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On Motion Control Design and Tuning Techniques

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***Abstract* - The conventional Proportional, Integral, Derivative (PID) controller, and some common linear variations to PID, are evaluated along with recently introduced Parameterized Loop-Shaping and Active Disturbance Rejection Control (ADRC). The evaluation is made using a simulation of a nonlinear DC servomotor driven "Pick & Place" positioning system that includes control input saturation, power amplifier saturation, motor current limit, viscous and coulomb friction, and gear backlash. The effects of adding torque disturbance, sensor noise, mechanical resonance, and load changes are examined for the various control methods. The relative merits of these control techniques in regards ease of controller design and tuning versus performance are summarized.**

I: INTRODUCTION

This paper concerns itself with transient position control with servo motors, a common yet difficult application, involving ever larger inertia mismatch, with zero tolerance for overshoot, and constant pressure for greater speed. The applications are nonlinear and environmentally severe in regards mechanical and electrical noise and disturbances. Linear Proportional, Integral, Derivative control (PID) is still the most widely used method for position control. Many position controller designers will use Ziegler-Nichols [1], or some variation [3,4] as a starting point for tuning, but will tune much more "aggressively" for command following [2,8], yielding controls which meet accuracy and speed requirements, yet wear out components with their constantly correcting control action, challenging everyone to significantly improve repair and quality costs.

The relative ease of tuning and subsequent performance of several linear PID modifications is evaluated, including motion profiling (which is used in all examples), lead/lag compensation, and cascade control with velocity feed forward. Two parameterized control design techniques, Loop-shaping [6] in the frequency domain, and Active Disturbance Rejection Control, or ADRC [5,6] using output state observer methodology, are also evaluated. Nonlinear techniques, such as gain scheduling, gain switching, nonlinear PID, and dither, among many, are not discussed, as they usually require more specific knowledge of the process and application to be effective, and thus are even more complex to apply [2,3,4]. Additionally, an extension of the Integral of Squared Error (ISE) performance index is proposed to improve the tuning process.

These nonlinear motion applications have been extensively studied and modeled [9,10,11], and this study will not extend that knowledge. It is demonstrated how the simplest simulation of the nonlinear process, particularly using relay control experiments [7] to simulate the dynamics of resonant modes, can yield significant improvements.

This paper is organized as follows: The simulation and setup of the nonlinear servo motor and transmission mechanism is described in section II. PID control and its variations used in the evaluation are reviewed in section III. The Loop-shaping controller is explained in section IV. Section V describes the construction of the ADRC control. The evaluation criteria, performance constraints, design and tuning description, and graphical results are presented in section VI, along with comments on these results. Finally, a general conclusion and recommendation for further research is presented in section VII.

II: PROBLEM AND SIMULATION SETUP

A nonlinear amplifier/motor/transmission Simulink model was used to compare controller types. This model added nonlinear saturation of amplifier gains, plus coulomb friction and backlash in the mechanics. The input voltage command is saturated at max specified value, as is the output of the power amplifier. The power amplifier gain is K_{pa}. The current feedback loop gain, K_{cf}, is chosen to limit the armature current to its specified max value when motor voltage is max. The motor torque constant, K_t, is given or calculated from the motor spec, as is the back emf constant, K_e. One assumption is that motor inductance, L_a, is small so as to be ignored, compared to the armature resistance, R_a. The transfer ratio, R_p, is as specified or calculated from the transmission specifications. The total inertia, J_t, must be calculated as part of motor sizing. A system resonant mode (as seen in the actual application) is also modeled at 50Hz, which is barely one decade above the desired operating bandwidth. The viscous friction coefficient, K_f, and a nonlinear coulomb friction offset may be calculated, but, like the resonant frequency, ω_r, and resonant damping ratio, ξ_r, most likely must be provoked and measured in the application.

The linear transfer function for the amplifier, servomotor and transmission from control voltage V_c to velocity *Vel* and position *Pos* output is:

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$$\frac{Vel}{V_c} = \frac{K_p}{(\frac{s}{\omega_p}+1)} \quad , \quad \frac{Pos}{V_c} = \frac{K_p}{(\frac{s}{\omega_p}+1)\cdot s} \tag{1}$$

where:
$$K_p = \frac{K_t \cdot R_p \cdot K_{pa}}{(R_{eff}\cdot K_f + K_t \cdot K_e)} \tag{2}$$

and
$$\omega_p = \frac{(R_{eff}\cdot K_f + K_t \cdot K_e)}{R_{eff}\cdot J_t} \tag{3}$$

and
$$R_{eff} = (R_a + K_{pa}\cdot K_{cf}) \tag{4}$$

The relevant component specifications used for these simulations were chosen to match those of an actual "pick and place" application in the metal forming industry which moved a 106.6 kg load 300mm at a 2sec cycle. A complete in and out motion profile is used for all simulations to account for the backlash. The external control and torque disturbance are applied at 0.5 and 1.5 seconds, when the system should be at steady state. See Figure1.

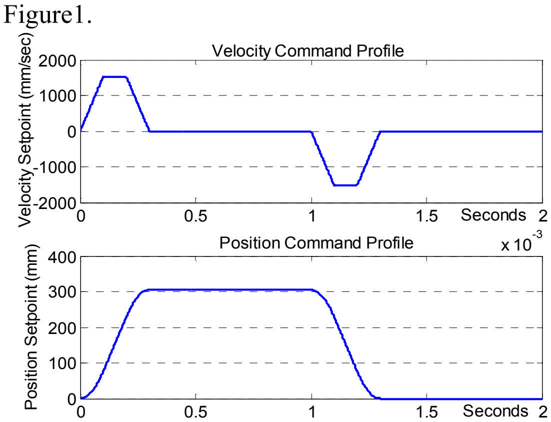

Figure 1. Velocity and Position Profiles

III: PID and VARIANTS

PID is the most commonly used method for control. The variants studied here are motion profiling (which is used in all examples), lead/lag compensation and cascade control with velocity feed forward.

The simple unity feedback PID control has the control output:

$$u(t) = K_P \cdot e(t) + K_I \cdot \int e(t)dt + K_D \cdot de(t)/dt \tag{5}$$

and a transfer function:

$$G_c(s) = \frac{(K_D \cdot s^2 + K_P \cdot s + K_I)}{s} \tag{6}$$

PID with Lead/Lag Compensation:

PID with lead/lag compensation is analogous to loop-shaping control, which is tedious and difficult to tune without benefit of a process model and parameterization. In the simulated system the properly tuned PID control managed acceptable command following and disturbance rejection, so a single stage lead compensator was the only necessary addition, to increase stability under load changes and still avoid the mechanical resonance. Additional compensation was deemed unnecessary so as to avoid provoking more serious issues, and to keep the number of choices during the setup and tuning process as small as possible.

The equation for PID control with a single stage lead compensation is:

$$G_c(s) = \left(\frac{K_D \cdot s^2 + K_P \cdot s + K_I}{s}\right)\cdot\left(\frac{s+\omega_{zero}}{s+\omega_{pole}}\right) \tag{7}$$

PID with Velocity Feed Forward:

Cascade control with feed forward velocity is a favorite of many motion control designers due to fast and accurate performance. The tradeoff for this improvement is a doubling of setup and tuning complexity, along with the increased sensitivity to disturbance and noise.

The Velocity Feed Forward controller transfer function is:

$$G_c(s) = \left(\frac{K_{Dvel}\cdot s^2 + K_{Pvel}\cdot s + K_{Ivel}}{s}\right) \cdot \left(\frac{(K_{Dpos}+1)\cdot s^2 + K_{Ppos}\cdot s + K_{Ipos}}{s}\right) \tag{8}$$

IV: PARAMETERIZED LOOP-SHAPING CONTROL

This paper evaluates two parameterized control design techniques, Loop Shaping and Active Disturbance Rejection Control [6]. One can design a parameterized position control following these techniques which meets the specified performance criteria, using the specification sheet parameters for the amp, the motor, the transmission, the sensor, the inertia of the load, and one or two tuning variables. As a matter of fact, if a resonant mode is present, the tuning value may be constrained and determined by that resonance. The performance of these controls, as with the PID techniques, may be optimized by more attentive and tedious tuning, but parameterization achieves the major goal of simplicity in application.

The loop shaping controller requires one to determine the plant's open loop transfer function, G_p , gain and natural frequency K_n = 20.2, ω_n = 3.2rad/sec, and then include an inverse of the plant transfer function, G_p^{-1}, in the controller, where:

$$G_c(s) = \left(\frac{s+\omega_1}{s}\right)^m \cdot \frac{1}{\frac{s}{\omega_c}+1} \cdot \frac{1}{\left(\frac{s}{\omega_2}+1\right)^n} \cdot G_p^{-1}(s) \tag{9}$$

The control used herein was calculated from component values used in the actual pick & place application.

V: LINEAR ACTIVE DISTURBANCE REJECTION CONTROL

Another alternative is the combination of a classic PD control with ADRC [5,6]. The philosophy using ADRC is that exact knowledge of the plant model is not required for design of a Linear Extended State Observer, LESO, as an approximation will be shown to work exceptionally well. (The derivation of a parameterized ADRC, for those unfamiliar with the theory, is not difficult, but beyond the scope of this paper. The reader is encouraged to read the references in order to construct the LESO.)

The ADRC defines an augmented state encompassing the unknown internal and external dynamics, such that now

$$x_1 = y \, , x_2 = \dot{y} \, , x_3 = f \, , \qquad (10)$$

and then

$$\ddot{y} = f + b_o u \qquad (11)$$

where

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= x_3 + b_o u \\ \dot{x}_3 &= h = \dot{f} \end{aligned} \qquad (12)$$

The Linear ESO is then defined by:

$$\dot{z} = [A - LC]z + [B, L]\begin{bmatrix} u \\ y \end{bmatrix} \qquad (13)$$

One then designs an ADRC control

$$u = \frac{(u_o - z_3)}{b_o} = \frac{K_P(r - z_1) + K_D(\dot{r} - z_2) - z_3}{b_o} \qquad (14)$$

This "augmented process" is a unit gain double integrator process which may be controlled by a PD controller.

VI: EVALUATION CRITERIA AND CONSTRAINTS

The prime acceptance criterion for positioning applications is heavily weighted by command following accuracy, not only for point to point measures, but for accuracy during the transient motion. Torque disturbance rejection, control signal disturbance rejection, and feedback sensor noise rejection are important criteria in all systems. The ability to handle significant changes in load (1/2 or 2 times rated) without adjustment and to tolerate/avoid resonant modes are very desirable features of any control system. Last, and very important, a system should be as simple to setup and tune as possible, with a minimum number of variables to measure or calculate during the design, or adjust iteratively during tuning. Tuning becomes progressively more complex as the number of parameters increase, due to the coupling.

TUNING CRITERIA AND RESULTS:

The setup and tuning of the controls was done using two methodologies, an empirical technique as practiced by the majority of control engineers, and a quantitative method using a performance cost function. Both techniques utilized the nonlinear motor model described in Section II. The performance targets were to achieve a 305 mm move in 0.3 seconds with <1% overshoot error and achieve <0.1% steady state error @ 1 second. The noise/oscillation passed through to the control signal was to be <1% of the max control signal. Disturbance rejection was also to be <1% overshoot and recovery <0.1% steady state error @ 3 cycles. No controller was able to meet all these constraints using the values from the actual application, however, the evaluation does illustrate the relative performance of the different methods.

Tuning PID:

The PID control was initially tuned for a linear positioning system simulated without any disturbance, noise, resonance, friction, backlash, or load variation. An empirical technique was followed to mimic the usual control designer practice wherein these compromising factors are unknown and no simulation is used. The resulting values were $K_P = 20$, $K_I = 10$, $K_D = 1$. The results are encouraging until the system encounters one or more of these compromising factors, in which case constant control cycling (most common) or instability (sometimes) may occur. The PID was then simulated with the nonlinear model (see Section II) yielding tuning values of $K_P = 20$, $K_I = 60$, $K_D = 0.2$. These values proved to be surprisingly robust, and still stable and usable when the load changed.

Tuning PID with Lead/Lag Compensation:

The application example used for these comparisons has natural parameters $K_n = 20.2$ and $\omega_n = 3.2$rad/sec, so a single stage lead compensator with zero at $\omega_{zero} = 10*\omega_n = 32$rad/sec, and a pole at $\omega_{pole} = \omega_r/5 = 62.8$rad/sec yielded $K_P = 20$, $K_I = 0$, $K_D = 0.5$ PID tuning for the linear simulation and $K_P = 20$, $K_I = 60$, $K_D = 0$ for the nonlinear simulation.

Tuning PID with Velocity Feed Forward:

Tuning the cascade control with velocity feed forward was accomplished iteratively, as in normal practice, using commercial internet based methods, and yielded $K_{Pvel} = 5$, $K_{Ivel} = 0$, $K_{Dvel} = 0$, $K_{Ppos} = 10$, $K_{Ipos} = 50$, $K_{Dpos} = 0$ for the linear model simulation, and $K_{Ppos} = 200$, $K_{Ipos} = 1000$, $K_{Dpos} = 0.2$, $K_{Pvel} = 0.1$, $K_{Ivel} = K_{Dvel} = 0$ for the nonlinear model simulation, a large difference. It is demonstrated how sensitive this technique can be to un-modeled dynamics, as the empirically tuned controller was quite sensitive to sensor noise and mechanical resonant modes.

Tuning Parameterized Loop-Shaping Controller:

A crossover frequency, $\omega_c = \omega_r/10 = 31.4$ rad/sec is chosen in the examples, such that it remains between any resonant frequency, $\omega_r = 100\pi$ rad/sec, and the desired frequency to meet the command following goals, $\omega_{desired} = 13.3$ rad/sec. High pass filters with zeroes at $\omega_l = \omega_c/2 = 15.7$ rad/sec and low pass filters with poles at $\omega_2 = 5*\omega_c = 157.1$ rad/sec were chosen herein, due to the narrow band between the desired crossover frequency and the resonant frequency. A preferred starting value would be a decade above the desired bandwidth and a decade below the resonance, which is not possible with the application. The value can be tuned as desired, the choice here gave more distance from the resonance. One decides how effective one wants the filters to be by choosing the number of zeroes at ω_l or poles at ω_2. In this comparison there are m = 2 zeroes and n = 3 poles.

Tuning Parameterized ADRC with PD Controller:

The empirical technique starts with any resonant frequency, $\omega_c = \omega_o = \omega_r / 10$ and increases ω_c to improve response. Or, if no resonance is present, uses the open loop natural frequency, ω_n, established using Z-N method, and $\omega_c = \omega_o = 10\omega_n$. We use $\omega_c = 3\omega_o = 3\omega_r / 10$ in the examples herein so that only one parameter, ω_c, is tuned.

A QUANTIFIED TUNING METHODOLOGY:

Design and Tuning Parameter Comparison	PID	PID + Lead Lag	PID + Feed Forward	Loop Shape	ADRC
Model Independent	yes		yes		yes
Design Variables		K_n, ω_n		K_n, ω_n m, n, ω_1, ω_2,	b_o, ω_o
Tuning Parameters	K_P, K_I, K_D	K_P, K_I, K_D, ω_1, ω_2, ...	K_{Pvel}, K_{Ivel}, K_{Dvel}, K_{Ppos}, K_{Ipos}, K_{Dpos}	ω_c	ω_c

Table 1 summarizes the setup and tuning ease of the various controllers:

Table 1 illustrates the qualitative nature of empirical tuning and the tradeoffs one faces in tuning complexity, leading one to conclude the advantages of PID, Parameterized Loop Shaping, and ADRC. The author proposes an Integral of Squared Control (ISC) performance index, analogous to the well known ISE performance index, be combined as part of a cost function to be minimized during tuning. A common MIMO Optimal Control Theory quadratic cost function [12] is:

$$J = \int_0^t \{e^T(\tau)W_e(\tau)e(\tau) + u^T(\tau)W_u(\tau)u(\tau)\}d\tau \quad (15)$$

where $W_e(\tau)$ and $W_u(\tau)$ are possibly time varying weighting matrices for the error and control respectively. The LTI equivalent for a SISO system is:

$$J = ISE \cdot W_e + ISC \cdot W_u \quad (16)$$

where the common performance indices are used:

$$ISE = \int_0^t e(\tau)^2 d\tau \text{ and } ISC = \int_0^t u(\tau)^2 d\tau \quad (17)$$

The weighting constants would be chosen such that $ISE{\cdot}W_e$ and $ISC{\cdot}W_u$ are of the same order, thereby balancing their effect on tuning. In the system being studied one wants errors on the order of 1% or less, and control actions on the order of 10V or less, so that $W_e \approx 1000{\cdot}W_u$.

As an aside, the Integral of Time weighted Absolute Error (ITAE) and Integral of Time weighted Squared Error (ITSE) are more commonly used for regulatory applications, where an offset error is avoided, but the ISE, and by extension the ISC, is well suited for transient processes such as this.

The tuning exercise then becomes a nonlinear programming optimization problem, using the proposed cost function *J(e,u,t)*. A very simple "downhill" search direction was used in conjunction with an iterative trust region reduction via computer simulation of the system to arrive at tuning parameters for the five controller types. A simple search was done because the system is nonlinear and simulated offline, so efficiency was not paramount. The tuning parameters generated are admittedly local optimums, but were consistent over a range of initial conditions for the search. Future research is proposed to study quasi-Newton or other trust region reduction methods, which might lead to more efficient and broader results.

Commercial Tuned Compare Data	% Point to Point Error	% Trans Error	% Offset	Offset Time %	Osc / Signal
Linear @ Load=1					
PID	0.8%	5.0%			
PID + LL	1.0%	5.3%			
PID + FF	0.3%	1.3%			
LoopShape	0.0%	0.5%			
ADRC	0.2%	0.8%			
Linear @ Load=0.5					
PID	0.4%	2.7%			
PID + LL	0.4%	3.0%			
PID + FF	0.2%	0.8%			
LoopShape	0.0%	0.5%			
ADRC	0.4%	1.7%			
Linear @ Load=2					
PID	2.1%	8.3%			
PID + LL	2.5%	10.0%			
PID + FF	0.6%	2.5%			
LoopShape	0.1%	1.3%			
ADRC	0.7%	3.3%			
NonLinear @ Load=1					
PID	1.7%	6.7%	2.5%	1666%	25%
PID + LL	1.8%	7.3%	3.0%	1666%	25%
PID + FF	2.1%	8.3%	2.5%	1000%	19%
LoopShape	1.7%	6.7%	2.5%	1666%	13%
ADRC	0.8%	3.3%	5.0%	833%	6%
NonLinear @ Load=0.5					
PID	3.3%	13.3%	5.0%	1666%	100%
PID + LL	0.8%	3.3%	3.0%	1666%	25%
PID + FF					
LoopShape	4.2%	16.7%	5.0%	1666%	100%
ADRC	1.7%	3.3%	2.5%	833%	6%
NonLinear @ Load=2					
PID	4.2%	16.7%	5.0%	1666%	25%
PID + LL	5.0%	23.3%	12.5%	1666%	25%
PID + FF	7.5%	16.7%	5.0%	1000%	13%
LoopShape	4.2%	16.7%	2.5%	1666%	6%
ADRC	2.1%	13.3%	7.5%	833%	6%

Table 2 summarizes the performance of the various commercially tuned controllers operating on linear and nonlinear systems under varying load conditions:

QUANTITATIVE MEASURES:

The criteria used in these comparisons are: point to point error as a percent of set point change, and transient following error as a percent of set point change, offset error as a percent of disturbance, offset recovery time as a percent of disturbance rise time, and control signal oscillation as a percent of total control signal. All measures are better when smaller.

All the comparison measures are taken under simulated conditions of a cyclic 10% control signal disturbance, a cyclic 10% max torque disturbance, 0.5% feedback sensor random noise, 10% viscous friction damping, 2% max coulomb friction offset, and 1° of total transmission backlash. The control system responses are finally compared when the load is halved, and then doubled, without retuning or otherwise altering the system design. (One would otherwise rebalance the system power/load ratio if the load changed by more than 2X.)

NOTE: It should be readily apparent the systems are much more consistent and stable when tuned using the proposed cost function. What is less obvious, and must be brought to the reader attention, are the consistently lower values for error and control in almost all instances and for all controller types when tuned using the cost function.

The empirically tuned velocity feed forward controller became unstable when the inertia load ($J_t = 0.5$) is reduced. No values are reliably measurable. In these cases the extreme values have been designated with a solid bar in the Table 2. Also note the velocity feed forward control tuned using the new cost function did not become unstable.

ISE & ISC Tuned Compare Data	% Point to Point Error	% Trans Error	% Offset	Offset Time %	Osc / Signal
Linear @ Load=1					
PID	0.1%	5.3%			
PID + LL	0.3%	5.7%			
PID + FF	0.2%	4.3%			
LoopShape	0.3%	4.3%			
ADRC	0.2%	1.7%			
Linear @ Load=0.5					
PID	0.3%	3.3%			
PID + LL	0.0%	3.0%			
PID + FF	0.0%	2.5%			
LoopShape	0.1%	2.0%			
ADRC	1.0%	4.3%			
Linear @ Load=2					
PID	0.6%	13.3%			
PID + LL	1.4%	14.3%			
PID + FF	1.1%	10.7%			
LoopShape	2.1%	14.7%			
ADRC	2.0%	6.7%			
NonLinear @ Load=1					
PID	0.1%	5.7%	2.3%	1500%	56%
PID + LL	0.3%	6.7%	2.6%	900%	59%
PID + FF	0.2%	5.0%	2.1%	733%	47%
LoopShape	0.3%	4.2%	1.6%	1267%	78%
ADRC	0.2%	1.8%	1.8%	1267%	34%
NonLinear @ Load=0.5					
PID	0.3%	3.3%	2.0%	1333%	156%
PID + LL	0.2%	3.3%	2.0%	833%	169%
PID + FF	0.1%	2.6%	1.6%	733%	281%
LoopShape	0.1%	2.0%	1.4%	833%	219%
ADRC	1.0%	4.3%	2.0%	800%	34%
NonLinear @ Load=2					
PID	0.6%	13.3%	3.0%	1667%	44%
PID + LL	1.4%	14.3%	3.5%	1000%	47%
PID + FF	1.1%	11.3%	2.8%	833%	41%
LoopShape	1.9%	15.0%	3.5%	1500%	59%
ADRC	2.1%	8.3%	5.0%	1067%	34%

Table 3 summarizes the performance of the various performance cost function tuned controllers operating on linear and nonlinear systems under varying load conditions:

Point to Point Error and Transient Following Error:

Point to point error is the common "overshoot" error for position transitions, a primary measure of set point following accuracy. It is measured as a percentage of the change.

Transient error is the error during transition from one set point to another, measured as a percentage of the transient signal and as such it is an indication of the phase lag in the system.

- PID control can be well tuned for a broad spectrum of conditions, as is evident here.
- PID with velocity Feed Forward did indeed respond exceptionally when the process remained linear. It is dependent on stable tuning and may be unstable when un-modeled dynamics occur, such as noise or resonance. This is evident when the load was halved.
- ADRC with PD control is also quite responsive to commands and remains stable with load changes and nonlinear dynamic stress while better following set point changes, particularly transient.
- The Parameterized Loop-Shaping control provides excellent set point following performance for the linear process, yet performs only as good as a well tuned PID for nonlinear and un-modeled dynamics.

Offset Error and Offset Recovery Time:

Offset error is measured similar to point to point error except that it is compared to the magnitude of the disturbance, whether a disturbance of the control signal, or a torque disturbance on the motor.

Offset recovery time is that time needed for any offset disturbance to reach the steady state error specification, divided by the rise time of the disturbance signal, measured as a percentage of the rise time.

- The ADRC with PD and the PID with velocity Feed Forward controllers are much faster responding controllers compared to the others.
- The well tuned PID controller, the PID control with lead compensation, and the Parameterized Loop-Shaping control recovery times are longer than desired, but manageable.

Control Signal Oscillation:

Control signal oscillation as a percentage of control signal is a measure of how much sensor noise is filtered by the controller. This is a very important measure as it relates to unnecessary control activity and the resulting accelerated wear and failure of components. The system mechanical resonance provoked the extreme responses seen in these examples, however, even small sensor noise does contribute cumulative oscillation, leading to eventual system degradation.

- Any measure >100% implies the control signal is saturated and oscillating alarmingly. This is the case for lightly loaded conditions for the simple PID or the Loop Shaping control.
- The velocity Feed Forward control is unstable under light load when provoked by sensor noise.

- The ADRC with PD control is least affected by resonance, and it is consistent with changes in loading and in the event of noise.
- The PID with lead compensation yields the results desired, more stability than simple PID under varying loads. The performance could be improved further with multiple stages of compensation, at the expense of simple tuning.

Cost Function Comparison:

Figure 2 illustrates the relative merits of the various control types as they are able to minimize the ISE and ISC performance indices.

- The ADRC control performs significantly better on both indices. The control signal itself, not shown here, is much less oscillatory than the others, as well as being lower in value.
- The Feed Forward and Lead/Lag compensated controls both improve on PID with more accurate tracking of commands, but at obvious expense in terms of control signal magnitude.
- The Parameterized Loop Shaping control surrenders some performance to the PID in trade for its single parameter tuning.

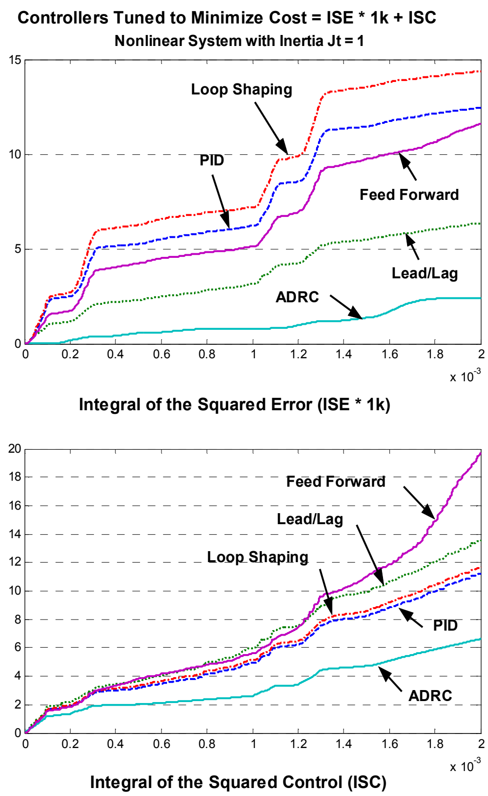

Figure 2. Control Performance Indices Comparison

VII: CONCLUSION

The controller comparisons illustrate superior performance of the model independent ADRC, versus other topologies, particularly in response to load changes in a nonlinear application. Both the error and the control effort for ADRC were significantly less, and the superior robustness in the presence of resonant modes is admirable. The model independent ADRC also proved comparable to the fundamental PID in terms of design and tuning simplicity. The improved performance of more complex topologies usually exacted a higher price in terms of design and tuning ease, or control effort.

The comparisons also illustrate the benefit of a performance index cost function used as a tuning criterion. All control topologies demonstrated improvement after minimizing the cost function.

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