Identification of the Mechanical Parameters for Servo Drive

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***Abstract*—This paper describes a novel identification technique of the moment of inertia and the viscous friction coefficient for a servo drive system. It is developed based on the basic dynamic equation of a mechanical system, and it uses the torque reference of a speed controller and the actual rotating speed of machine for the identification. Regardless of the simplicity of the techniques it is robust to external disturbances such as measurement noise, coulomb friction, mechanical resonance of the drive system. The identified inertia and friction can be used for auto-tuning of the gains in the speed controller and the disturbance observer. The effectiveness of the proposed method is proved by the computer simulation and experimental results.**

Keywords— inertia;viscous friction; identification; auto-tuning;

I. INTRODUCTION

The performance of a servo system is highly influenced by uncertainties of unpredictable mechanical parameter variations and external load disturbances. To achieve high dynamic performance of the speed control, the gains of the speed controller should be increased as much as possible, and some kind of feed forward control technique, like a feed forwarding of the derivative of speed reference multiplied by system inertia which is the torque for acceleration, should be adopted. To design these controllers, the mechanical parameters such as the moment of inertia and the viscous friction coefficient are very essential. Furthermore, the speed observer should be needed for low speed operation with relatively low resolution encoder or resolver, and the disturbance observer may be used to reduce the influence of disturbance torque. The inertia and the viscous friction coefficient are also needed at the design stage of these observers. There have been innumerable researches about this topic, and some of them show reasonable performance. Most of them are based on the observer theory with parameter adaptation algorithm [1-4]. Those results, however, are very complex to implement, and strongly affected by adaptation gains. Moreover, if an initial value of parameter is far from the actual one, their convergence rates are somewhat disappointing. The other approaches have been tried to overcome these drawbacks [5-6]. They are based on the dynamic equation of mechanical model, and request relative simple calculation. But they need a special trajectory of speed command, and bi-directional rotation.

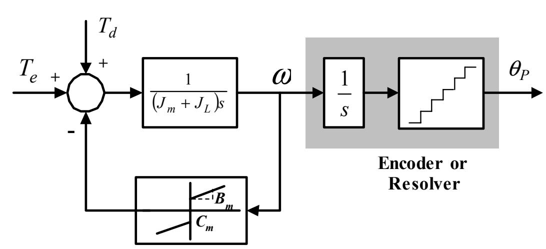
In this paper, a novel identification technique of the moment of inertia and the viscous friction coefficient for high dynamic servo drive systems is proposed. It uses a dynamic equation of a simple mechanical system, a torque reference of a speed controller, and an actual rotating speed of machine. After some calculation, the proposed method identifies the moment of inertia and the viscous friction coefficient on real time, so that the controller and the observer gains can be auto-tuned for high dynamic performance of the speed control. The technique can be applied to a drive system where only uni-directional rotation is possible. The effectiveness and feasibility of the proposed method are verified by the computer simulation and experimental results.

II. MODELONG OF MECHANICAL SYSTEM

Fig. 1 shows the model of the mechanical system, and this model can be expressed by (1).

$$T_e = (J_m + J_L)\frac{d\omega}{dt} + B_m\omega + sign(\omega)C_m - T_d \,, \qquad (1)$$

where ω is the mechanical angular speed of the rotor (rad/s) and J_m, J_L are the inertia of the motor and the load, respectively (Kg·m^2). B_m is the viscous friction coefficient



Fiure 1. Modeling of mechanical system

(N·m·s/rad), C_m is the coulomb friction (N·m), and T_e, T_d are the driving and the disturbance torque, respectively (N·m).

The encoder or the resolver, which is widely used for measuring rotational speed, is a position sensor, thus the output of the mechanical system is a quantized position θ_P . The rotational speed can be obtained by a simple differentiation of the rotor position or estimation using a speed observer.

III. Identification Strategy of Mechnical Parameters

Fig. 2 shows the typical control block diagram of servo drive system, where ω^* is speed reference, T_e^* is torque reference, $\hat{\omega}$ is estimated speed, and $\hat{T}_d$ is estimated disturbance torque. The speed estimator can be either a speed observer or a differentiator, as mentioned in the previous section, the disturbance observer is used to reduce the influence of the disturbance torque, and the speed controller consists of a simple Proportional Integral (PI) controller and a reference feed forward. The parameter estimator identifies the total equivalent inertia of the motor and the mechanical load with respect to the motor, and the viscous friction coefficient. The coulomb friction torque can be regarded as an external disturbance torque, thus only two parameters are considered for identification.

Multiplying both sides of (1) with the derivative of the actual rotational speed yields

$$T_e \frac{d\omega}{dt} = (J_m + J_L)\left(\frac{d\omega}{dt}\right)^2 + B_m \omega \frac{d\omega}{dt} + T_d' \frac{d\omega}{dt}, \tag{2}$$

where $T_d' = sign(\omega)\, C_m - T_d$.

Assuming a slowly varying disturbance torque and taking a definite integral of (2) give (3).

$$\int_{t_1}^{t_2} T_e \frac{d\omega}{dt} dt = (J_m + J_L)\int_{t_1}^{t_2}\left(\frac{d\omega}{dt}\right)^2 dt + B_m \int_{t_1}^{t_2} \omega \frac{d\omega}{dt} dt + T_d' \int_{t_1}^{t_2} \frac{d\omega}{dt} dt, \tag{3}$$

where t_1 represents the starting time of identification and t_2 is finishing time.

The second term of the right side of (3) can be expressed as (4),

$$B_m \int_{t_1}^{t_2} \omega \frac{d\omega}{dt} dt = \frac{B_m}{2}\left(\omega_{at\ t=t_2}^2 - \omega_{at\ t=t_1}^2\right). \tag{4}$$

Thus, from (3) and (4), the total moment of inertia can be described by (5) with the assumption of $T_e = T_e^*$,

$$J_T = J_m + J_L = \frac{\int_{t_1}^{t_2} T_e^* \frac{d\omega}{dt} dt}{\int_{t_1}^{t_2}\left(\frac{d\omega}{dt}\right)^2 dt} - \frac{B_m}{2}\frac{\left(\omega_{at\ t=t_2}^2 - \omega_{at\ t=t_1}^2\right)}{\int_{t_1}^{t_2}\left(\frac{d\omega}{dt}\right)^2 dt} - T_d' \frac{\left(\omega_{at\ t=t_2} - \omega_{at\ t=t_1}\right)}{\int_{t_1}^{t_2}\left(\frac{d\omega}{dt}\right)^2 dt}. \tag{5}$$

The derivative of the actual speed in (5) can be filtered by a low pass filter in order to avoid amplifying the noise from the digital quantization or measurement.

From (5), it can be noted that if the identification time is sufficiently long, and if the derivative of actual speed is not zero during the identification time, the second and the third term of (5) can be neglected because the denominator will grow very large compared to the numerator. In case that the interval of the identification time is not sufficiently long, if the speed of t_1 and t_2 is equal or close to each other, the numerator of those terms can be set to zero. Thus the moment of inertia can be simply obtained as in (6), regardless of the viscous friction and the disturbance.

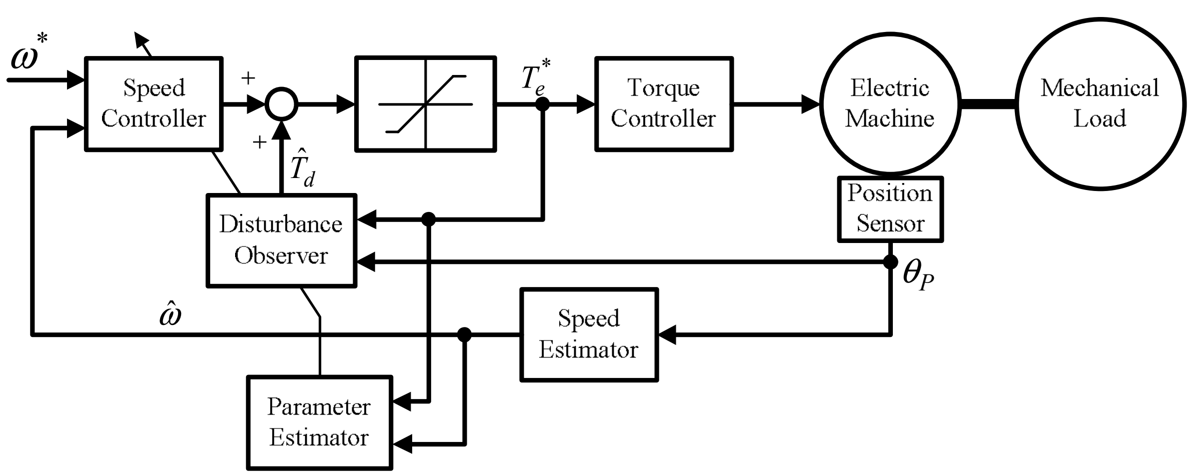


Figure 2. Block diagram of the typical speed control system

$$J_T = \frac{\int_{t_1}^{t_2} T_e^* \frac{d\omega}{dt} dt}{\int_{t_1}^{t_2} \left(\frac{d\omega}{dt}\right)^2 dt}. \quad (6)$$

Moreover, this method does not need bi-directional rotating operation, thus it can be very useful especially in the system such that only the uni-directional rotation is allowed.

Similarly, the viscous friction coefficient can be obtained. Differentiating both sides of (1), multiplying them with the derivative of the actual speed, and assuming a slowly varying disturbance torque yield (7)

$$\frac{dT_e^*}{dt}\frac{d\omega}{dt} = (J_m + J_L)\frac{d^2\omega}{dt^2}\frac{d\omega}{dt} + B_m\left(\frac{d\omega}{dt}\right)^2, \quad (7)$$

and taking a definite integral of (7) gives (8),

$$\int_{t_1}^{t_2} \frac{dT_e^*}{dt}\frac{d\omega}{dt} dt = J_T \int_{t_1}^{t_2} \frac{d^2\omega}{dt^2}\frac{d\omega}{dt} dt + B_m \int_{t_1}^{t_2} \left(\frac{d\omega}{dt}\right)^2 dt. \quad (8)$$

The second term of the right side of (8) can be expressed as (9)

$$J_T \int_{t_1}^{t_2} \frac{d^2\omega}{dt^2}\frac{d\omega}{dt} dt = \frac{J_T}{2}\left(\left(\frac{d\omega}{dt}\right)^2_{at\,t=t_2} - \left(\frac{d\omega}{dt}\right)^2_{at\,t=t_1}\right). \quad (9)$$

Thus, from (8) and (9), the viscous friction coefficient can be obtained as (10)

$$B_m = \frac{\int_{t_1}^{t_2} \frac{dT_e^*}{dt}\frac{d\omega}{dt} dt}{\int_{t_1}^{t_2} \left(\frac{d\omega}{dt}\right)^2 dt} - \frac{J_T}{2}\frac{\left(\left(\frac{d\omega}{dt}\right)^2_{at\,t=t_2} - \left(\frac{d\omega}{dt}\right)^2_{at\,t=t_1}\right)}{\int_{t_1}^{t_2} \left(\frac{d\omega}{dt}\right)^2 dt}. \quad (10)$$

In (10), the derivatives of the actual speed and the torque reference can be obtained by low pass filtering of both signals in order to avoid amplifying the digital quantization noise and the measurement noise. From (10), it can be known that if time interval between t_1 and t_2 is sufficiently long, and if the derivative of actual speed is not zero in this interval, the second term of (10) can be neglected because the denominator will grow indefinitely. Also, if the acceleration rate at t_1 is equal or close to that at t_2, the numerator can be set to almost zero. Thus the viscous friction coefficient can be obtained by simple calculation of (11)

$$B_m = \frac{\int_{t_1}^{t_2} \frac{dT_e^*}{dt}\frac{d\omega}{dt} dt}{\int_{t_1}^{t_2} \left(\frac{d\omega}{dt}\right)^2 dt}. \quad (11)$$

Once these parameters are identified correctly, they can be used for the adjustment of the gains of controller and observer.

In the speed controller, the output of the (PI) controller is calculated in (12)

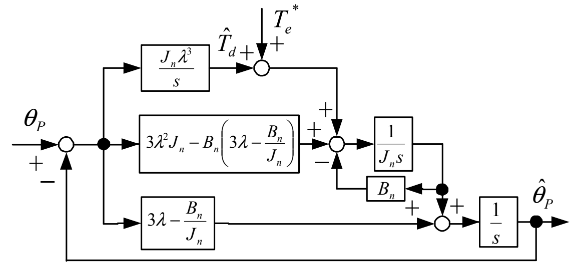


Figure 3. Block diagram of the disturbance observer

$$T_{PI} = K_V\left(1 + \frac{1}{T_i s}\right)\cdot\left(\omega^* - \omega\right), \quad (12)$$

where K_V is proportional gain and K_V/T_i is integral gain. By tuning the proportional gain in proportion to the total moment of inertia, the dynamics of the speed feedback loop can be made constant.

The identified parameters can be used for observers also. Fig. 3 shows the block diagram of the disturbance observer, where λ is real triple root of the observer. Nominal moment of inertia J_n and nominal viscous friction coefficient B_n can be tuned to the identified values.

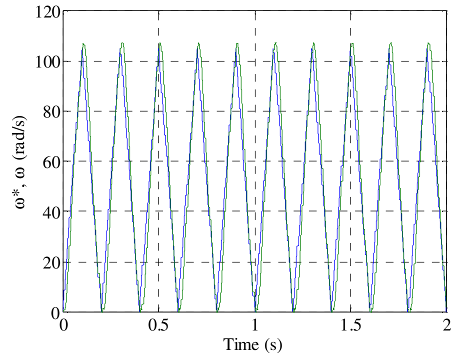
When some reference feed forward control with model parameters is used, the identified moment of inertia and viscous friction coefficient can be used for tuning the model parameters.

IV. Simulation and Experimental Results

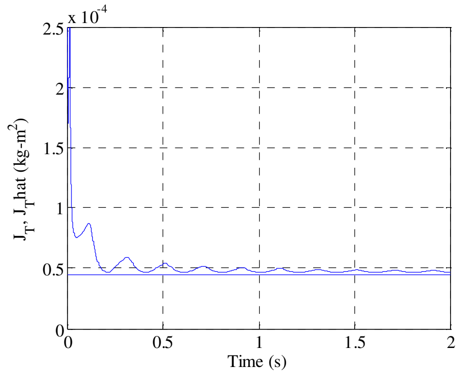
Simulation and experiments have been performed to prove the effectiveness of the proposed method.

The proposed method was applied to a servo drive system. The system consists of a linear stage KR4620A+540L, a 200W AC servomotor SGMAH02A, its servo driver, and a PC. The resolution of an encoder is 3.835×10^{-4}(rad) and the encoder is connected to a counter board in the PC. The moment of inertia and the viscous friction coefficient are identified in the PC. The stage consists of a coupling, a ball screw, and a table, that have highly nonlinear characteristic. Total moment of inertia J_T is 0.442×10^{-4}(Kg·m2) and the viscous friction coefficient is 0.5×10^{-3}(N·m·s/rad). In the simulation, coulomb friction is 0.02(N·m).

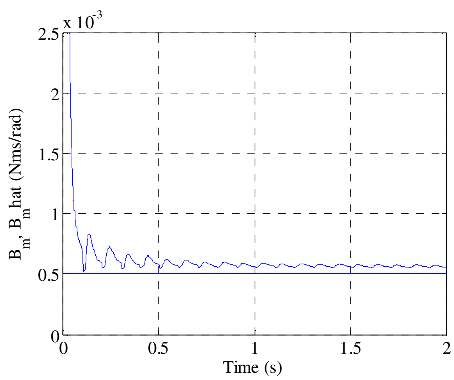
Fig. 4 and Fig. 5 show the simulation results and experimental results of the parameter identification using the proposed method. The speed reference accelerates up to 104.7 rad/s in 100ms and decelerates to zero in 100ms. The motor rotates normally and does not rotate reversely. Even for such uni-direction rotation, the identified values of parameters are almost close to the real values. The identification error of inertia is within 10% and the error of viscous friction is within 20%, and the identification time is less than 2 seconds.



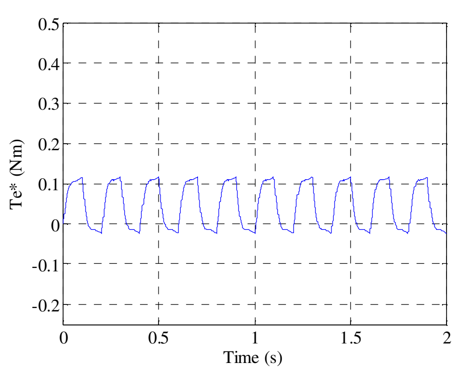
(a) Speed reference and actual speed



(b) Identified inertia

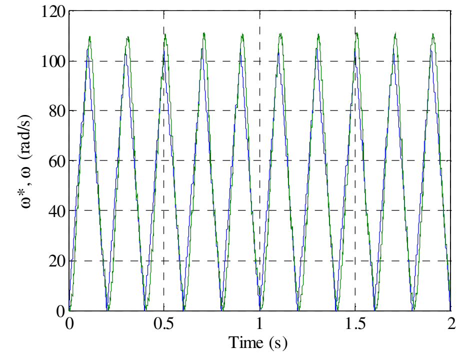


(c) Identified viscous friction coefficient

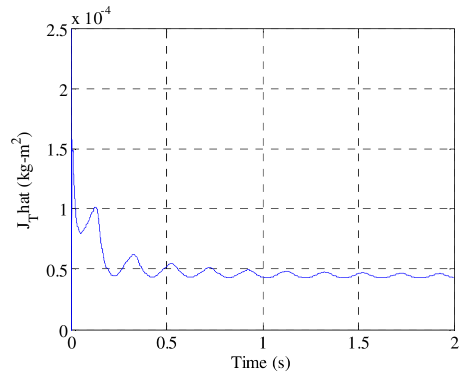


(d) Torque reference

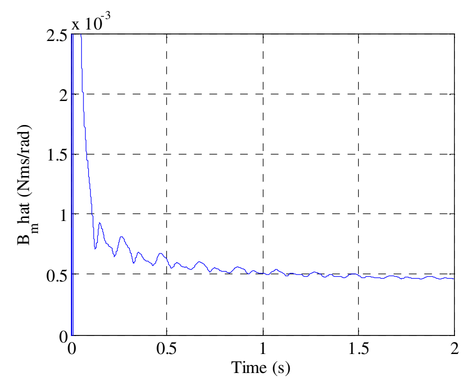
Figure 4. Simulation results of the proposed method



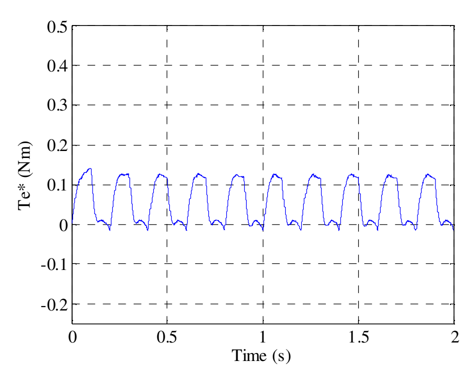
(a) Speed reference and actual speed



(b) Identified inertia

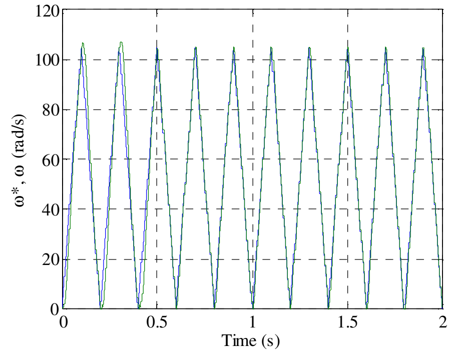


(c) Identified viscous friction coefficient

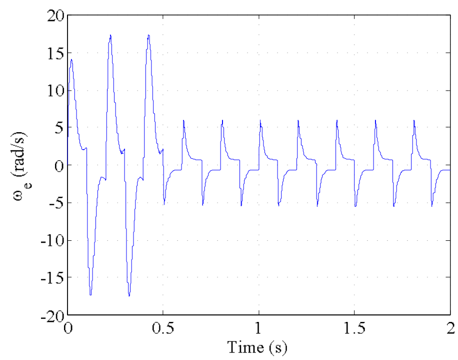


(d) Torque reference

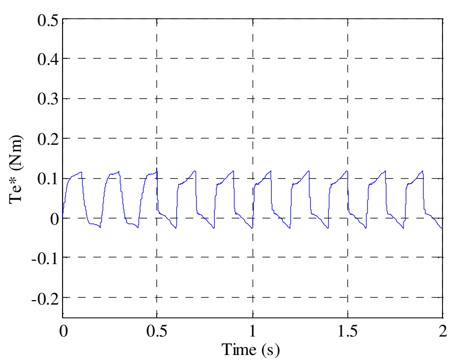
Figure 5. Experimental results of the proposed method



(a) Speed reference and actual speed



(b) Seed error



(c) Torque reference

Figure 6. Simulation results of proposed method with auto-tuned controller

The effectiveness of the auto-tuned controller and observer has been proved in simulations. Fig.6 shows the simulation results of proposed method with auto-tuned controller. In the controller, the proportional gain K_V is set proportional to a nominal moment of inertia J_n from the start to 0.5 second. Because the nominal J_n is less than a fourth of the real moment of inertia, the speed error between speed reference and actual speed is large and vibrating until 0.5 second. After 0.5 second, the proportional gain K_V is set in proportion to the identified moment of inertia and the speed error decreases to a third of that during the first 0.5 second, and does not vibrate.

Fig. 7 shows simulation results of proposed method with auto-tuned controller and disturbance observer. As shown in Fig. 2, the output of the speed controller is modified by the estimated disturbance torque $\hat{T}_d$. The step disturbance torque 0.2 N·m is added at 0.8 second. In the disturbance observer, the root λ is 628.3(rad/s), the nominal moment of inertia J_n is 0.106x10^{-4}(Kg·m^2), and the nominal viscous friction coefficient B_n is 0.2x10^{-2}(N·m·s/rad). The speed error and the torque reference in Fig. 7 vibrate because the output of the speed controller is modified to make the actual speed similar to the nominal model's behavior.

Fig. 8 shows simulation results of proposed method with auto-tuned controller and auto-tuned disturbance observer. In the disturbance observer, nominal moment of inertia J_n and nominal viscous friction coefficient B_n are tuned to the identified values. The speed error after 0.9 second in Fig.8 is almost the same as that in Fig.6 and does not vibrate unlike in the case of Fig.7.

V. Conclusion

This paper proposes a novel identification method of the moment of inertia and the viscous friction coefficient for high dynamic servo drive systems. It uses the dynamic equation of a simple mechanical system, the torque reference of a speed controller, and the actual rotating speed of machine. Using these, the proposed method can identify the moment of inertia and the viscous friction coefficient on real time by performing some simple calculations without being affected by external disturbances such as quantization noise, and coulomb friction. The identified inertia can be used for auto-tuning of the gains in the speed controller, the speed observer, and the disturbance observer, in order to obtain high dynamic system performance. The effectiveness and feasibility of the proposed method are verified by the computer simulation and experimental results. The simulation and experimental results show that the error of the identification is within 20% after 2 seconds.

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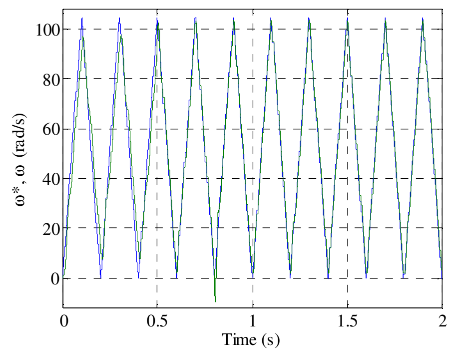
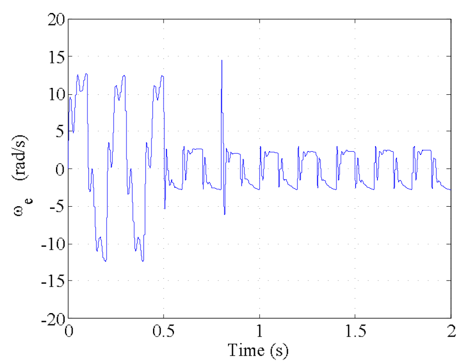
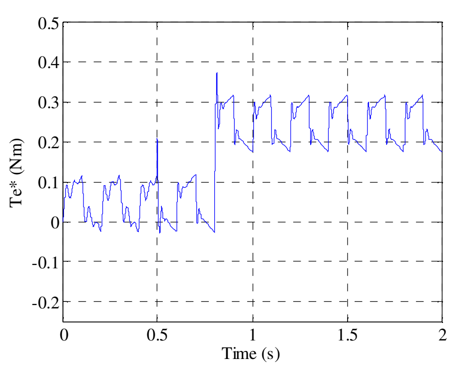


(a) Speed reference and actual speed

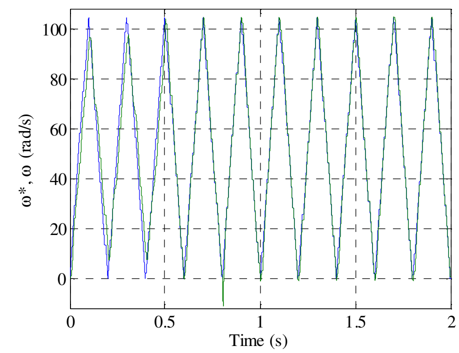


(b) Seed error

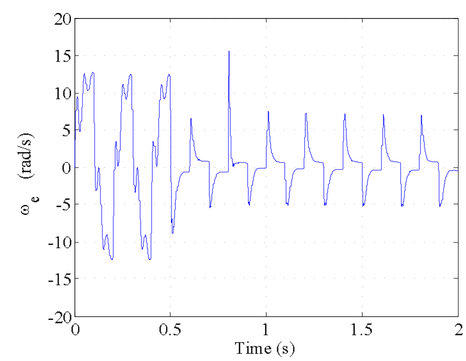


(c) Torque reference

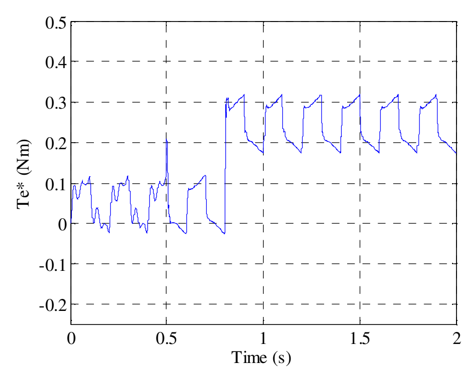
Figure 7. Simulation results of proposed method with auto-tuned controller and disturbance observer



(a) Speed reference and actual speed



(b) Speed error



(c) Torque reference

Figure 8. Simulation results of proposed method with auto-tuned controller and auto-tuned disturbance observer