ADAPTIVE FILTERING FOR UNBALANCE VIBRATION SUPPRESSION

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ABSTRACT

A new method for suppressing the unbalance vibration of a rotor suspended by magnetic bearings is introduced, which uses two kinds of adaptive filtering techniques. The 1st filter separates the rotor displacement signal into rotation-synchronized and rotation-asynchronous components. The former is the output of the rotor system plant due to the unbalance exciting force, which can be cancelled by the output of the 2nd filter. The 2nd filter is a disturbance estimator which uses the output of the 1st filter together with the impulse response of the plant measured on-line at high-speed rotation. These filters compose a feed-forward control system and do not affect the original plant stability. Both simulation and experimental results are shown to describe the basic and practical effectiveness of the proposed method.

INTRODUCTION

One of the most important technical areas in applying magnetic bearings to industrial use is vibration suppression. This consists of two kinds of problems -- one is the forced vibration due to rotational unbalance force and the other is the self-excited vibration or control system instability. Both must be controlled for the safe and reliable operation of magnetically suspended rotational machinery, and many kinds of vibration suppression techniques have been presented or implemented. [1,2,3]

In constructing a vibration suppression system, it is important to check effectiveness, robustness, reliability, simplicity and cost. While it is desirable to model the rotor and control system as accurately as possible for effective reduction of vibration amplitude, it is also necessary to allow some modelling errors to keep the system robust and reliable. When applying an unbalance suppression technique, care should be taken not to affect the system stability in order to avoid self-excited vibrations. Moreover, the rotor characteristics to be modeled change with rotational speed due to the gyroscopic effect and eddy currents generated in the rotor. These effects have a great influence on the control system including vibration suppression.

In this paper a new method of suppressing the unbalance vibration is introduced which uses adaptive filtering techniques to generate an unbalance force cancelling signal. This method uses system characteristics actually tested on-line and serves as a feed-forward system which does not affect the original system stability.

ADAPTIVE FILTERING STRUCTURE

Since the unbalance vibration is a steady state output of the rotation-synchronized exciting force, which cannot be directly measured, it is necessary to estimate the exciting force as a periodic disturbance using the rotor displacement signal (the plant output) and the information of the system characteristics. This can be done by applying the technique of adaptive filters for periodic disturbance. [4,5] Fig.1 shows the system block diagram of adaptive filtering. It consists of two parts : a 1st synchronized adaptive filter or "line-enhancer" and a 2nd adaptive filter for disturbance estimation.

The rotor displacement signal is used in the 1st filter to separate and extract the rotation-synchronized components of the primary frequency and higher harmonics apart from other components of eigenvalues or random signals. The filter is of the FIR type with a finite impulse response of $w_{1,k}(i)$ at a time $t=kT/n$, where T is the primary rotational period, and n

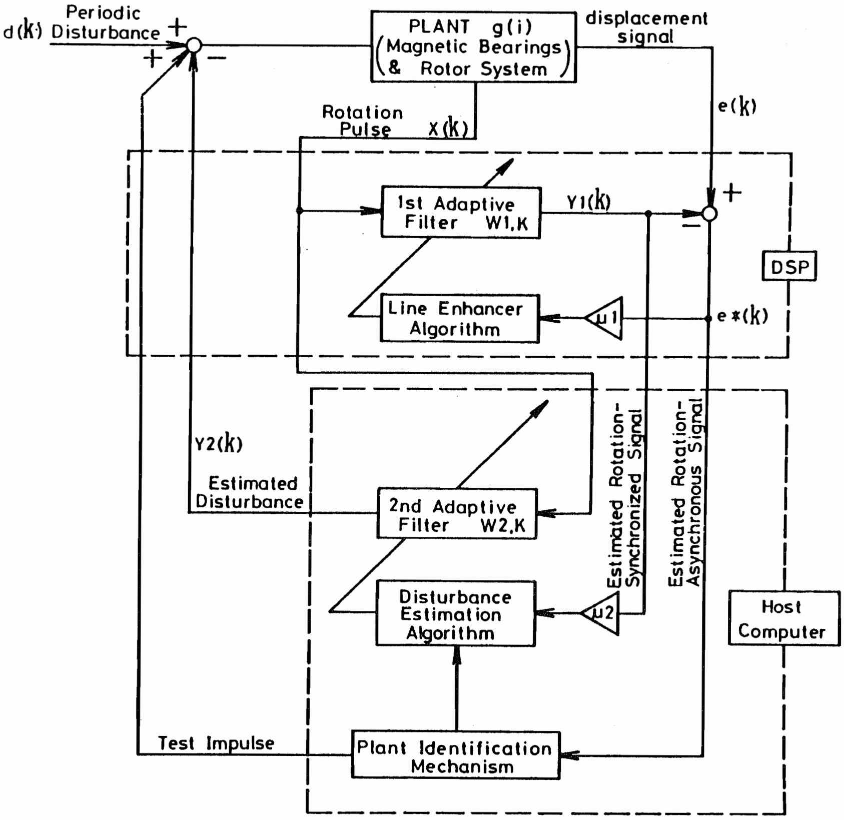

FIGURE 1: Overall System Block Diagram

is the division number. The filter output $y_1(k)$ is obtained by the convolution of input $x(k)$ and the impulse response $w_{1,k}(i)$ as:

$$y_1(k) = \sum_{i=0}^{n-1} w_{1,k}(i)x(k-i) \tag{1}$$

The input to the filter $x(k)$ is taken to be a moving impulse with the rotational period T (1 PPR):

$$x(k-i)=\begin{cases} 1 & : i=k_n \\ 0 & : \text{otherwise} \end{cases} \tag{2}$$

where k_n is the remainder of k divided by n.
This choice of $x(k)$ greatly simplifies eq.(1) to :

$$y_1(k)=w_{1,k}(k_n) \tag{3}$$

The ordinary LMS (least mean square) adaptive algorithm is used to modify the impulse response of the filter:

$$w_{1,k+1}(k_n)=w_{1,k}(k_n)+2\mu_1 e*(k) \tag{4}$$

$$e*(k)=e(k)-y_1(k) \tag{5}$$

where μ_1 is the correction factor and $e(k)$ is the plant output or the rotor displacement signal, which is to be minimized by the total adaptive system.

This 1st filter works actually as a periodically averaging process with the correction coefficient $2\mu_1$.

The 2nd filter is a disturbance estimator using the rotation-synchronized signal estimated by the 1st filter and the information of the plant characteristics. The plant of magnetic bearings and rotor system is assumed to have known characteristics represented by an infinite impulse response of $g(i)$. Then the plant output $e(k)$ is expressed as :

$$e(k) = \sum_{i=0}^{\infty} g(i)\{d(k-i)-y_2(k-i)\} \tag{6}$$

where $d(k)$ is the periodic disturbance of the unbalance exciting force to be estimated, and $y_2(k)$ is the output of the 2nd filter or the estimated disturbance. The input to the 2nd filter is again taken to be the moving unit impulse $x(k)$, the same as the input to the 1st filter. This leads to :

$$y_2(k)=w_{2,k}(k_n) \tag{7}$$

and, considering the periodicity, the gradient of the square of the error has a form as follows :

$$\partial e^2(k)/\partial w_{2,k}(j)=-2e(k) \sum_{m=m_0}^{\infty} g(k_n-j+mn) \tag{8}$$

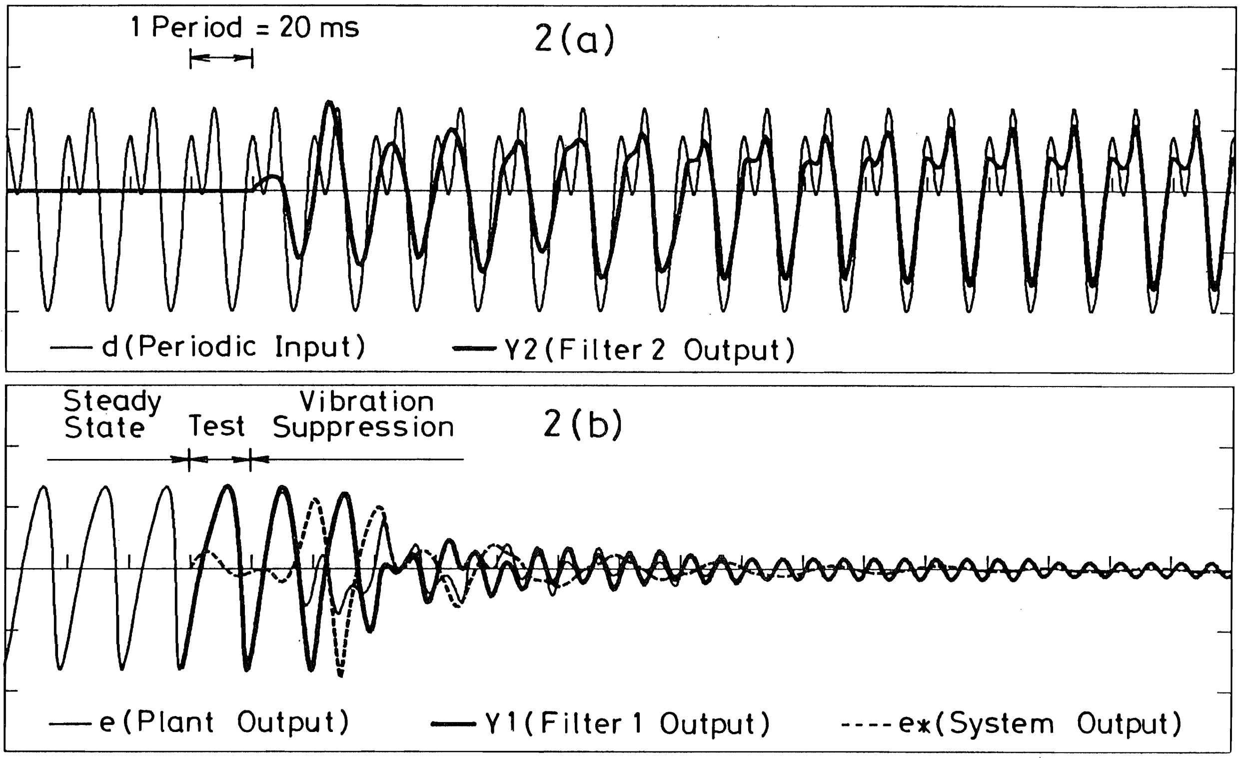

FIGURE 2: Simulation Result

where $m_0 = 0 : 0 \leq j \leq k_n$
$\quad\quad 1 : k_n < j \leq n-1$

Therefore, the correction scheme of the impulse response $w_{2,k}(j)$ is expressed as :

$$W_{2,k+1}(j) = w_{2,k}(j) + 2\mu_2 e(k) \sum_{m=m_0}^{\alpha} g(k_n - j + mn) \quad (9)$$

and μ_2 is the correction factor for the 2nd filter. The infinite response characteristic of the plant requires infinite summations in eq.(9), but actually it should be limited to a finite value. The number of impulse responses to be modified at a time k is only one for the 1st filter and n for the 2nd filter. Besides, the modification scheme for the 2nd filter (9) is much more complicated. This means that the 2nd filter takes more time and becomes more difficult to be performed with a sufficient number of divisions n as the rotational speed increases. However, both filters are very simple compared to the usual adaptive filters due to the use of the moving impulse as the input.

The plant is a rotor system suspended by magnetic bearings with a fundamental control system. Although its characteristic g(i) is assumed to be known, modelling of the rotor and control system always includes some errors, and moreover, the characteristic itself changes with the rotational speed due to the gyroscopic effect and the eddy currents induced in the moving metal. Therefore, it is necessary to measure the actual present characteristic during high-speed rotation. As indicated in Fig.1, this can be performed by imposing a test impulse and obtaining its response as the rotation-asynchronous signal e*(k) which is another output of the 1st filter.

NUMERICAL SIMULATION

The actual number of summations in eq.(9) has an influence on the calculation speed and structual simplicity of the 2nd filter. Some numerical simulations were conducted with the objectives of (1) ascertaining the basic effectiveness of the proposed method, (2) clarifying the criteria of the summation number, and (3) investigating the effect of the correction factor.

Fig.2 shows a typical simulation result. The plant is approximated with a second-order low pass filter with the cut-off frequency of 50 Hz and Q-value of 1.4. This corresponds to a rigid rotor model suspended by magnetic bearings so that it has an eigen-frequency of 50 Hz and damping ratio ζ of 0.35. The fine line in Fig.2(a) is the periodic disturbance d(k) of 50 Hz with its second harmonic of 100 Hz mixed together which should be approximated by the 2nd filter output $y_2(k)$ indicated by the bold line in Fig.2(a).

The fine line in Fig.2(b) is the plant output e(k), the periodic component of which should be estimated by the 1st filter output $y_1(k)$ indicated by the bold line in Fig.2(b). One increment of the horizontal time coordinate is 20 ms, the primary cycle period of the disturbance, and the 1st 3 cycles show a steady state where only the 1st filter is working. As the disturbance frequency coincides with the plant eigen-frequency, the system is in a resonant state.

At the beginning of the 4th cycle, a test impulse is imposed on the plant, and its response is measured as an output of the 1st filter $e*(k)$, which is shown by the dashed line in Fig.2(b). Then from the 5th cycle, the 2nd filter begins to work to estimate the periodic disturbance using the information of the plant characteristic just obtained. It can be seen that the 2nd filter output $y_2(k)$ approaches d(k) including the harmonic, and the plant output e(k) smoothly converges to zero. In this case, only one period is used for measuring the plant impulse response, and this means that only one summation is used in eq.(9).

Fig.3 shows the effect of the number of summations in eq.(9). The vertical coordinate is the number of iteration cycles for the plant output to converge to within 5% of its steady state RMS value. Two cases of rotational speeds as the disturbance frequency are plotted in Fig.3, both indicating little effect of the summation number. Therefore, for simplicity of the software, just one summation would be sufficient.

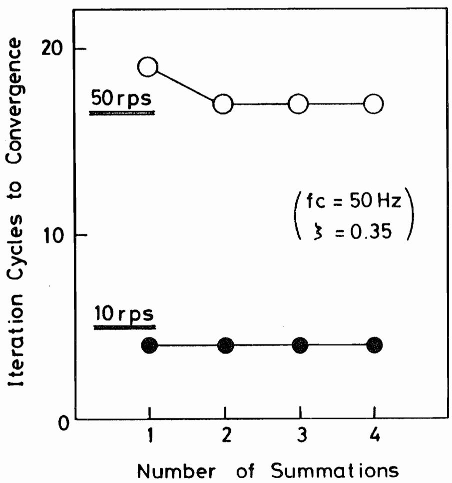

FIGURE 3: Effect of the summation number

The value of the correction factor μ_2 for the 2nd filter has much more influence as indicated in Fig.4. It has an optimum value which minimizes the number of convergence cycles, and a higher limit for stability. The optimum value increases with the disturbance frequency and the plant damping ratio.

The modification of the 2nd filter response using eq.(9) is presently performed at every sampling or for each value of k. However, when the two kinds of filtering tasks are performed by different computers, say, a DSP and a personal computer, it is more convenient to calculate eq.(9) n times at the end of a rotational period. This method of calculation has been also simulated, and the result is that it also works effectively, but is slightly slower in convergence (about 50 % more convergence cycles) and needs a slightly lower (≃0.8 times) correction factor.

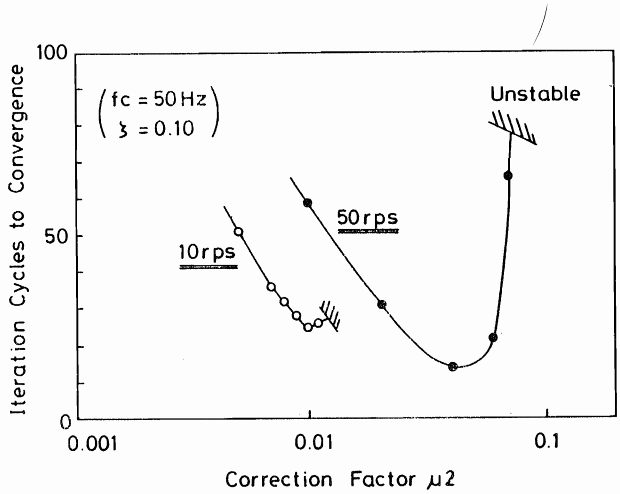

FIGURE 4: Effect of Correction Factor

EXPERIMENTAL RESULTS

Fig.5 shows the experimental apparatus of a flexible rotor. The rotor is supported by magnetic bearings; an axial bearing, two radial bearings, and an auxiliary radial bearing. A reluctance motor is built in for driving. The design speed is 30,000 rpm and the first bending-mode critical speed is 24,000 rpm. The adaptive filtering technique proposed was applied to the auxiliary bearing, and synthesized rotor displacement signals in two perpendicular directions were estimated at the location of the auxiliary bearing.

As shown in Fig.1, a DSP was used to perform the 1st filtering function together with the output of the disturbance cancelling signal and the test impulse. This task is rather simple, but fast action is required. The minimum unit sampling time of the DSP was 32 μs. A personal computer was used as a host computer which performs the 2nd filter disturbance estimation, plant identification, and overall control. They communicate with each other by hand-shaking and transfer data through a 2-port RAM.

As mentioned in the previous section, the calculation scheme of the 2nd filter after one rotational period was adopted for convenience. The plant output data for one period is stored in the 2-port RAM, and the host computer uses it for calculating eq.(9) not impeded by the necessity of adjusting the timing with the DSP.

Typical data is shown in Fig.6. The rotational speed

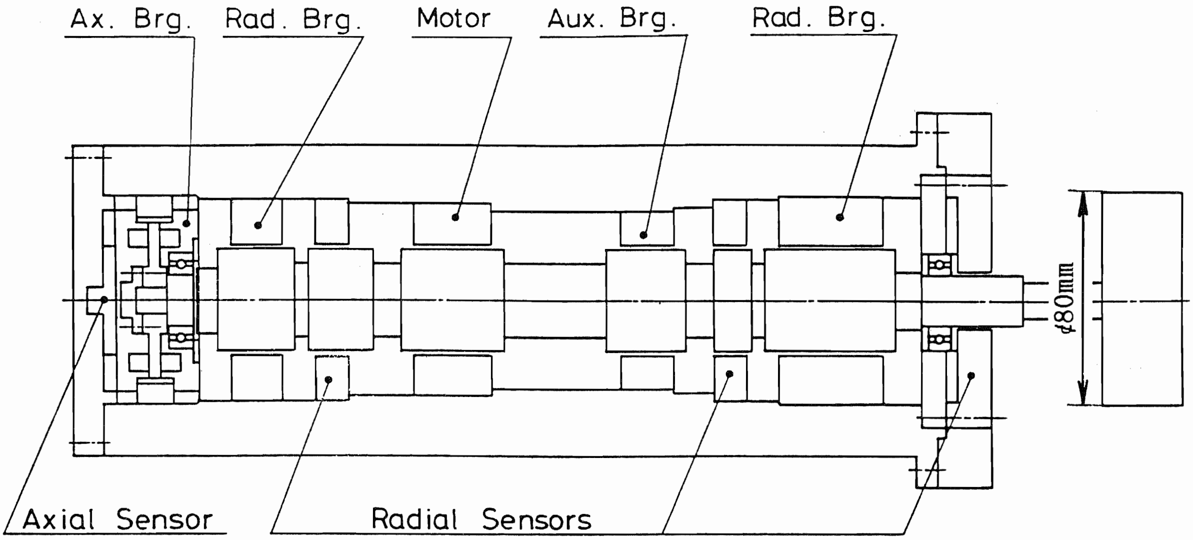

FIGURE 5: Test Apparatus

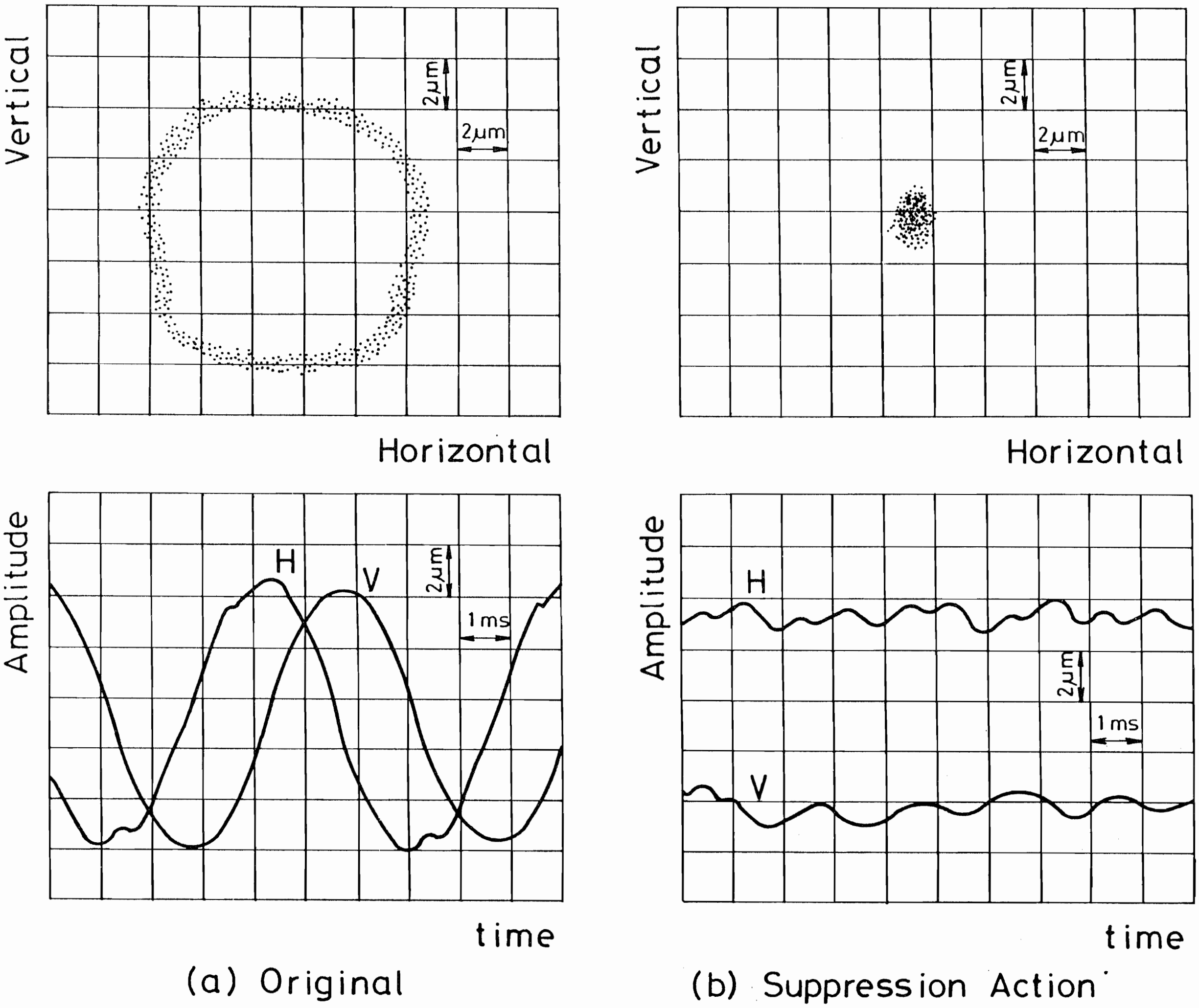

FIGURE 6: Vibration Amplitude

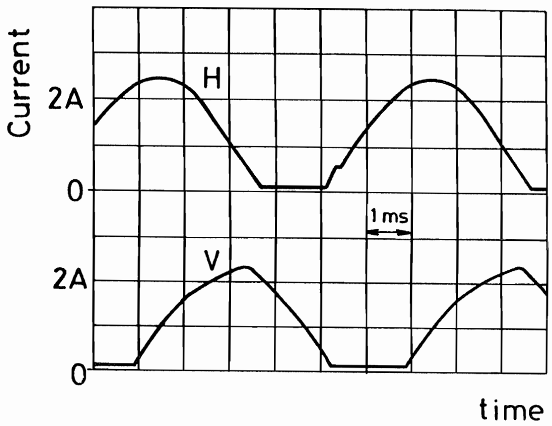

FIGURE 7: Control Currents

is 10,000 rpm. The original vibration amplitude is about 10 µm,p-p, and the waveform is slightly distorted (Fig.6(a)). Fig.6(b) shows the situation after adaptive filtering was performed for 27 iteration cycles. The amplitude decreased to less than 1/10 of the original value, indicating the practical effectiveness of the method.

Fig.7 shows the current waveforms flowing in the auxiliary bearings during the suppression action. They were originally constant bias currents of 1A. In the suppression action they vary from 0 to over 2A, saturated and distorted, showing that the method works well even involving the non-linearity.

The iteration cycle time of the 2nd filter was several seconds, and the action response time was much larger compared to the 1st filter. However, faster and more powerful work stations or another DSP can be used for quicker action of disturbance suppression.

The rotor system described here is considered to be a black box of single-input and single-output for both horizontal and vertical directions. It can be expanded into a multiple-input and multiple-output system which can deal with signals for two or more bearings.

CONCLUSIONS

A new method of suppressing the unbalance vibration was introduced which uses adaptive filtering techniques for a rotor system suspended by magnetic bearings, and the following conclusions were obtained.

(1) The method works well in a resonant state not only for the primary vibration component but also for higher harmonics.
(2) An infinite summation process due to the infinite response of the plant was proved to be unnecessary.
(3) The correction factor used in the adaptive filter for estimating the periodic disturbance has an optimum value which varies with the plant eigenvalues and the disturbance frequency.
(4) The practical effectiveness of the method was experimentally verified, including the non-linearity of the system.

REFERENCES

1. Kanemitsu,Y.,Ohsawa,M., and Watanabe,K., "Real Time Balancing of a Flexible Rotor Supported by Magnetic Bearing", 2nd Int. Symp. on Magnetic Bearings, 1990, Tokyo
2. Higuchi,T.,Otsuka,M.et.al., "Application of Periodic Learning Control with Inverse Transfer Function Compensation in Totally Active Magnetic Bearings", 2nd Int. Symp. on Magnetic Bearings, 1990, Tokyo
3. Beale,S.,Shafai,B.,et.al.,"Adaptive Forced Balancing for Magnetic Bearing Systems", 3rd Int. Symp. on Magnetic Bearings, 1992, Alexandria
4. Widrow,B. and Stearns,S.D., "Adaptive Signal Processing", PRENTICE-HALL
5. Obata, "Synchronous Adaptive Filter", Transactions of SICE, Vol.19, No.3,(1983) (in Japanese)