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Modeling and Control System Design for Tape Drive Loader Mechanism

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***Abstract*----In this paper, the modeling and control system design for the loader mechanism in a tape drive/video cassette system is presented. The system is driven by a voltage controlled Pulse-Width-Modulated(PWM) servo controller. During the entire range of the loader operation, the system inertia, damping, and the flexibility of the coupling between the drive and the load vary considerably. A Proportional-Integral-Derivative (PID) controller is designed around a nominal operating point using pole-placement technique. This allows the controller gains to be set directly based on the desired integrator action and bandwidth requirements. The performance of the sensor based motion control of the loader mechanism is then presented.**

I. INTRODUCTION

The loader mechanism is an integral part of cartridge drives used in tape automation and tape libraries. For applications with frequent cartridge replacement that use tape libraries to provide access to volume of data to many users via networks, a loader mechanism in tape drive is very critical. The loader mechanism must be able to load and unload cartridges many times without jamming or other failures. A failure of the loader mechanism may damage a tape cartridge, and it makes the drive unusable until repaired or replaced. Typical design parameters call for the loader mechanism to continue to operate successfully for at least 300,000 to 500,000 loading/unloading cycles. All the reliability requirements make the loader mechanism complex and calls for reliable motion control algorithms. There are a number of loader mechanisms in operation in tape automation loader mechanisms in operation in tape automation industry .

Fig. 1 shows a schematic of a typical loader mechanism. The loader consists of a motor driven cam gear mechanism to pull-in a cartridge to the Cartridge In (Cart In) position, raise the reel motors to Off Head position and finally move the loader to the On Head position. This is the load operation. The unload operation is just the reverse of this namely moving the loader from the On Head position to the Off Head position, lowering the reel motors to the Cart In position and finally ejecting the cartridge. Fig. 2 shows the position versus time for the loader motion. The On Head to Off Head motion is used to pull the media away from the head. This is required on three occasions, namely rewind, high-speed search and when the tape is sitting idle for several seconds. The tape is moved off the head while sitting against it for elongated periods because the head operates at much higher temperature than the normal storing temperature of the tape so that the detrimental effects of high temperature in tape media is minimized. Fig. 3 shows the typical variation of load torque during the load operation. The load torque varies dramatically along the path of the loader motion. In addition, the effective inertia seen by the drive motor, the damping and flexibility of the coupling between the drive motor and load varies considerably.

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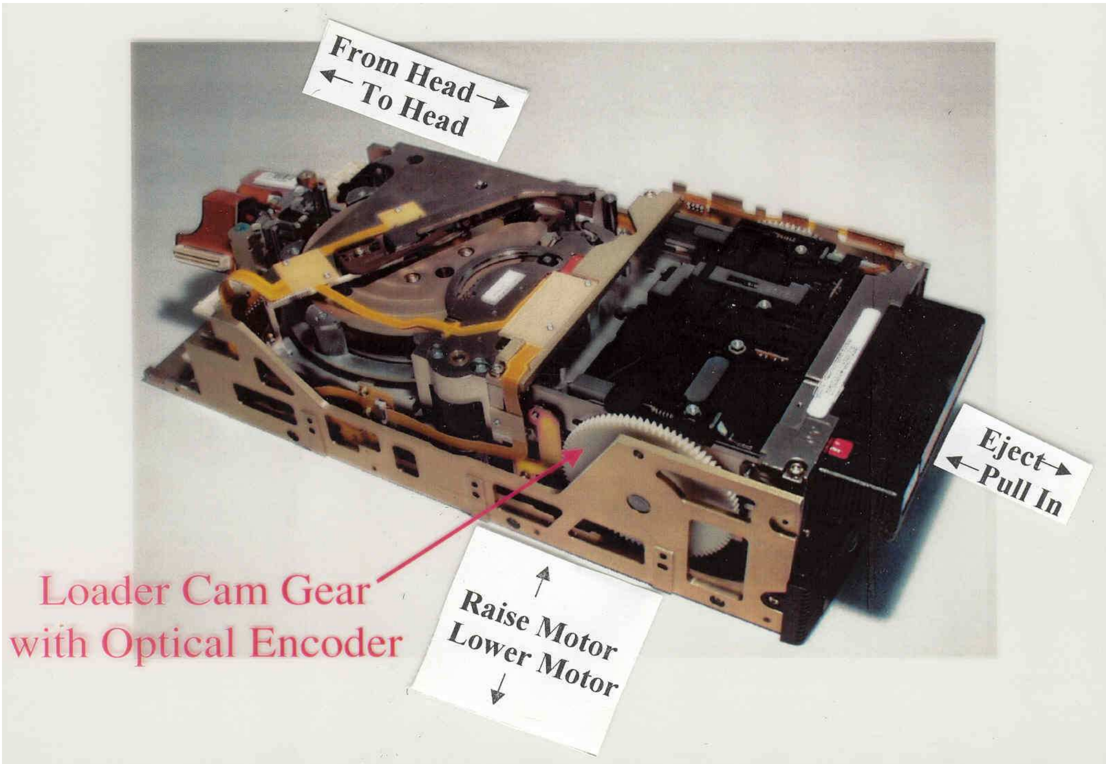

Fig. 1. Typical Loader Mechanism

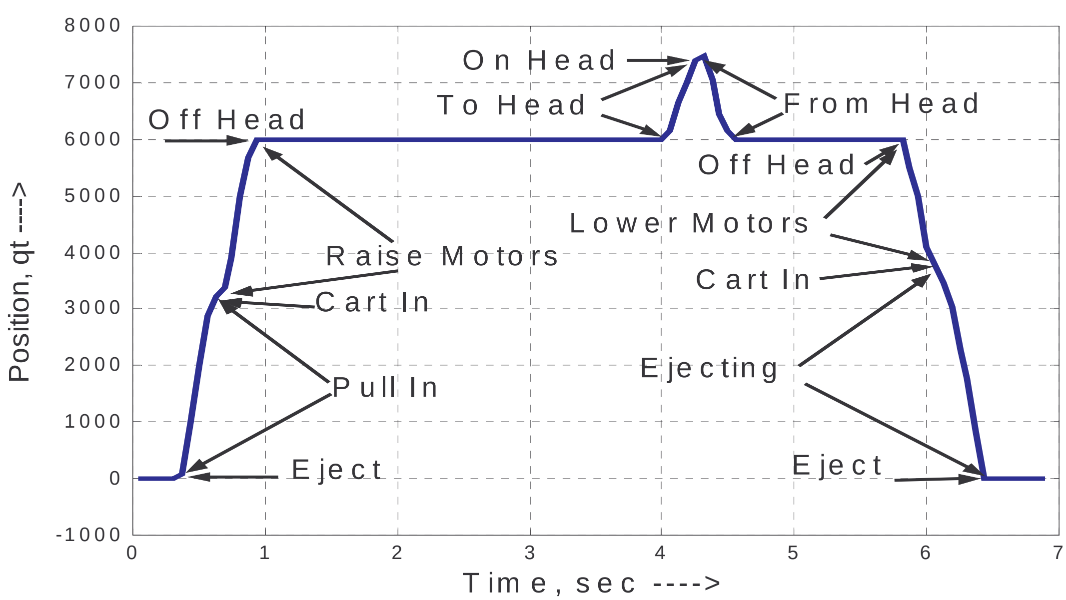

Fig. 2. Position-Time Relationship for various states in Loader Mechanism

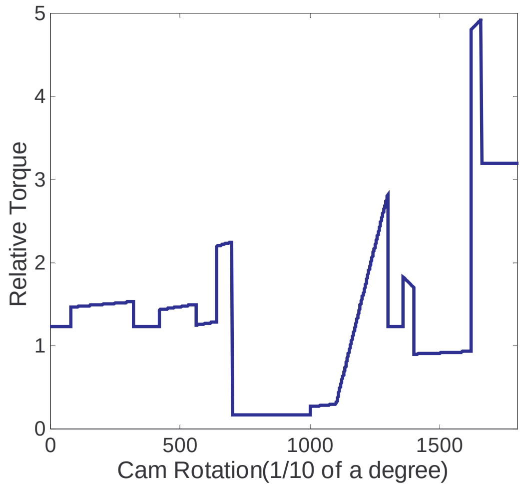

Fig.3. Torque Requirements for Constant Speed

There are a number of loader mechanism/system in operation in the tape automation industry [3], [4]. Many of the loaders are operated open loop trying to match the input command (control effort) to the load profile similar to the one shown in Fig. 3. This results in uneven operation and gives rise to motion profiles that had the potential to cause tape damage and reduced life of the mechanism. With the addition of an optical encoder along with the sensors that demarcate the different regions of operation, the closed loop operation is now possible. This paper is concerned with the feedback controller design along with the resulting closed loop operation; and is organized as follows. In Section II, the system model is presented. The system is actuated by a DC motor driven by a voltage controlled Pulse-Width-Modulated (PWM) power amplifier [1]. Section III discusses the design of a classical controller, namely the Proportional-Integral-Derivative (PID) controller that is designed by the pole-placement technique. The motion control algorithm for loader mechanism whose states are sensor based is presented in Section IV along with test results for the loader mechanism using the PID controller. Finally, conclusions are presented in Section V.

II. SYSTEM MODEL

Fig. 4 shows the linearized model of the voltage controlled PWM drive with position feedback.

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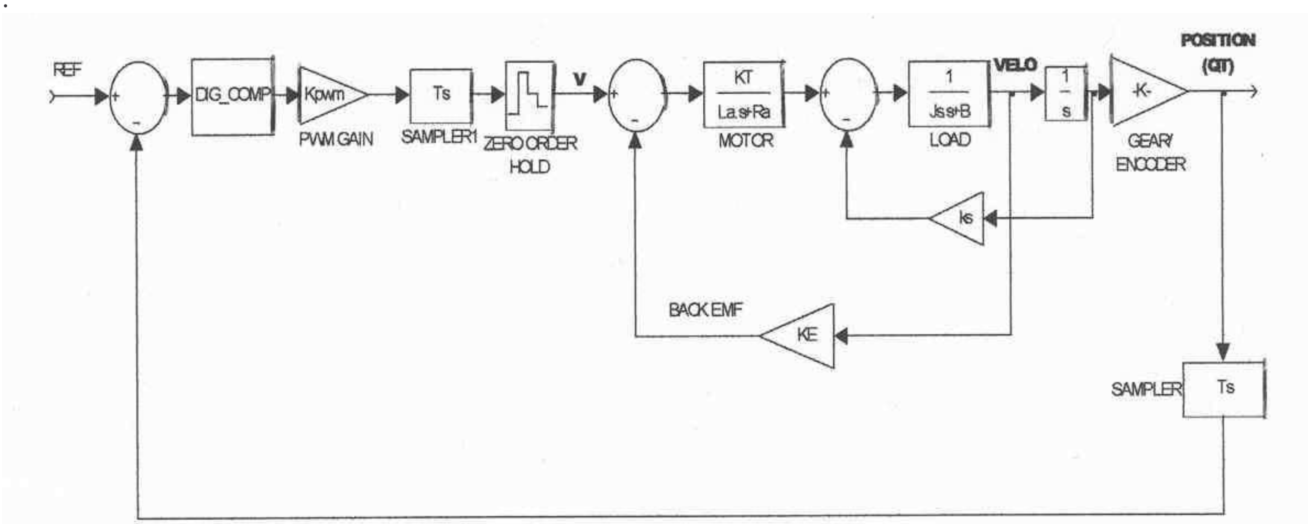

Fig.4. Linearized Model of a Voltage Controlled PWM Drive with Position Feedback

R_a = Motor resistance, ohm

L_a = Motor Inductance, henry

J = Total Inertia seen by the Drive(Motor + Load), oz-in/(rad/sec^2)

B = Viscous Damping Coefficient, oz-in/(rad/sec)

k_s = Torsional Spring Constant for the coupling between Motor and Load , oz-in/rad

K_T = Motor Torque Constant, oz-in/amp

K_E = Back EMF Constant, volts/(rad/sec)

ω = Angular velocity, rad/sec

K_{en} = Encoder gain, quarter-tach(qt)/rad

K_{pwm} = Pulse Width Modulator Amplifier Gain, volts/count

N_{em} = Gear Ratio, Encoder to Motor

θ_q = Position Output, qt

V = Voltage applied to Motor, volts

Variations:

$2.1\text{x}10^{-4} \le J \le 8.4\text{x}10^{-4}$ (oz-in/(rad/sec^2))

$0.017 \le B \le 0.075$ (oz-in/(rad/sec))

$0 \le k_s \le 186$ (oz-in/rad)

For the nominal plant model, we take J = 5.25 oz-in/(rad/sec^2), B = 0.045 oz-in/(rad/sec) and k_s = 93 oz-in/rad.

III. PID CONTROLLER DESIGN

For the nominal plant shown in Fig. 4, voltage command to the position transfer function is given by

$$G(s) = \frac{\theta_q(s)}{V(s)} = \frac{K_{en}K_T}{N_{em}} \frac{1}{\left[(sL_a + R_a)(Js^2 + Bs + k_s) + K_E K_T s\right]} \quad (1)$$

Without the electromechanical coupling, the electrical time constant is L_a / R_a ($\approx$0.5 milliseconds) which is much smaller than the approximate mechanical time constant J / B ($\approx$ 20 milliseconds). Thus L_a could be neglected without affecting the dynamics by much. Neglecting the sL_a term, we get

$$G(s) = \frac{\theta_q(s)}{V(s)} = \frac{K_{en}K_T}{N_{em}R_a} \frac{1}{\left[Js^2 + (B + K_E K_T / R_a)s + k_s\right]} \quad (2)$$

The plant is now approximated to a second order system with equivalent inertia of J and equivalent viscous damping of $(B + K_E K_T / R_a)$ and flexibility of k_s. When augmented by a position error integrator, the plant becomes a third order system. When a controller for such a plant is designed via pole-placement compensator, the three gain terms correspond to the gain terms of a PID controller. Thus the gains of the PID controller are obtained uniquely by the pole-placement technique. The performance of the controller is then checked against the full model given in (1) so that the zero db crossover frequency, phase margin requirements, etc., are satisfied.

The controller design is carried out as follows. The state space description of the reduced order system augmented by an integrator is given by

$$\dot{x} = Ax + Bu \quad (3)$$

$$y = Cx \quad (4)$$

with state vector given by $x = [\, x_1 \;\; x_2 \;\; x_3 \,]^T$, with $x_1 = \theta_q =$ Loader Position Error, $x_2 = \omega_q =$ Loader Velocity Error, $x_3 = \theta_{qI} =$ Loader Position Error Integral. The input vector is $u = u_1 =$ V=Voltage applied to motor. The output vector is $y = y_1 =$ Loader Position Error. The system matrices are given by

$$A = \begin{bmatrix} 0 & 1 & 0 \\ 0 & -(\frac{B}{J} + \frac{K_E K_T}{R_a J}) & -\frac{k_s}{J} \\ 1 & 0 & 0 \end{bmatrix},$$

$$B = \begin{bmatrix} 0 \\ \frac{K_{en} K_T}{N_{em} R_a J} \\ 0 \end{bmatrix}, \quad C = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \quad (5)$$

The system is completely controllable, however, it is not completely observable because of the augmented state variable for the position error integrator.

A sampling frequency of 1.67 kHz (sample period =0.6 msec) is used. Matrices F, G, and H, are discrete-time versions of the A, B, and C matrices leading to the discrete-time plant equations:

$$x(k+1) = Fx(k) + Gu(k) \quad (6)$$

$$y(k) = Hx(k) \quad (7)$$

The linear controller based on pole-placement technique gives rise to the control law

$$u(k) = -Kx(k) \quad (8)$$

with

$$K = [k_1 \quad k_2 \quad k_3] \quad (9)$$

Because of the manner in which the state variables are chosen, the controller gains correspond to $k_1 = k_P$ (proportional gain), $k_2 = k_D$ (derivative gain), and $k_3 = k_I$ (integral gain). Thus a direct design of PID controller is done via pole-placement technique. The controller performance such as zero db cross-over frequency, phase margin, gain margin, and settling time are verified using the full plant model (1). Fig. 5 shows the open loop magnitude response of the plant and compensator at two different positions of the loader. These have zero db crossings of 25 Hz (PULL IN POSITION) and 7 Hz (TO HEAD POSITION). In each of these position the phase margin margin is 36 degrees. Fig. 6 shows the magnitude and phase response for the loader at another position (RAISE MOTOR POSITION) where the zero db crossing is 6 Hz and phase margin is 74 degrees. At this position we see a resonance at 75 Hz which is possible due to k_s becoming too large as the motor gets coupled to the load. We have designed one PID controller for the entire range of motion and tested the performance with only one controller, the parameter variations are thus evident from these plots.

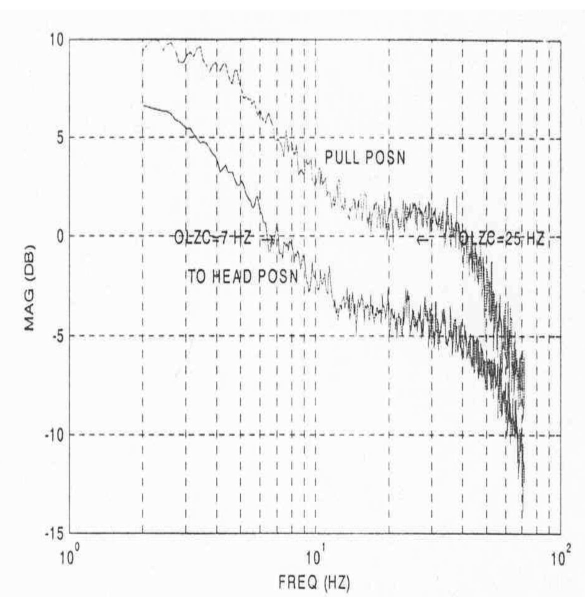

Fig. 5. Open loop (plant plus compensator) magnitude response of the loader

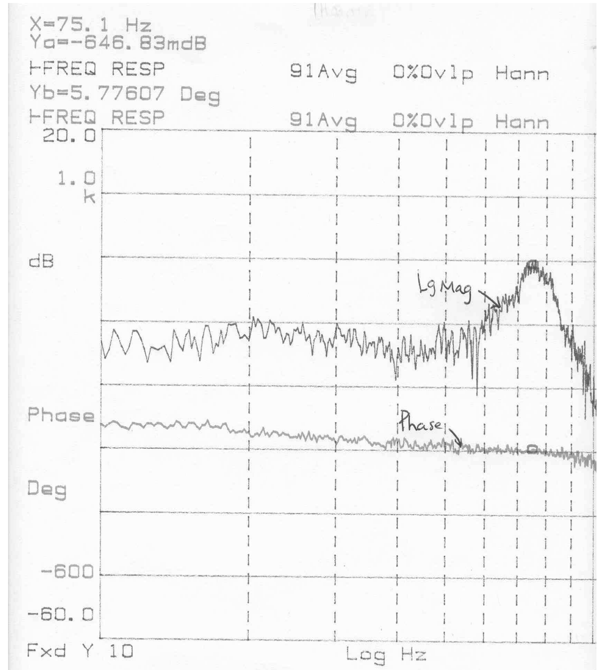

Fig. 6. Open loop (plant plus compensator) frequency response of the loader at RAISE MOTOR POSITION

IV. MOTION CONTROL ALGORITHM AND TEST RESULTS

For the loader mechanism, the motion control is not strictly point-to-point, rather it is sensor based. The sensors are active over a small range of position. For this reason, the motion control has a profile as shown in Fig. 7. The destination sensor is reached while hunting at a low velocity V_h. For generating the profile, the following parameters are normally given.

d = Minimum distance traveled to reach the destination sensor

T = Time required to attain the maximum velocity

V_h = Hunting velocity

With a parameter 'p' defined as $V_h / V_{\max}$, we calculate

$$p = -\frac{d}{TV_h} + [(\frac{d}{TV_h})^2 + 2]^{0.5} \qquad (10)$$

$$V_{\max} = V_h / p \qquad (11)$$

After obtaining p, the required acceleration and deceleration for profile generation are calculated. The acceleration and deceleration are also used for feed forward PWM command calculations.

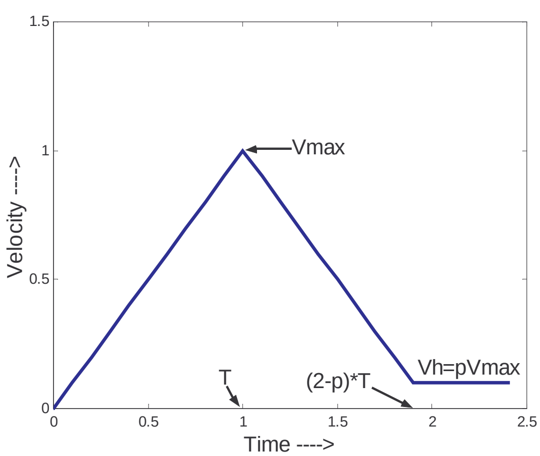

Fig. 7. Motion Control – Velocity Profile

Fig. 8 shows the performance of the loader mechanism as it completes as it completes the motion in the forward direction, i.e., the PULL IN, RAISE MOTOR, and the TO HEAD motions. Fig. 9 shows the motion in the backward direction, i.e., the FROM HEAD, LOWER MOTOR, and EJECT motions. The position error (output) and the voltage command (input) are plotted against time. The line voltage available for PWM purposes is $\pm$12 volts. With the present controller the loader motions are accomplished with 70 % of the maximum control effort.

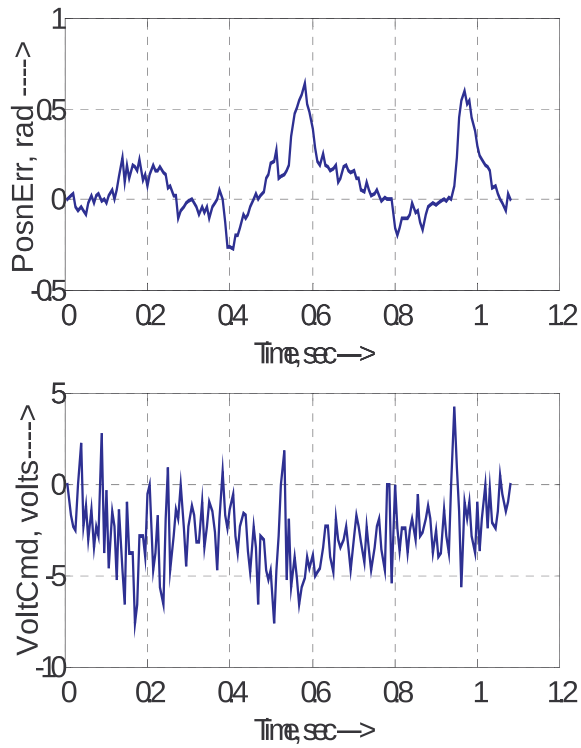

Fig. 8. Position error and voltage command profile in the forward direction (PULL IN, RAISE MOTOR & TO HEAD)

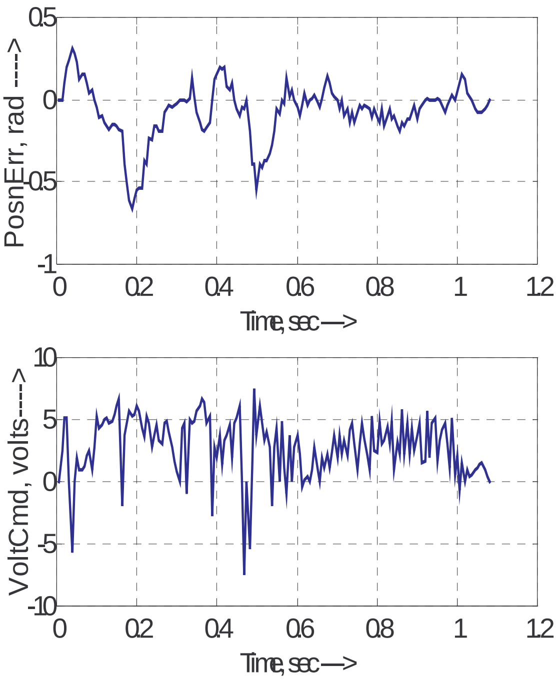

Fig. 9. Position error and voltage command profile in the backward direction (FROM HEAD, LOWER MOTOR & EJECT)

V. CONCLUSIONS

In this paper, we have presented the modeling and control system design of a loader mechanism commonly used in tape drive systems. Using the nominal plant parameters, a PID controller is designed using the pole-placement technique. The performance is shown to be satisfactory based on the fact the use of closed loop control along with an innovative motion control algorithm has resulted in smooth and gentle motion profiles in spite of huge variations in inertia and load during the entire range of loader motion. The parameter variations as loader moves along different positions is brought out and a robust controller design which would include uncertainties and variations in plant parameters will be reported in a future paper. This is expected improve the performance and further reduce the mean time between failures.

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