High-Precision Control System Design for Servo Actuators Based on Optimized Command Shaper

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***Abstract*—Servo actuator provided with harmonic drive gearing has been used for positioning loads in many applications. In industrial applications, high-precision motion systems not only require high speed specifications but also high precision. However, these mechanisms involve non-linear disturbances and transmission errors which degrade their performance. Proportional controller cascaded with proportional plus integral controller (P-PI) is used as an efficient cascaded control scheme in many applications but it fails to provide good performance when applied to high precision motion systems equipped with harmonic drive gearing. Accordingly, reshaping the velocity command of the inner control loop has been proposed in this study using an optimal gain-delay-shaper as a new methodology. Parameters of the introduced shaper have been obtained using genetic algorithm (GA) optimization. The proposed scheme has been verified using numerical simulation. In order to assess the effectiveness of the proposed techniques, the results have been compared with those of other control system such as conventional P-PI controller.**

Keywords*—*Harmonic drive gear, transmission error, cascaded control, command shaping, optimal state feedback; genetic algorithm.

I. INTRODUCTION

Nowadays, design of high-precision servo mechanisms is an essential requirement in motion control mechanisms, Servo actuators provided with harmonic drive (HD) gearing are used in many applications even industrial, aerospace, robotics or medical ones. For those actuators, due to permitting quite high reduction ratio in a very compact and lightweight package, harmonic drive (HD) gearing are used to replace the classical gear. HD gears utilize a unique operating principle as its operation is based upon the elastic mechanics of metals [1]. The HD consists of three basic parts, as shown in Fig.1. The design of those parts combined with its operating principle permit quite high reduction ratio in a very compact and lightweight package, as will be described in the following paragraph [2]. The 1st part is wave generator (WG), which is comprised of a thin raced ball bearing fitted onto an elliptical hub. The WG is used to receive the transmitted mechanical motion generated by the servo actuator then convert the generated torque efficiently to the second part which is called the flexspline (FS). The FS is a thin cylindrical cup made from alloy steel with external teeth on the open end of the cup. The *FS* is radially compliant where it is torsionally stiff. When the *WG* is inserted into the *FS*, the *FS* takes the *WG* shape i.e. an elliptical shape. To convert the torque to the load it is connected to the FS. The 3rd part of the HD is a circular spline (CS), which is a rigid ring with internal teeth. When the HD parts are assembled, the teeth of the FS are engaged across the major axis of the WG. Moreover, the FS has two teeth less than the CS which is fixed to a gear housing, accordingly, when the wave generator moves clockwise the flexspline moves in anti-clockwise direction as for a one cycle of WG rotation, the FS go ahead with two teeth anti-clockwise [1-5]. The operating principle of HD gearing results in high performance features such as zero backlash, high torque, compact size, excellent positional accuracy and repeatability. The schematic diagram of the under study servo actuator, FHA-8C-50-E200-C, configuration is illustrated in Fig. 2. However, those servo actuators include some nonlinear characteristics e.g. nonlinear friction forces, transmission error, and transmission compliance.

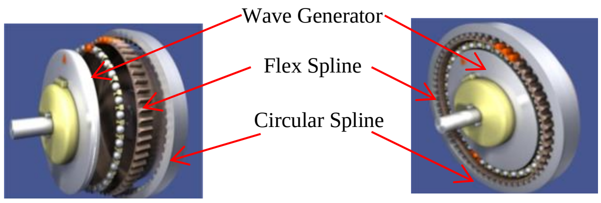

Fig. 1. Harmonic Drive Gearing assembly and its component.

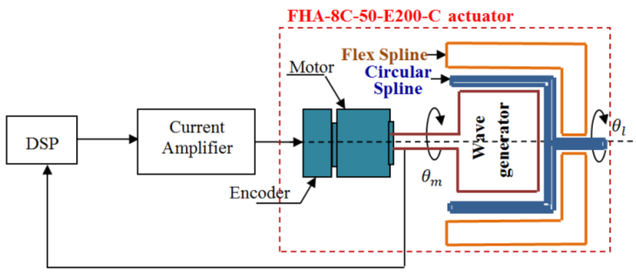

Fig. 2. Schematic Diagram of the Servo Actuator Configuration.

Many studies in the literature addressed those nonlinearities. It is stated from these studies that, in order to improve the positioning precision, the effect of those nonlinearities must be compensated. In [6], a modified model predictive control was designed, the transient characteristics have been improved whereas there is a steady state error of the load positioning still exists. Authors of [7] applied a robust torque control and using a model-based friction compensation

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algorithm, the positioning performance was improved as the nonlinear friction characteristics were compensated. Where in [8], the effect of the angular transmission error was reduced by compensating motor shaft synchronous components which occur in synchrony with the motor position.

From the literature it can be concluded that, multi-function systems are difficult to be controlled using single loop controllers. However, controlling those systems utilizing multi-nested loops is considered an effective methodology to fulfill the demanded performance [2, 6]. Figure 3 illustrates a cascade motion control arrangement including two loops; the inner loop (velocity loop) and outer loop (position loop), output of the position controller which has been designed as proportional controller (P) drives the set point of velocity controller which has been designed as a proportional plus integral controller (PI). Accordingly, the position variable is controlled by updating the command of the inner control loop as the velocity impacts the load position through the process [1].

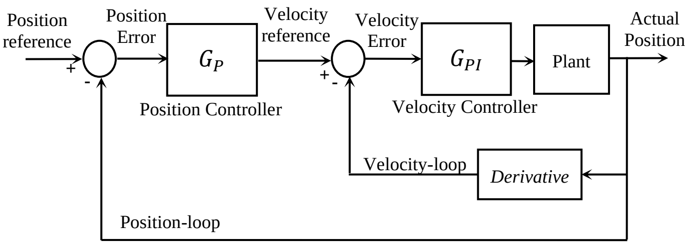

Fig. 3. Block diagram of a cascaded control system with two-loops.

Moreover, command shaper is used as a methodology for generating a particular trajectory for a specific control system based on state feedback (SFB) i.e., a desired reference command is updated in order to improve the system performance. The reference command is improved so as to minimize the disturbance effect which cannot be compensated by implementing conventional controllers [9-10]. Besides, command shapers are considered effective solution for suppressing vibration of equipments with flexible structure [11-13]. However, information about some characteristics of the controlled system e.g. resonance frequencies and damping ratios are essential for shaper design, in addition, sequence of impulses should be implemented to produce the command that is able to provide the desired performance of the system [14]. As presented by [14], a modified version of command shaping control scheme which depended on genetic algorithms (GA) for controlling the vibration generated from vehicles that associated with flexible structure. Also authors of [15] introduced these kinds of shapers for providing minimum vibration and vibration derivative relaying on implementing the shaper with uniformly distributed delays as an alternative for the classical lumped delays. The gains and delays of the designed shaper were provided using spectral technique. The presented shaper offer enhanced features in comparison with the conventional shapers.

In this article, effectiveness of utilizing optimal designed velocity command shaper for compensating the nonlinear dynamics of an *AC* servo actuator has been illustrated. Section II presents the details of a servo actuator supplied with strain wave gearing. The applied cascade control and the proposed command shaper have been formulated in Sections III. In Section IV, numerical simulated results have been introduced to evaluate effectiveness of the proposed control scheme on the system performance. Finally, conclusions are given in section V.

II. Servo Mechanism Modeling

The under consideration servo actuator (positioning system) can be described as a load driven by an *AC* servo motor through strain wave gearing (harmonic drive gear or HD) which is used to transmit the angular motion and torque of the *AC* motor to the load. A sketch of the servo actuator configuration is displayed in Fig. 4. As illustrated in the figure and due to *WG* is attached to the motor shaft, *WG* angular position, θ_{wg} , acts as the rotation of motor shaft θ_m , in addition, as the load is attached to *FS*, the load angular displacementθ_l acts as the angular displacement of the *FS*, θ_{fs}. Because of fixed *CS*, its angular displacement, θ_{cs}, can be set to zero. Moreover, relations between displacement, velocity, and torque can be formulated as follows [2, 16, and 17].

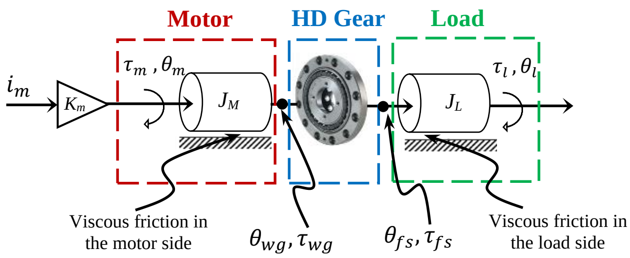

Fig. 4. Schematic of the servo mechanism.

When taking the ideal model of HD gearing into account, the HD can be considered as a classical gear with a reduction gear ratio N and the angular displacement and angular velocity of the HD components can be expressed by (1) & (2), respectively, as:

$$\theta_{wg} = (N+1)\theta_{cs} - N\theta_{fs} \quad (1)$$

$$\omega_{wg} = (N+1)\omega_{cs} - N\omega_{fs} \quad (2)$$

where θ_{wg} is the angular displacement of the WG, θ_{cs} is the angular displacement of the CS, and θ_{fs} is the angular displacement of the FS and ω_{wg}, ω_{fs}, and ω_{cs} are the angular velocity of the WG, the FS, and the CS, respectively. HD gear's torque transmission can be calculated as follows:

$$\tau_{wg} = -\frac{1}{N}\tau_{fs} \quad (3)$$

$$\tau_{cs} = \tau_{fs} + \tau_{wg} \quad (4)$$

where τ_{wg}, τ_{fs}, and τ_{cs} are the transmission torque across the WG, the FS, and the CS, respectively. By balancing the forces of inertia at the motor and load sides, the dynamics of positioning mechanism can be formulated as follows:

$$\ddot{\theta}_{wg} = \frac{1}{J_m}\left(\tau_m - D_m\dot{\theta}_{wg} - \tau_{f\_wg} - \tau_{wg}\right) \tag{5}$$

$$\ddot{\theta}_{fs} = \frac{1}{J_l}\left(-D_l\dot{\theta}_{fs} - \tau_{f-fs} - \tau_{fs}\right) \tag{6}$$

where τ_m is motor torque, τ_{f_wg} is a frictional torque at the bearings of the WG, τ_{f_fs} is a frictional torque at the load. A block diagram of the nonlinear servo system is shown in Fig. 5, where the effect of nonlinear friction, motor shaft synchronous component, and angular transmission error are considered.

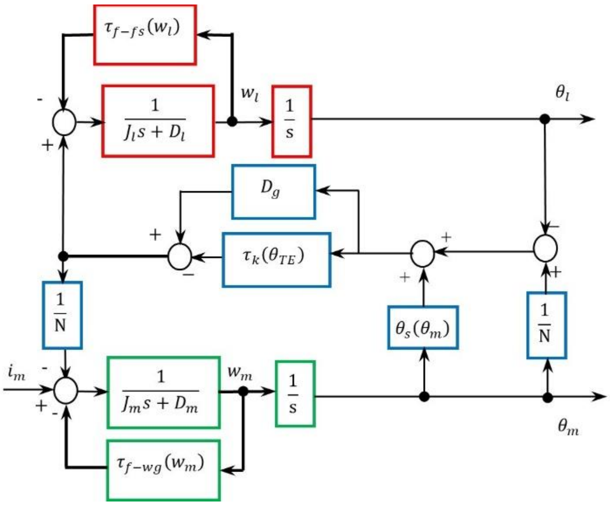

Fig. 5. Block diagram of the non-linear model of HD servo actuator.

At low velocities, the nonlinear friction forces τ_{f_wg} and τ_{f-fs} can be represented as functions of motor and load velocities as follows [1-2]:

$$\tau_{f\_wg}(\omega_m) = q_m \cdot tanh(p_m \cdot \omega_m) \tag{7}$$

$$\tau_{f\_fs}(\omega_l) = q_l \cdot tanh(p_l \, . \, \omega_l) \tag{8}$$

where ω_m, and ω_l are angular velocities of the motor and load, respectively, and q_m, p_m, q_l, and p_l are constants. The motor toque can be obtained as $\tau_m = K_m . i_m$, where $K_m$is the motor torque constant and i_m is the motor current. Nonlinear dynamics of the HD gear can be represented as the dynamics of a series damper and nonlinear spring, as expressed by (9), where the stiffness torque τ_k is a function of the transmission error, θ_{TE} , which can be given by (10) [2-3, & 6].

$$\tau_{fs} = D_g\dot{\theta}_{TE} + \tau_k(\theta_{TE}) \tag{9}$$

$$\theta_{TE} = \left(\frac{\theta_m}{N} - \theta_l\right) + (\theta_s + \theta_h) \tag{10}$$

where D_g is the HD gear damping coefficient, θ_s is the synchronous component, and θ_h is the nonlinear elastic component. Refereeing to [1 & 6], nonlinear spring model can be formulated as:

$$\tau_k(\theta_{TE}) = K_1 \cdot \theta_{TE} + K_2 \cdot \theta_{TE}^2 + K_3 \cdot \theta_{TE}^3 \tag{11}$$

where θ_s is caused by kinematic error in FS and CS teeth, and errors in gearing and load shaft assembly, and can be given by (12) as a periodical pulsation of θ_m [1 & 4]:

$$\theta_s(\theta_m) = \sum_{i=1}^{n} A_i \cdot cos\,(i \cdot \theta_m + \varphi_m) \tag{12}$$

where i is the harmonic order for motor angular displacement, A_i is the amplitude of the harmonics, and $\varphi_m$is the phase of the harmonics.

The nonlinear elastic deformation in micro-displacement region of motion causes the nonlinear elastic component of the transmission error, which can be expressed as a hysteresis model [1]. Accordingly, θ_h can be expressed as a function of the displacement after velocity reversal, δ, as follows:

$$\theta_h(\delta) = \begin{cases} sgn(\omega_m).\,(2\theta_0 g(\xi) - |\theta_{h0}|) \\ \qquad : \delta \le \theta_r and |\theta_{h0}| \le \theta_0 \\ sgn(\omega_m).\,\theta_0 \\ \qquad : \delta > \theta_r or |\theta_{h0}| > \theta_0 \end{cases} \tag{13}$$

$$g(\xi) = \begin{cases} \frac{1}{2-\varepsilon}.\,(\xi^{\varepsilon-1} - (\varepsilon-1)\xi) & : \varepsilon \ne 2 \\ \xi(1 - Ln\,\xi) & : \varepsilon = 2 \end{cases} \tag{14}$$

$$\delta = |\theta_m - \delta_0|, \; \xi = \delta/\theta_r \tag{15}$$

where $sgn(\omega_m)$ is a sign function to determine the velocity direction, $2\,\theta_o$ is the level fluctuation after the non-stable region, θ_{h0} is the transmission error at the velocity reversal, δ_0 is the motor displacement at the velocity reversal, and ε is the hysteresis width.

By neglecting HD nonlinearities, the relationship between the motor angular displacement and the input motor current can be defined by (16), where parameters of the model are defined in [1]:

$$\frac{\theta_m}{i_m} = \frac{K_t \cdot (J_l s^2 + (D_l + D_g)s + K_g)}{s \cdot (b_3 s^3 + b_2 s^2 + b_1 s + b_0)} \tag{16}$$

where, $b_3 = J_m J_l$,

$$b_2 = J_m(D_l + D_g) + J_l\left(D_m + \frac{D_g}{N^2}\right),$$

$$b_1 = D_m(D_l + D_g) + \frac{D_l . D_g}{N^2} + K_g\left(J_m + \frac{J_l}{N^2}\right),$$

$$b_0 = K_g\left(D_m + \frac{D_l}{N^2}\right).$$

J_m and J_l are the motor inertia and the total load inertia, respectively, in kg/m^2, D_m and D_l are the viscous damping coefficient at the motor side and the load side, respectively, in Nm/(rad/sec) and K_g is the gear spring constant in Nm/ rad.

III. Control Mechanism Design

Velocity command shaping technique has been proposed in this study to achieve the objectives of the control system, which can be summarized as follows: for high-speed and high-precision positioning, the angular displacement of the load should have an accuracy of ±30 load arc-sec ($1^{\circ} = 3600$ arc-sec) from the set point, θ_{ref}, within a settling time less than 0.3 sec. Moreover, the applied current to the motor, i_m, should be kept within the range, $|i_m| \leq 0.64$ A.

In order to achieve a precise positioning performance of the servo actuator, a control technique based on optimal state-space feedback was proposed in [1] that scheme relying on the generating an auxiliary control signal to be added to the effort of the outer control loop (positioning-loop) to modify the velocity command of the inner loop (velocity-loop). The auxiliary control signal is produced using a state-feedback-based linear quadratic regulator (LQR) and Kalman filter [1]. For obtaining a robust tracking performance, and improving the behavior of the under study servo actuator in the presence of its elastic dynamics and transmission error, a feedback control system based on command shaper is introduced. The proposed control system is shown in Fig. 6, which includes an optimal state feedback control signal in addition to the conventional P-PI controller.

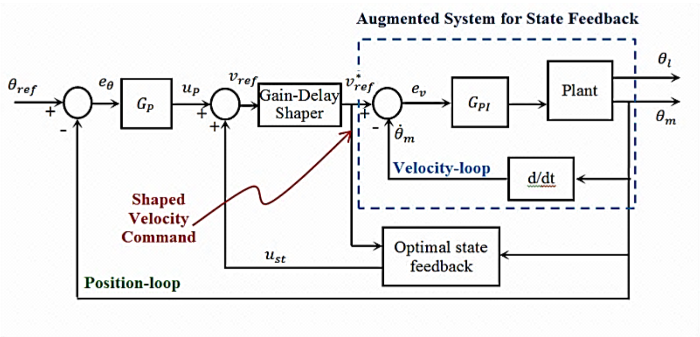

Fig. 6. Block diagram of the Proposed Control System

A Gain-delay shaper has been used to reshape the velocity command (the reference signal of the inner loop) and generating a shaped command v^*_{ref} which has the ability to enhance the transient characteristics of the load displacement and provide a robust tracking performance. The implemented proposed shaper is called gain-delay shaper as its design is based on utilizing the gain and delay elements to reshape the velocity command v_{ref}. The unshaped reference signal is passed through multiple delay units τ_i then multiplied by gain factors k_i. The effectiveness of the suggested technique depends on a suitable selection of the number of delayed signals and their corresponding delay and gain values [18].

A. Structure of the Gain-Delay Shaper

The structure of the n numbers gain-delay shaper is illustrated in Fig. 7. The velocity command v^*_{ref} can be obtained in time domain by (17):

$$v^*_{ref}(t) = \sum_{i=1}^{n} k_i v_{ref}(t-\tau_i) \quad (17)$$

By converting (17) to *S*-domain, the shaped velocity reference can be expressed by (18):

$$V^*_{ref}(s) = \sum_{i=1}^{n} k_i V_{ref}(s) \cdot e^{-s\tau_i} \quad (18)$$

where, $v_{ref}(t) = u_p(t) + u_{st}(t)$

The delayed units τ_n are represented as N_{dn} multiples of the sampling time T_s that is $\tau_n = N_{dn}T_s$, where N_{dn} is an integer number. The selection of the gains and delay units should be done under certain constraints i.e. the selected values of the gains must sum to one in order that with the shaped command the system achieves the same level of its response as well as with the unshaped one i.e. the shaped command reaches the same final value as the unshaped command [19-20]. This constraint can be expressed as:

$$\sum_{i=1}^{n} K_i = 1 \quad (19)$$

Furthermore, to minimize the delay in system's response, the first delay unit must be set to zero, i.e.,$\tau_1 = 0$ [19]. The remaining delay units $\tau_2, \tau_3, \ldots, \tau_n$ and all gain values $K_1, K_2, \ldots, K_n$ are selected via genetic algorithm (GA) optimization technique. The GA is used to optimize the values of the gains and delay units so as to minimize the elastic dynamics of the system and transmission error effect.

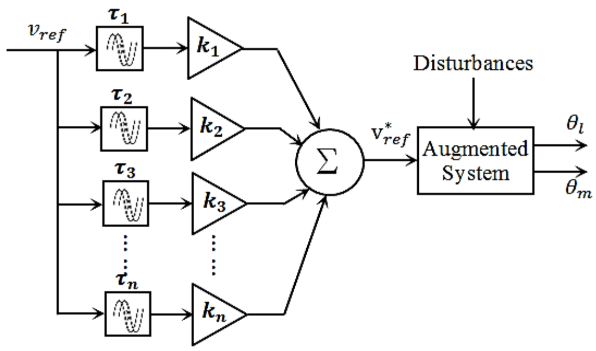

Fig. 7. Block diagram of the gain-delay shaper.

B. Genetic Algorithms

Generally, GAs fall under the multiple directions searching methods to solve for problem variables, as GAs inherently include three evolutionary operations: reproduction, crossover, and mutation, which indicate that GAs have high probabilities to escape from a local minimum. The steps applied for real coded GA optimization to generate the optimized solution can be summarized as follows: The GA performance is mainly affected by crossover and mutation operators. The crossover operation is to make the generated chromosomes cross mutually and randomly, and mutation is to provide a possible random mutation on selected chromosomes. The reproduction of new generation continually happen during a number of generations and stopped when some conditions are satisfied such as reaching the maximum number of generations, the value of the fitness function of the best solution in the current population is less than or equal a predetermined fitness value, or the change in the best fitness function value over the generations is less than a predetermined tolerance value [21].

Structure of the GA-based command shaper is shown in Fig. 8. The objective function of the GA is to choose optimal values shaper gains and delay units by minimizing the fitness function *f(x)* which is formulated by (20):

$$f(x) = \alpha \sum_{i=1}^{k} e(i)^2 + \beta\, T_{set} \qquad (20)$$

where $e(i)$ is the positioning error at the sample i, which can be expressed as in (21), i is changed from 1 to the total number of samples k. T_{set} is the positioning settling time which is calculated according to 2% criteria.

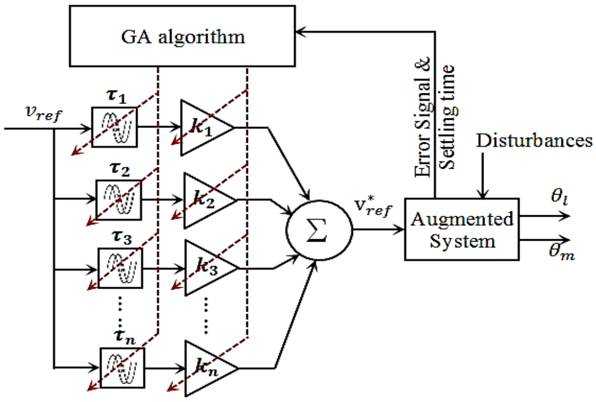

Fig. 8. GA-based command shaping scheme using gain-delay units

$$e(i) = \theta_{ref}(i) - \theta_m(i) \qquad (21)$$

In (20) α and β are weighting factors, when one of these factros increases, the fitness function will be more sensitive to the changes in its term. The weighting factors α and β have been chosen as 0.1 and 1, respectively, in order to provide equally fitness function sensitivity against the positioning error e and the settling time of the load response.

IV. Results and Discussion

To illustrate that the requirements were not met when utilizing the P-PI scheme, and to verify the effectiveness of the suggested control methodology, simulations have been conducted using MATLAB/Simulink software package. Simulations were performed using the nonlinear model of the servo actuator shown in Fig. 5. The P-PI controller has been designed to provide a position control loop with proportional (P) controller in addition to a velocity control loop with a proportional plus integral (PI) controller. As shown in Fig. 9, performance of the servo system when applying the P-PI controller is not acceptable, indicating that system nonlinearities are not adequately compensated. Parameters of the P-PI controller have been calculated with consideration of the stability of the positioning control system as follows: the proportional gain of the position controller $k_{\mathrm{pp}} = 312$ rev/sec, the velocity proportional gain $k_{vp} = 0.0133$ A/(rad/sec), and the velocity integral gain $K_{vi} = 0.1575$ A/rad [3] & [6].

Considering the proposed gain-delay shaper design, the number of the gains have been chosen to be $n = 4$, which is sufficient to reshape the velocity command and producing an adequate load response. Thus, the number of the delay units which have to be determined is three. The GA optimization program has the following options: population size of 20 chromosomes, where each chromosome is considered as a vector containing a solution of parameters of gain-delay shaper parameters e.g. $\{K_0, K_1, K_2, \tau_1, \tau_2, \tau_3\}$ as $\tau_0 = 0$ and $K_3 = 1 - \sum_{i=0}^{2} K_i$. Those chromosomes are generated by the crossover and the mutation processes. The best two chromosomes are directly reproduced to the new generation. Where the crossover factor is 0.8 and the mutation factor is randomly added to each entry of the parent vector, depending on a Gaussian distribution. After around 100 iterations, the fitness function was converged to its best value, and the final tuning of gain-delay shaper was obtained. The final values of the shaper gains and delay units have been finally synthesized as in Table 1. Figure 9 shows the positioning reference trajectory (indicated as black solid lines), and the load displacement with different control systems. The positioning trajectory is indicated related to the load side i.e. θ_{rev}/N. It has noticed from the figure that, with the gain-delay shaper (indicated as solid pink lines), the reference and the load positions are aligned, and good tracking performance has been achieved comparing to those achieved with other control systems i.e. the conventional P-PI controller, and the P-PI control system with auxiliary SFB-based signal which are indicated as dashed red lines and dotted blue lines, respectively. The effectiveness of the control mechanism with shaped velocity command is pointed out by getting a lower peak overshoot (PO), less settling time (T_{set}), and zero steady state error (e_{ss}) i.e., the shaped velocity command accomplish the required specifications of load performance. Figure 10 shows the velocity trajectory (which is obtained by differentiating the positioning reference, θ_{ref} and indicated as black solid lines), and velocity commands, v^*_{ref}(command to the velocity loop) which are generated with different applied control schemes. The velocity commands that are generated using the conventional P-PI controller, SFB-based control system, and SFB-based control system with velocity command shaping are indicated as dashed red lines, dotted blue lines, and solid pink lines, respectively. Figure 11 illustrates the actuator efforts in amperes. The figure shows that the input current still within the rated range with all the applied control system. The signals generated using the gain-delay shaper are indicated as solid pink lines, where those obtained using the optimal SFB-based control system and P-PI controller are indicated as dot blue lines and dashed red lines, respectively. The resultant transmission error is shown in Fig. 12. The signals generated using the gain-delay shaper are indicated as solid pink lines, where those obtained using the optimal SFB-based control system and classical P-PI controller are indicated as dot blue lines and dashed red lines, respectively. Based on Fig. 9, it is clear that, the effect of the non-linear transmission errors that are involved in the mechanism, shown in the Fig. 12, has been compensated successfully when applying the proposed controllers however, the conventional controller failed to provide the desired performance characteristics i.e. the mechanism combined with the optimal shaper offers faster response in comparison with those have conventional controller or proposed without velocity command shaping.

V. Conclusion

In this paper, a new design of a controller based on a command shaper has been investigated for a precise positioning servo mechanism provided with a harmonic drive gear. The velocity command of the inner control loop has been reshaped using optimal gain-delay-shaper. Whereas a multi-objective function is utilized to find the optimal solution, but the algorithm is sensitive to the weighting factors and needs complicated computations. Numerical simulations have been carried out in order to validate the effectiveness of the proposed schemes. The proposed controller has been tested for the servo actuator nonlinearities. A comparison between performances of the proposed controller and conventional P-PI and state-feedback-based P-PI controllers has been done. The obtained results clarify that the closed loop system with the proposed optimal command shaper has achieved better performance comparing to the latter two controllers.

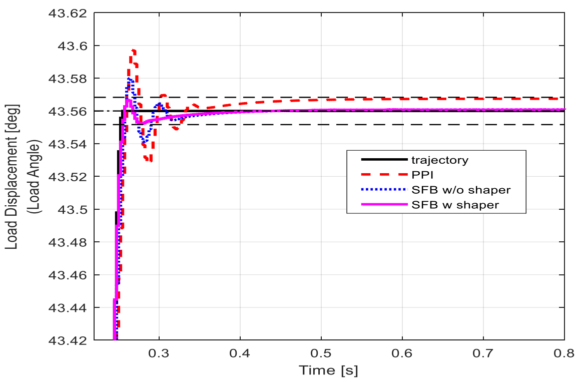

Fig. 9. Load Displacement

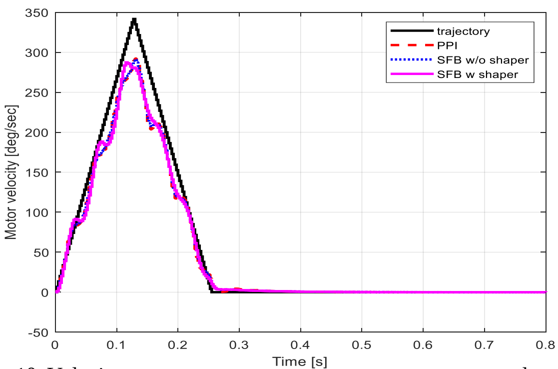

Fig. 10. Velocity command

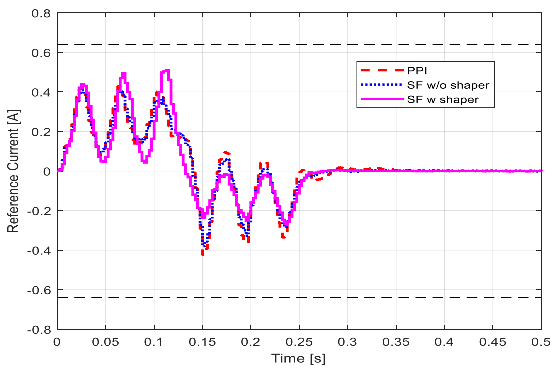

Fig. 11. Control Input to the servo Actuator

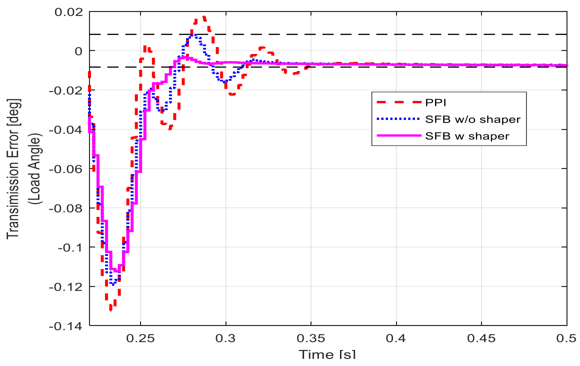

Fig. 12. Transmission Error

Table 1. The final gain and delay values of the Shaper.

Delay [Samples #]	**Gain**
$N_{d0} = 0$	$K_0 = 0.646$
$N_{d1} = 10$	$K_1 = 0.120$
$N_{d2} = 2$	$K_2 = 0.194$
$N_{d3} = 1$	$K_3 = 0.049$

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