e(t) is written by,

$$\begin{aligned} e(t) &= y(t) + d(t) + Kr(t) \\ &= A\{\alpha(t)\sin(\Omega(t)t+\theta) \\ &\quad + \beta(t)\cos(\Omega(t)t+\theta)\} \\ &\quad + \alpha_d(t)\sin(\Omega(t)t) \\ &\quad + \beta_d(t)\cos(\Omega(t)t) \\ &\quad + K\{\alpha(t)\sin(\Omega(t)t) \\ &\quad + \beta(t)\cos(\Omega(t)t)\} \end{aligned} \quad (4)$$

e(t) is multiplied by the harmonic function,

$$\begin{aligned} \bar{n}_1(t) &= e(t)\sin(\Omega(t)t) \\ \bar{n}_2(t) &= e(t)\cos(\Omega(t)t) \end{aligned} \quad (5)$$

And then,

$$\begin{aligned} \bar{n}_1(t) &= 0.5A\{\alpha(t)[\cos\theta - \cos(2\Omega(t)t+\theta)] \\ &\quad + \beta(t)[\sin(2\Omega(t)t)+\theta) - \sin\theta]\} \\ &\quad + 0.5\{\alpha_d(t)[1-\cos(2\Omega(t)t)] \\ &\quad + \beta_d(t)\sin(2(\Omega(t)t)\} \\ &\quad + 0.5\{\alpha(t)[1-\cos(2\Omega(t)t)] \\ &\quad + \beta(t)\sin(2(\Omega(t)t)\} \end{aligned}$$

Here, the cutoff frequency of the low pass filter should be ωB<<2Ω(t)t, in this adaptive algorithm, according to this procedure, the high frequency component of 2Ω(t) becomes small, the final output from the low pass filter becomes as follows;

$$\begin{aligned} n_1(t) &= 0.5A\{\alpha(t)\cos\theta - \beta(t)\sin\theta\} \\ &\quad + 0.5\alpha_d(t) + 0.5K\alpha(t) \end{aligned} \quad (6)$$

Similarly,

$$\begin{aligned} n_2(t) &= 0.5A\{\alpha(t)\sin\theta + \beta(t)\cos\theta\} \\ &\quad + 0.5\beta_d(t) + 0.5K\beta(t) \end{aligned} \quad (7)$$

Here, the adaptive rule does not depend on phase θ is applied, α(k)and β(k) are defined by

$$\begin{Bmatrix} \alpha(k+1) = \alpha(k) - \mu_1(k+1)n_1(k) \\ \beta(k+1) = \beta(k) - \mu_2(k+1)n_2(k) \end{Bmatrix} \quad (8)$$

Here, μi is the step size. Furthermore, n1 (t) and n2 (t) are becomes as

$$\begin{aligned} n_1(k+1) &= n_1(k) \\ &\quad - 0.5A\{\mu_1(k+1)n_1(k)\cos\theta \\ &\quad - \mu_2(k+1)n_2(k)\sin\theta\} \\ &\quad + 0.5K[-\mu_1(k+1)n_1(k)] \\ &\quad + 0.5\Delta\alpha_d(k+1) \end{aligned} \quad (9)$$

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

where,

$$\begin{Bmatrix} \mu_1(k+1) = \mu_1(k)\,\mathrm{sgn}(n_1(k-1)^2 - n_1(k)) \\ \mu_2(k+1) = \mu_2(k)\,\mathrm{sgn}(n_2(k-1)^2 - n_2(k)) \end{Bmatrix} \quad (11)$$

From Eqs. (9) and (10), it can be found that the output signal of the filter is dependent on the amplitude of the disturbance variation signal. Δαd(k+1), Δβd(k+1) are the changes in the amplitude of disturbance in time k ~k +1. respectively. However, because they changes at one sampling time when the frequency shows little change, they might be near zero therefore can be omitted. Furthermore, for different θ, if μi meets a certain condition, the asymptotic stability of the correction term of the non-linear system can be guaranteed [(2)].

2.2 Asymptotic stability of non-stationary periodicity disturbance adaptive control algorithm

According to figure 1, the conditions of the asymptotic stability of a closed loop control system are considered. Herein, the stable theorem based on theory of Lyapunov is employed.

The following equation can be obtained form Eqs. (9) and (10)

$$\begin{aligned} N(k+1) &= n_1(k+1) + n_2(k+1) \\ &= N(k) \\ &\quad - 0.5A\{\mu_2(k+1)n_2(k)[\cos\theta - \sin\theta + \frac{K}{A}] \\ &\quad + \mu_1(k+1)n_1(k)[\cos\theta + \sin\theta + \frac{K}{A}]\} \end{aligned} \quad (12)$$

Here, we assume

$\gamma(k+1)N(k)=[\mu_1(k+1)n_1(k)+\mu_2(k+1)n_2(k)]$

and

$|\mu_2(k+1)| \le |\mu_1(k+1)| \le |\mu_1(0)| \le |\gamma(0)|$

γ as follows

$$|\gamma(k+1)| = |\mu_1(k+1)n_1(k) + \mu_2(k+1)n_2(k)| / |N(k)| \leq |\gamma(0)| |n_1(k)+n_2(k)| / |N(k)| \leq |\gamma(0)| \quad (13)$$

Here, we assume $V(k)=N^2(k)$ is the Lyapunov function candidacy, then, the conditions of asymptotic stability of the adaptive algorithm is $V(k+1)-V(k)<0$. from Eq.(12), we can obtain

$$V(k+1)-V(k) = N^2(k+1) - N^2(k) \leq \{-A(\sqrt{2}+\frac{K}{A})\gamma + [\frac{A}{2}(\sqrt{2}+\frac{K}{A})\gamma^2]\}N^2(k) \quad (14)$$

we assume $\eta = \frac{A}{2}(\sqrt{2}+\frac{K}{A})\gamma$, if $0<\eta<2$ then $V(k+1)-V(k)<0$,we obtained

$$|V(k+1)| < \frac{4}{\sqrt{2}A+K} \quad (15)$$

The asymptotic stability of adaptive algorithm can be guaranteed if the initial values for μ1(k+1) and μ2(k+1) are given as follows;

$$|V(0)| < \frac{4}{\sqrt{2}A+K} \quad (16)$$

Equation (16) indicates that the conditions for asymptotic stability changes with the parameter K.

3. DESIGN OF ADAPTIVE CONTROL OF AN UNBALANCE VIBRATION SUPPRESSION TYPE AND CONSUMPTION CURRENT SUPPRESSION TYPE

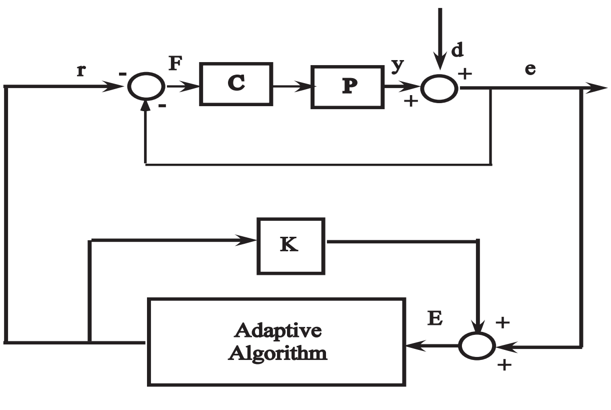

Fig.2 Proposed concept of new non-stationary adaptive vibration control system block diagram

(a) An unbalance vibration suppression type design

In the case of K= 0, the vibration control belongs to a general adaptive vibration control type. The purpose of adaptive algorithm is to generate the input signal r for the plant to yield an output signal y that can counteract the disturbance signal d to make error e=y+d approach zero asymptotically.

(b) Consumption current suppression type design

To control consumption current, a control strategy is designed as illustrated in Fig.2. Where, E is defined as

$$E = e + Kr \quad (17)$$

The total input signal of the plant is F

$$F = e + r \quad (18)$$

If F approaches zero, the control current supplied to the plant P will decrease sharply. In addition, in the light of the asymptotic stability of algorithm depicted in chapter 2, it can be expected that E in equation (17) becomes zero for an arbitrary K within a certain range. However, in order to realize F=0, it is necessary to set K at 1, and to make E=r+e equal to F. Under these conditions, F in equation (18) approaches zero, making the algorithm a consumption current control one.

(c) The design of unbalance vibration suppression and consumption current suppression type

The above-mentioned adaptive control algorithm deal with either unbalance vibration by setting K at 0 or current control by setting K at 1, respectively. In fact, in addition to vibration control, properly controlling the coil current is of practical significance. For this purpose, a series of experiment are performed to adjust values of K between 0 and 1.

4. EXPERIMENTAL RESULT OF STATIONARY VIBRATION CONTROL AND NON-STATIONARY VIBRATION CONTROL

Adaptive vibration control experiments were performed for stationary disturbance and non-stationary disturbance, respectively. In this research, adaptive vibration control was applied to all the radial of magnetic bearings. By mounting the algorithm on the DSP, unbalance vibration was adaptively controlled. First, the rotor turning at a high speed was stably surfaced with the zero bias PID controller; subsequently, a series of experiments were done at a stationary rotating frequency of 80Hz, and at non-stationary rotating frequencies varying from 75Hz to 80Hz, respectively.

4.1 Experimental result of stationary vibration control

The experiment was carried out for stationary disturbance at frequency of 80 Hz. The experimental results obtained without unbalance vibration control are shown in figure 3. And the results for K=0, 0.3, 1 are illustrated in figures from 4 to 6. Shown on the left-top

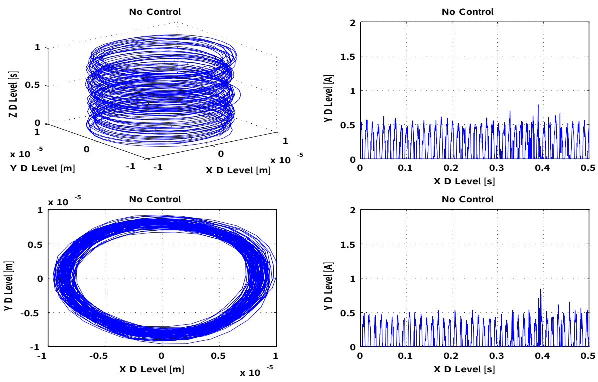

Fig.3 stationary periodicity disturbance suppression control system

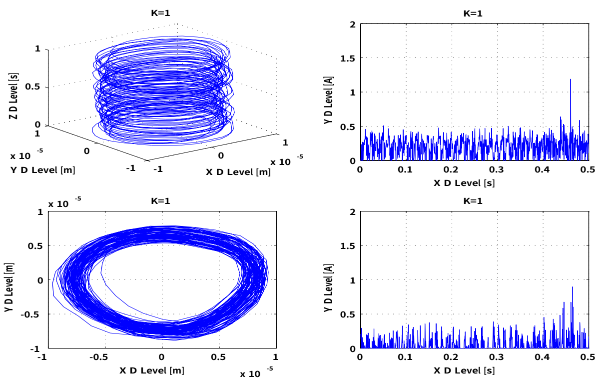

Fig.4 stationary periodicity disturbance suppression control system (K=1)

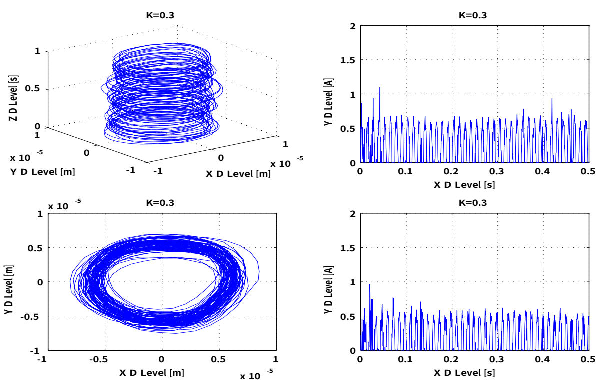

Fig.5 stationary periodicity disturbance suppression control system(K=0.3)

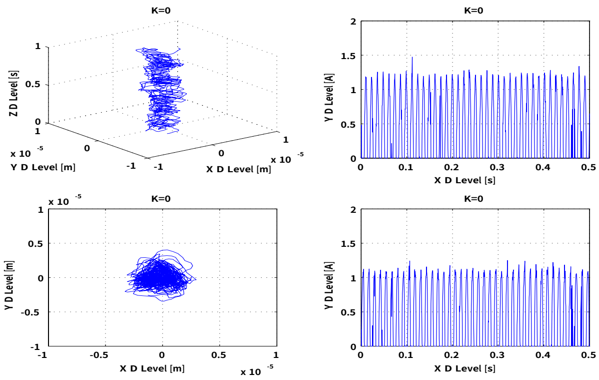

Fig.6 stationary periodicity disturbance suppression control system (K=0)

Side of each figure is the three-dimensional (involving both spatial and temporal dimensions) amplitude of the controlled rotor, and on the left-bottom side, the two-dimensional projection graph on the X-Y plane. Illustrated on the top-and bottom-sides are the output currents along the radial X axis. From these results, it is found that following the increase in K, although the amplitude of error signal is augmented, the consumed control current is pronouncedly reduced. For instance, in the case of k=0, the amplitude of error signal is very small but with a markedly increased control current. In contrast, in the case of k=1, the amplitude of error signal shows little difference but with a significant reduction in control current by up to 50 percent in comparison with the experimental results obtained without unbalance control.

4.2 Experimental result of non-stationary adaptive vibration control

Considering that the frequency of the secondary sympathetic vibration of the rigid mode is about 80Hz, experiments on non-stationary adaptive vibration control were carried out with the vibration frequency changing from 75 Hz to 80Hz. The experimental results for K=0, 0.3 and 1 are plotted in figure 8~10, respectively. Shown in figure 8~10 , (a) is the results for acceleration speed of 15 rpm/s, and in figure (b) is the results for acceleration speed of 30 rpm/s. For K=0, the adaptive vibration control is stable with very little amplitude of error signal under stationary condition; whereas, under non-stationary condition, the experimental results show markedly unstable oscillation of error signal, which is considered to result from the time needed to determine the transient frequency since it may elicit errors of frequency that are very likely to deteriorate the outcome of control. In contrast, if K is set at a nonzero value, such as 0.3 and 1 defined in the present experiments, rotations of the rotor are stably controlled under non-stationary condition, which indicates that the new control algorithm proposed in this study is suitable for non-stationary vibration control. Moreover, the fact, from the experimental results,

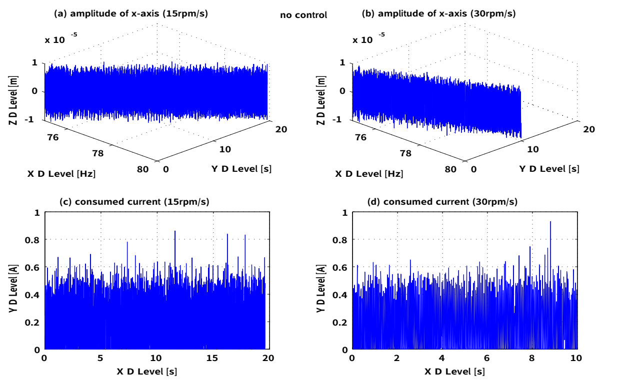

Fig.7 Non-stationary periodicity disturbance suppression control system

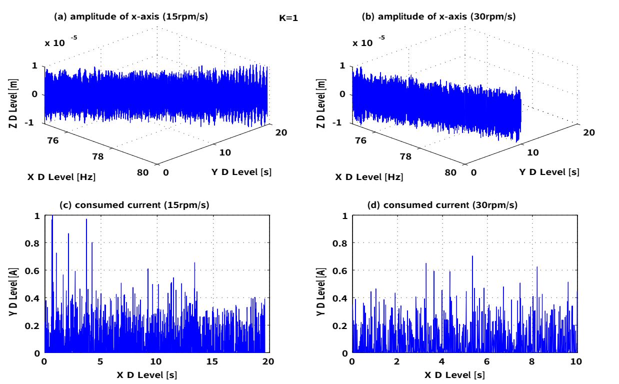

Fig.8 Non-stationary periodicity disturbance suppression control system (K=1)

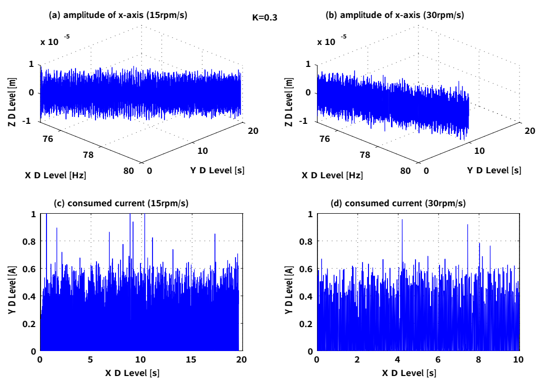

Fig.9 Non-stationary periodicity disturbance suppression control system (K=0.3)

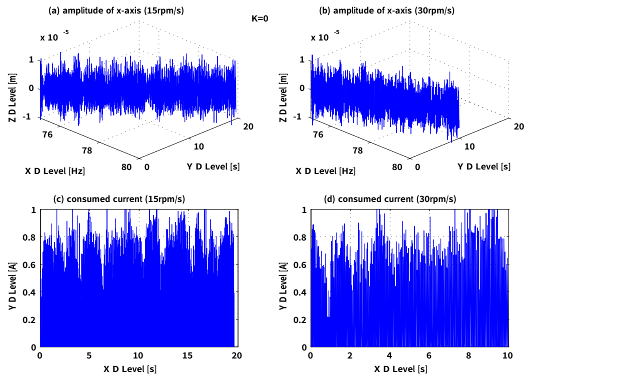

Fig.10 Non-stationary periodicity disturbance suppression control system (K=0)

that the present control algorithm help achieve simultaneous reduction in the amplitude of error signal and the magnitude of consumed control current in non-stationary vibration control, implicates that the present algorithm is of practical significance.

5 Conclusions

In the present study, the unbalance control with K=0, and the adaptive control with K=1 were devised and tested by vibration control experiments. For a nonzero K, the asymptotic convergence conditions of adaptive vibration control of the closed loop of system were mathematically deduced. Subsequently, in the experiments, the effect of K on control stability and consumed control current was systemically investigated by changing K from 0 to 1. Conclusions drawn in the present study are summarized as follows:

(1) For stationary periodicity disturbance, in the case of K= 0, signal error is limited at the lowest level; the control is classified as an unbalance control type; and if K=1, the control changes into a consumed current control type when consumed current is controlled to be the smallest. When changing the value of K from 0 to 1, both the amplitude of signal error and the magnitude of consumed current can be, to some extent, reduced to relatively low levels, leading to a satisfactory stable vibration control.

(2) For non-stationary periodic disturbance, when K=0, even under a semi-ideal condition that the rotor accelerates slowly, the control becomes extremely unstable. In contrast, if K≠0, under a variety of rotor-acceleration conditions, the control can be well stabilized. Especially, in the case of k =1, the smallest consumed current can be achieved.

(3) In the future work, it is of significance to find a way to enable the application of the adaptive vibration control algorithm with K=0 to non-stationary periodic disturbance.

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