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Drivetrain resistance and starting performance of a small wind turbine

Jerson R.P. Vaz [a,*], David H. Wood [b], Debanik Bhattacharjee [c], Erb F. Lins [a]

[a] Faculty of Mechanical Engineering, Institute of Technology, Federal University of Pará, Av. Augusto Correa, N 1, Belém, PA, 66075-900, Brazil
[b] Department of Mechanical and Manufacturing Engineering, Schulich School of Engineering, University of Calgary, Calgary, T2N 1N4, Alberta, Canada
[c] Department of Mechanical Engineering, University of Alberta, Edmonton, T6G 2R3, Alberta, Canada

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ABSTRACT

Most small wind turbines do not have pitch adjustment of the blades. This makes starting at low wind speed a serious challenge which is magnified by the drivetrain resistance caused by bearing friction, generator cogging torque and so on. Typically the resistive torque is much less than the rated generator torque so drivetrain resistance is safely ignored once power production commences but must be considered when the rotor torque is low. This occurs during starting and when the turbine approaches the runaway condition of no output load. Equations are derived here for the drivetrain resistance of a small turbine as part of an analysis of starting and runaway which is compared to wind tunnel measurements. This paper focuses on the resistance due to the bearings in the drivetrain, especially on the transition from high static to significantly lower dynamic resistance as high static resistance increases the wind speed at which a turbine starts. The measurements were made using a three-bladed turbine with no other loads so they include both starting performance and runaway. The results demonstrate a static resistive torque about seven times the dynamic one, giving a theoretical starting wind speed of 4.20 m/s which is 6% higher than the measured value. Good agreement is found between the analysis and measurements of rotor angular velocity over the whole operating range from starting to runaway, highlighting the importance of this work for estimating the minimum wind speed for the starting of small wind turbines.

1. Introduction

It is well known that the use of horizontal-axis wind turbines (HAWTs) has increased due to their low environmental impact. Researchers around the world have dedicated much effort to improving models applied to HAWTs, e.g. Refs. [1–3]. Although several scientific works have been devoted to improving HAWT design methodologies, only a few have studied starting behavior. Melício et al. [4] studied the influence of wind speed disturbances and a pitch control malfunction on the quality of energy injected into the electricity grid, considering the transient behavior of wind turbines with different power-electronic converter topologies. A new control strategy was proposed for the variable speed operation with permanent magnet synchronous generators. However, rotor starting was not reported in their work. Nagai et al. [5] described HAWT performance in terms of rotational speed, generator output, and its stability when the wind speed changes. The modeled system behavior was confirmed under real wind conditions. Starting was not evaluated.

Drivetrain resistance is particularly important for small HAWTs, where the blades rarely have pitch adjustment, because of the high cost, Wood [6]. Stationary blades experience high angles of attack and so lift and blade torque are low. Consequently, it is difficult to start the blades at low wind speed, and it is common for the wind speed at which they start to be considerably higher than the cut-in wind speed as determined by a standard power performance test [6,7]. An important demonstration of this is provided by Sarkis & Pinilla [8] who show that the wind speed for maximum efficiency of a waterpumping windmill is less than the starting speed. The only way a windmill can reach this desirable operating condition is when the wind speed has decreased after the blades have started. Wright & Wood [7] and Wood [6] show that blade element theory (BET) accurately predicts starting behaviour when typical rotor acceleration is slow enough for a quasi-steady analysis to be accurate. These authors considered turbines whose dominant resistance was the cogging torque of the permanent magnet generator or the frictional torque of the gearbox. Both these have a much

* Corresponding author.
E-mail address: jerson@ufpa.br (J.R.P. Vaz).

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smaller variation between static and dynamic values than the bearings considered here. To our knowledge, there has been no analysis in the open literature of the effect of bearing friction on starting and runaway. Assuming no power extraction, which is equivalent to ignoring axial and rotational induction, gives simple expressions for the rotor torque. After subtracting the drivetrain resistance, the angular acceleration of the rotor is determined. This analysis ignores the rotational kinetic energy of the rotor which must come from the wind. For the cases analyzed in Refs. [6] and [7], the rotor kinetic energy at the end of starting is very small compared to the kinetic energy that passed through the rotor during the long starting period.

Clearly any resistive torque in the drivetrain can have a large influence on wind turbine starting. For turbines with permanent magnet generators, the cogging torque of the generator can be the major resistance. Wood [6] developed the rule of thumb that the resistive torque must be less than 1% of the rated torque for there to be minimal impact on starting. For all turbines, the bearings in the drivetrain can also contribute. The effects of bearing resistance on a small vertical axis wind turbine was recently studied by Aso et al. [9]. Measurements showed that for a judicious choice of bearings, the required aerodynamic torque could be reduced 70%, reducing the starting wind speed by approximately 62%. A detailed review on tribological issues for the main shaft and generator bearings, and gearboxes in wind turbines was given by Kotzalas and Doll [10], as well as by Fernandes et al. [11], who studied five fully wind turbine gear oils with the same viscosity grade but different formulations. Stammler et al. [12] studied the behaviour of large wind turbine pitch bearings using the friction model of the company Rothe Erde GmbH. This model does not incorporate the difference between static and dynamic friction associated with the Stribek effect described in the next paragraph, so it has not been investigated here.

Even though the present work treats wind turbine drivetrains, the methodology can be used for other turbine types, including hydrokinetic turbines (HTs) [13,14]. The transient behavior of HT drivetrains was studied by Mesquita et al. [15] who used BET combined with models of the friction for each component. Lopes et al. [16] used the same methodology to model the transient behavior of a small HT using the Palmgren [17] model of the frictional bearing torque as reported by Harris and Kotzalas [18]. Lopes et al. [16] compared their simulation with measured data from a field test, yielding good agreement. The Palmgren model [17] used in Ref. [16], however, is applicable only for dynamic friction. For starting, it is necessary to extrapolate to the static friction. Generally, and unfortunately for small wind turbines and HTs, the static resistance is often significantly higher than the dynamic one. The difference is usually attributed to the Stribeck effect, [19], arising from limited bearing lubrication at low or zero speeds.

This work describes an analysis of the drivetrain resistance due primarily to bearing friction and its effect on the starting performance of a small HAWT including the Stribeck effect. The context of this work is an experimental and computational investigation of a small HAWT based on a reproduction of the 24-bladed horizontal-axis windmill tested by Weegeref [20]. Part of the present program is a study of solidity effects, see Fagbenro et al. [21], so the model was designed to allow variable blade numbers. For the present purposes, solidity is a complication so the 0.68 m diameter model was used with three blades. Good agreement is found between the analysis and measurements.

The remainder of this paper is organized as follows. The next section describes the dynamic model, and the equations for the bearing frictional torques. Section 3 details the windmill experimental setup and blade element analysis. Section 4 explains the two separate experiments on the bearing resistance and shows the final form of the model for the Stribeck effect. Section 5 contains the results and discussion, where the Stribeck effect is evaluated, as well as the starting behavior of the turbine driveline. The conclusions of this study are given in Section 5.

2. Dynamic model

The wind turbine consists of an aerodynamic rotor with mass-moment of inertia J_T connected by a shaft to the drivetrain with two deep-groove ball bearings, as illustrated in Fig. 1, that are part of a magnetic particle brake. The balance of the drivetrain has an inertia of J_S.

The torque balance on the wind turbine is

$$T_T - T_D = J_{total}\frac{d\omega_T}{dt} \tag{1}$$

where T_T is the aerodynamic torque which is calculated using BET as explained in Section 3. T_D is the drivetrain resistive torque, in this case caused partly by the frictional torque of the bearings, and ω_T is the turbine angular speed. The total mass-moment of inertia is $J_{total} = J_T + J_S$ where the former is the rotor inertia and the latter is the (usually much smaller) inertia of the remaining rotating components. The blades start only when $T_T > T_D$.

2.1. Bearing frictional torques

The bearing friction is analyzed for two cases: dynamic and static. Palmgren [17] and Harris and Kotzalas [18] separated the frictional torque into a load-dependent component (T_L) and a load-independent component (T_V) which is influenced by the lubricant type, the amount of the lubricant employed and bearing speed. T_L and T_V are given by empirical formulas. Using the subscript “P” to denote “Palmgren”:

$$T_{D,P} = T_L + T_V \tag{2}$$

T_L is given by

$$T_L = f_L F_\beta d_m \tag{3}$$

where F_β is described below, d_m is the bearing pitch diameter, and f_L is a factor depending on the bearing design and relative bearing load:

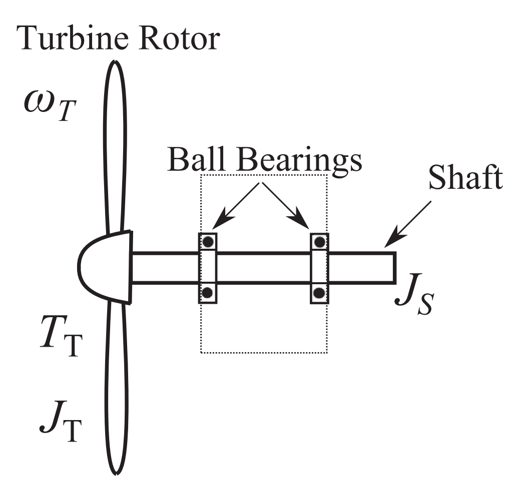

Fig. 1. The rotor and drivetrain.

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