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Benchmark Tests of Active Disturbance Rejection Control on an Industrial Motion Control Platform

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Abstract— **The improvement of Active Disturbance Rejection Control over the existing industry controller is benchmarked with the state of the art industrial automation equipment, using the common PLC, drive, and the mechanical transmissions of belt, gear, and direct coupling. All together 168 tests are run at various line speeds, different levels of pulse-like torque disturbance, and with various viscous friction. The results are tabulated for comparison, in terms of integrated absolute error and maximum error observed, and the RMS and peak torque required from the drive motor. In the worst case scenarios, across all variations of mechanical configuration, we observe the performance improvement from just under 20% to 290%, with comparable control efforts. Together with simplicity in the control algorithm itself, the ease of tuning and operation, such improvement establishes ADRC as a viable replacement of the existing industry controller in manufacturing.**

I. INTRODUCTION

All manufacturing processes involve motion, that is, moving objects from one location to another. In the endless drive towards higher productivity and lower cost, the advance in motion control has played a key role, from better sensors, such as the optical encoders that eliminated much noise from the position measurement, to better actuators, transmissions, and faster computer processors that regulate all aspects of motion. With low profit margins and relentless pressure of reducing cost, continuing upgrade of motion control technology is a matter of survival in manufacturing industry.

If one takes notes of all technological progresses made in the last few decades in motion control, one can not but notice a glaring exception in feedback control technology that has been dominated to this day by the Proportional-Integral-Derivative (PID) control from the 1920s [1]. It is as though someone puts an engine of Model T from early 1900s in a 2008 automobile. However simple, powerful, versatile, and user friendly PID is, it is a piece of analog technology that leaves much room for improvements in the digital world of today. Screamingly fast processors, rich sources of data, and modern data processing technologies are all here for us to take advantage of.

To be sure, much progress has been made in control theory since 1940s, giving us a much deeper, rigorous, understanding of feedback control systems and a pool of advanced design techniques to choose from. But to replace PID, a new technology must 1) improve on its key weakness; 2) achieve a significantly higher performance; 3) be tolerant of large uncertainties in a physical process; and 4) be easy to tune and operate by an average factory operator.

A key problem in motion control and a major weakness in PID is disturbance rejection. A measure of control system performance is how fast it rejects disturbances in a manufacturing environment and tightly regulate the process variables to within their tolerance. In this regard, PID is an error-driven technique that only reacts to disturbances when they have already caused process output to deviate from the setpoint significantly.

In recent years, a promising new technology, Active Disturbance Rejection Control (ADRC), has been proposed and continuously developed to address this very issue, to the degree that all four requirements in replacing PID are now largely met. The key idea in ADRC, that of actively seeks out (estimates) disturbance and cancel it out with the control action, was first proposed by Han in 1995 [2]-[3], formally presented in the English literature in 2001 [4]-[5], and its design philosophy fully articulated in 2006 [6], and then in 2009 [7].

It turns out that ADRC is a particularly suitable solution for motion control problems and numerous papers have shown its effectiveness in both software simulation studies and hardware test results. See, for examples,[4], [8]-[9]. But these studies were carried out largely in an academic setting with little input from practitioners. The purpose of this paper is to quantify the performance of ADRC in an industrial setting, as it moves towards industry-wide applications. To give realism to the results, we adopt the common practice in industry into performance measures, quantification of improvements, hardware configurations used, etc. We also used the motors, transmissions, drives and motion controller that are commonly found in a manufacturing line. In doing so, we hope that the results will be transparent to both engineers and management in industry.

The paper is organized as follows. The existing industry controller and ADRC are described in Section II, followed by the description of the test setup in Section III. The performance matrix is discussed in Section IV and the results of the tests are shown in Section V. Finally, concluding remarks are given in Section VI.

II. CONTROL SYSTEM DESIGN

A. Problem Formulation

Based on the Newtonian Law of motion, a simplified mathematical description of a motion control problem can

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be written as a second-order linear differential equation

$$\ddot{y} + a\dot{y} = bu \tag{1}$$

where u is motor torque,y is the position output, and a, b are friction coefficient and torque constant, respectively. And most motion control solutions today are derived based on this model.

In reality, however, there is much nonlinearities and disturbances in the plant, such as nonlinear friction, the gear backlash, torque disturbance, *etc*. A more realistic plant description is

$$\ddot{y} = g(\dot{y}, d) + bu \tag{2}$$

where $g(\cdot)$ is an unknown nonlinear function, d is an external disturbance, and coefficient b varies with load changes.

A servo control is designed to force the output of the plant to track the reference signal, denoted as r, and its performance is usually quantified using the Integral Absolute Error(IAE) and, Maximum Error(ME). The control effort needed is normally viewed in terms of the root mean square(RMS) torque and the peak torque. In general, a good controller results in smaller errors and control efforts but there is a trade-off between performance and control effort.

B. Industrial PID Design

The PID based industrial motion control system is shown as follows.

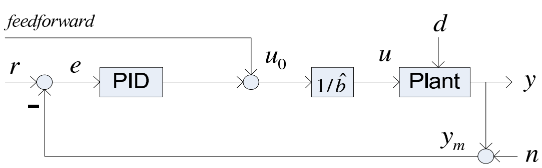

Fig. 1. Block Diagram of PID Control Systems

If we normalize the plant in Fig. 1 to a unit gain by

$$u = u_0/\hat{b} \tag{3}$$

where $\hat{b}$ is a given approximate value of b, then the PID controller can be shown in general as

$$u_0 = k_p e + k_d \dot{e} + k_i \int e \tag{4}$$

where $e = r - y$ is the tracking error, k_p, k_d, and k_i are proportional, derivative, and integral gains, respectively. Note that in the actually implementation of PID in industry, practitioners, over the decades, have added various additional components to improve its performance, such as feedfoward, nonlinear gains, anti-windup, self-tuning, etc. Fully characterizing such vast, incremental, improvements over the years is beyond the scope of this paper. These details in the industry controller are not discussed here but were fully employed in the process of evaluation for fairness. The presence of such array of various of add-ons to PID perhaps also indicates the need for a better understanding the inadequacy of this controller and the need for a fundamental change in the control algorithm itself.

C. ADRC Controller

With the wide spread use of optical encoders in motion applications, clean, relatively noise free, position measurement is often available, from which the velocity signal can be obtained using numerical differentiation. In this case, motion plant (2) becomes

$$\dot{v} = g(v, d) + bu \tag{5}$$

where v is velocity.

In ADRC design, the controller has two parts: 1) the PID controller for the ideal plant of

$$\dot{v} = \hat{b}u \tag{6}$$

and 2) a disturbance rejection mechanism that estimate and rejection any discrepancies between the design model (6) and the real plant (5), denoted as the *total disturbance*

$$f = g(v, d) + (b - \hat{b})u \tag{7}$$

Specifically, the plant (5) can be rewritten as

$$\dot{v} = f + \hat{b}u \tag{8}$$

which shows that the unknown dynamics, the external disturbance and the inaccuracy of the $\hat{b}$ are lumped together in f to be estimated and canceled out by the control action. If the cancellation is successful, such control system will have inherent robustness against dynamic change, superior disturbance rejection properties, and it reduces the need for re-tuning when the parameter b in the real plant varies in a certain range during the operation, which is a issue in the industrial controller.

Incorporating the state observer technique from Modern Control Theory, f is estimated as follows. Let $x_1 = v$ and $x_2 = f$, and the augmented state space form of (8) is

$$\begin{cases} \dot{x} = Ax + B\hat{b}u + E\dot{f} \\ y = Cx \end{cases} \tag{9}$$

where:

$$A = \begin{bmatrix} 0 & 1 \\ 0 & 0 \end{bmatrix}, B = \begin{bmatrix} 1 \\ 0 \end{bmatrix}, E = \begin{bmatrix} 0 \\ 1 \end{bmatrix}, C = \begin{bmatrix} 1 & 0 \end{bmatrix}$$

With the augmented state space form of the plant, ADRC design is straightforward, shown as follows.

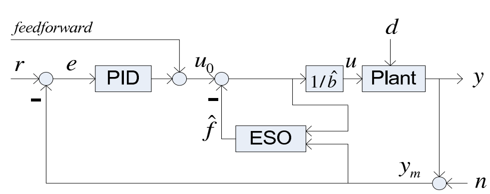

Fig. 2. Block Diagram of ADRC Control Systems

As shown in Fig. 2, an Extended State Observer is now constructed to estimate both states and the *total disturbance*:

$$\dot{z} = Az + B\hat{b}u + L(\dot{y} - z_1) \tag{10}$$

where the observer gain vector $L = \begin{bmatrix} 2\omega_o & \omega_o^2 \end{bmatrix}^T$ is chosen to let all the observer eigenvalues locate at $-w_o$ for the ease of tuning. With a proper selection of w_o, z_1 and z_2 approximate v and f respectively.

Then the total disturbance is rejected with

$$u = u_0 - z_2 \tag{11}$$

forcing the plant to behave as the ideal plant (6), which can be easily controlled using the existing controller described in (4). In fact, in comparison of the two controllers, ESO and disturbance rejection are superimposed on the existing PID control structure to make the least amount of changes in the existing control system.

III. Testbed Setup

To systematically evaluate the improvement of ADRC over existing motion control method in a typical industry automation equipment, we first build a test bed consists of two low inertia motors(a drive motor and a load motor), coupled with three different kinds of mechanical transmissions: belt, direct coupling, and gear. The two motors are controlled by two industry standard servo drives respectively. The industrial PID control algorithm and ADRC control algorithm are uploaded to the driving drive. The reference profiles, configurations, and the test procedures are uploaded via a computer to the Programmable Logic Controller (PLC) module that is interfaced with both drives to provide reference profiles to and collect test data from the drives. The test bed hardware configuration is shown in Figure 3.

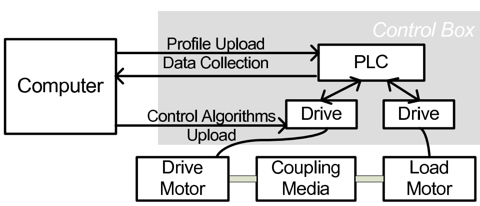

Fig. 3. Test Bed Configuration

The machine is designed so that the drive motor forces the load to move according to the given reference position profile. The position measurement is obtained through an optical encoder in the drive motor and the controller correspondingly generates a torque command in the drive. To emulate operating conditions in a manufacturing process, the load motor is operated in open-loop with the corresponding profiles, generating a bidirectional torque pulse to emulate external disturbances and a torque proportional to the motor speed to emulate viscous friction.

IV. Performance Matrix

The objective of this study is to make a comparison of two different control algorithms under the adversarial conditions similar to those found on a manufacturing floor. The performance of the control system is evaluated under different production line speeds at 50%, 70% and 100% of the maximum speed, sudden bi-directional external torque disturbances at 40% and 100% of the maximum strength, and in the presence of viscous frictions, emulated using a disturbance torque that is programmed to be proportional to the line speed. Altogether there are twelve tests run for each controller and transmission configuration, numbered as shown in the table below. There are three different transmissions between the drive motor and the load, including belt, gear and direct coupling, each has its own variations. The total number of mechanical configurations is seven and the total number of tests run is 84 for each controller.

TABLE I
Test Number Matrix

	50% speed	70% speed	Full speed
No disturbance	1	2	3
40% torque disturbance	4	5	6
100% torque disturbance	7	8	9
add viscous damping	10	11	12

We show the comparison of ADRC with PID in four graphs for each transmission configuration. In each graph, we will always display, for convenience, the ratio of the industry controller data over the ADRC data on the vertical axis and the test number (1-12) on the horizontal one. For example, in the position IAE or ME figure, if the bar display is 1.2 for test 3, it means that the industry controller has an error measure that is 1.2 times that of ADRC at full speed; in the RMS torque or peak torque figure, if the display is 0.9, it means that the industry controller uses 90% of the torque required by ADRC.

In each transmission configuration the drives are tuned for the industrial controller first. After the twelve tests for the industrial controller are done, the code is switched to ADRC and then tested without re-tuning.

V. Test Results

In this section, the presentation of test results is arranged according to the transmission used, from belt, to gear, to direct coupling. Toward the end, we also perform a frequency sweep to characterize the disturbance sensitivity of the two controllers in a Bode plot.

A. Belt Test

Two timing belts are tested: one is wide and the other is narrow. The drive motor and the load motor are coupled with a timing belt each time, as shown in Fig. 4. The pulley radius ratio between drive motor and load motor is 3:1. The belt tension is measured by a sonic belt tension meter and adjusted to a constant value. The reference is a ramp profile.

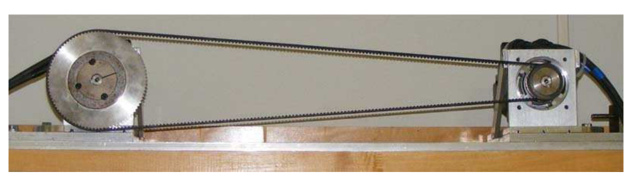

Fig. 4. Belt Configuration

The position IAE and ME ratios between the industrial controller and the ADRC controller are shown in Fig. 5 and Fig. 6 respectively.

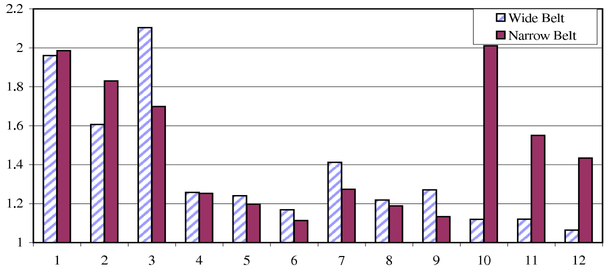

Fig. 5. Position IAE Ratio of Belt Tests

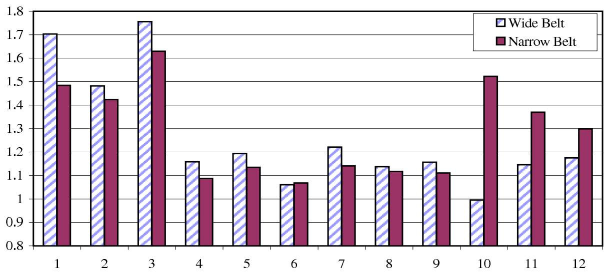

Fig. 6. Position ME Ratio of Belt Tests

The position error results show that position IAE is improved with ADRC, especially for the case without external disturbance and the case with the Narrow belt and viscous friction. The RMS torque and peak torque ratios are shown in Fig. 7 and Fig. 8 respectively, indicating that ADRC needs a little bit more torque to achieve the performance improvement.

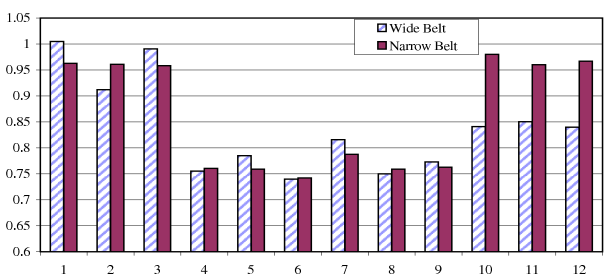

Fig. 7. RMS Torque Ratio of Belt Tests

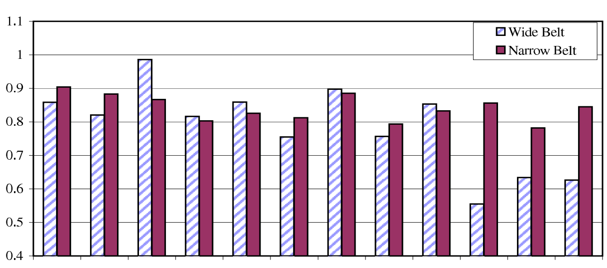

Fig. 8. Peak Torque Ratio of Belt Tests

B. Gear Test

In the gear tests, the drive motor and the load motor are coupled with gears in a 1:1 ratio, as shown in Fig. 9. Three different backlashes are used: low backlash , medium backlash , and high backlash.

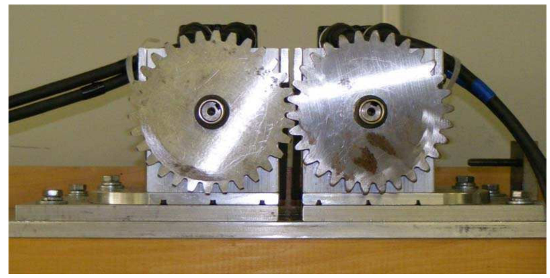

Fig. 9. Gear Configuration

The position IAE and ME ratios between the industrial controller and the ADRC controller are shown in Fig. 10 and Fig. 11 respectively.

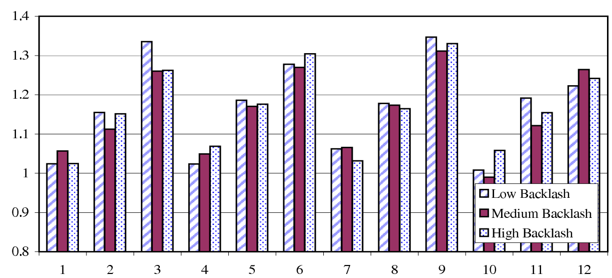

Fig. 10. Position IAE Ratio of Gear Tests

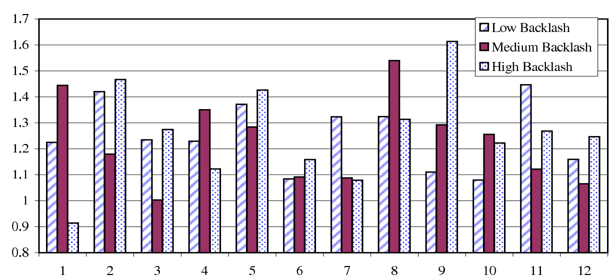

Fig. 11. Position ME Ratio of Gear Tests

The test results show that the position IAE is improved with ADRC controller and the improvement is bigger for higher speed. And in most case (except for case 1 of high backlash), the position ME is improved with ADRC controller. The RMS torque and peak torque ratios for gear tests are shown in Fig. 12 and Fig. 13 respectively, indicating that ADRC needs a little bit more torque to achieve the performance improvement. In some cases, ADRC even requires less peak torque.

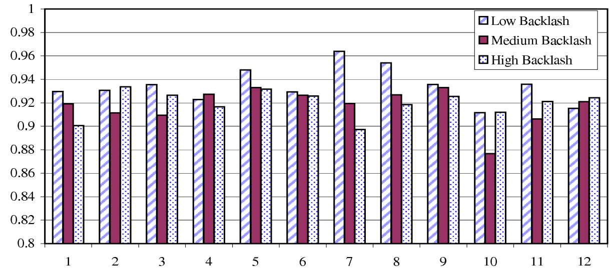

Fig. 12. RMS Torque Ratio of Gear Tests

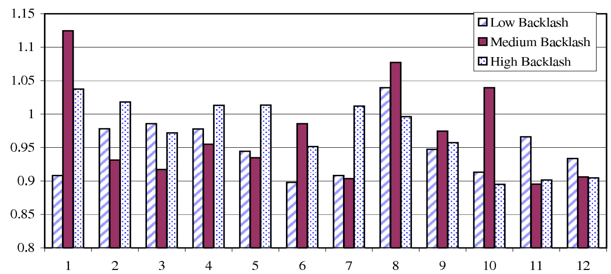

Fig. 13. Peak Torque Ratio of Gear Tests

C. Direct Coupling Test

In the direct coupling tests, the drive motor and the load motor are first coupled with a large coupling, as shown in Fig. 14. The drives are tuned for the industrial controller and tests are implemented for both the industrial controller and the ADRC controller. Then in order to investigate the tolerance of inaccuracies or variations in b, the motors are coupled with a small coupling without re-tuning (keep using the $\hat{b}$ tuned for large coupling configuration), and the industry controller and ADRC are tested for comparison.

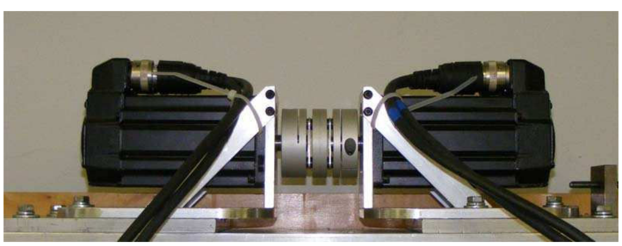

Fig. 14. Direct Coupling Configuration

The position IAE and ME ratios between the industrial controller and the ADRC controller are shown in Fig.e 15 and Fig. 16 respectively.

The test results show that, with a more accurate $\hat{b}$ (in the large coupling tests), the industrial PID controller yields two times IAE and 1.5˜2.5 times ME over ADRC controller. When $\hat{b}$ is quite different from b(in the small coupling tests), ADRC yields much better performance than the industrial controller: up to four times better IAE when the speed is 100%, and 2˜3 times better ME. The RMS torque and peak torque ratios are shown in Fig. 17 and Fig. 18 respectively, which indicate that ADRC consumes less RMS torque in most cases and less peak torque in more than half cases.

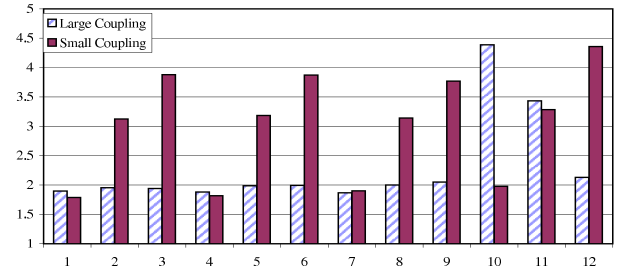

Fig. 15. Position IAE Ratio of Direct Coupling Tests

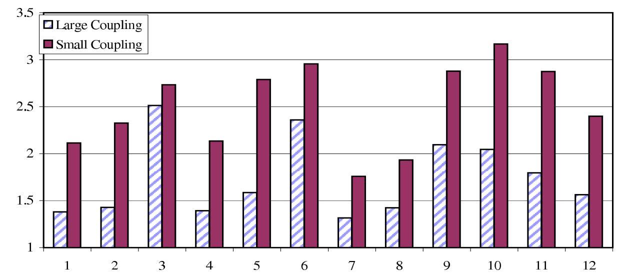

Fig. 16. Position ME Ratio of Direct Coupling Tests

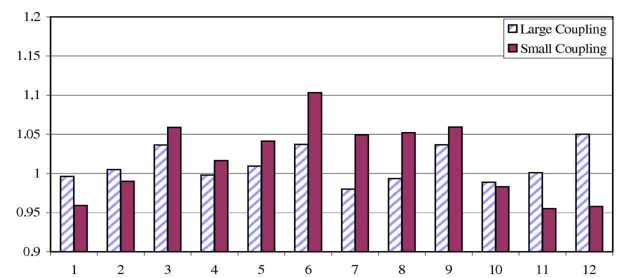

Fig. 17. RMS Torque Ratio of Direct Coupling Tests

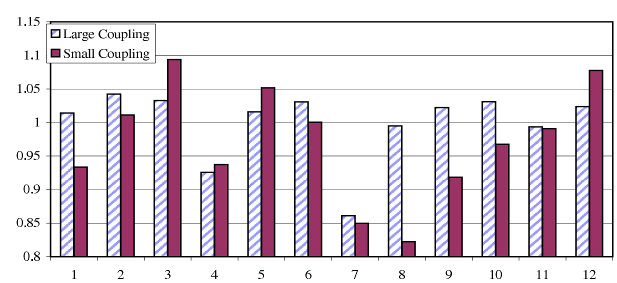

Fig. 18. Peak Torque Ratio of Direct Coupling Tests

D. Maximum Absolute Error Comparison

Directly related to the material cost and product quality assurance in a manufacturing process, the maximum error is an essential criterion for control system performance where any substantial improvement could lead to enormous economic benefits. Of course the system performance varies as operating conditions change, as shown in all 12 tests for seven variations in mechanical configuration. But it is the worst case scenario, the bottom line, that determines the machine tolerance and specifications. Here, let's try to characterize the improvement of such bottom line.

The worst case scenario of both controllers for each of the seven mechanical configurations shows the improvement of the bottom line: from just under 20% to 290%, as shown in the following figure.

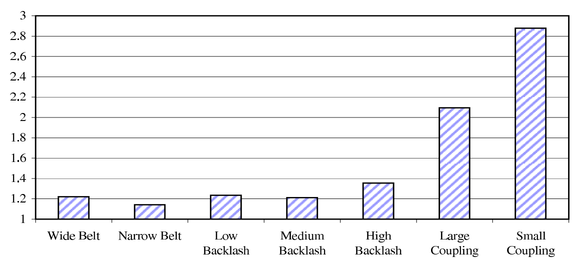

Fig. 19. Maximum Error Ratio in All Cases

E. Disturbance Rejection Ability

To characterize the disturbance rejection ability of the industrial controller and ADRC in frequency domain, a sinusoidal sweep is applied as a disturbance signal to the drive motor free of coupling or load. The disturbance rejection sensitivity is captured and plotted in the following figure, showing as much as 20 dB improvement in ADRC within the bandwidth of the closed-loop control system, which is close to 200 rad/s. Perhaps this helps to shed some light on why we are getting a varying degree of improvement shown above. It may also help us to understand the potential and limitations of active disturbance rejection control.

The Bode plot clearly shows that the most advantage of ADRC is found at the low frequency range, from just a few dBs to 20 dB. If the actual mechanical configuration and external disturbance result in a total disturbance whose main frequency components are in this range, then we'll see a significant reduction in disturbance sensitivity in ADRC. If the total disturbance is outside the loop bandwidth of 200 rad/sec, however, the figure shows that ADRC and the industry controller are very close, with no clean winner.

In light of this discovery, perhaps we could see that with the direct coupling, the mechanical system is the most rigid with the least amount of high frequency components in disturbance, particularly in the change of inertia where we see the largest amount of improvement. On the other hand, the belt transmission is cheap, dirty and inexpensive. But its imperfections such as elasticity and slippage may introduce a total disturbance in a higher frequency range, hence the smaller payoff in performance improvement. Of course in flexibility and cost, the gear is somewhere between the belt and the direct coupling and it gets the corresponding improvement.

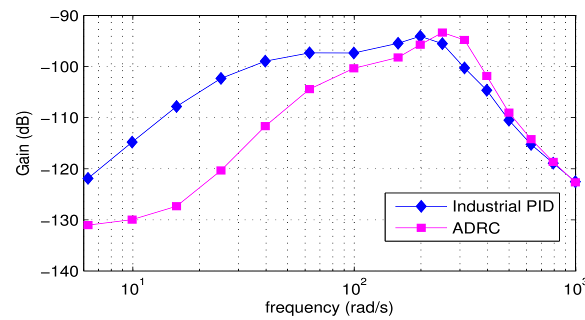

Fig. 20. Frequency Sweep of Disturbance Rejection

VI. Concluding Remarks

In this paper, we present the detailed results from total of 168 tests, performed on the state of the art PLC and drives, characterizing the performance of both the existing industry controller and ADRC. Every attempt is made so that the comparison of the two controllers is fair and comprehensive. To avoid cherry picking, we selected, at the end, the worst case performance of both controllers in terms of maximum tracking error in all seven variations of mechanical configurations, and we observed improvement in ADRC from close to 20% to 290%. The Bode plot of the disturbance sensitivity obtained from a sinusoidal sweep in hardware also confirmed and illuminated the test results as a whole, showing that as much as 20dB improvement in disturbance rejection within the bandwidth of control loop. This rather realistic benchmark evaluation of the two controllers shows the potential of ADRC as a viable method of choice in the manufacturing industry.

VII. Acknowledgments

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