

SELF-TUNING ADAPTIVE CONTROLLER USING ONLINE FREQUENCY IDENTIFICATION

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ABSTRACT

A real-time adaptive controller has been designed and tested successfully on a fourth order laboratory dynamic system which features very low structural damping and a non-colocated actuator-sensor pair. The controller, implemented in a digital minicomputer, consists of a state estimator, a set of state feedback gains, and a Frequency-Locked-Loop (FLL) for real time parameter identification. The FLL can detect the closed-loop natural frequency of the system being controlled, calculate the mismatch between a plant parameter and its counterpart in the state estimator, and correct the estimator parameter in real time. The adaptation algorithm can correct the controller error and stabilize the system for more than 50% variation in the plant natural frequency, compared with a 10% stability margin in frequency variation for a fixed-gain controller having the same performance at the nominal plant condition. After it has locked to the correct plant frequency, the adaptive controller works as well as the fixed-gain controller does when there is no parameter mismatch. The very rapid convergence of this adaptive system is demonstrated experimentally, and can also be proven with simple root-locus methods.

I. INTRODUCTION

A controller using Kalman filter and full state feedback usually has good performance, provided a very accurate model of the plant is known. But such controllers are very sensitive to parameter variation, especially when the plant has very low inherent damping, and when the sensor is not colocated with the actuator.

A two-disk laboratory model, consisting of two inertia disks connected by a torsion rod, which has a structural damping of 0.004, and with separated sensor and actuator locations was constructed to test several adaptive controller designs. The form of the equations of motion of the model is known due to the ease of analysis of the lumped system; but the lack of accurate knowledge about the natural structural frequency during controller design corresponds to a plant parameter uncertainty or variation; and this uncertainty is what the adaptive controller handles.

It has been proposed by Kopf, Brown, Marsh (Ref.1) and Macala (Ref.2) to use a Phase Locked-Loop to implement tuned damping and notch filtered command torque, so that the feedback control force near the structural frequency can be adjusted properly according to the natural frequency of the plant. Rosenthal and Cannon (Ref.3) have implemented such a kind of controller for the two-disk experimental system.

Under the same research project, a different approach using a Frequency-Locked-Loop (FLL) to identify the plant frequency was developed. This paper describes in detail how the FLL identifies the unknown plant parameter and updates the controller in real time.

II. DESCRIPTION OF THE TWO-DISK PLANT AND FIXED-GAIN CONTROLLER

The plant to be controlled is a mechanical system which consists of two horizontal steel disks connected by a vertical elastic steel rod. The two disks are supported by bearings which allow rotational motion only. A low-friction DC motor is attached to the lower disk, and an RVDT sensor detects the angular position of the upper disk.

If structural damping is neglected*, the state equation of motion of this system can be expressed as

$$\begin{pmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{pmatrix} = \begin{pmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -\omega_n^2 & 0 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{pmatrix} + \frac{1}{J} \begin{pmatrix} 0 \\ 1 \\ 0 \\ -1 \end{pmatrix} u, \quad (1)$$

where x_1 and x_3 are the position states of the rigid body mode and the structural oscillation mode respectively, x_2 and x_4 are rates of those states respectively; ω_n is the natural frequency, J is the total moment of inertia of the two disks, and u is the control torque from the DC motor.

The sensor output is

$$y = x_1 + x_3. \quad (2)$$

A first-order high-pass filter with 100 Hz cutoff frequency is used to differentiate the position sensor output and provides the pseudo-rate of the top disk.

If all the parameters of the plant are known accurately, an LQG design (Ref.4) will result in a set of state feedback gains C for regulation and estimator gains L for state estimation. However, if the plant natural frequency ω_n is not known by the controller designer, and a value ω_e is used in the estimator, the stability of the whole system has to be analyzed by augmenting the system state equations with those of the estimator states, and finding the modal frequencies and dampings of the system (Ref.5)

Using the same penalty weightings for control effort and state errors, an LQG design produces different feedback gains C and L for different natural frequencies ω_n of the plant. Analysis shows that the stability of the whole system is less sensitive to those feedback gains than to the parameter ω_e used in the estimator, since an error in the latter parameter corresponds to a modeling error, while variations in the former ones correspond to different weightings in the LQG design process. In the experiment described here, feedback gains C and L are chosen for the nominal plant frequency, and are kept constant in order to demonstrate the adaptation of the controller by correcting ω_e in the estimator.

From the analysis of the augmented system state equations, the frequency ω_c of the most unstable closed-loop mode can be found as a function of ω_n and ω_e , if all other parameters are kept constant. This function

$$\omega_c = f(\omega_e, \omega_n), \quad (3)$$

will affect the closed-loop performance of the adaptation process, and has to be taken into account in the design process. The two-disk model has a nominal frequency of 13.3 rad/sec, and the function described in equation (3) can be shown approximately as in Fig. 1, and can be approximated as

$$\omega_c - \omega_n = (\omega_n - \omega_e) + 0.6 \quad (4)$$

for $|\omega_n - \omega_e| < 1.5 \text{ rad/sec}$.

III. FREQUENCY IDENTIFICATION USING FREQUENCY-LOCKED-LOOP

A Phase-Locked-Loop (PLL) was initially proposed to be used to detect the vibration frequency. PLLs have been used widely in locking onto high-frequency signals in electrical engineering applications, but it

* It is actually 0.004

is only beginning to be used in locking onto low-frequency signals in mechanical systems. A PLL has the ability to identify the phase and frequency of a signal contaminated by a relatively large amount of noise at other frequencies. Several signal components at different frequencies can be identified by using several PLLs.

The traditional PLLs are nonlinear elements for which the performance is hard to analyze and predict; and they have limited locking ranges due to their nonlinearity. Besides, PLLs are more sensitive to the phase than to the frequency of their driving signal, which makes them unsuitable for frequency identification because the identification will be disturbed by the phase in the sensor signal every time a new position command or an external disturbance is applied to the system, even though a PLL has identified the correct plant frequency already.

A modification is made to a PLL to eliminate its sensitivity to phase in the input signal and make the input/output relation linear in a larger tracking range, so that it works better for frequency identification, while retaining the other virtues of PLLs. The final product, called a Frequency-Locked-Loop (FLL), is shown schematically in Fig. 2, and its input/output relation can be seen from the functional block diagram in Fig. 3, where ω_i is the frequency of the input signal and ω_o is the output signal – the frequency detected by the FLL. Also shown in the same block diagram are ω_s , the starting oscillation frequency; $\Delta\omega$, the correction on the output; and ω_{er} , the error of the output of the FLL.

The character of the block $G(s)$ can be chosen arbitrarily by the designer as long as it can update the output frequency of the FLL according to its error ω_{er} . If a simple integrator $\frac{G}{s}$ is chosen as the element $G(s)$, then the FLL will have a pole at $-K$ where

$$K = \frac{G(a-b)}{ab} \quad (5)$$

Parameters a and b should be determined with the following restriction

$$\omega_n > a > b > |\omega_i - \omega_o| \quad (6)$$

In the present case,

$$\omega_n = 13.3 \text{ rad/sec}, \quad (7)$$

and the linear search range is chosen to be

$$|\omega_i - \omega_o| = \frac{\omega_n}{4} = 3.3 \text{ rad/sec}. \quad (8)$$

The pole location $s = -K$ should be determined as the result of a compromise between speed of response and noise rejection, at the nominal locking frequency range. In this case, the parameters of the FLL are chosen as

$$a = 6.0, \quad b = 4.0, \quad G = 20.0, \quad \Rightarrow \quad K = 1.67, \quad (9)$$

to work in the range of 1 to 3 Hz.

With parameters chosen as above, the block diagram in Fig. 3 can be simplified to the transfer function

$$Q(s) = \frac{\Omega_o(s)}{\Omega_i(s)} = \frac{K}{(s + K)} \quad (10)$$

Fig. 4(a) shows the test result of the FLL output when the frequency of the input signal is changed stepwisely. The response for small input change (the first change in Fig. 4(a)) is similar to the step response of a first-order filter with pole at $-K$, as shown in Fig. 4(b). The response for a larger input change (the second

change in Fig. 4(a)) experienced some nonlinearity at the beginning because its internal structure is not linear; however, the FLL still tracked the input signal and provided the correct output in a reasonable time.

IV. CORRECTION OF PARAMETER ERROR IN THE CONTROLLER

Because eigenvalues are properties of the system, they are independent of the instantaneous value of state variables and are influenced only by changes of parameters. The relation between ω_e and ω_c , as shown in Eqn. 4, can be expressed as in Fig. 5. Using the difference between $(\omega_o - 0.6)$ and ω_e to update - through the integrator $\frac{H}{s}$ - the parameter ω_e in the controller, the closed loop dynamics of the parameter variation, identification, and correction can be expressed as in Fig. 6. The characteristic equation of the closed parameter adaptation loop is

$$1 + \frac{HK}{(s+H)(s+K)} = 0, \quad (11)$$

or,

$$s(s+K) + (s+2K)H = 0, \quad (12)$$

which can be written in Evan's form as

$$\frac{s(s+K)}{(s+2K)} = -H. \quad (13)$$

The root locus of Eqn. 13 vs. the positive value of H with $K = 1.67$ is shown in Fig. 7, and the value of $H = 9.9$ is chosen obviously to maximize the adaptation rate. The change of the slope in Fig. 1 corresponds to a variation in the gain in Eqn. 4, and Eqn. (11) can be modified as

$$1 + \frac{rHK}{(s+H)(s+K)} = 0, \quad (14)$$

where $2 > r > 0$, and the root locus shown in Fig. 8 guarantees the stability of the system over the range of the gain "r".

Any sensor measurement, controller state variable, or linear combination thereof can be chosen as the input signal to drive the FLL, so long as the signal contains the modal frequency of interest (the larger the better!). The error between the sensor rate and the estimate of it is chosen to drive the FLL, since there is less error signal if all parameters in the controller are correct.

The FLL must be turned off if its input signal is too small, in order to reject the influence from random noise.

A PDP-11/23 minicomputer was used to implement the controller and the FLL at 25 Hz sample rate. The test results of this adaptive system are summarized in the following section.

V. EXPERIMENTAL RESULTS

Fig. 9 shows the natural oscillation of the uncontrolled disk system. The frequency of oscillation is 2.11 Hz. with 0.004 damping. (The long-period motion is caused because the disk system is hung from the ceiling with a long steel wire to reduce the axial thrust on bearings. This model is approximated as a rigid body mode in the controller design analysis.)

Fig. 10 shows the step response of a nonadaptive control system designed with the LQG method. The response is very good (Fig. 10) when there is no modeling error in the controller design. However, as Fig. 11 shows, the system becomes unstable when there is 10% modeling error in frequency in the designing of the nonadaptive controller.

When the FLL is used in the adaptive control, the system can detect and correct a controller's parameter error of 50% or more in frequency. Figs. 12 (a) through (e) show the sensor output in different tests. The instability due to the initial parameter error is shown when the control system was just turned on, and the system was then stabilized after the adaptation algorithm had corrected the controller's error. The initial turn on of the control system and the time when position commands are changed are marked on those recordings.

Fig. 13 shows the comparison of the impulsive disturbance response, between the nonadaptive controller with no modeling error and the adaptive one after its parameter error has been corrected. The comparison shows almost no difference between their performances.

VI. DISCUSSION

(A) Frequency-Locked-Loop

The FLL is a nonlinear element, but its input/output relation is almost linear. It behaves linearly for 40% changes in input signal frequency, and still works for 100% change in frequency in the nonlinear region. The test recorded in Fig. 4 attests to the discussion above. The linear range can be chosen by selecting parameters properly.

The FLL still works when the amplitude of its input signal is as weak as two quantization intervals of the A/D converter, if it is free of noise and bias; but in real applications it must be turned off at small level of input signal to reduce the effect of noise.

The FLL can identify the plant characteristic in a small window of the frequency spectrum, so that the effects of other parts of the system dynamics do not have to be taken into account if they are not critical to the overall performance. It can only detect modes that are either only slightly damped or unstable, since they can provide oscillatory signals for detection; however, heavily damped modes are usually robust to parameter uncertainty and don't need adaptive control.

(B) Parameter Error Correction Loop

The parameter error correction scheme can be determined by root-locus analysis, or even by the LQG method, since the FLL has a linear characteristic.

Fig. 12 shows some small-amplitude vibration building up due to the lack of signal to lock the FLL, but the parameter estimate error was soon corrected and vibration suppressed.

By examining the response to command change and to disturbances, it is found that the Self-Tuning Adaptive Controller behaved almost the same as the correct fixed optimal controller, except for the few cycles of vibration at the beginning when the parameter error was being corrected.

It is better to use the error of an estimated sensor output to drive the FLL, since it is undisturbed by the control force during a new command change if the model is correct.

Both the identification and error correction are running in real time while the controller is doing its job. Any change in the plant can be tracked and adapted to rapidly.

VII. CONCLUSION

The use of FLL in identifying system vibration frequency and adapting controller parameters is promising. All kinds of controllers, such as Kalman filter and state feedback, band-pass, or notch filters can have their parameter errors corrected in a similar way. It is expected that system with many vibration modes can

be handled with several FLLs.

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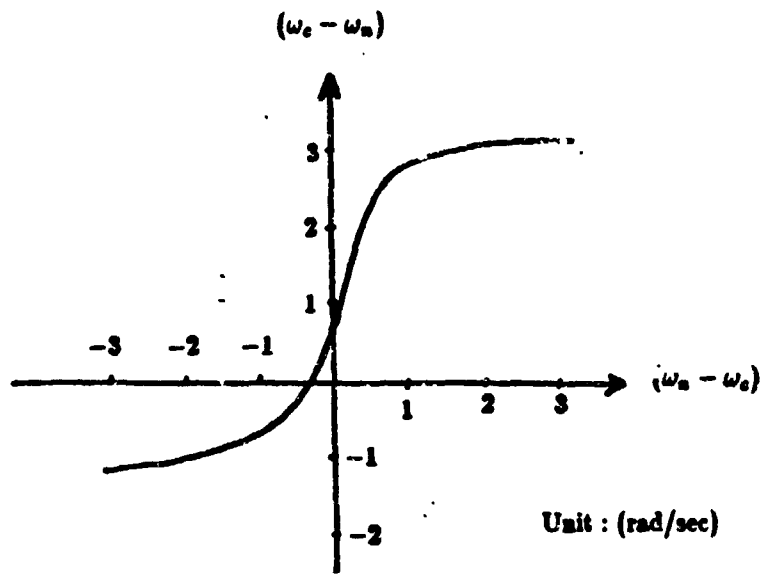
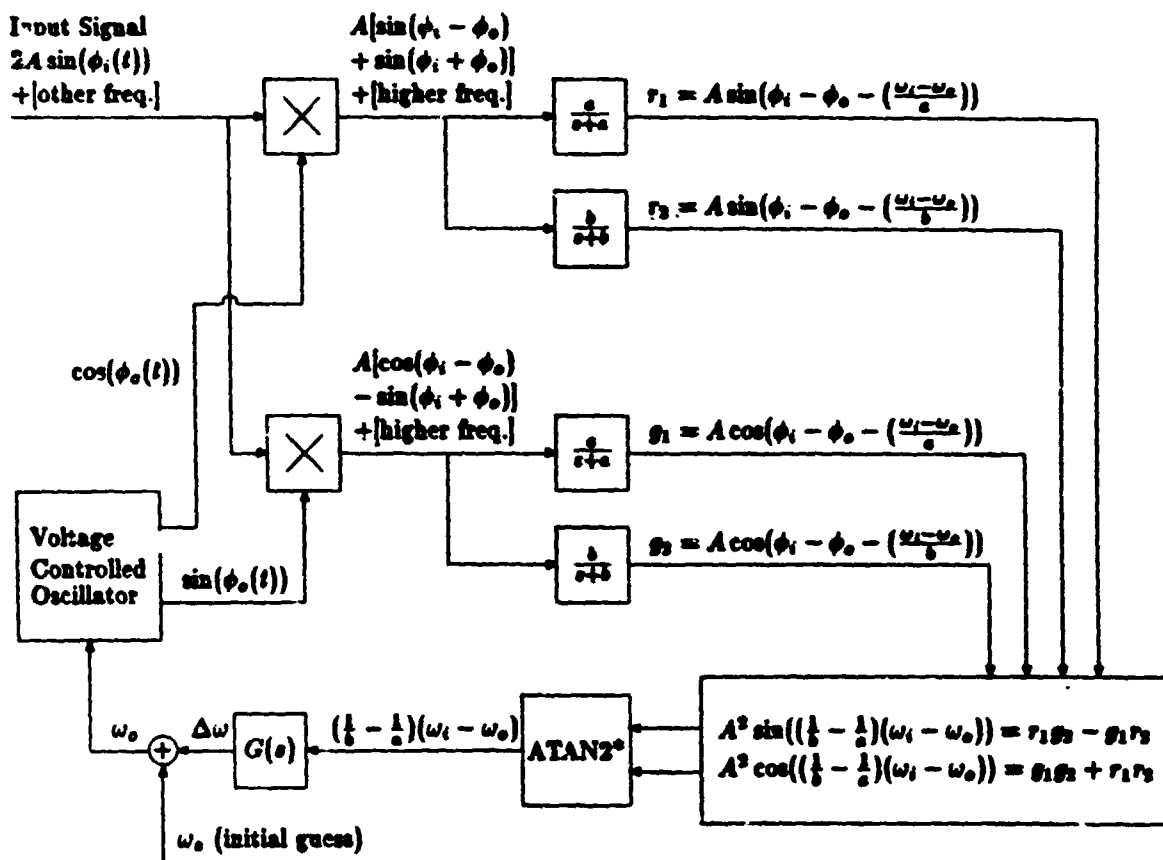


Fig. 1 Relationship between ω_n , ω_c , and ω_s .



Choose $a > b$

*ATAN2 is a FORTRAN arctangent function which keeps tracking the correct quadrant of the angle.

Fig. 2. Diagram of the FLL Implementation.

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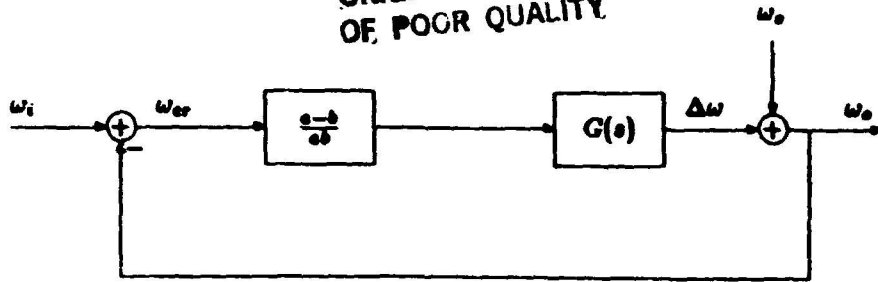


Fig. 3 Functional Block Diagram of the FLL.

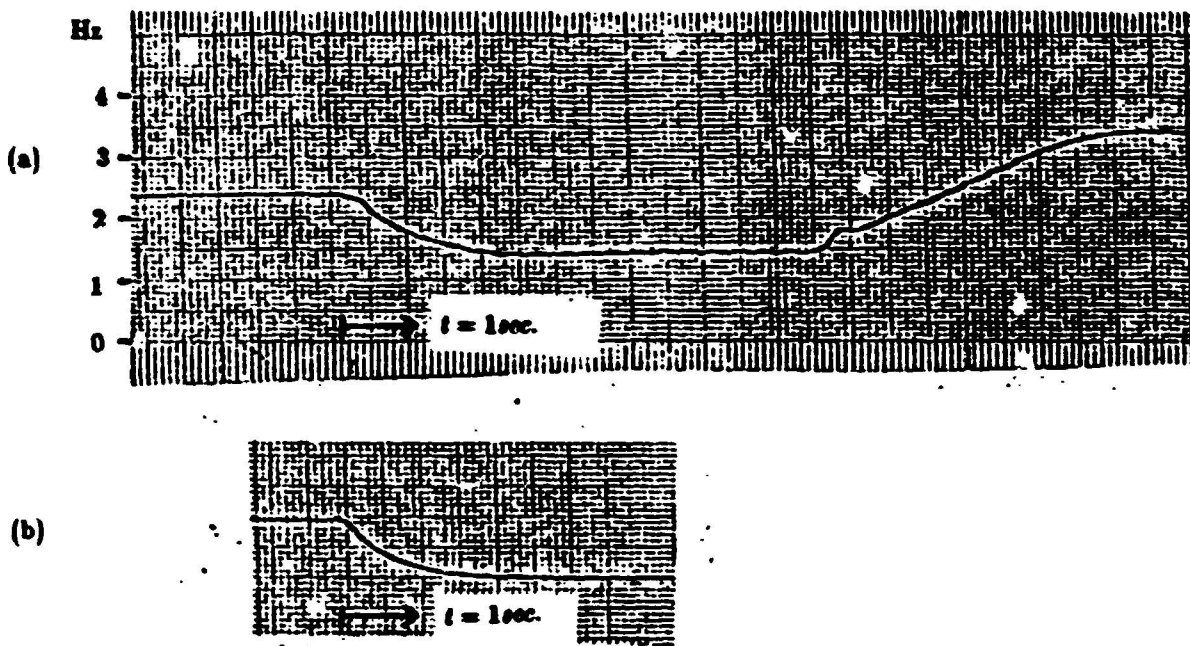


Fig. 4 Step Response of the FLL (a), Compared with That of a First-Order System (b) with Pole at $s = -1.67\text{sec}^{-1}$.

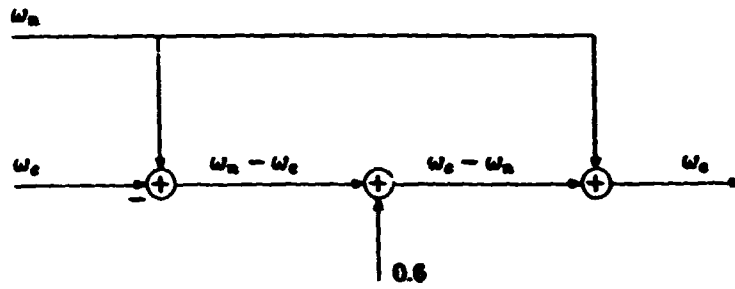


Fig. 5 Block Diagram of the Relation Between ω_n , ω_e , and ω_c .

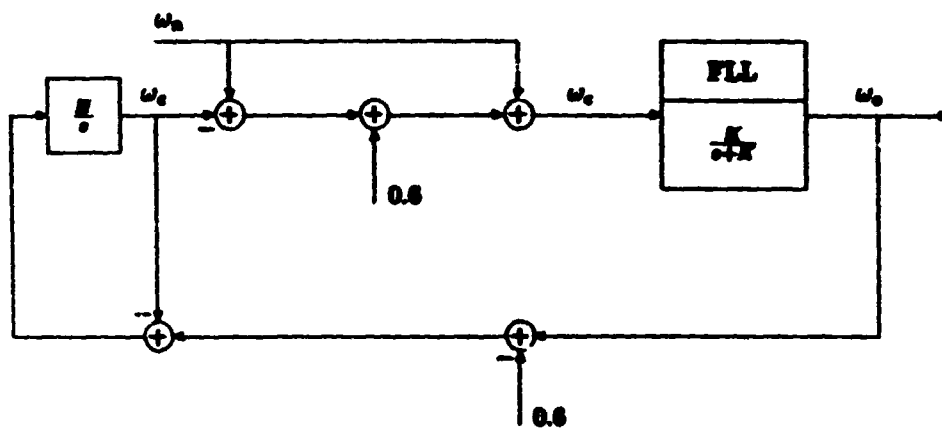


Fig. 6 Closed-Loop Dynamics of Parameter Variation, Identification, and Correction.

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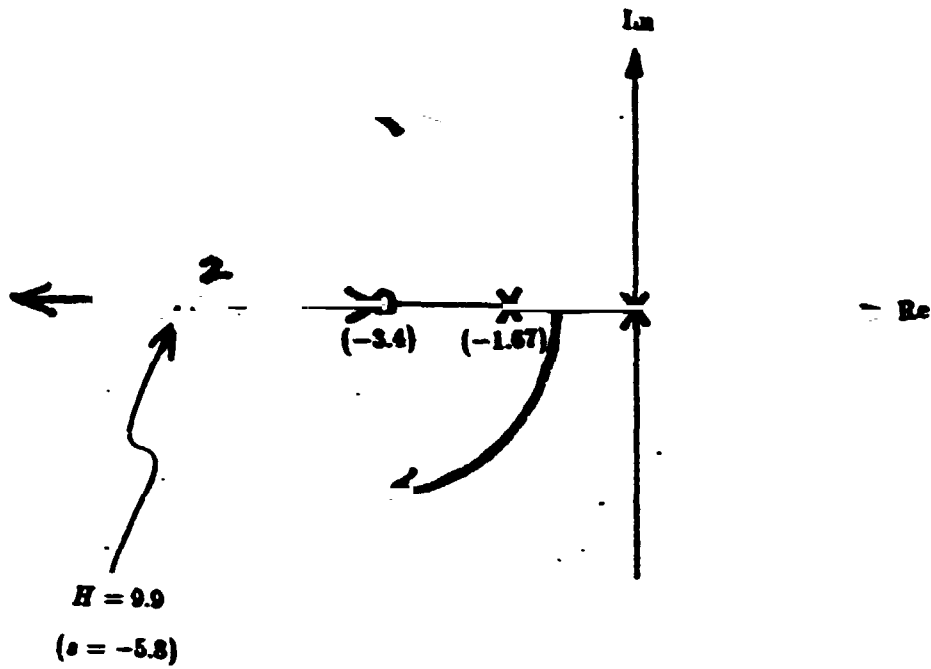


Fig. 7 Variation of Poles of the Closed Parameter Loop versus the Selection of the Value of H .

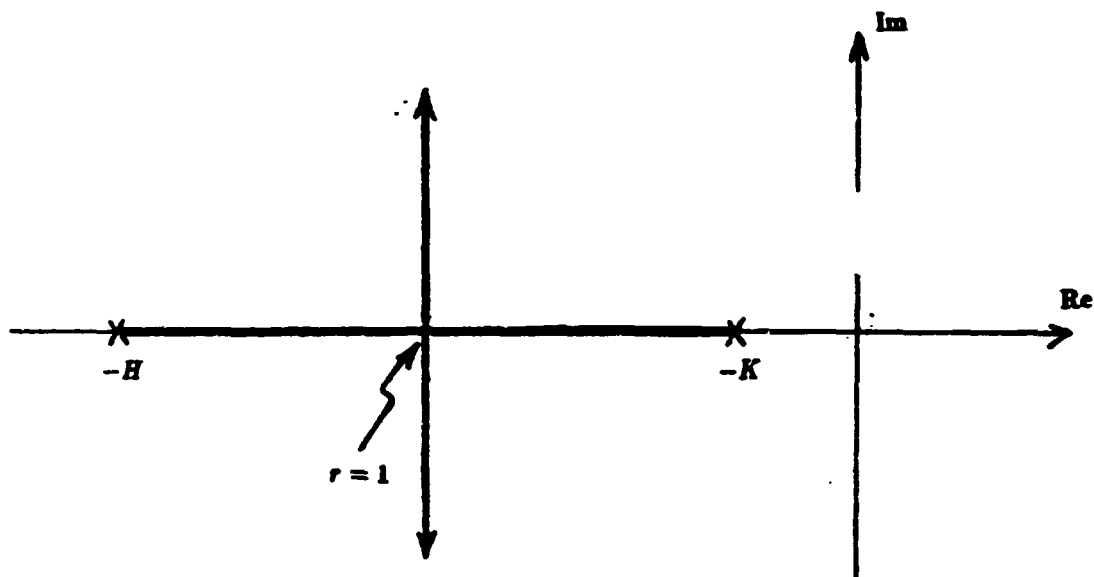


Fig. 8 Variation of Poles of the Closed Parameter Loop versus r , the Changing Slope in Fig. 1.

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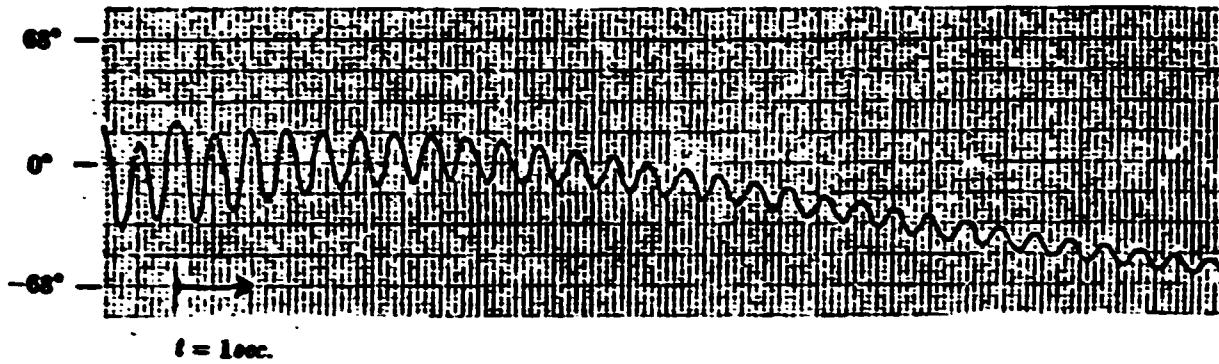


Fig. 9 Natural Vibration of the Plant (Opened Loop) at 2.11 Hz. with Damping = 0.004.

* The long-period motion is caused because the disk system is hung from the ceiling with a long steel wire to reduce the axial thrust on bearings.

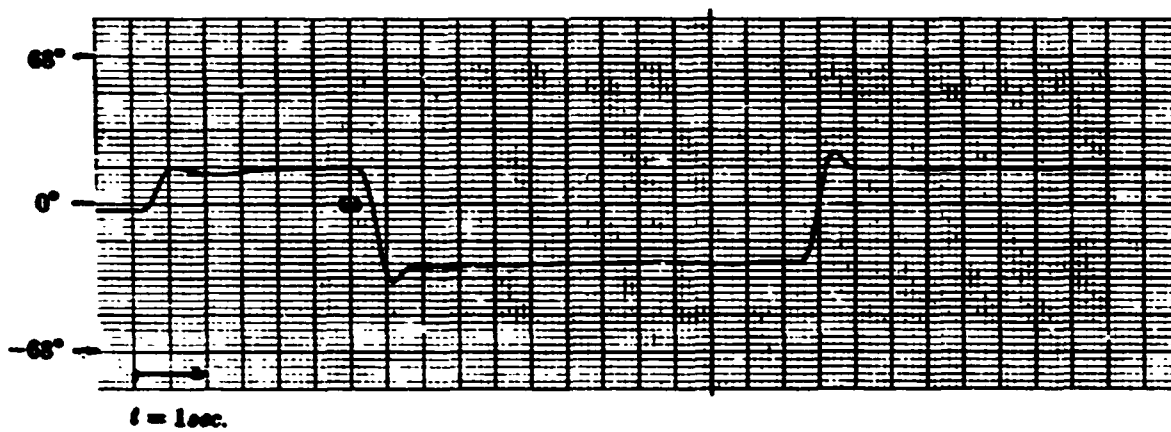


Fig. 10 Step Response of the Closed Loop with a Fixed "Optimal Controller".
(No model error in the Kalman filter.)

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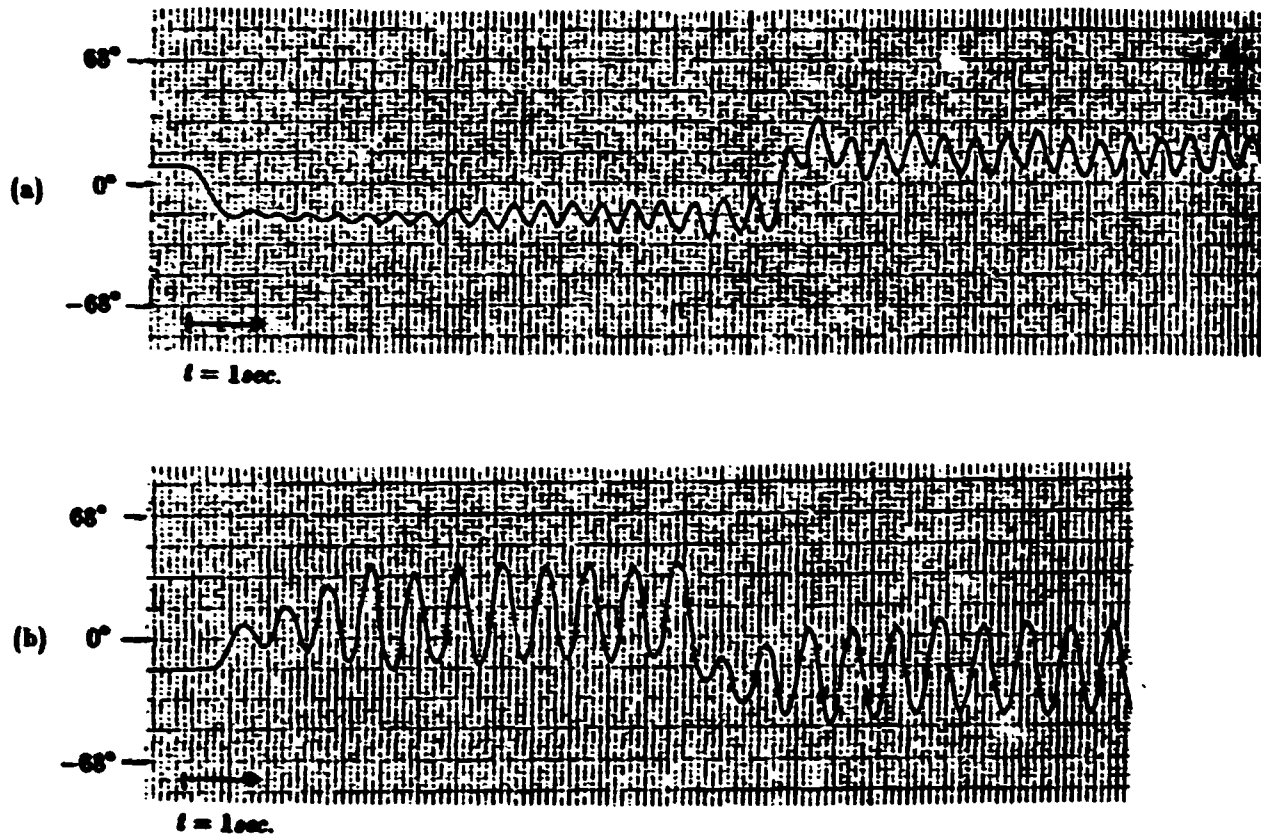
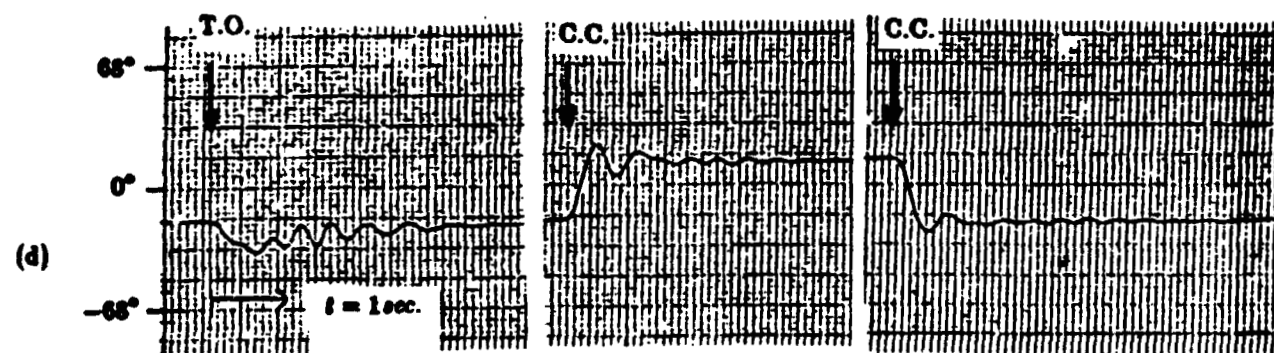
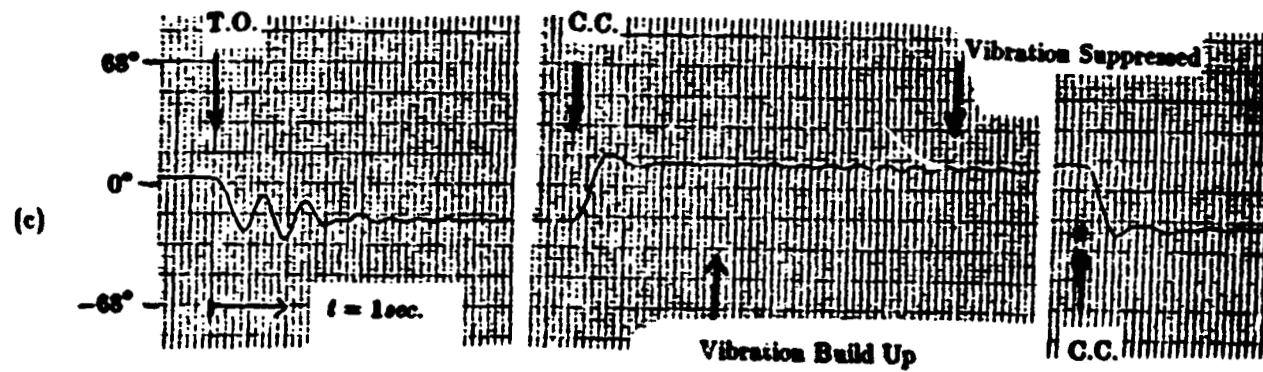
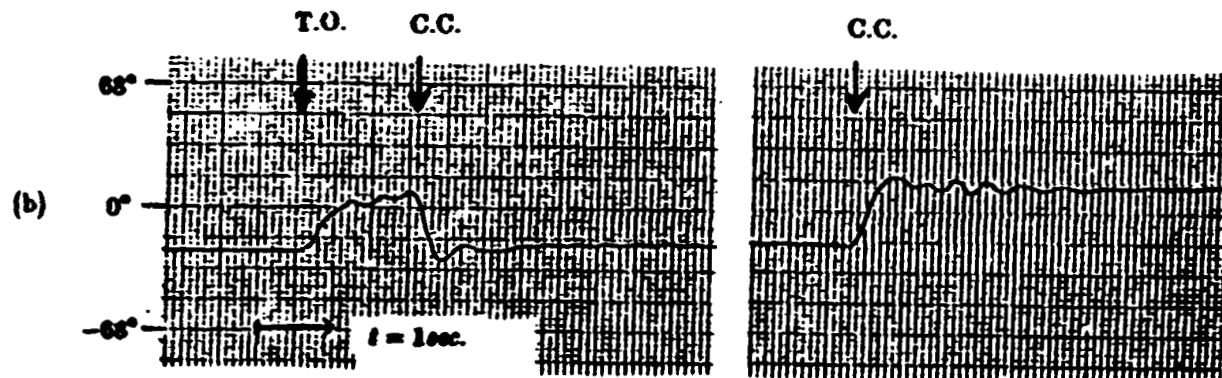
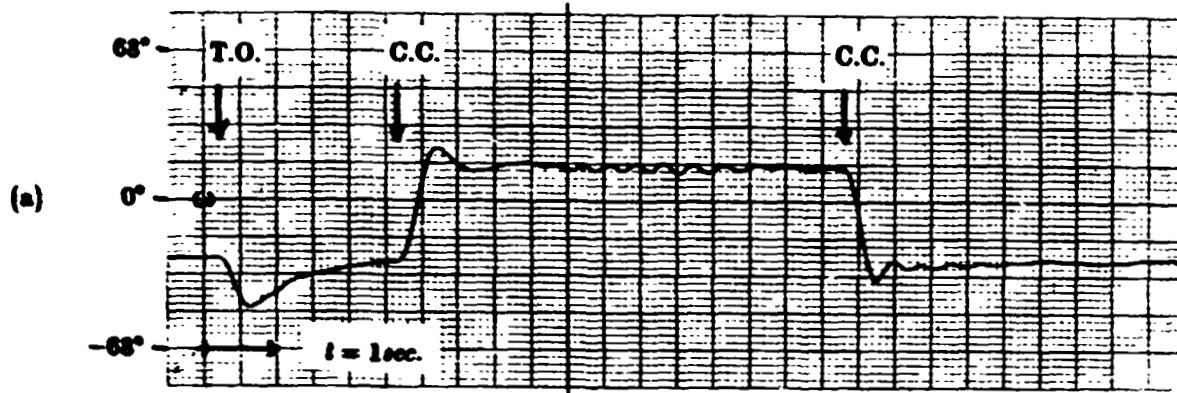


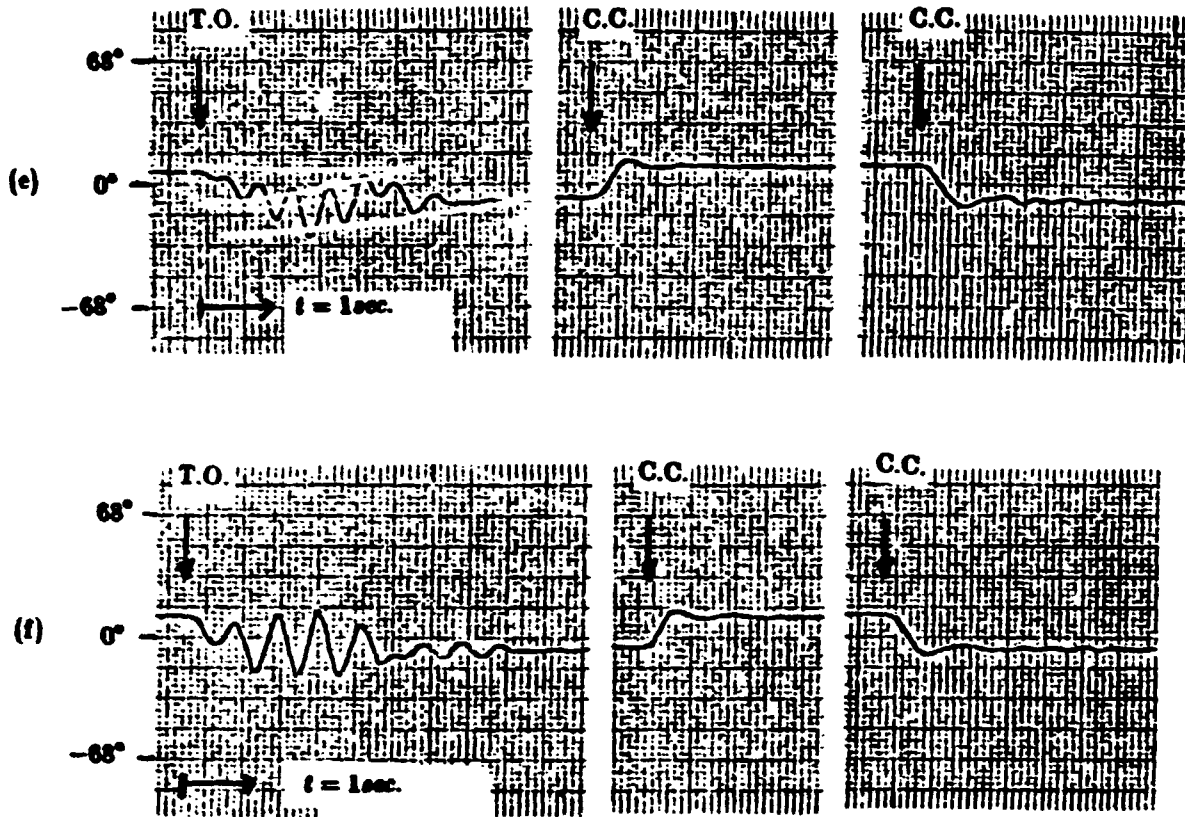
Fig. 11 Step Response of Closed Loop with Fixed "Optimal Controllers".

- (a) Vibration frequency was assumed to be 1.9 Hz. (-10% error in frequency) in Kalman Filter.
- (b) vibration frequency was assumed to be 2.5 Hz. (+20% error).

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- T.O. : initial turned on.
- C.C. : step command change.

Fig. 12 Step Response of the Adaptive Controller with FLL Detecting Initial Modeling Error in Plant Frequency.

- (a) no error.
- (b) -10% error.
- (c) +25% error.
- (d) -25% error.
- (e) -50% error.
- (f) +50% error.

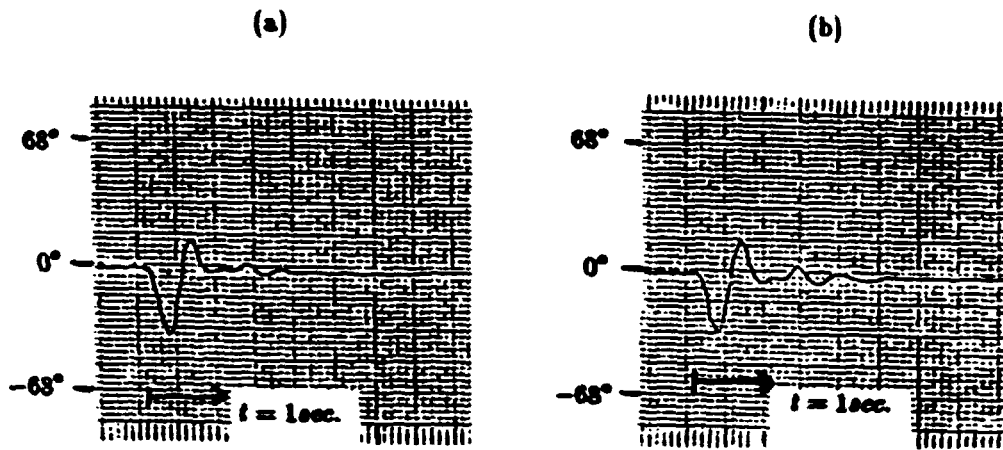


Fig. 13 Comparison of the Impulsive Disturbance Response between (a) the Nonadaptive Controller with No Modeling Error and (b) the Adaptive One after Its Parameter Error Has Been Corrected.