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Adaptive Robust Coordinated Control for Over-actuated Cutter-head Driving Systems of Hard Rock Tunnel Boring Machines

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Abstract: The cutter-head system of tunnel boring machines (TBM) is one of the key components for rock cutting and excavation. In this paper, the coordinated control problem is investigated for the over-actuated cutter-head driving system of TBM. A generalized nonlinear time-varying dynamic model is developed for the hard rock TBM cutter-head driving system. An adaptive robust control law integrated with control allocation technique is proposed, which controls the cutter-head to desired velocities while driving torques are well coordinated. Specially, the adaptive robust control (ARC) is used to deal with the negative effects of various uncertainties and variable loads acting on the cutter-head, and a coordinated control scheme with appropriate control allocation is developed to distribute the driving torque of each motor evenly. Comparative simulations are carried out to verify the excellent performance of the proposed scheme. The results suggest that, with the proposed control scheme, the motion performance is guaranteed and the driving torques are also well coordinated, validating the effectiveness and the performance improvement of the proposed control strategy.

Keywords: Adaptive robust control, Over-actuated system, Control allocation, TBM, Tunnel boring machine, Cutter-head

1. INTRODUCTION

Hard rock tunnel boring machine (TBM) is the major technical equipment badly needed in national infrastructure construction, resource development and national defense construction. The TBM has been widely applied in tunnel constructions for decades(Du and Du, 2011). The TBM is a kind of mobile machines constituted by many subsystems, shown in Figure.1. The cutter-head driving subsystem is driven by multi-motors shown in Figure.2, which generates enough torque to excavate the rocks, so it plays an important role in hard rock TBM.

Even though some researchers and publications have paid attention to the TBM cutter-head driving system, many problems still remain to be solved. At present, most publications generally focus on the structure and composition of the TBM cutter-head driving system, excavating performances, field operation experiences, and the shield tunneling methods (Rostami, 2008). Recently, a mathematical model of the cutter-head driving system with consideration of gear frequency cycle error, backlash, and the parameter difference of driving motors was proposed in (Li et al., 2010, 2013), in which the effects of some physical parameters on the system performance was analyzed as well. The dynamic models of TBM cutter-head driving system provide a useful guidance for better understanding and analyzing the cutter-head driving system in theory, achieving multi-motor control of the cutter-head. Even though some researchers and publications pay attention to the TBM cutter-head driving system, many control problems still remain to be solved.

For the cutter-head driving system, most of the controllers used in practice are designed to control each driving motor independently and the controller of each axis receives no information from others, which is essentially a decoupled controller designed without considering the whole dynamic model of cutter-head driving system. For the multi-motors driving systems, the decoupled controller usually leads to a degraded performance, due to the lack of coordination. As opposed to the traditional decoupled controls, the master-lave control strategy is commonly used as well (Perez-Pinal et al., 2004; Koren, 1980). An alternative approach is the cross-coupled control strategy (Lin et al., 2012; Chen and Chen, 2012), which is widely used in coordinated control of multi-axes machines. Based on the cross-coupled technique in Sun et al. (2010), a ring coupled control strategy is proposed for multi-motor synchronization of TBM cutter-head based on the idea of parallel control with compensation. However, these control designs did not consider the negative effect of the physical constraints among the driving motors, which may result in uneven driving torques.

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The objective of this paper is to integrate the recently developed ARC strategy (Yao and Tomizuka, 1997; Yao, 1997) with control allocation technique (Alwi and Edwards, 2008; Oppenheimer et al., 2006; Chen and Wang, 2014) to develop practically implementable and yet high performance coordinated controller for TBM cutter-head driving system. Theoretically, the idea of adaptive robust control (ARC) provides a rigorous theoretic framework for the precision motion control of systems under both parametric uncertainties and uncertain nonlinearities. The effectiveness to attenuate the negative influence of load variations has also been verified through various application studies (Hu et al., 2010a; Sun et al., 2013). The proposed control strategy also addresses synchronous control scheme with control allocation to regulate the unbalance of the driving force, whose excellent performance has also been verified for over-actuated system (Li et al., 2014; Prasad et al., 2013). The rest of the paper is organized as follows. In Section II, the mechanical coupling among the cutter-head driving systems is analyzed and the system dynamics are derived. In Section III, the ARC technique integrated with the control allocation is developed. Comparative simulation results are presented in Section V to illustrate the effectiveness of the method, with conclusions drawn in Section V.

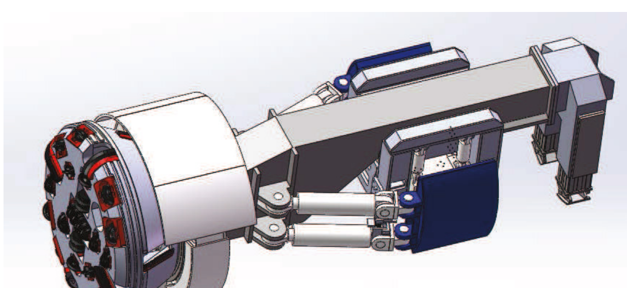

Fig. 1. Hard rock Tunnel boing machine

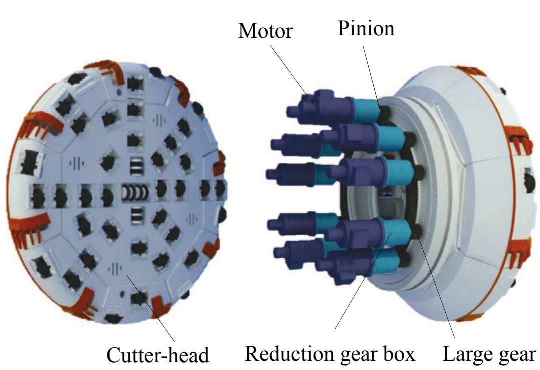

Fig. 2. Hard rock TBM cutterhead driving system

2. SYSTEM DYNAMICS AND PROBLEM FORMULATION

2.1 Nonlinear Cutterhead Driving Model

The cutterhead driving system structure is shown in Fig.3. According to paper(Li et al., 2010, 2013), the dynamic model of TBM cutter-head driving system can be obtained.

$$J_{d,i}\ddot{\theta}_{d,i} + b_{d,i}\dot{\theta}_{d,i} + M_1^i = T_{d,i} \tag{1}$$

$$\theta_{d,i} = q\theta_{p,i}, M_2^i = qM_1^i \tag{2}$$

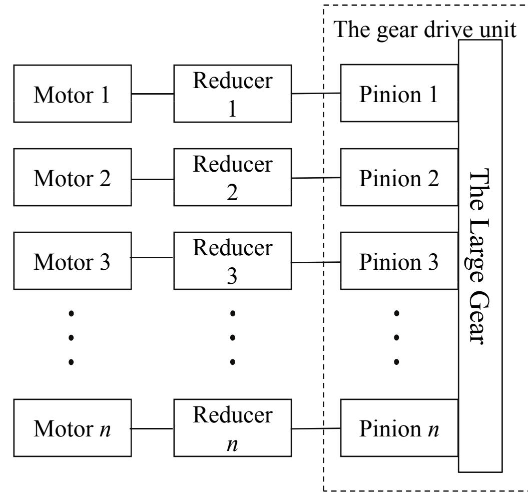

Fig. 3. Hard rock tunnel boing machine

$$J_{p,i}\ddot{\theta}_{p,i} + b_{p,i}\dot{\theta}_{p,i} + M_{p,i} = M_2^i \tag{3}$$

$$M_{p,i} = k_{t,i}\varphi + c_{t,i}\dot{\varphi} \tag{4}$$

$$J_c\ddot{\theta}_c + b_c\dot{\theta}_c + T_L = \sum_{k=1}^{n} M_{p,k} i_{c,k} \tag{5}$$

where $u_{d,i}$ are the control inputs that produce the electromagnetic torque $T_{d,i} = k_i u_{d,i}$. $J_{d,i}$ is inertia of the ith coupling between motor$i$ and reducer$i$,$b_{d,i}$ is the viscous damping ratio of the ith coupling between motor-i and reducer$i$, $\theta_{d,i}$ is the angular displacement of ith motor rotor, q is the speed reducer ratio, $\theta_{p,i}$ is the angular displacement of ith active pinion, $J_{p,i}$ is ith active pinion inertia after equivalent coupling-i and $b_{p,i}$ is ith active pinion damping ratio after equivalent coupling-i, θ_c is the cutter-head angular displacement, J_c is cutter-head inertia and b_c is cutter-head viscous damping ratio, $M_{p,i}$ is the mesh torque between the pinion and large gear, and T_L is total load torque of TBM cutter-head. Total load torque is affected by many factors such as geology conditions, cutter-head design parameters, and physical structure. However, the actual load torque is varying with the geology conditions. Gear mesh process is a nonlinear transmission process, φ in expression of mesh torque M_c is chosen as follow in this paper.

$$\varphi(z) = \begin{cases} z - \Delta_i, z \geq \Delta_i \\ 0, -\Delta_i \leq z < \Delta_i \\ z + \Delta_i, z < -\Delta_i \end{cases} \tag{6}$$

where $z = \theta_{p,i} - i_{m,i}\theta_m - e_i(t)/r_i$, Δ_i is the ith pinion total gear backlash, $e_i(t)$ is ith pinion transmission error, and r_i is the radius of ith active pinion.

2.2 Reduced Dynamic Model

Since the mesh backlash was taken into consideration, the aforementioned dynamic model of cutter-head driving system is a kind of 4-order model. For controller design, the mesh process can be treated as ideal transmit process, so the following equations can be obtained.

$$\theta_{p,i} = i_c\theta_c, M_{c,i} = i_c M_{p,i} \tag{7}$$

Substituting (7) into (3), we can obtain the second order dynamic model of the cutter-head driving system.

$$J_E\ddot{\theta}_c + b_E\dot{\theta}_c + T_{ld} = k_v u_v + \tilde{d} \tag{8}$$

where $J_E = i_c^2 J_{e,i} + J_c$ represents the equivalent moment of inertia of cutter-head driving system, $J_{e,i} = q^2 J_{d,i} + J_{p,i}$ and