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Hierarchical modeling method and dynamic characteristics of cutter head driving system in tunneling boring machine

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ABSTRACT

Traditional modeling method is difficult to establish the fully coupled model of the cutter head driving system in tunneling boring machine (TBM) owing to the structural complexity. In this paper, a generalized hierarchical modeling method is proposed to establish the entirely dynamic model of the TBM cutter head driving system. The components are equivalent to beam element with the same mass and stiffness characteristics, and the nonlinear coupling factors such as time-varying mesh stiffness, transmission error and gear backlash of both the multiple pinions driving and planetary gearbox are considered. The overall dynamic equation is formed by assembling the unit matrixes of basic finite elements according to the coupling relationships. Based on the proposed model, the vibration modes of driving system are classified into axial modes, rotational modes and lateral modes with distinct characteristics. The effects of coupling parameters on the natural frequency of driving system are investigated as well. The dynamic model is also simulated to analyze the dynamic response of driving system under two load conditions, cyclic external loads and simulated external loads. The main response frequency components and the effect of damping coefficients on vibration amplitudes are investigated. The proposed modeling method can be generalized to other complex driving system, and the obtained results provide data support for the dynamic design of driving system in TBM.

1. Introduction

TBM is widely used in the tunnel engineering for water conservancy, transport and municipal due to advantages such as high driving speed, environmental protection and high efficiency. As a critical component of TBM, the cutter head driving system is a complex nonlinear system, which contains multiple interfaces, redundant gear transmission and planetary gears in restricted space (Sugimoto and Sramoon, 2002). The strong shock excitation caused by rock breaking (Ramoni and Anagnostou, 2011; Delisio et al., 2013) may lead to excessive vibration (Zhang et al., 2012), which posed many challenges to the equipment (Zhao et al., 2007).

In recent years, few researchers focused on the multiple pinions transmission in TBM cutter head driving system. The traditional lumped mass model is established by considering both the linear and nonlinear factors such as gear backlash, time-varying stiffness and transmission error, larger gear inertia and the speed-torque characteristics of variable frequency motor. On this basis, Zhang et al. (2010) simulated the dynamic characteristics of TBM tunneling in mixed-face conditions; Wei et al. (2013) studied the load-sharing characteristic of multiple pinions driving; Li et al. (2010) predicted the dynamic response of the driving system and refined some important issues that should be given closer attention.

Generally, in current research, the planetary gearbox is always ignored or equivalent to mass element in the established dynamic model of TBM cutter head driving system. However, numerous of studies have shown that, the planetary gears have distinct vibration mode structure because of their cyclic symmetry (Lin and Parker, 1999; Kiracofe and Parker, 2007), and the varying contact conditions at the gear tooth mesh interface may induce noise and vibration problems (Vijaya and Parker, 2007). Therefore, it needs to take into account the dynamic effects of planetary gearbox for studying the dynamic characteristics of the whole TBM cutter head driving system.

As regards the lamped parameter model of planetary gears, three different modeling approaches: constant mesh stiffness models with linear time-invariant equations (Velex and Ajmi, 2006), fluctuating mesh stiffness models with linear time-varying equations (Lin and Parker, 2002), and fluctuating mesh stiffness and tooth contact loss with nonlinear time-varying equations (Al-Shyyab and Kahraman, 2007), are performed in current researches.

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Mesh phasing was found that it had a dramatic impact on the static and dynamic behavior of planetary and epicyclic gears (Parker and Lin, 2004). On this basis, Abousleiman and Velex (2006) presented a hybrid 3D finite element/lumped parameter model which enables the simulation of the three-dimensional dynamic behavior of planetary/epicyclic spur and helical gears. Sun et al. (2014) established the bending-torsional coupling model of multi-stage planetary gears, and obtained the distinct natural frequency and coupled mode characteristics.

However, it is difficult to establish the fully coupled dynamic model of the TBM cutter head driving system using the traditional methods because of the complex coupling relationships. In this paper, a novel modeling method is proposed to establish the fully coupled finite element (FE) model of TBM cutter head driving system. Taking a TBM working in a water project as an example, both the modal characteristics and vibration response of the driving system, as well as their sensitivity to the dynamic parameters such as coupling stiffness and damping coefficients are investigated. It provides the data support in the dynamic design of TBM driving system, and the modeling method can be generalized to other complex driving system.

2. Hierarchical modeling method of cutter head driving system

The TBM cutter head driving system provides steady rotary power for cutter head to cut rock continuously. It is a driving system with multiple pinions, and the ring gear is driven by six, or eight, or ten or more driving motors via planetary gears. Fig. 1 shows the structure diagram of the cutter head driving system in TBM.

2.1. Hierarchical principle

Obviously, the TBM cutter head driving system can be seen as an integrated system which is assembled by several basic subsystems and components with couplings such as gear mesh, bolt connection and spline connection. It is suited to use hierarchical principle to establish the FE model of TBM cutter head driving system. The main idea of this theory is to define the unit matrix of different basic elements and load vector firstly, and then assemble them via the coupling elements to obtain the overall mass, stiffness, damping matrix and load vector, which forms the dynamic equation of the whole system.

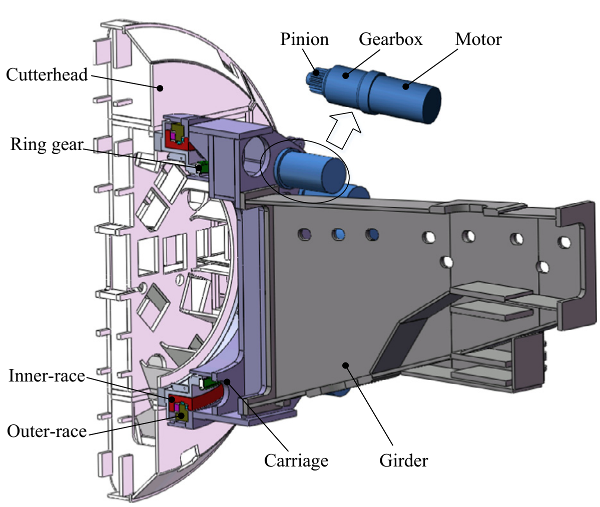

Fig. 1. Structure diagram of the driving system in TBM.

Fig. 2 shows the modeling process of TBM cutter head driving system using hierarchical principle. It contains the following three steps:

(1) *Splitting*. Firstly, the driving system is divided into one cutterhead-bearing subsystem and several identical driveshaft subsystems.
(2) *Equivalent*. The main components such as the cutterhead, the flange, the pinion and the motor in each subsystem are equivalent to Timoshenko beam elements with the same mass and stiffness characteristics. The dynamic model of planetary gears is established using the lumped mass method. The supporting between the cutterhead and rock and the connections such as that between the cutterhead and the flange can be equivalent to the spring-damper elements.
(3) *Coupling*. Coupling the FE model of cutterhead-bearing subsystem with particular number of driveshaft subsystems by gear mesh elements and bolt coupling with the equivalent beam elements of carriage and girder by bolt connection, the dynamics model of the whole driving system can be established finally.

Therefore, there are four kinds of basic finite elements in the FE model of driving system, beam element, gear mesh element, bearing/bolt/spline coupling element and the dynamic model of planetary gears.

2.2. Characteristics of basic finite elements

2.2.1. Beam element

Fig. 3 shows the coordinate definition of beam element with two nodes A and B. Each node has 6 degrees of freedom, which is defined as $\boldsymbol{q}_b = [x_A, y_A, z_A, \theta_{xA}, \theta_{yA}, \theta_{zA}, x_B, y_B, z_B, \theta_{xB}, \theta_{yB}, \theta_{zB}]^T$, where $x_i, y_i, z_i, \theta_{xi}, \theta_{yi}, \theta_{zi}$ $(i = A,\ B)$ are the linear and rotational displacements along the three coordinate axes directions respectively.

The stiffness matrix $\boldsymbol{K}_b$ and mass matrix $\boldsymbol{M}_b$ of Timoshenko beam element can be deduced according to elasticity theory (Wang, 2003). Hutchinson calculated the shear coefficients in Timoshenko's beam theory for various cross sections (Hutchinson, 2001). The damping matrix $\boldsymbol{C}_b$ is defined as Rayleigh damping, $\boldsymbol{C}_b = \alpha\boldsymbol{M}_b + \beta\boldsymbol{K}_b$, where, α and β are the Rayleigh scaling factors.

Therefore, neglecting the rotate effect, the dynamic equation of beam element is

$$\boldsymbol{M}_b\ddot{\boldsymbol{q}}_b + \boldsymbol{C}_b\dot{\boldsymbol{q}}_b + \boldsymbol{K}_b\boldsymbol{q}_b = 0 \quad (1)$$

where $\boldsymbol{q}_b$ is the force vector of the beam element.

2.2.2. Gear mesh element

Fig. 4 shows the equivalent dynamic model of gear mesh in the space coordinate. The displacement vector of both gear node and pinion node is $\boldsymbol{q}_m = [x_r, y_r, z_r, \theta_{xr}, \theta_{yr}, \theta_{zr}, x_p, y_p, z_p, \theta_{xp}, \theta_{yp}, \theta_{zp}]^T$, where the subscript "$r$" represents the gear node, and "p" represents the pinion node.

The deflection of tooth mesh along the line of action is

$$\delta_{rp} = \boldsymbol{V}\cdot\boldsymbol{q}_m + e(t) \quad (2)$$

where $e(t)$ is the time-varying, unloaded static transmission error; $\boldsymbol{V}$ is the projection vector, which can be defined as

$$\begin{aligned}\boldsymbol{V} = [&-\sin\psi_{rp}\cos\beta_b,\ \cos\psi_{rp}\cos\beta_b,\ -\sin\beta_b,\\ &-r_{br}\sin\psi_{rp}\sin\beta_b,\ r_{br}\cos\psi_{rp}\sin\beta_b,\ r_{br}\cos\beta_b,\\ &-\sin\psi_{rp}\cos\beta_b,\ -\cos\psi_{rp}\cos\beta_b,\ \sin\beta_b,\ -r_{bp}\sin\psi_{rp}\sin\beta_b,\\ &-r_{bp}\cos\psi_{rp}\sin\beta_b,\ r_{bp}\cos\beta_b]\end{aligned} \quad (3)$$