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Research on dynamical characteristics of wind turbine gearboxes with flexible pins

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ABSTRACT

Wind energy is believed as one of the most efficient clean renewable energies and is explored more often currently. The dynamic behaviors of wind turbine gearboxes are concerned by engineers since they affect the whole working performances and the service lives of wind turbines. In this work, a coupled dynamic model is developed for a wind turbine gearbox with flexible pins. For the planetary gear stage, the sun gear is floated and the planetary gears are flexibly supported by flexible pins. The sub-transmission system of the gearbox, which consists of a planetary gear stage and a parallel gear stage, and the body sub-system are coupled through the bearings which are simulated as springs. The dynamic behaviors of the system are studied using a lumped parameter model, in which the stiffnesses of the gearbox body is predicted using the finite element method, the results is also verified by experiments conducted on a test rig.

1. Introduction

Due to the large gear ratio, the compact structure and the good power split ability, the planetary gear sets are commonly applied in wind turbine gearboxes. The unavoidable manufacturing errors and the assembly errors of the gear sets would cause the load sharing of the planetary gears uneven. Hence, adding flexibilities in the planetary gear sets are good choices to alleviate effects of the uneven load sharing. Wind turbine gearboxes with flexible pins not only allow the sun gear floating, but also allow the planetary gears floating by the aid of flexible pins. In this way, the load would be shared more evenly between planets, and the structure space would saved compared with the straddle type. The number of planetary gears can be increased with the flexible-pin type and the power density of the gearbox can be improved. However, since wind turbine gearboxes work in extreme conditions such as continuous variations of the rotating speed and the load, the requirements of the dynamic behaviors of gearboxes should be higher to maintain acceptable working performances and service lives.

The dynamic problems of wind turbine gearboxes have been studied recently. Peeters investigated the dynamical behavior of planetary stages, also analyzed the behaviour of wind turbine gearboxes by means of a coupled finite element and multi-body [1,2]. Xing et al. investigated the effect of a spar-type floating wind turbine nacelle motion on drivetrain dynamics [3]. Helsen focused on the dynamics of the gear units of wind turbine [4]. Link et al. establish a database of gearbox failures and investigate condition monitoring methods to improve reliability in drive trains that particular is gearboxes [5]. Guo and Parker studied nonlinear tooth wedging behavior and its correlation with planet bearing forces by analyzing the dynamic response of a planetary gear [6] and then they analyzed the nonlinear effects introduced by bearing clearance using the lumped-parameter and the finite element method [7]. Montestruc showed the load sharing characteristics of different Hicks type flexible planet pin using the finite element method [8]. Very recently, Velex et al. developed a lumped-parameter model to study effect of planet position error and eccentricity errors on dynamic behaviors of planetary systems [9,10]. Qin et al. applied a mixed flexible-rigid multi-body model to investigate dynamic behaviors of wind turbine [11,12]. Zhu et al. focused on the load sharing characteristic of flexible-pin wind turbine gearbox [13,14]. Feng et al. conducted fault diagnosis for a wind turbine planetary gearbox via demodulation analysis based on ensemble empirical mode decomposition and energy separation [15]. Those studies provide valuable guidance for the optimum design of wind turbine gearboxes. Those studies

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provide valuable guidance for the optimum design of wind turbine gearboxes. However, the double floating structure both planet gear with flexible pin and floating sun gear is considered in the dynamic model, this is rarely mentioned in the previous publications. Due to the complicated structures of flexible-pin wind turbine gearboxes, it is difficult to reflect the dynamic responses of the gearbox systems when the speed or the load vary continuously by only applying the finite element method. In the present work the lumped-parameter method and the finite element method are combined to study the dynamic behaviors of a multi-flexible-pin wind turbine gearbox.

2. Basic structure of the gearbox

Fig. 1 shows the diagram of a 1.5 MW flexible-pin wind turbine gearbox, in which the planets are supported through flexible pins. The input planetary carrier c is connected with the blade spindle through main shaft, while the output stage is connected with the generator. The symbols shown in Fig. 1 are defined as below: s represents the sun gear, r is the internal gear ring, c is the carrier, Pn ($n = 1,2,3,4$) represents the nth planets, $g1$ and $g2$ are inter-mediate stage gears while $g3$ and $g4$ are high-speed stage (output) gears. The main parameters of this gearbox are listed in Table 1.

The structure of a flexible pin is shown in Fig. 2(a). The pin consists of a planetary carrier 1, a central shaft 2, and a planet gear shaft 3 [16,17]. The force diagram of the pin is shown in Fig. 2(b). The planet shaft 3 keeps balance under the combined action of the external load f and the reacting force and the torque M_1. In terms of the end side of the central shaft, the moment equations read

$$\begin{bmatrix} M_1 \\ M_2 \end{bmatrix} = \begin{bmatrix} f \times l/2 \\ f \times (l/2 + l/2) \end{bmatrix} - \begin{bmatrix} 0 \\ M_1 \end{bmatrix} \tag{1}$$

where f represents the external load applied on the planet shaft, M_1 means the torque on the planet shaft, M_2 is the torque on the central shaft, and l represents the length of the central shaft.

From Eq. (1) it can be seen that $M_1 = M_2$, which means the bending moment at the central point of the central shaft should be zero. The two ends of the central shaft can be assumed as two cantilevers which would show S shape deflections without any lean. This characteristic assures the planets keeping load shared evenly though some manufacturing errors or assembly errors introduce torques on planets. The supporting stiffness k_{pb} of the planets can be assumed as the series of the stiffness of the flexible pin k_p and the bearing stiffness of the planet k_{b}

$$k_{\mathrm{pb}} = 1/(1/k_{\mathrm{p}} + 1/k_{\mathrm{b}}) \tag{2}$$

3. The dynamical model

The wind turbine gearbox is divided into the transmission sub-system and the body sub-system. The transmission sub-system, which consists of a planetary gear stage and a parallel gear stage, is developed using the lumped-parameter method, while the body sub-system model is developed using the finite element method. The two sub-systems are coupled using bearings when the boundary conditions of the two sub-systems are defined.

3.1. Dynamical model of the transmission sub-system

The dynamic model of the transmission sub-system, which is shown in Fig. 3, is developed using the lumped-parameter method. All gears are defined as spur gears, whose movements are determined through a rotation degree-of-freedom (DOF) u and two translational DOFs x and y. oxy represents the fixed coordinate system of the ring, with o representing the center of the ring. The direction x indicates the direction toward the first planet from the origin o. ort is the moving coordinate system fixed on the planet carrier. $x_i, y_i (i = r, c, s, g1, g2, g3, g4)$ represent the vibration amplitude of the component i in the x and y direction, respectively. u_i is the rotational displacement of component i. t_n,r_n($n = 1,2,3,4$) are the vibration amplitude of the nth planet along the tangential and the radial direction, respectively. u_n is the rotational displacement of the nth planet. k_b^i, $c_b^i (i = r, c, s, g1, g2, g3, g4)$ are the supporting stiffness and the damping of the component i, respectively. k_u^i and c_u^i are the rotational stiffness and the damping of the component i, respectively. $k_{sp}^n(t)(n = 1,2,3,4)$, $k_{rp}^n(t)$, $k_{g1g2}(t)$ and $k_{g3g4}(t)$ are the meshing stiffness of the sun gear and the nth planet ($s - P_n$) pair, the ring-the nth planetary ($r - P_n$), the inter-mediate gear pair ($g1 - g2$) and the high speed gear pair ($g3 - g4$), respectively. c_{sp}^n, c_{rp}^n, c_{g1g2} and c_{g3g4} are the meshing damping of those gear pairs. k_{pb}^n and c_{pb}^n are the supporting stiffness and the damping of the planets. ω_c is the input angular velocity of the planet carrier. The flexible pins and the floating of the sun gear are simulated as low-stiffness springs.

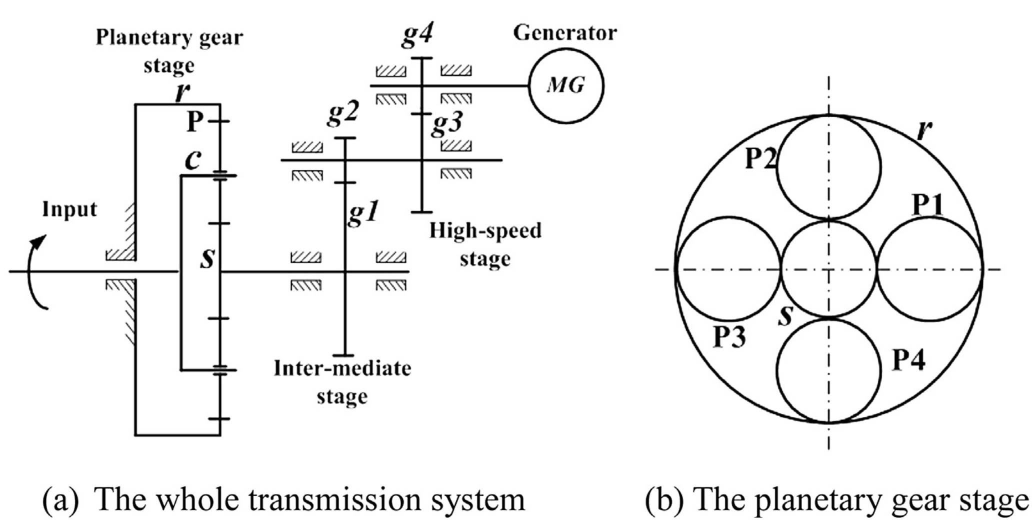

Fig. 1. Planetary gear train of wind turbine gearbox with flexible pins.