A Nonlinear PID Control Scheme for Hard Disk Drive Servosystems

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Abstract — The aim of this paper is to put forward a novel design of a nonlinear PID controller type based on carefully choosing nonlinear gains to enable reducing the settling time, minimizing the overshoot and improving the required control effort during the functions of the read/write (R/W) head positioning servomechanism in hard disk drives. The controller is capable of adjusting a driving force, a damping effect and an integral action according to the position of the actual system output with respect to the desired output. This is carried out by utilizing the three actions of the PID controller which are tuned by different nonlinear functions to provide the system with the needed level of a driving force, a damping and a steady-state error correction action. Simulation results, including step function response, control signal history, and a disturbance rejection capability are presented to show the efficiency of the proposed controller.

I. INTRODUCTION

In hard disk drives (HDDs), data is written on and read from rotating disks coated with a thin magnetic layer (recording medium). The disks rotate at a high speed, driven by a spindle motor. Data bits are arranged in concentric circles called tracks. Data is magnetically read or written by an almost radially traversing read/write heads.

The recording heads are moved by the voice coil motor (VCM) across the disk surface to reach a specific track. The head positioning servomechanism is a control system which positions the heads over a desired track with minimum deviation from the track center and repositions the heads from one track to another in minimum time. For computer system performance reasons, it is desirable to reach any track as quickly as possible.

In general, the two main functions of the R/W head positioning servomechanism in disk drives are track seeking and track following. During the track seeking stage, the R/W head moves from its present track to a specified destination track in minimum time using a bounded control effort. The system then switches to the track-following stage to precisely position the R/W head on the target track center and subsequently keep it as close as possible to the track center as information is being read from or written on the disk.

Current hard disk drives use a combination of classical control techniques, such as the time optimal control technique in the track seeking stage and lead-lag compensators and linear proportional-integral-derivative (PID) compensators in the track following stage. However, because of higher demands of a good performance, a number of modern control approaches was employed to realize more precise positioning. For examples, Weerasooriya and Phan [1] presented the discrete-time LQG/LTR design of a disk drive track following servo system. Zhang and Guo [2] introduced a method that incorporated the time optimal control idea into the time-varying sliding mode control design. Hirata et al. [3] proposed a design method of R/W head-positioning controller using a multi-rate sampled-data H∞ control theory. In addition to that the adaptive control techniques had been used to control tracking in the HDD systems [4] & [5].

Among all the proposed control strategies for a hard disk drive for improving its servo performance, there was also a trend towards using nonlinear control schemes. Wang et al. [6] presented a nonlinear PID control scheme for track following. This was done by using a track differentiator and an extended state observer to reduce noises in a similar fashion to an original work by Han [7]. Venkataramanan et al. [8] & [9] presented a composite nonlinear controller made of the combination of a linear and a nonlinear feedback parts to overcome possible problems associated with mode-switching, in a similar way to an earlier work by Lin et al. [10]. Li et al. [11] proposed a nonlinear PID controller, where a combination of linear proportional and linear derivative terms was used with an integral term whose gain was tuned by a nonlinear Gaussian function. As a result, the settling time was shortened by reducing the overshoot caused by the integrator. Peng et al. [12] introduced a servo system design by using an enhanced composite nonlinear feedback control technique with a simple friction and nonlinearity compensation scheme. In a recent work, Hawwa and Masoud [13] proposed a nonlinear PID servo controller for reducing the settling time and lowering the required control effort. Only the derivative term contains nonlinearity while other control law terms were linear. Hence, the controller was capable of adjusting damping effect according to how the actual output is moving with respect to the desired track. This was carried out by tuning the derivative action with a
nonlinear function in the form of the product of the error and its derivative.

This paper aims at introducing a novel design of a nonlinear PID controller for hard disk actuating systems so that an improved system response is achieved with a minimum settling time and free from an overshoot. The appeal of our proposed controller emanates from the fact that the PID scheme represents a simple methodology which has proven the ability to meet the needed requirements for improving the system performance.

II. CONTROLLER DESIGN

A. The Controlled Plant

The controlled plant consists of the voice coil motor (VCM), which sets an E-block arm in rotational motion, and in consequence actuates the load beam, which carries the read/write head at its tip. The control input \( U \) is a voltage which passes through a current amplifier for the VCM and the measurement output \( Y \) is the head position in the tracks. The model of the HDD can be described mathematically by the following electromechanical governing equations

\[ K_m \cdot i = J \frac{d\theta^2}{dt^2} + B \frac{d\theta}{dt} \]

\[ U = R \cdot i + L \frac{di}{dt} \]

where \( \theta \) is the angular position, \( i \) is the electrical coil current, \( K_m \) is the torque constant, \( R \) is the coil resistance, \( L \) is the coil inductance, \( J \) is the moment of inertia of the actuator mass and \( B \) is the friction coefficient of the motor bearing and arm pivot.

The dynamical model of the electromechanical plant is made a fifth-order one by complementing it with a second-order resonance block to model the Head-Gimbal Assembly (HGD). The frequency response of the mechanical plant is planned to have its first resonance peak at 2 kHz. Since the normal practice in HDD servo designs is to "mask" the effect of higher-order resonance frequencies with the use of notch filters, attention is only given to the fundamental frequency. The block diagram model of the HDD system which represents the open loop system is shown in Fig. 1.

Assuming that all initial conditions are zeros and taking the Laplace transform for the above equations, as well as adding the second-order resonance block will lead to obtain the following transfer function of the plant which relate the input voltage \( U \) with the actual head position \( Y \)

\[ G(s) = \frac{Y(s)}{U(s)} = \frac{K_f(s)}{s(\tau_1 s + 1)(\tau_2 s + 1)} \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \]

where \( \tau_1 = J / B \), \( \tau_2 = L / R \) and \( K_f = K_m / B \cdot R \).

B. The Proposed Controller

The design target for the closed-loop control system is to move the head fast from one track to another one and to maintain the actual position on the center of the track without any variation. The PID controller has been a tool of choice in such applications but increasing of system complexity and highly needed performance accuracy put some limitations on its performance.

Hence, we believe that the use of nonlinear gains in the PID control should provide with more tuning capabilities and so better tracking performance. The general form of the nonlinear PID controller is given by

\[ U = K_p(\cdot)e + K_i(\cdot)\int e dt + K_d(\cdot)\dot{e} \]

where \( K_p(\cdot), K_i(\cdot) \) and \( K_d(\cdot) \) are time-varying controller gains, which may depend on the system error and/or its derivative.

To introduce an efficient design of the nonlinear PID controller, each term should have its own nonlinear gain with a suitable logic. The proposed design is introduced in the next paragraphs

Design of the Nonlinear Derivative Gain: The derivative gain represents the additional damping added by the controller. If it is introduced at suitable periods during system response, the performance will be enhanced clearly. The needed derivative gain can be constructed in a way to increase damping relative to that response achieved by the linear PID controller. This is done by injecting damping at suitable periods of system response. The periods of adding high damping can be illustrated in a clear way as in Fig. 2.

![Figure 1. The open-loop HDD system in a block diagram.](image)

![Figure 2. Illustration of the high damping periods in a system step response.](image)
The system response is shown when the head goes from one track to another one and it is divided into different time periods. High damping must be added in the shaded periods of the response in order to eliminate any overshoot while in non-shaded periods damping should not be added since the driving force and the nature of the system takes the output to its target track. So, the damping should have high, low and zero value according to the position of the actual output and its direction toward or away from the desired track.

To adjust response according to the previous discussion, let us propose the following derivative nonlinear gain

$$K_D = k_D \cdot [f(e, \dot{e}) \cdot g_D(e) + h(e)]$$  \hspace{1cm} (5)

where $k_D$ is a constant, and the nonlinear functions $f(e, \dot{e})$, $g_D(e)$ and $h(e)$ are given, respectively, as follow

$$f(e, \dot{e}) = s(e \times \dot{e}),$$  \hspace{1cm} (6)

$$g_D(e) = k_{0D} + k_{1D} \cdot (1 - \text{sech}(k_{2D} \cdot e)))$$  \hspace{1cm} (7)

$$h(e) = \frac{k_s}{(1 + \mu \cdot e^e)}$$  \hspace{1cm} (8)

where $s$ is the step function which has zero or one values for determining when to apply damping. Utilizing the error value ($e$) and its speed ($\dot{e}$) in this way was originally proposed by Hawwa and Masoud [13]. The second nonlinear function $g_D(e)$ is a hyperbolic function which has a small value when error is small and a large value when error is large and it changes smoothly with the error value ($e$). It is introduced in this way in order to prevent any additional damping more than needed. In this function, the values $k_{0D}$, $k_{1D}$ and $k_{2D}$ are user defined positive constants. Such hyperbolic function was used before by Seraji [14] and Xu et al. [15]. Fig. 3, shows a typical variation of this nonlinear function versus the error $e$. This function will control only the magnitude of damping.

The amount of damping added by the above form does not guarantee a response with free overshoot, because of that the second nonlinear term $h(e)$ is added in a form as in (8). It is a type of an exponential function which was utilized by different researchers such as Shahruz and Schwartz [16] and Garrido et al. [17]. It is designed to have a zero value in a long period and a positive value when error is small. The variation of the total nonlinear gain with respect to the error value and its speed is shown in a 3D plot as in Fig. 4.

The question now is "When to apply the function $g_D(e)$ and when to eliminate it?". In order to answer this question, one has to note that if the product of the error with its derivative is negative, imposed damping is not needed because it is implied that either the positive error is taken care of by a negative error rate or the negative error is dealt with by a positive error rate. So, damping is needed when the product of the error with its derivative is positive. In order to plan according to the above outlined reasoning, a tuning function has been chosen in the form of a step function $s(e \times \dot{e})$, then multiplied by the error-dependent, nonlinear hyperbolic gain $g_p(e)$ to provide the suitable magnitude of damping when it is needed.

The Design of the Nonlinear Proportional Gain:

The importance of the proportional gain comes from its responsibility for adding the suitable amount of driving force. It is noticed from Fig. 2, that large value of driving force is needed when the head moves from its present track to the target track as in the period from time zero to time $t_0$. This will help in reaching the target track rapidly and in a very short time. To achieve this, the proportional nonlinear gain is designed to have the following form

$$g_p(e) = 1 + k_{1P} \cdot (1 - \sec h(k_{2P} \cdot e))$$  \hspace{1cm} (9)

Figure 3. The variation of the nonlinear gain $g_p(e)$ which used in the form of the derivative nonlinear gain.

Figure 4. A 3D plot for the variation of the total nonlinear derivative gain.
which is similar to the form of \( g_d(e) \), but with some modification. Here it has two adjusting parameters \( k_{1p} \) and \( k_{2p} \). This gain will provide high value of driving force when error is large and decreases smoothly to a small value as output comes near its target track.

Design of the Nonlinear Integral Gain: The integral action is introduced to eliminate the steady-state error problem caused by the proportional control gain and other disturbances. Sometimes it causes saturation problems (known as integrator windup) during transient response. On the other hand, when error is small the integral gain is preferred to take large values in order to eliminate steady-state error. For these reasons, the integral gain should be designed carefully to take both actions and to change gradually between large and small values. To achieve the above requirements the following form of the integral nonlinear gain is used

\[
g_i(e) = \frac{1}{1 + \exp[\beta (-\alpha e - 0.1)]} - \frac{1}{1 + \exp[\beta (-\alpha e + 0.1)]}
\]

Where the parameters \( \alpha \) and \( \beta \) are positive constants which determine the profile of the gain. The variation of the integral gain as a function of the error is shown in Fig. 5.

This form of exponential functions was also utilized by Garrido and Soria [18] where they integrated it in the derivative gain to control damping in servomechanisms. When this form is used in the integral term, it takes positive values when error is in a small range to eliminate the steady-state error while outside that range it takes zero values to avoid the negative affects.

Hence, the proposed nonlinear PID controller shall take the form

\[
U = k_p \cdot g_p(e) \cdot e + k_i \cdot g_i(e) \int_0^t e \cdot dt + k_{D} \int f(e, \dot{e}) \cdot g_D(e) + h(e) \cdot \dot{e}
\]

III. SIMULATION RESULTS

A closed-loop model is developed within the MATLAB/SIMULINK simulation environment. The continuous time model for the VCM actuator is contained inside a “Plant” block, with the property \( K_f = 0.05 \ N/m^2/sec^2/A \), the time constants are \( \tau_1 = 1 \times 10^{-3} \) sec and \( \tau_2 = 0.05 \) sec, the damping ratio is \( \zeta = 0.03 \) and the first natural frequency is \( \omega_n = 2\pi f \ rad/sec \), where \( f = 2000 \) Hz. The track density per unit radius (track per inch, TPI) is considered to be 60,000.

A. Comparison between Linear and Nonlinear Controllers

In order to check performance characteristics of the proposed nonlinear PID against those of its linear PID counterpart, the nonlinear controller parameters are chosen as follow, \( k_{1p} = 4.167, k_{2p} = 5.9, k_p = 4560, \alpha = 12000, \beta = 15, k_f = 50, k_{0D} = 16, k_{1D} = 120, k_{2D} = 5.9, \mu = 6000, k_h = 1.3 \) and \( k_D = 73 \). The controller parameters are manually tuned for the best possible nonlinear controller’s performance but optimization tools based on a suitable performance function can be used to choose them automatically. The simulation for the linear PID controller is carried out also for best performance and the linear parameters are chosen to be \( k_p = 4500, k_f = 20 \) and \( k_D = 85 \). Comparison between the performances of both controllers is given in the form of the step responses as shown in Fig. 6. The nonlinear PID controller moves the R/W head twenty tracks and settles on the target track in less than 0.045 second, while the linear PID controller is able to move the R/W head to the desired track in 0.17 second. The superior performance of the nonlinear PID controller over the linear PID controller in realizing a shorter settling time is quite obvious.

Figure 6. The step response of the proposed nonlinear controlled system.

It is also of significance to compare the control effort spent by each controller. Fig. 7, shows the control signals in volts as a function of time for both controllers where it is shown that the nonlinear PID controller consumes more control signal than that of the linear PID controller because high driving force and damping are
needed to give a good system performance. Although large value of control signal is needed but it stills acceptable and sets within $-3$ to $+3$ volts which is a practical range in many real hard disk drive systems.

**B. Disturbance Rejection**

Disturbances affecting HDD servomechanisms are mainly represented by repeatable runouts (RRO). RRO are caused by imperfections in the spindle motor construction or functioning. Typical example is eccentricity. The performance of the closed loop system in the presence of RRO is verified by injecting a simulated disturbance. The disturbance signal is generated as a linear combination of the sinusoidal waves at the fundamental frequency of disk rotation and its harmonics, representing the disturbances caused by spindle runout and disk flutter. The disturbance input is assumed to take the form

$$d(t) = 0.4 + 0.1 \cos(300 \cdot t) + 0.05 \sin(600 \cdot t)$$  \hspace{1cm} (12)

The response of the proposed controller against the imposed disturbance is shown in Fig. 8, where it is observed that the controller significantly attenuates the repeated runout disturbance and remains stable. Fig. 9, shows a magnified picture of the step response, indicating that the R/W head is acceptably following the target data track.

**IV. CONCLUDING REMARKS**

A novel nonlinear PID servo controller was introduced to meet the higher demands of designing efficient and not complicated track seeking/following controller in the hard disk drive systems of large capacity storage. The controller was designed to implement selective driving force and damping amount on the actuator activities by adjusting the strength of the proportional and derivative terms of the control signal. This was performed by adding nonlinearities in the controller gains which based on the system error and its speed. In addition to that, nonlinearity was added in the integral term which acted to correct the steady state error. A good selection of the nonlinear gains in the proposed nonlinear PID scheme was found to significantly shorten the response settling time without getting any overshoot when compared to the corresponding classical linear PID controller. The robustness of the nonlinear PID controller to handle system performance and stability against a repeated runout disturbance was confirmed positive. In a future work the optimization tools will be introduced as an extension for design of the proposed nonlinear PID controller.

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