Fully-digital tension control system with PID algorithm for winding ultra-fine enameled wires

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Abstract. An active fully-digital tension control system with PID algorithm is proposed. Only digital signals are involved and processed throughout the closed-loop control system, which employs the micro-controller unit (MCU) dsPIC33EV256GM102 as the main controller with PID algorithm, incremental photoelectric encoder as the angular sensor and AC servo motor as the actuator. A rod-spring mechanism is indispensably constructed to convert the change of tension to the variation of rod’s swing angle. Characteristics of the controlled object are tested and analyzed, from results of which the mathematical model is theoretically deduced. The PID coefficient set is determined by Ziegler and Nichols method. Its practicability is initially validated in simulation using SIMULINK/MATLAB. The prototype is also fabricated and experimented on with ultra-fine enameled wires (0.08mm). In order to enhance the practical performance, PID coefficients are further adjusted in experiments. The results show that the proposed system performs well both in transient process and steady stage. Meanwhile, it has good anti-interference capability as well.

1. Introduction

Tension control techniques have enormous applications in not only civil industry but also national defense, where precise tension regulation is highly essential to guarantee both quality and productivity. In winding process, as an example, the surface strength of engine shells and pressure vessels in a rocket may be deteriorated as huge as 30% due to bad tension regulation [1]. In reeling process, the reeling quality of either paper or band steel hugely relies upon the quality of the control of tension that is conducted by adjusting the torque of pay-off and take-up motors. Moreover, in cutting process of paper or thin film, the tension on materials changes drastically due to the fluctuation of motor velocity, variation of winding diameter and material unevenness. Thus, it can be concluded that proper tension control is very necessary for quality assurance and cost saving.

Passive dancers and hysteresis brakes are tension controllers that are initially designed and developed. Ebler et al. analyzed the use of dancer roll regulation system in paper machinery as early as in 1993 [2]. V. Gassmann et al. demonstrated a method to maintain web tension by using a pendulum dancer (PD) with an $H_\infty$ controller [3]. Kang et al. continued putting work forward by presenting a dancer system with a reduced-order observer [4].

However, passive devices are more likely to lose control precision due to their limitations on instantaneous adaption to fast tension variation, so that the quality of products will be considerably deteriorated. Consequently, in order to cope with aforementioned deficiencies, active tension controllers that can automatically and quickly regulate tension are developed.
Zhou and Gao proposed an active control system that is able to deal with the dynamic changes in the system and reject the external disturbance occurred in web process line [5]. Since system friction and rotational roll inertia cannot be overlooked when the speed of web process is high, Lin et al. proposed observer-based feedback controllers to ensure the estimation of web tension by compensating friction and inertia [6]. Knittel et al. further presented a multivariable $H_\infty$ based controller which can separately track properties and reject perturbations [7]. Some other schemes are also applied in tension controllers to improve the control precision and efficiency. Iterative Learning Control (ILC) is one of the typical schemes to deal with periodic external disturbance and discussed in many researches [8-11]. Especially, Lu et al. put forward an Iterative Learning Sliding Mode Controller (ILSMC) scheme with the employment of a disturbance observer to estimate the wire tension for sensorless wire tension control [11]. However, the external disturbance is not always periodic. Alternatively, Proportion Integration Differentiation (PID) scheme turns out to be another good option. Easy tuning and flexible implementing are two major merits of PID scheme, which further widen its spectrum of applications in industry [12-13], such as heat exchange systems [14-15], pressure control systems [16-18] as well as tension control systems [19-22]. Zhang employed the traditional discrete PID algorithm in filament winding [19]. Pan et al. proposed a high-speed initiative electronic tension control system for winding fine enameled wires with fuzzy PID control algorithm that efficiently improves the flexibility and rapidity of the controller [20]. Zhang et al. proposed an online self-optimizing fuzzy PID controller based on robust extended Kalman filter for systems that are influenced by nonlinear and uncertain factors. Both simulation and experimental results show the proposed controller can reduce the control error and enhance the tension control precision [22].

Many researches on tension control techniques have been reviewed above, however, it is found that the existing researches on wire tension control techniques are surprisingly few, even though enameled wires are a kind of important raw materials that are widely used for electromagnetic winding in motors, household appliances and electronic instruments [23]. In case of the variation of electrical properties induced by poor tension control, some tension controllers that are exclusive for winding machines are developed to ensure the compactness and smoothness of winding enameled wires.

Nowadays in domestic market, magnetic, electronic and servo tension controllers are three main types for wire winding. Magnetic tension controllers regulate tension along the wires by adjusting integrated magnetic dampers in order to change the magnetic field intensity, while electronic tension controllers change the field current to control tension by following the principle of hysteresis. Comparatively, servo tension controllers have an advantage in control precision because of the utilization of servo motors to actively control the tension by tracing the linear velocity of the wires being wound. However, servo tension controllers are always bulky and expensive. One major reason is that the embedded control system is too complex, especially when analog signals are introduced and processed. Redundant circuits for filtering and processing analog signals significantly not only increase the cost, but also lower the integration level of the mechanical structures. Furthermore, as discussed in [24-26], in the new era of industry, real-time monitoring and cloud-based big data processing are foundations in the state-of-the-art technologies, such as Industrial Cyberphysical Systems (ICPS). Whereas, most tension controllers in the market are in lack of real-time monitoring technique, which leads to the situation that neither the real-time tension data can be observed nor the data is traceable to ensure the consistency between different winding operations.

All aforementioned complexities and difficulties bring out the motivation of the research presented in this paper. In the proposed tension control system, digital sensor is employed to ensure only digital signals are involved so that the control system is fully digital. As a result, the control circuit is simple and highly integrated. Moreover, the cost of the proposed tension control system is saved through the employment of the cheap micro-controller unit (MCU) after fully evaluating the hardware requirements of the system. PID algorithm with experimentally-adjusted coefficients is applied to rapidly and precisely regulate the tension along the wires. Control performance is simulated in MATLAB and further evaluated through a series of experiments. In addition, Human Machine
Interface (HMI) is built up by introducing a touch screen that provides the access to online monitoring and data manipulation.

2. Mechanical structure and control system

![Figure 1. Sketch of mechanical structure. A: Enameled Wires Source, B1-B5: Wire-Guiding Wheel, C: Pay-Off Wheel, D: Pay-Off Motor, E: Spring, F: Rod, G: Ceramic Ring, H: Take-Up Wheel, I: Winding Mechanism, \( \theta \): Swing Angle, O: Pivot.]

2.1. Mechanical structure

During the winding process, enameled wires are stretched due to the tension that is generated from the torque difference between pay-off and take-up mechanisms. However, the real-time tension is technically difficult to be precisely measured because it keeps changing fast and subtly. As a solution, a rod-spring mechanism is constructed in this paper to magnify the small stretched deformation of the wires so as to convert the tension to the angle that rod swings. The sketch of the proposed mechanical structure is depicted in Figure 1.

As shown, the rod is fixed at pivot O and hooked by a spring in the x-z plane. Enameled wires travel from the source to the take-up wheel with the guide of guiding wheels. With the tension along the enameled wires, the rod swings \( \theta \) that is the angle between rod’s real-time position and its initial vertical position. Since the dimensions of the structure are all fixed, when the rod stabilizes at a certain position without acceleration, it can be deduced from torque equilibrium equation that tension along the enameled wires \( T \) is uniquely determined by corresponding \( \theta \) as

\[
T = f_1(\theta)
\]

In terms of \( \theta \), it can be further expressed as

\[
\theta = f_2 \int [V_1(t) - V_2(t)] dt
\]

Where \( V_1(t) \) is the line velocity of take-up mechanism, \( V_2(t) \) is the line velocity of pay-off mechanism. Both are (1) and (2) established only when the enameled wires remain under tension without snapping and no relative slippage occurs. To be specific, the rod reaches the equilibrium position and \( \theta \) remains constant only if \( V_1(t) = V_2(t) \). Otherwise, no matter \( V_1(t) > V_2(t) \) or \( V_1(t) < V_2(t) \), the swing angle \( \theta \) will either increase or decrease because of the change of tension \( T \) so that the rod will keep swinging back and forth and the system remains unstable. In conclusion, based on (1) and
(2), the swing angle $\theta$ is set as the controlled variable of the system with the ultimate intention to regulate the tension $T$. Besides, the line velocity $V_2(t)$ of the pay-off mechanism is set as the control variable in this paper.

2.2. Control system

As the one-to-one correspondence between tension $T$ and swing angle $\theta$ interpreted in (1), it is reasonable to control $\theta$ to maintain the rod at the desired position instead of directly regulating the tension. Generally, potentiometric angular transducer, Hall-Effect angular transducer, magnetic voltage division angular transducer and photoelectric encoder are several options for the purpose of accurate angle measurements. However, the first three aforementioned transducers all output analog signals that need to be filtered for inevitable noise and require extra time-consuming analog-to-digital conversion (ADC) for further processing in the main controller. Differently, the photoelectric encoder output digital signals that can be directly processed by controllers, such as programmable logic controller (PLC) and micro-controller unit (MCU). In this proposal, a high-precision 10-bit incremental photoelectric encoder is employed, which is able to yield as many as 2048 digital level pulses per 360°. The output $Z$ of the encoder is utilized as a detector for compensating counting errors.

In terms of actuator, an alternating current (AC) servo motor is engaged in the proposal as the pay-off mechanism. The reason is that AC servo motor has advantages over step motor and direct current (DC) servo motor in acceleration ability, control precision, responsiveness speed and operating performance, all of which are strictly required by the proposed tension control system. The motor is actuated by Pulse-Width Modulation (PWM) signals from the main controller. Moreover, in the motor, position control is a preference for further improvement of control precision.

![Figure 2. The main control circuit.](image)

To conclude aforementioned factors, the main controller is supposed to possess at least two Capture/Compare/PWM (CCP) modules for capturing and comparing outputs from the encoder, one PWM module for generating pulse signals to actuate the pay-off servo motor with corresponding frequency and one receiver-transmitter module for communication. By taking integration level and cost saving into consideration, dsPIC33EV256GM102 is selected as the main controller. It has maximum 70MHz central processing unit (CPU) frequency which can sufficiently ensure the instantaneity and precision of control. Practically, two CCP modules and a 16-bit PWM module in dsPIC33EV256GM102 are used. The square-wave with different duty cycles is generated from the PWM module to actuate the pay-off motor to operate in the speed range from 0 to 4000rpm. The main control circuit is shown in Figure 2, which is highly-integrated with very few components. Besides, a WEINVIEW touch screen (host) is introduced to communicate with the controller (slave) for online display and parameters manipulation. Especially, Siemens PPI protocol is adopted to encapsulate the controller to a Siemens PLC because of its universal applications in domestic industry. Thus, real-time
data observation can be realized so that winding consistency will be maintained between operations by chronologically tracing tension fluctuation and comparing history winding data.

The overall flow diagram of the proposed control system is displayed in Figure 3. A closed-loop is built in the system with the feedback. Due to the inseparable cooperation between the pay-off motor and rod-spring mechanism, these two parts are defined as the controlled object of the system. The complete operating procedures can be concluded as follows. Prior to the start of operation, the desired tension is primarily calibrated by using a digital force gauge to drag the enameled wires. Based on (1), once the tension reaches the desired value, the position of the rod is immediately marked and instant reading of the accumulator in the encoder is assigned as the set value $SV(t)$. Secondly, after the operation begins, controller compares $SV(t)$ with the currently measured value $MV(t)$ from the encoder. The difference, also defined as the error value $e(t)$, is processed in the controller and corresponding rev value is consequently yielded by the PID algorithm. Thereafter, the controller delivers a PWM signal with specific frequency related to the rev value to drive the actuator (AC servo motor). Since the rev change of pay-off motor results in the variation of swing angle $\theta$, the rod consequently swings towards the set position. The aforementioned procedures will be repeatedly executed until the error $e(t)$ stabilizes at zero. It is noteworthy that only digital signals are introduced and processed throughout the flow. Although the proposed system is fully digital, initially, it still can be theoretically analyzed using traditional method for analog system as illustrated in the following section.

![Flow diagram of the tension control system.](image)

### 3. Mathematical model and PID coefficients determination

#### 3.1. Mathematical model

In order to determine the transfer function, the control system is under test for obtaining its angular displacement step response. In particular, after the rod is stabilized at a position where tension along the wires is maintained, the pay-off motor is driven by the controller to operate clockwise at 3000rpm for 1ms. Based on the result shown in Figure 4, it is positive to classify the system being a first-order inertial system with pure time delay since the plot largely fits in the typical curve displayed in [27]. Therefore, the transfer function of the controlled object can be modeled as

$$G(s) = \frac{V_o(s)}{V_i(s)} = \frac{K_A}{T_c s + 1} e^{-\tau s} \quad (3)$$

Where $V_i(s)$ is the input angular displacement, $V_o(s)$ is the output angular displacement, $K_A$ is the amplification coefficient, $T_c$ is the time constant and $\tau$ is the delay time. $K_A$ can be firstly determined as $K_A = 0.41$ since it is the ratio between the input step change from the pay-off motor ($\Delta m$) and the output change from the controlled object ($\Delta \theta$). In addition, $\tau = 7.5 ms$ and $T_c = 9.5 ms$ can be determined from diagramming the step response plot as shown in Figure 4. With all parameters determined, the transfer function of the controlled object can be ultimately ascertained as
The angular displacement step response of (4) is simulated in MATLAB and compared with measured result in Figure 5 in the premise of the equal input. As the graph demonstrates, the tendencies of two response plots are largely identical and both of them converge on the same value. These results positively verify the assumption that first-order inertial system with pure time delay is a suitable model for the proposed system.

\[
G(s) = \frac{0.4}{0.0095s + 1} e^{0.0075s}
\]  

(4)

3.2. PID coefficients determination

As demonstrated in the introduction, PID algorithm is employed in the proposed control system. From the mathematical model determined in the last section, the controlled system is confirmed as first-order so that the Ziegler and Nichols method is appropriately to be applied to theoretically determine PID coefficients [27-28] with the equation set

\[
\begin{align*}
K_p & = \frac{6T_c}{5L} \\
K_i & = \frac{3T_c}{5L^2} \\
K_D & = \frac{3T_c}{5}
\end{align*}
\]

(5)

Where \(L\) is delay time, \(T_c\) is time constant, \(K_p\), \(K_i\) and \(K_D\) are coefficients for proportional, integral and derivative controls respectively. Since \(L = \tau = 7.5ms\) and \(T = 9.5ms\) have been already determined above, theoretical PID coefficients can thus be derived as \(K_p = 1.52\), \(K_i = 101.33\) and \(K_D = 0.0057\) after simple substitutions. The set of PID coefficients is then applied in the simulation carried by SIMULINK. The simulation model is constructed as shown in Figure 6 and the plot of small step response with PID control is plotted in Figure 7.
Figure 6. SIMULINK model of the system with PID control.

It can be confirmed that the theoretically-tuned PID coefficient set does work for the control of the system, because the system traces the input precisely and quickly stabilizes at the set angle as graphed in Figure 7.

Figure 7. Angular displacement step response with theoretically-tuned PID coefficient set.

Figure 8. Assembled prototype of tension controller and take-up mechanism.

Subsequently, with the intentions of eliminating ambient analog noise and improving operating speed, pure digital discrete PID algorithm is recommended. The PID algorithm with four-point central differential method [29] for the control system can be expressed as

\[
v_0 = K_p \left\{ e(k) + \frac{T_S}{T_I} \sum_{n=0}^{k} e(k) + \frac{T_D}{6T_S} [e(k) + 3e(k-1) - 3e(k-2) - e(k-3)] \right\}
\]  

(6)

Where \(T_I\) is the integral time constant, \(T_D\) is the differential time constant, \(T_S\) is the sample time of the system. In the proposed fully-digital system, it is noticeable that the controlled variable \(e(t)\) in unit of angle has been replaced by \(e(k)\) in unit of pulse of encoder in experiments. Namely, PID coefficient set tuned in the simulation has to be further adjusted for experiments. As a result, after taking the transformation from angle to pulse and the discrete sample time \(T_S = 340 \mu s\) into account, a new set of PID coefficients is correspondingly derived to be \(K_p = 8.65\), \(T_I = 0.015s\) and \(T_D = 0.00375s\). This set is defined as the theoretically-tuned PID set and its feasibility is experimented in several aspects in Section 4.
4. Experiment

Performance of the proposed system in both transient process and steady stage is experimented. Specifically, ultra-fine enameled wires with diameter of 0.08mm are used for winding on the prototypical tension controller as shown in Figure 8.

A round take-up wire wheel is attached to the shaft of take-up motor as a simple winding mechanism. The relative rest between the wires and pay-off wheel is effectively guaranteed by inserting a rubber ring with high friction coefficient into the V-shaped groove of the pay-off wheel to provide extra friction. During winding process, the tension fluctuation on the wires will directly influence the winding quality. As the one-to-one correspondence previously mentioned in Section 2.1, although the rod-spring mechanism can convert the tension $T$ to the swing angle $\theta$ for more straightforward observation, it is preferable to directly evaluate the control performance in terms of the fluctuating level of tension $T$. Thus, under the circumstances that the pay-off motor is out-of-operation and enameled wires have no relative slippage with the pay-off wheel, a digital force gauge is used to parallel drag the enameled wires from guiding wheel B5 as shown in Figure 1. Consequently, the tension value is calibrated and plotted in Figure 9, when the swing angle $\theta$ counter-clockwise increases within the typical operation range from 30° to 60°. Moreover, because the scatter plot of tension $T$ in Figure 9 verges on being linear, thus a one-degree polynomial fitting curve is established in MATLAB and also depicted in the same figure. Therefore, the angle value $\theta$ obtained during winding process can be mathematically converted to tension value in the controller based on the fitting curve equation in the operation range, so that performance can be evaluated in terms of the degree of tension fluctuation. In all following experiments, the set value $S^V(t)$ of tension $T$ is set as 31.6g, which corresponds to the swing angle $\theta$ being 44°.

![Figure 9. Fitting curve presenting the correspondence between swing angle $\theta$ and tension $T$ from MATLAB.](image)

![Figure 10. Comparison of transient performance in 3s between systems with theoretically-tuned and experimentally-adjusted PID sets.](image)

4.1. Transient response

When the take-up motor starts operating from the static state, the pay-off motor must simultaneously change its speed to trace up the take-up line velocity with the purpose of regulating the tension on the wires. The time that system responds to the instantaneous change is a crucial performance index in transient process. In the experiment, the take-up motor is programmed to drive the take-up wheel to accelerate from 0rpm to 3000rpm. The original diameter of the frame is 66.3mm and the error value $e(k)$ is sampled every 10ms. Primarily, the theoretically-tuned PID set is applied. As presented in Figure 10, a maximum 35.4g tension is captured at 0.11s, which is 3.8g beyond the set tension value 31.6g. During the whole start-up period of the transient process from 0 to 0.3s, it is obvious that...
tension on the wires fluctuates significantly. At 1.02s, the rod reaches the set angle for the first time and then enters the steady stage. However, the range of tension fluctuation is as large as 0.90g around the set value throughout the steady stage. Therefore, with the aim of reducing overshoot and shortening transient duration, $K_p$ is supposed to be increased theoretically. Simultaneously, as compensation, it is necessary to correspondingly increase $T_I$ and decrease $T_D$ to alleviate fluctuation in steady stage. After fine tuning through considerable experiments, the experimentally-adjusted set of PID coefficients is determined as $K_p = 11.5$, $T_I = 0.021s$, and $T_D = 0.00266s$. Under the same initial condition, performance comparison in transient process between systems with different PID sets is shown in Figure 10. The result interprets that the maximum tension overshoot is decreased 26.3% to 2.8g at 0.1s after the PID coefficients are adjusted. It is also clear from viewing the graph that a gentler start-up of transient process with slighter fluctuation is achieved. Besides, after the system enters the steady stage from 1.02s, the steady error range is narrowed 33.3% with a minor fluctuating range of 0.60g around the set tension value. To sum up, it is positive that the experimentally-adjusted PID makes more significant contributions than the theoretically-tuned PID set on stabilizing the system throughout the whole operation.

4.2. Steady response

Generally, in the flat-winding situation, after the winding machine passes through the short transient process at the start, it stably operates for the rest time with small fluctuations until the product is completely wound. Therefore, the importance of looking into the steady stage for a relatively long time should be seriously considered so as to evaluate the steady performance of the system. Specifically, the take-up motor uniformly accelerates from 0rpm to 3000rpm and then maintains the speed for 30s in the experiment. The result is presented in Figure 11, where the tension $T$ is sampled every 100ms. As the plot interprets, after the tension is eventually stabilized at 1.1s, the system enters steady stage with minor fluctuation around the set tension value. The maximum tension is captured as 32.0g and minimum tension is 31.2g, namely the maximum error ratio is only 1.27% throughout the steady stage in the experiment. Furthermore, during the whole experiment, the diameter of the take-up frame increases from 74.8mm to 77.7mm with an average winding line velocity at 11.98m/s.

![Figure 11. Steady performance of the system with rev at 3000rpm for 30s.](image)

![Figure 12. Transient performance of the system with interference.](image)

4.3. Performance with interference

Considering the potential application in complex industrial occasions, it is necessary to conduct tests on anti-interference capability of the system. In the experiment, after the tension stabilizes and system completely enters the steady stage at 2500rpm, a 250rpm speed increment is introduced to the take-up motor to abruptly break up the equilibrium at a random time. In Figure 12, responses of the system to
two introduced interference are exhibited. At 5.25s, the take-up motor accelerates to 2750rpm, thereafter it takes the system 0.15s to re-stabilize the tension. The maximum tension overshoot is captured being 0.7g at Q1. The other interference takes place at 10.15s, when the take-up motor takes another velocity increment to 3000rpm. The system therewith spends 0.2s to regulate the tension and another 0.8g maximum tension overshoot at Q2 is captured. During the whole process, the diameter of take-up wheel increases from 73.5mm to 75.2mm with an average winding line velocity of 11.81m/s. Thus, conclusions can be drawn from aforementioned results that the proposed system is able to handle the abrupt interference with immediate adaption and small overshoot.

5. Conclusion
A fully-digital tension control system with PID algorithm is proposed in this paper. According to the analysis on its step response characteristics, the system is classified as being first-order. Thus, the set of PID coefficients is theoretically tuned by using the Ziegler and Nichols method and its controlling capability is validated in simulations. Moreover, the PID coefficients are further adjusted in experiments and discrete PID control algorithm is practically engaged in consideration of the instantaneous responsiveness taking priority in the proposed system. A tension controller as the prototype is also fabricated and assembled for evaluating both transient and steady performance. From the results, it can be concluded that the control system passes through the transient period quickly with only 2.8g tension overshoot at maximum. Meanwhile, when the tension controller operates stably, the winding speed can be as high as 11.98m/s and the tension fluctuation is restrained within 1.27%. Meanwhile, it has good anti-interference capacity when interference is randomly introduced, which is an important merit in complex industrial applications.

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