Filtered proportional controller

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***Abstract*—Previously, it was shown that P and PI control could be optimized for simple and consistent operation of the servo creel system. However, both control schemes have the shortcoming of high peak torque due to step velocity response. This new method looks to reduce the peak torque induced by changing the form of controller to a first order filter.**

I. Introduction

Proportional control on the servo creel system has been shown to be a very effective way to control dancer position through empirical testing. There are a few key strengths that are properties of the P controller and a couple weaknesses.

Pros:

1. ~~Guaranteed~~ Zero overshoot if current saturation does not occur
2. Larger dancer displacements at larger speed reducing motor requirements
3. Simple implementation and prediction of response
4. Easy tuning based on dancer stroke and motor capabilities

Cons:

1. High torque peaks during step response
2. Tension variation with speed due to spring

The goal of this paper is to perform optimization around peak torque reduction during operation with minimal additions to the control scheme.

II. S-Domain Controller Solve

In order to generate a controller, the response of the input velocity to the output velocity was set to be a second order. This is due to the fact that second-order systems have an initial slope of zero, allowing torque to continuously change instead of jump. Recall in the previous controller design paper Equation (1) and (2) which define the characteristics of the plant.

Where
v_o is the feed velocity
v_i is the tow surface velocity leaving the spool
x_d is the dancer displacement from the initial position

The system dancer response can then be written in Laplace domain as shown in Equation (1).

$$x_d\ s = \frac{v_o\ s - v_i\ s}{2s} \tag{1}$$

$$G\ s = \frac{1}{2s} \tag{2}$$

The transfer function between v_0 and v_i is a function of the controller structure. Setting this to a second order allows the controller to be solved in the Laplace domain.

$$\frac{v_i}{v_o} = \frac{G}{CG+1} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{3}$$

Solving for C gives the following control structure

$$C\ s = \frac{\omega_n}{\zeta} \cdot \left(\frac{1}{\frac{1}{2\zeta\omega_n} s + 1} \right) \tag{4}$$

$$\tau = \frac{1}{2\zeta\omega_n} \tag{5}$$

Since we want the behavior to be responsive without overshoot, simply set $\zeta = 1$. Now in order to get long term performance similar to the proportional method the steady state gains are set based on proportional control scheme calculations.

Therefore $\frac{\omega_n}{\zeta} = K_p r_s$ in steady state.

Let $\zeta = 1$
Let $K_e = K_p r_s$

$$\omega_n = K_e \tag{6}$$

This means that the steady state displacement is completely determined by the quantity ω_n since the damping ratio is set to $\zeta = 1$.

III. Simulation

In order to see the percentage improvement, we compare the proportional controller with gain $K_e = 25 rads^{-1}$ with the filtered controller corresponding to parameters $\omega_n = K_e$ and $\zeta = 1$. In Figure 1 a plot of the simulated dancer and torque response is shown.

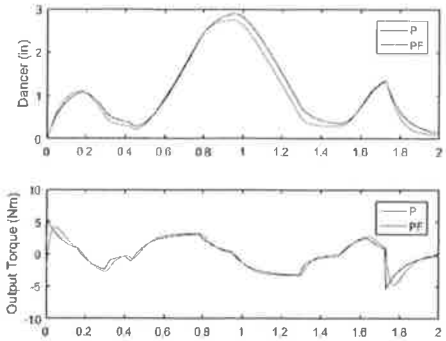

Figure 1 – Simulation of PF response vs. P response

Clearly, the torque ramps up to the maximum, and the maximum torque is also reduced when a spar simulation was conducted. In this case, the maximum torque was reduced by 22% with basically no significant effect on the dancer response as expected. Notably, the peak torques occur when a step velocity input occurs as expected.

IV. Overshoot-Peak Torque trade-off

When designing this type of system there is a need to understand the tradeoffs between certain parameters in the system. A filter is applied to the P controller in this case allowing the torque to ramp up slowly during a step velocity output. Intuitively, the larger the time constant of the filter, the slower the ramp and the lower the peak torque.

However, since the control structure has the damping ratio in the time constant expression in Equation (5), there are serious consequences of raising the time constant to reduce peak torque. The filter time constant is directly related to both the natural frequency and the damping ratio. Supposing that the steady-state controller gain is specified this gives the following expression for the time constant in terms of the damping ratio in Equation (8)

$$K_e = \frac{\omega_n}{\zeta} \tag{7}$$

$$\tau = \frac{1}{2K_e\zeta^2} \tag{8}$$

Therefore, the damping ratio can be expressed as proportional to the inverse square root of the time constant.

$$\zeta = \sqrt{\frac{1}{2K_e\tau}} \tag{9}$$

In Equation (9) the effect of larger time constants is the reduction of damping and therefore increase of overshoot. It is possible that some overshoot can be tolerated in the interest of torque reduction. In the interest of testing, we can easily calculate the damping ratio corresponding with a maximum overshoot we are willing to accept. From experience, roughly 2% is reasonable since there are oscillations and safety margins in the system.

$$\zeta = \sqrt{\ln 0.02^{\ 2}/(\pi^2 + \ln 0.02^{\ 2}} = 0.78$$

In this case $\omega_n = 19.5 rads^{-1}$ is adjusted while $K_e = 25 rads^{-1}$. Although only 2% overshoot results with this damping ratio, the peak torque is reduced by 30% compared to the original P control scheme and 10% compared to the critically damped PF scheme.

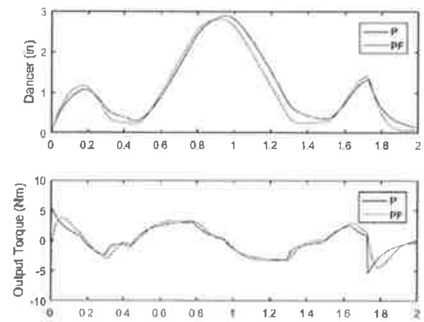

Figure 2 – Simulation of under-damped PF response vs. P response with $\zeta = 0.78$

It should be noted that even if the dancer was arbitrarily long, the peak torque can only be reduced to the peak torque caused by the acceleration ramp. This is due to the fact that in steady state, the accelerations will be matched. Thus, it is meaningless to decrease ζ below values of around 0.7 in any situation.

V. Implementation

In order to implement a filter on the controller, the sampling time T_s must be known. A bilinear approximation can easily be calculated to filter with any time constant. Therefore, the velocity command v_k can be written as function of previous values and the current error measurement in Equation (11).

$$s \approx \frac{2}{T_s}\left(\frac{z-1}{z+1}\right) \tag{10}$$

$$v_k = \frac{K_e(x_{e_k} + x_{e_{k-1}}) - \left(1 - \frac{2\tau}{T_s}\right)v_{k-1}}{T_s(1 + \frac{2\tau}{T})} \tag{11}$$

VI. Conclusion

It has been shown that smoothing the proportional controller can greatly affect the peak torque induced by the system. Reductions of up to 30% can be realized with minimal impact on the dancer response characteristics with a relatively simple implementation and the addition of one extra parameter. Testing must be done to see whether a controller based on a difference equation can be used to smooth torque in practice.

PF Controller Verification

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***Abstract*—Combining P control with a first order filter to the velocity control scheme was shown to theoretically reduce peak torques during step velocity payouts. This development allows for higher steady state gain and lower peak torque. This can potentially be used to downsize the motor and gear ratio or run at higher speeds.**

I. Introduction

An implementation of the controller difference equation was implemented in the Elmo system to test the feasibility and measure the required torque from the motor. Using the new controller the test apparatus was set to simulate step and ramp payouts.

II. Implementation

The control scheme is largely the same with the addition of extra state variables to store the previous dancer error and velocity commands. Necessary constants are added to make parameters more readable. The difference equation was applied using the Tustin's method to approximate a first order system.

In order to verify the similarity of performance, a bode plot was constructed on both the continuous time and discrete time equations.

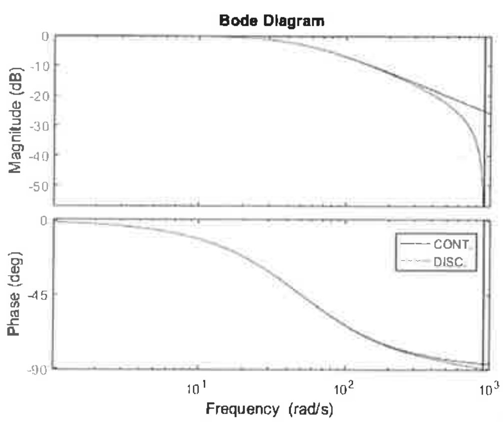

Figure 1 – Comparison of discrete time and continuous time filters

Clearly, the approximation is close until roughly the $500 rads^{-1}$ point which is far beyond the corner frequency of operation of $50 rads^{-1}$.

III. Initial Results

In order to compare the original response of the system the gain constant was set to $K_c = 25 rads^{-1}$. Using the same K_c and tuning ζ to give no overshoot, the torque response of the system was compared. In short, experimental results have shown that up to 50% reduction was realized by simply adding a first order filter. Tests of effective damping ratio as low as $\zeta = 0.5$ were deemed safe.

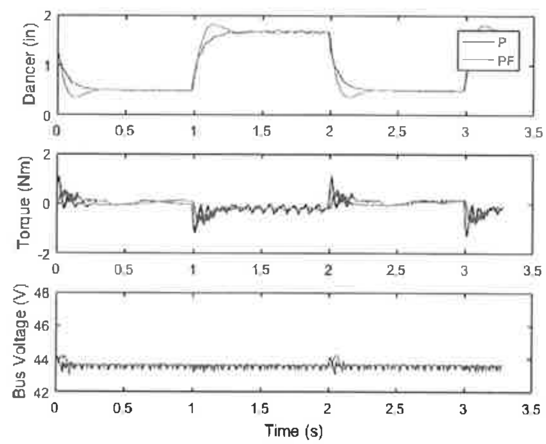

Figure 2 – Experimental payout test with cycled step velocity payout at 2000in/min with a 7.5in spool.

Notably, since the custom power supply is under development, a prototype with 42V nominal was used with a 4700uF capacitor to reduce voltage rise during deceleration. The voltage rise was limited to around 1V which is in the acceptable range.

IV. Conclusion

The theoretical simulation showing a large reduction of torque at the expense of dancer overshoot has been verified. The implementation currently in the Elmo Drive system must eventually be converted to the Animatics system using interrupt based programming to ensure consistent sample times as integrating action must be applied accurately.

More testing can be done in order to construct an understanding of the optimal selection of steady-state gain and damping ratio. In general, the steady state gain changes the dancer stroke and the damping ratio affects peak torque. High steady-state gain reduces dancer stroke in any situation and increases peak torque and lowering the damping ratio can offset the peak torque at the cost of more overshoot.

Trigger based ultrasonic sensor noise filtering

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***Abstract*— Previously a combination of median and moving average filtering had been tested by their ability to remove ultrasonic signal artifacts. Although combining the two results in very accurate as well as robust performance, the computational time penalty is simply too great to be used with the current implementation.**

I. Introduction

The key of this implementation is that it is fast, memory efficient and robust. However, this method does introduce problems regarding false readings during spool replacement, thus a reset routine must be implemented to compensate for such errors.

II. Assumed Artifact Behaviours

During all testing conducted on multiple ultrasonic sensors, readings would fluctuate to a high signal and then return to its nominal value. Each step was for a brief period of time (fractions of a second), so that the diameter could not feasibly change drastically in such a short period. Below is an example of voltage spikes causing a sharp decrease in diameter measurements.

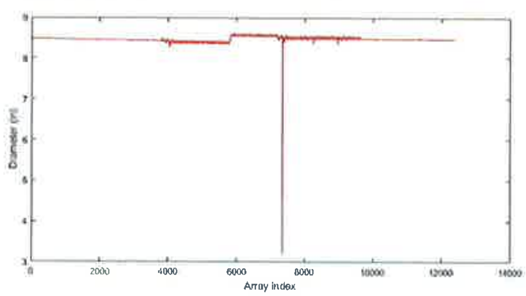

Figure 1 – Diameter reading artifact caused by voltage spike

III. Filter Characteristics

The filter simply uses a condition to determine whether the new reading should be taken or rejected and simply holds the last good reading until a new reading that passes some trigger condition is met.

The specific implementation follows the procedure below:

1. Calculate the difference between current and previous reading
2. If the difference exceeds a statistically improbable value (3 Sigma) set trigger
3. During trigger, hold last good value of diameter for calculations
4. Reset trigger when difference exceeds the trigger threshold again and resume taking new readings

IV. Trigger Definition

During operation, there always exists a fluctuation in the diameter readings due to the imperfect spool geometry as well as wobble due to eccentricity. The effects of noise is very small compared to the aforementioned factors. During operation, the 3 standard deviation value was approximately 0.25 inch diameter variation. Thus, any reading corresponding in 0.25 inch changes or more will be rejected with the previous value used.

V. Code

The code used in the SmartMotor language pertaining to the filter is given below.

```
a = INA(V1,2)
aaa = a-
aa    ' Take the difference reading

IF ABS(aaa) > 200
    a = aa
ELSE
    aa = a
ENDIF
```

VI. Long Payout Test

In order to test the robustness of the filter, a payout test conducted for 5 minutes with two ultrasonic sensors was done. Figure 2 is a plot of the payout test. It can be seen that all large voltage spikes have been removed during operation. Thus, corresponding gain errors due to artifacts have also been removed. The standard deviation was 3mV.

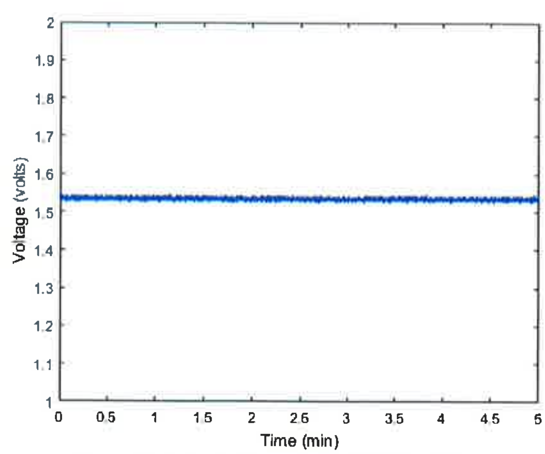

Figure 2 - Extended test with shielded cable

VII. Conclusion

This method will be used during tool changer testing in the absence of other smoothing filters as low material triggers are not the main objective of study.

Trigger-based switching controller

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Abstract—**Slight refinements control structure allowed for the reduction of peak torque requirements of the motor. However, using a single effective gain in the system creates inefficiencies by using more torque than is necessary for many scenarios. This controller which switches between two linear controllers allows two effective gains to be present in the system, allowing more dancer stroke and less stress on the motor.**

I. Introduction

The problem that this control structure aims to remedy is first and foremost the issue of the film material not being wrapped properly and causing failures due to tangling.

The failure has been observed to occur under the condition that most of the material on the spool has been run. After video review, the mechanism by which the failure occurs has been assumed below.

1. Material is run through the system, eventually significantly increasing the inertia of the film spindle
2. The large increase in rotational inertia exceeds the torque able to be provided via friction.
3. Slipping occurs between the carbon and film material, allowing the film spindle to spin at unsynchronized surface speed to the material spindle
4. Slack is formed since the film material spins slower than the material spindle.

Assuming this process of failure there are numerous ways to reduce and remove this failure entirely:

1. Increase tension
    a. Larger tension increases the friction linearly eventually allowing the film spindle to spin without slipping
2. Reduce maximum material spindle acceleration
    a. If this acceleration is smoothed out, the maximum acceleration that the film material must undergo is also reduced
3. Reduce inertia of the film spindle

All three can be implemented simultaneously, however, it is important to note the drawbacks. They are listed below:

1. Higher tension to some extent can be accepted but has also been thought to cause many other issues downstream of the creel inside the business end. Thus, an increase of this tension should be limited.
2. Maximum acceleration of the material spindle is specified by a gain constant. Reducing gain will necessarily increase dancer stroke, and subsequently overall tension. It also reduces the maximum speed of safe operation below the target specification.
3. Reduction of the inertia of the film spindle technically has no drawbacks although its effects are hypothesized to be small unless large design changes are made. The problem is that injection molds are used for the current design allowing very cost-effective manufacturing. The redesign would likely incur large initial costs or simply larger variable costs in the presence of uncertain performance gains.

II. Velocity profile assumption

It has been discussed that the machine is assumed to have a payout velocity that has an initial step input to 2000in/min. Although in reality, this is not possible, it is a good model of the process so that the gain can be tuned.

Note that as mentioned before, it is in this rapid acceleration that the film material tends to lose tension. Lowering the steady-state gain has two functions in this case. More dancer stroke is used, creating higher tension. Simultaneously, lower initial acceleration is used, minimizing the amount of torque that needs to be transmitted to the film spindle.

At this point, the only problem with lowering the gain is that at higher speeds, the gain must be increased in order to eliminate dancer over travel. This creates the motivation for the switching controller system.

III. Modified control structure

In essence, the new control structure modifies the existing one by adding an additional feedback path after the dancer displacement reaches a threshold value. This effectively doubles the steady state gain for higher speeds if the threshold is set correctly.

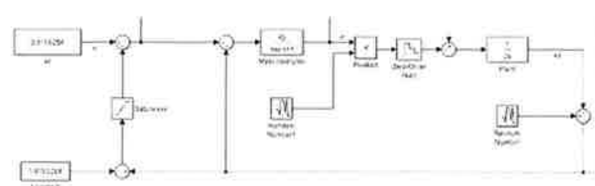

Figure 1 - Switching controller structure

The steady-state displacement is written in Eq. 1.

$$x_{ss} = \frac{x_r}{2} + \frac{x_t}{2} + \frac{v_o}{2K_e} \tag{1}$$

IV. Parameter selection

In order to test the new control system, the first thing was to reduce the gain as much as possible. A selection with $K_e = 17.5 rads^{-1}$ was observed to significantly reduce slack in the film material. This is down from a nominal value of $25 rads^{-1}$ which would imply a 30% peak acceleration reduction.

The next step is to check the steady state displacement under 2000in/min.

$$\frac{0.85ms^{-1}}{17.5rads^{-1}} = 1.91in$$

Therefore, we must select a threshold greater than 1.91in otherwise the steady state displacement changes during an add which creates unnecessary oscillation and torque spikes. We select a threshold $x_t = 2in$ to add a margin for error. Nominally, the set point is $0.25in$. At this point, the steady state gain at rated speed can be calculated.

$$x_{ss} = \frac{1}{2}\left(0.25 + 2 + \frac{1.693}{17.5}\right) = 3.03in$$

This is under the overall constraint that the dancer stroke is only 3.5in total giving approximately 0.25in off the full stroke which is acceptable.

V. Simulation

Using tow velocities emulating 3777X spars combined with noise artifacts the response of the system was simulated. From

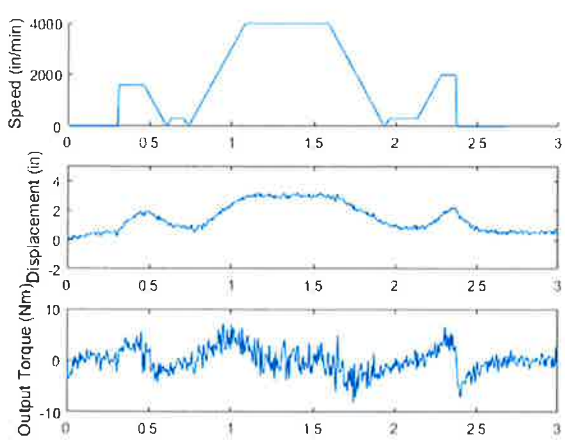

Figure 2 - Simulation of system performance with switching controller

Simulation results show that the torque is within the feasible range. Also, it is clear that the step velocity profile generates very comparable torques to the constant acceleration profile. This shows that the system is balanced properly as one is not significantly more than the other. We can there is no discernable jump in the torque due to switching at around 2 inches of dancer travel which affirms that no Zeno conditions exist that would potentially cause oscillation and subsequent failure.

VI. Experiment

An experiment was conducted using the same velocity profile shown in Figure 2. The performance was as expected and had less noise than the original simulation. The result is shown in Figure 3.

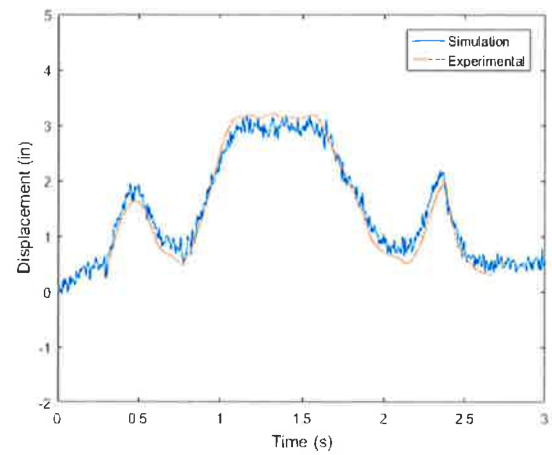

Figure 3 - Comparison between simulation and experiment

As from before, there is acceptable agreement between theory and experiment. It is thus, in principle possible that a PLC using a predictive model can check whether there are errors in the tension control system in real time by comparing dancer travel to predicted behavior. Dancer displacement characteristics in response to feed velocities are independent of spool diameter as long as diameter measurements are accurate which simplifies the implementation immensely. Thus, small or drifting deviation from projected behavior can likely mean sensors must be recalibrated. A real time algorithm using a simple regression check can be used to detect errors in the future.

VII. Conclusion

Changes in the control structure has helped reduce the likelihood of film material related failures. This combined with a slightly higher overall tension and small adjustments to the film spindle are likely to completely eliminate tangling failures in the future. The control structure behaves closely to predictions which allows for possible real time safety checks to be implemented.

Feed motor torque prediction

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***Abstract*—One of the added benefits of the servo creel is that the inertia of the spool should be unseen by the feed motor. As a result, the torque variation is significantly reduced and behaves in a predictable pattern. This paper investigates how torque variation can be predicted and used to aid control efforts at the business end.**

I. Introduction

The experimental setup uses the same apparatus from servo creel testing. This time data is recorded from the feed motor as opposed to the spindle motor.

The key data are listed:

1. Velocity command
2. Actual velocity
3. Motor torque

In a simplified dynamic model of the system, there are only a few main considerations that affect motor torque. The dancer spring tension, dancer mass and friction are the main elements creating opposing torque to the motor that can vary. In this test, a constant velocity with a superimposed sinusoidal velocity was used to generate data to build a simple model to predict torque variation at the feed motor side.

II. Theory

Although spring tension and dancer acceleration are varying factors in the torque requirements, the characteristics of the servo control loop there is a direct linear mapping of commanded velocity to steady-state dancer displacement. This allows for easy prediction of dancer position and acceleration. One problem with this simplification is that the new control algorithm has two distinct behaviors that trigger based on dancer travel. Thus, at high speeds, there will inherently be error

To predict the steady state displacement there are two equations which depend on the velocity command (low or high speed). Equation 1 describes the displacement at low speed, and Equation 2 at high speed.

$$x_{ss} = x_r + \frac{v_o}{K_e} \quad (1)$$

$$x_{ss} = \frac{x_r}{2} + \frac{x_t}{2} + \frac{v_o}{2K_e} \quad (2)$$

The condition at which the operational mode switches is approximated in Equation 3.

$$x_r + \frac{v_o}{K_e} > x_t \quad (3)$$

Thus, the threshold which separates what is colloquially called low or high speed is shown in Equation 4.

$$v_{threshold} = K_e(x_t - x_r) \quad (4)$$

This forms a piecewise linear function of steady-state displacement and feeds velocity. The threshold in the current control system is given as

The parameters used in this test are given below:

$$K_e = 17.5rads^{-1}$$
$$x_t = 2in = 0.0508m$$
$$x_r = 0.1in = 0.00254m$$
$$v_{threshold} = 0.845ms^{-1} = 1995ipm$$

Note that this threshold velocity calculated so that adding material which at most can be done at 2000ipm would see a low control gain using more dancer stroke. This is important since for more aggressive accelerations a reduction in gain proportionally reduces motor torque at the cost of dancer stroke.

Since the spring has a preload and a stiffness constant the torque from the dancer displacement is given in Equation 5.

$$F_d = F_0 + kx_d \quad (5)$$

The dancer acceleration can also be added to produce the overall torque. If we view the problem a curve fitting problem then torque T can be written in a linear 2 variable expression.

$$T = k_0 + k_1 x_d + k_2 a_d \quad (6)$$

Note that due to the tension control system, there should be two expressions for T, one for low speed and another for high speed.

III. Procedure

Below are the basic steps used to collect data.

1. Feed tow through the dancer system
2. Calibrate dancer and diameter sensors
3. Run tension control algorithm
4. Run 2000ipm feed with superimposed 1000ipm sinusoidal velocity at $10rads^{-1}$
5. Record velocity, actual velocity and motor torque of the feed motor

IV. Results

As expected the torque requirement from the feed motor was relatively low. Even with the superimposed sinusoidal velocity which had a maximum of approximately 0.5G acceleration, torque stayed within a 0.17Nm band. The feed motor accelerates tow with no gearbox (direct drive) which means that load torque is not scaled. In the current system a 7:1 reducer is used on a 2000W motor. This motor has an instantaneous stall torque of approximately 9Nm. Using the data to approximate what the feed motor would experience in the worst case scenario results in 1.7Nm which is approximately 20% of instantaneous stall torque.

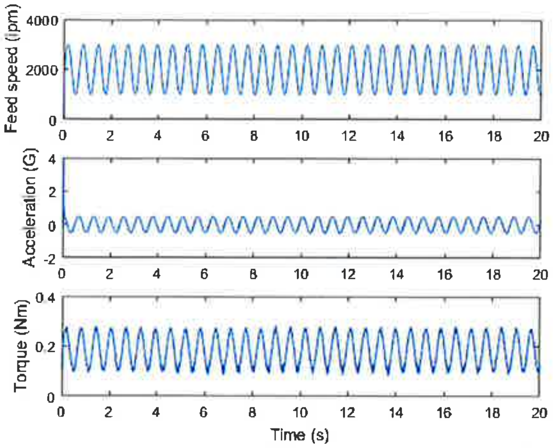

Figure 1 – (top) Measured feed speed converted to ipm; (middle) calculated dancer acceleration; (bottom) measured torque in Nm

Plotting torque versus the velocity and acceleration shows an elliptical plot with points approximately on a plane. This is exactly expected.

Using the curve fitting tool we can find constants in Equation 6. In Figure 2, the planar fit is shown.

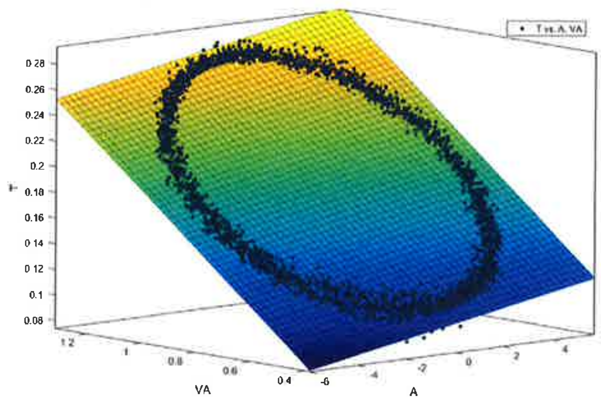

Figure 2- Planar curve fit on torque plot

Following are the constants with their 95% confidence intervals.

$$k_0 = 0.02527 \quad 0.02471{,}0.02583$$
$$k_1 = 0.1894 \quad 0.1887{,}0.190$$
$$k_2 = 0.003594 \quad 0.003538{,}0.00365$$
$$RMSE = 0.006363Nm$$

As it turns out even though the torque expression should be piecewise, the RMSE is sufficiently small.

V. Suggested Implementation

After the machine is built, a calibration routine to calculate the torque equation constants in Equation 6 can be made. This would require each lane to be fed individually, with a few sinusoidal velocity cycles. A simple way to calculate the constants uses an assumed overdetermined system. We can use the data to create matrices $\boldsymbol{A}, \boldsymbol{T}$ and solve for linear constants in k.

$$\boldsymbol{A} = \begin{pmatrix} 1 & x_i & y_i \\ \vdots & \vdots & \vdots \\ 1 & x_n & y_n \end{pmatrix} \tag{7}$$

$$k = \begin{pmatrix} k_0 \\ k_1 \\ k_2 \end{pmatrix} \tag{8}$$

$$\boldsymbol{T} = \begin{pmatrix} T_i \\ \vdots \\ T_n \end{pmatrix} \tag{9}$$

Thus, the overdetermined linear system can be written as in Equation 11.

$$\boldsymbol{A}k = \boldsymbol{T} \tag{10}$$

$$k = inv\ \boldsymbol{A}^T\boldsymbol{A}\ \boldsymbol{A}^T\boldsymbol{T} \tag{11}$$

Another issue is the estimation of dancer acceleration. Currently, the MATLAB implementation uses 4th order finite difference approximation constants for equally spaced data.

VI. Conclusion

From experiment, it has been shown that using a simple linear approximation that torque can be estimated with reasonably small error (3% RMSE). Thus, in practice, a calibration routine can be written to predict torque from each lane so that resultant adjustments to feed motor torque can be added. From the data however, the maximum torque measured which occurred at 3000ipm only generated a motor torque of approximately 0.3Nm. In practice the feed motor does not advance the tow to full speed, thus for this application the motor is oversized. For intermittent operation, this motor can provide roughly more than 10 times the torque required. This is expected since no changes were made to the business end and 16 servo motors have been added to handle the inertia of the spools.

Elongated Dancer Tuning and Simulation

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***Abstract*—New mechanical modifications to the dancer design have allowed for greater stroke within the same footprint of the old design. This means that the gain can be reduced, allowing both lower torque and higher speed operation.**

I. Introduction

Although the servo creel system has shown to be far more consistent in tension control whilst running lower tension, the top speed is limited by the no-load speed of the motor. This is unlike the passive pneumatic system which requires the feed motor or machine axes to draw out material and only applies a drag torque to the system.

For very flat parts that do not require rapid direction changes while depositing material, the pneumatic system has the advantage since the stopping force is higher. The feasibility of this concept has been shown in linear speed testing, where speeds of up to 8000in/min on the TOP heads was achieved when depositing on tape. In the context of the creel, this test shows that the capabilities of the pneumatic system in high top speed may be sufficient.

In hopes to develop the servo creel system as a true next-generation machine, optimizations are considered in this paper to configure the system for both high speed and high acceleration operation.

II. Mechanical Changes

The two main elements in the mechanical system that are under consideration are the reducer and the dancer. The reducer is straightforward as the series of gearboxes have multiple gearing ratios that are possible. For future generation heads instead of using a Wittenstein TP004, a custom Harmonic HPG-14 is under consideration. The reason is that multiple gearing ratios (7,8,9,10:1) are all possible whereas with the TP004 does not have those options. Both gearboxes have the necessary static moment load rating to support rapid C-axis acceleration.

The longer dancer stroke is the main mechanical change that allows for a lower reduction to be used. Since lower torque is necessary, the reduction is reduced subsequently extended the maximum speed of operation. The maximum rotational speed the input of the gearbox will see is calculated below in Eq. 1.

$$\omega_{max} = G\left(\frac{v_{max}}{\pi d_{min}}\right) \tag{1}$$

For the spools used, the minimum diameter is $3.5in$. In this case, we are considering a 8:1 reduction with 5500in/min max speed.

$$\omega_{max} = 8\left(\frac{5500}{3.5}\pi\right) = 4000rpm$$

Empirically, through a 10:1 reducer 4000rpm is consistently achievable on the SM23166MT motors. This is why 8:1 reduction will be used as a starting point for simulation.

III. Calculation of Control Constants

From a previous paper, in Figure 1 we have the switching controller structure. The structure switches from a single feedback loop to two feedback loops to increase the gain of the system. It can essentially be modeled as a 2 DOF controller with two unity gain paths.

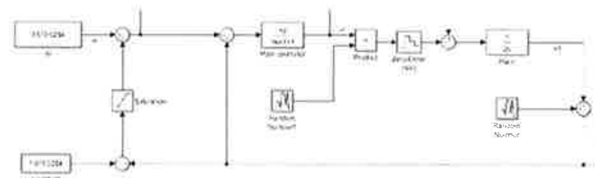

Figure 1 - Switching controller structure

The steady-state displacement is written in Eq. 2.

$$x_{ss} = \frac{x_r}{2} + \frac{x_t}{2} + \frac{v_o}{2K_e} \tag{2}$$

The key to calculating steady-state displacement is to first find the region which the controller is operating.
The main variables that are important to calculating an appropriate threshold to switch are the material addition speed, controller gain, and the reference point.

The control regime is separated into two modes, high and low-speed operation. What defines high and low speed is mainly the initial material addition speed. In order to reduce the torque demands we want a small gain to existing during this process and therefore the threshold has to be above the steady-state displacement that is caused by the maximum addition speed.

$$x_t > x_r + \frac{v_i}{K_e} \tag{3}$$

$$v_i = 2000in/min \tag{4}$$

In general x_r is the initial displacement of the dancer and therefore also sets the initial tension. In practice, values around 0.1in are good values since small oscillation will not cause any lost tension.

At this point, we can set the x_{ss} to be the value of the maximum dancer travel we wish to allow. Keep in mind the control structure has a damping constant to guarantee no overshoot during the process. Therefore, setting this to the maximum "safe" value off the top of the dancer stroke is done. In this case we,, use a value of 4.25in.

$$x_{ss} = 4.25in \tag{5}$$

$$v_o = 5500in/min \tag{6}$$

The calculation is then done to minimize the gain under the Equations 1-5.

We can simplify the calculation by setting $x_t = x_r + \frac{v_i}{K_e}$

$$x_{ss} = \left(x_r + \frac{1}{2K_e}(v_i + v_o) \right) \tag{7}$$

$$K_e = \frac{1}{2} \frac{v_i + v_o}{(x_{ss} + x_r)} = 14.5rad/s \tag{8}$$

Eq. 6 gives the optimal gain with the threshold assumption made. Note that this is not optimal in a traditional optimal control sense since penalty equation was made. This simply allows low torque and high-speed operation within the dancer stroke.

IV. Simulation profile

In order to test both acceleration performance and top speed performance the spars simulation previously used to test the torque requirements of the motor is used. Instead, however, the target speed is 5500in/min. This is approximately 50% faster than the top speeds of current spars machines and is a good starting point since it is likely that other parts of the system such as the heater may bottleneck performance.

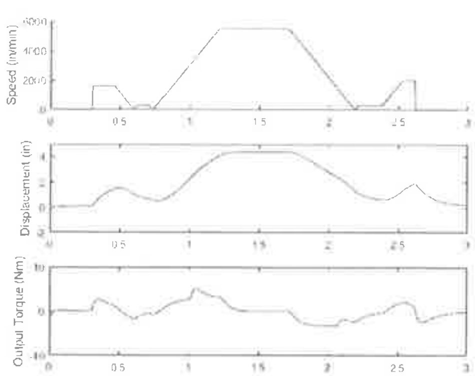

Figure 2 - Modified payout speed profile

The simulation shows that the torque required at the output during acceleration is maximum at 5Nm and is significantly less than the 7.2Nm max for the SM23166MT at the 1000rpm operating point of operation.

V. Conclusion

Initial simulation shows that the torque speed curve of the SM23166MT combined with a 8:1 reducer (versus currently 10:1) is sufficient to run 8.5 inch diameter spools at a top speed of 5500in/min. This means that with the exact same control architecture within the motor, and the exact same commands from the PLC can be used in a seamless way. The only difference is that the software loaded onto the SmartMotor initially will have different control constants.

Thus, the servo creel can likely be used for different aircraft parts appropriately reaching extremely high top speeds whilst maintaining the ability to accelerate at the rate of the machine acceleration.

Diameter standard deviation based triggers

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Abstract—**On the 4017 Top heads it was found that interference between the diameter sensors was present. Currently, it has been shown that using shielded cable can reduce the interference between the sensors. The exact mechanism by which the interference occurs is still unknown at the time of writing. This paper describes a standard deviation based trigger which, when tuned, can be used to identify regions where the readings have interference.**

I. INTRODUCTION

It has been found that from the previous study, ultrasonic sensors do exhibit noise and sometimes spiking in readings when using unshielded cable. Figure 1 taken from "Ultrasonic sensor noise filtering" is shown. Clearly, the shielded cable reduces large spikes as well as general random noise. As a result, shielding was suggested as the first trial solution to the problem. Ben Wiggins has investigated the issue and has shown that ungrounded shielding has definitely reduced the interference between the sensors, although large spikes are still present. Thus, this trigger was designed to identify potential problem areas where additional filtering or rejection can be run in case shielding could not completely remedy the issue.

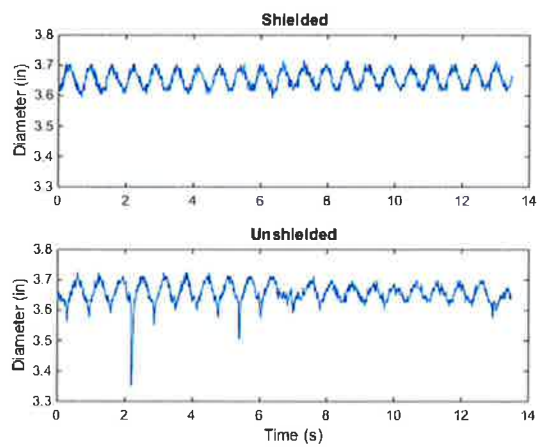

Figure 1 - Shielded vs. Unshielded cable diameter readings

II. WIRING

An important finding was that PIN 4, the synchronization pin for the BUS 004J, should be grounded to reduce large spiking. This has not been done on the 4017 heads and has been left floating which is why I believe there are still large spikes.

Figure 2 is taken directly from the datasheet showing that on a high input voltage the sensor is deactivated. This should be grounded since we want the sensors on at all times.

Synchronization
You can synchronize as many sensors as you like.

- Apply a square-wave signal to the sync-input with pulse width t_i and repetition rate t_p (Fig.3 and technical data).

A high level on the sync-input will deactivate the sensor.

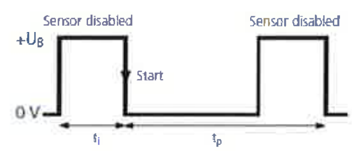

Figure 2 - Synchronization pin should be grounded

III. STANDARD DEVIATION ASSUMPTION

Assuming that ambient noise is present the standard deviation is some small number during normal operation without interference. As shown in Figure 1 there is a sinusoidal wave that is superimposed on the diameter reading since there is a wobble. Notably, on the test setup, the alignment was poor relative to the real machine. However, imperfections in the carbon spool itself will cause the reading to have a larger standard deviation than the deviation due to standard noise.

Thus, there are three elements to the standard deviation.

$$\sigma_e = \sigma_c + \sigma_m + \sigma_s \qquad (1)$$

Where
σ_e is the effective standard deviation
σ_c is standard deviation due to variations in carbon spool diameter
σ_m is the standard deviation caused by misalignment
σ_s is the ambient sensor noise that is always present

During operation all three elements of standard deviation are present. However, while the spool does not rotate, then only the sensor noise should be present. The main argument is that the effective standard deviation should be approximately constant during operation, and that changes in this standard deviation are a result of sensor failure. Although standard deviation is generally used to talk about random variation, these standard deviations are certainly not random as they follow a cycle with a period dependent on the rotational speed of the spool.

Notably, changing the spool will change the standard deviation calculated from a moving standard deviation window. Currently, there is a method to resolve this issue implemented by Dustin Schmidt using dual moving average buffers. At this time, it is assumed that similar methods can be used in parallel with standard deviation trigger.

IV. Algorithm

Since the standard deviation should be constant throughout the payout, all we need to do is sample it sufficiently to get a good estimation. At this point, the window used for the standard deviation estimation is 100 samples. This was done through empirical testing with no theoretical basis.

There are three basic steps to the procedure:

1. Calculate the standard deviation in a window
2. Calculate the derivative of the standard deviation
3. Take the absolute value of the derivative signal
4. Smooth the signal further
    a. Suggest a unity gain first order filter for ease of calculation and computation time
5. Set a threshold of the change in standard deviation used to trigger additional filtering or rejection of readings

Notably, the derivative does not have to be with respect to time. It can be done as a difference calculation or equivalently thought of as a numerical derivative with unity step size.

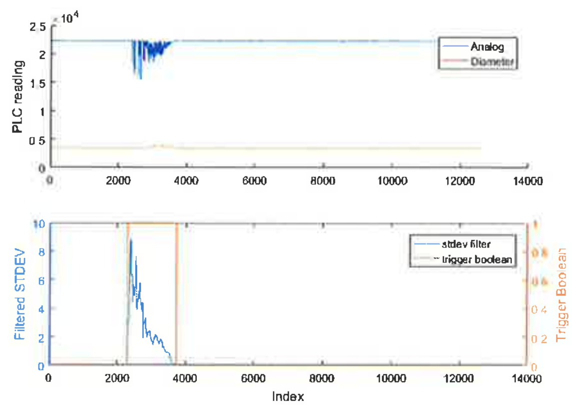

Figure 3 – Standard deviation trigger used to find signal interference

V. Conclusion

The necessity of such a filter has yet to be proven especially after the proper hardware changes have been made. Currently, there are plans to move to shielded and grounded cables which will likely reduce or completely eliminate the issue. Currently, shielded cable has already eliminated the noisy looking interference waveform in the 30-minute window testing. In the case where only large spikes are present, the current filter will likely prevent these issues from adversely affecting performance in a meaningful way. If need be, a good way to completely eliminate these spikes is to use a median filter combined with a moving average, which has been shown to work in a previous paper.