A Hydraulic Wind Power Transfer System: Operation and Modeling

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Abstract—Conventional wind power plants employ a variable speed gearbox to run a generator housed on top of a tower. A new topology can remove some of the weight from the tower and centralize the wind power generation. This new topology uses a hydraulic wind power transfer system to connect several wind turbines to the generation unit. This paper demonstrates a mathematical modeling of this wind power transfer technology and its dynamic behavior. The flow response, angular velocity, and pressure of the system obtained from the mathematical model are compared with test results to demonstrate the accuracy of the mathematical model. Several speed-step responses of the system obtained from the mathematical model demonstrate a close agreement with the results from the prototype of the hydraulic wind power transfer unit.

Index Terms—Hydraulic wind power transfer, mathematical modeling, wind turbine.

I. INTRODUCTION

TILIZATION of renewable energies as an alternative for fossil fuels is growing considerably due to the exhaustion of natural hydrocarbons and the related environmental concerns [1], [2]. Potential sources of renewable energy available around the world, if harvested, can meet all power demands and eliminate the negative effects of fossil fuels [3]. Recent advancements in wind turbine manufacturing have reduced production costs of the wind energy harvesting units and have resulted in the expansion of the application of wind power plants by 30% [5], [6]. Consequently, wind turbines can become one of the major power sources contributing to the world’s energy demands [4]. However, the harvesting technology has remained in its traditional topology. Typical horizontal axis wind turbines include a rotor to convert the wind energy into the shaft momentum [7]. This rotor is connected to a drivetrain, a gearbox, and an electric generator, which are integrated in a nacelle located at the top of the tower. These components, specifically the variable speed gearbox, are expensive, bulky, and require regular maintenance, which keeps wind energy production expensive. In addition, since the gearbox and generator are located on the top of the tower, their installation and maintenance are time consuming and expensive.

Moreover, although the typical expected lifetime of a utility wind turbine is 20 years, the gearboxes require an overhaul within 5–7 years of operation, and their replacements could cost approximately 10% of the turbine cost [8]. Accumulation of the wind energy from several wind turbines in one central unit at the ground level is an innovative solution to address the above deficiencies. In this novel system, each wind turbine harvests wind energy and converts it to a high-pressure fluid. The flows from several wind turbine towers are combined and fed to the central unit. At this unit, the combined fluids are split between a main generator and an auxiliary generator. This technology will eliminate the weight from the tower which reduces the maintenance time and cost. Moreover, instead of having one generator and one variable gearbox for each wind tower, multiple wind turbines are integrated to ultimately reduce the capital costs.

A hydraulic transmission system (HTS) is identified as an exceptional means of power transmission in applications with variable input or output velocities such as manufacturing, automation, and heavy-duty vehicles [9]. It offers fast response time, maintains precise velocity under variable input and load conditions [10], and is capable of producing high forces at high speeds [11]. Moreover, HTS offers decoupled dynamics, allowing for multiple-input, single-output drivetrain energy transfer configurations [12]. Earlier research has shown the possibility of using this type of power transfer technology in a wind power plant, even though it is not feasible in its electrical counterpart [20]–[22], [30].

Simulation tools have been developed for hydraulic circuits [14] and used for modeling and control of turbines [15] and hydraulic transmissions [16]. Closed-loop hydraulic transmission lines have similarly been modeled by the use of governing equations [17], [18] and by modeling fluid compressibility [19]. Mathematical models of HTS wind turbine power plants are required to understand the dynamic behavior of the system, to investigate the performance of the plant, and to improve their design and controls. However, no validated mathematical model is available for the hydraulic transmission of wind power.

This paper introduces a mathematical model of a hydraulic wind power transmission system and demonstrates the performance of its operation at different speed ratios. This model was developed based on the models and governing equations of hydraulic circuit components that include wind-driven pumps, generator-coupled hydraulic motors, hydraulic safety components, and proportional flow control elements. The dynamic operation and step response of the system were modeled and verified with the experimental results gained from a prototype of the wind power plant.
This paper is organized as follows. Section II explains the overall hydraulic power transfer system and its system components. Section III presents the dynamic model of the HTS. A pressure loss calculation model is introduced in Section IV. Section V includes the mathematical model verification with computer simulations, the experimental data, and a discussion.

II. HYDRAULIC WIND POWER TRANSFER SYSTEM

The hydraulic wind power transfer system consists of a fixed displacement pump driven by the prime mover (wind turbine) and one or more fixed displacement hydraulic motors. The hydraulic transmission uses a hydraulic pump to convert the mechanical input energy into pressurized fluid. Hydraulic hoses and steel pipes are used to transfer the harvested energy to the hydraulic motors [16].

A schematic diagram of a wind energy HTS is illustrated in Fig. 1. As the figure demonstrates, a fixed displacement pump is mechanically coupled with the wind turbine and supplies pressurized hydraulic fluid to two fixed displacement hydraulic motors. The hydraulic motors are coupled with electric generators to produce electric power in a central power generation unit. Since the wind turbine generates a large amount of torque at a relatively low angular velocity, a high displacement hydraulic pump is required. The pump might also be equipped with a fixed internal speed-up mechanism. Flexible high-pressure pipes/hoses connect the pump to the central generation unit.

The hydraulic circuit uses check valves to ensure the unidirectional flow. A pressure relief valve protects the system components from the destructive impact of localized high-pressure fluids. The hydraulic circuit contains a specific volume of hydraulic fluid, which is distributed between the hydraulic motors using a proportional valve. In Section III, the governing equations of the hydraulic circuit are obtained.

III. MATHEMATICAL MODEL

The dynamic model of the hydraulic system is obtained by using governing equations of the hydraulic components in an integrated configuration. The governing equations of hydraulic motors and pumps to calculate flow and torque values [20], [25], [30] are utilized to express the closed-loop hydraulic system behavior. Note that all parameters are measured in British Engineering Units. If necessary, appropriate conversion factors were applied when dealing with lbf and lbf/ft as well as rpm and in³/rev.

A. Fixed Displacement Pump

The flow that is delivered by the hydraulic pump is determined by [29]

\[ Q_p = \frac{D_p \omega_p}{\eta_p, \rho_0} - k_{L_p} P_p \]

where \( Q_p \) is the pump flow delivery, \( \omega_p \) is the pump angular velocity, \( D_p \) is the pump displacement, \( k_{L_p} \) is the pump leakage coefficient, and \( P_p \) is the differential pressure across the pump defined as

\[ P_p = P_{tp} - P_{qp} \]

where \( P_{tp} \) and \( P_{qp} \) are the gauge pressures at the pump terminals. The pump leakage coefficient is a numerical expression of the leak probability and is expressed as follows:

\[ k_{L_p} = \frac{K_{HP, p}}{\rho \nu} \]

where \( \rho \) is the hydraulic fluid density and \( \nu \) is the fluid kinematic viscosity. \( K_{HP, p} \) is the pump Hagen–Poiseuille coefficient and is defined as [29]

\[ K_{HP, p} = \frac{D_p \omega_{nom, p} (1 - \eta_{vol, p}) \nu_{nom, p} \rho}{\eta_{nom, p}} \]

where \( \omega_{nom, p} \) is the pump’s nominal angular velocity, \( \nu_{nom} \) is the nominal fluid kinematic viscosity, \( P_{nom, p} \) is the pump’s nominal pressure, and \( \eta_{vol, p} \) is the pump’s volumetric efficiency. Finally, torque at the pump-driving shaft is obtained by

\[ T_p = \frac{D_p P_p}{\eta_{mech, p}} \]

where \( \eta_{mech, p} \) is the pump’s mechanical efficiency and is expressed as

\[ \eta_{mech, p} = \frac{\eta_{total, p}}{\eta_{vol, p}} \]

B. Fixed Displacement Motor Dynamics

The flow and torque equations are derived for the hydraulic motor using the motor governing equations. The fluid leakage within its gears and casing reduces the shaft speed from ideal speed. Thus the hydraulic flow supplied to the hydraulic motor can be obtained by [29]

\[ Q_m = D_m \omega_m + k_{L_m, P_m} \]

where \( Q_m \) is the motor flow delivery, \( D_m \) is the motor displacement, \( k_{L_m} \) is the motor leakage coefficient, and \( P_m \) is the differential pressure across the motor and is expressed as

\[ P_m = P_{tm} - P_{qm} \]

where \( P_{tm} \) and \( P_{qm} \) are the gauge pressures at the motor terminals.
The motor leakage coefficient is a numerical expression of the leak probability and is expressed as follows:

\[ k_{L,m} = K_{HP,m} / \rho \nu \]  

(9)

where \( \rho \) is the hydraulic fluid density and \( \nu \) is the fluid kinematic viscosity. \( K_{HP,m} \) is the motor Hagen–Poiseuille coefficient and is defined as [29]

\[ K_{HP,m} = \frac{D_m \omega_{nom,m}(1 - \eta_{vol,m}) \nu_{nom,m}}{P_{nom,m}} \]  

(10)

where \( \omega_{nom,m} \) is the motor’s nominal angular velocity, \( \nu_{nom,m} \) is the nominal fluid kinematic viscosity, \( P_{nom,m} \) is the motor nominal pressure, and \( \eta_{vol,m} \) is the motor’s volumetric efficiency. Finally, torque at the motor driving shaft is obtained by

\[ T_m = D_m P_m \eta_{mech,m} \]  

(11)

where \( \eta_{mech,m} \) is the mechanical efficiency of the motor and is expressed as

\[ \eta_{mech,m} = \frac{\eta_{total,m}}{\eta_{vol,m}}. \]  

(12)

The total torque produced in the hydraulic motor is expressed as the sum of the torques from the motor loads and is given as

\[ T_m = T_I + T_B + T_L \]  

(13)

where \( T_m \) is the total torque in the motor and \( T_I, T_B, \) and \( T_L \) represent the inertial torque, damping friction torque, and load torque, respectively. This equation can be rearranged as

\[ T_m - T_L = I_m (d\omega_m / dt) + B_m \omega_m \]  

(14)

where \( I_m \) is the motor inertia, \( \omega_m \) is the motor angular velocity, and \( B_m \) is the motor damping coefficient.

C. Hose Dynamics

The fluid compressibility model for a constant fluid bulk modulus is expressed in [19]. The compressibility equation represents the dynamics of the hydraulic hose and the hydraulic fluid. Based on the principles of mass conservation and the definition of bulk modulus, the fluid compressibility within the system boundaries can be written as

\[ Q_c = (V / \beta) (dP / dt) \]  

(15)

where \( V \) is the fluid volume subjected to pressure effect, \( \beta \) is the fixed fluid bulk modulus, \( P \) is the system pressure, and \( Q_c \) is the flow rate of fluid compressibility, which is expressed as

\[ Q_c = Q_{fp} - Q_m. \]  

(16)

Hence, the pressure variation can be expressed as

\[ dP / dt = (Q_p - Q_m) / \beta / V. \]  

(17)

D. Pressure Relief Valve Dynamics

Pressure relief valves are used for limiting the maximum pressure in hydraulic power transmission. A dynamic model for a pressure relief valve is presented in [26]. A simplified model to determine the flow rate passing through the pressure relief valve in opening and closing states [19] is obtained by

\[ Q_{prv} = \begin{cases} \frac{k_v (P - P_c)}{A_{disc} k_s}, & P > P_v \\ 0, & P \leq P_v \end{cases} \]  

(18)

where \( k_v \) is the slope coefficient of valve static characteristics, \( P \) is the system pressure, and \( P_v \) is the valve opening pressure.

E. Check Valve Dynamics

The purpose of the check valve is to permit flow in one direction and to prevent back flows. Unsatisfactory functionality of check valves may result in high system vibrations and high-pressure peaks [27]. For a check valve with a spring preload [28], the flow rate passing through the check valve can be obtained by

\[ Q_{cv} = \begin{cases} C_l \frac{(P - P_v) A_{disc}}{k_s}, & P > P_v \\ 0, & P \leq P_v \end{cases} \]  

(19)

where \( Q_{cv} \) is the flow rate through the check valve, \( C \) is the flow coefficient, \( l \) is the hydraulic perimeter of the valve disc, \( P \) is the system pressure, \( P_v \) is the valve opening pressure, \( A_{disc} \) is the area in which fluid acts on the valve disc, and \( k_s \) is the stiffness of the spring.

The overall hydraulic system can be connected as modules to represent the dynamic behavior. Block diagrams of the wind energy transfer using MATLAB Simulink are demonstrated in Figs. 2 and 3. The model incorporates the mathematical governing equations of individual hydraulic circuit components. The bulk modulus unit generates the pressure of the system.

IV. PRESSURE LOSS CALCULATION

The energy in the hydraulic fluid is dissipated due to viscosity and friction. Viscosity, as a measure of the resistance of a fluid to flow, influences system losses as more viscous fluids require more energy to flow. In addition, energy losses occur in pipes as a result of the pipe friction. The pressure loss and friction loss can be obtained by continuity and energy equations (i.e., Bernoulli’s equation) for individual circuit components such as transmission lines, pumps, and motors [29].

The Reynolds number determines different flow regime of a fluid (laminar or turbulent). It can be used as a design principle
for the system hose sizing. The Reynolds number of a fluid is obtained by

$$
Re = \frac{\rho v L}{\mu} = \frac{v L}{\nu}
$$

(20)

where \( \rho \) is the density of the fluid, \( L \) is the length of the pipe, \( \mu \) is the dynamic viscosity of the fluid, \( \nu \) is the kinematic viscosity, and \( v \) is the average fluid velocity and is expressed as

$$
v = \frac{Q}{A_{\text{pipe}}}
$$

(21)

where \( Q \) is the flow in the pipe and \( A_{\text{pipe}} \) is the inner area of the pipe. The energy equation is an extension of Bernoulli’s equation by considering the frictional losses and the existence of pumps and motors in the system. The energy equation is expressed as [29]

$$
z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}
$$

(22)

where \( z \) is the elevation head, \( v \) is the fluid velocity, \( P \) is the pressure, \( g \) is the earth gravity, \( \gamma \) is the specific weight, \( H_p \) is the pump head pressure, \( H_m \) is the motor head pressure and is calculated by the compressibility equation, and \( H_L \) is the head loss.

The pipe head loss is calculated by Darcy’s equation, which determines loss in pipes experiencing laminar flows by [29]

$$
H_L = \frac{f L v^2}{D D 2 g}
$$

(23)

where \( D \) is the inside pipe diameter, \( v \) is the average fluid velocity in the pipe, and \( f \) is the friction factor and is defined for a pipe experiencing laminar flow as

$$
f = \frac{64}{Re}.
$$

(24)

The energy equation is utilized along with Darcy’s equation and the compressibility equation to calculate the pressure loss at every pipe segment (both horizontal and vertical) and the head of each pump in the system.

V. SYSTEM OPERATION

One benefit of the proposed hydraulic transfer system is having one central generation unit. The key point in this system is the usage of a proportional valve to make this design a possibility. In a conventional wind power plant, individual gearboxes in each wind tower interface the low- and high-speed shafts of the turbine and the generator and provide an amplified linear speed follower mechanism.

In the new design, hydraulic energy transfer in the form of pressurized fluid replaces the gearbox compartment, allowing for energy collection from multiple turbines. This technique utilizes hydraulic pumps and hydraulic motors connected through high-pressure pipes to transfer the energy from the wind turbine to a central ground level generator. Hydraulic valves are utilized to control and regulate the hydraulic flows while propelling a loaded generator at the desired frequency.

Multiple-turbine configuration requires multiple hydraulic pumps driven by individual turbines. The energy from each turbine in this system is transferred through pipes and is collected in a fluid combiner unit at the central ground level power generation unit. In the central power generation unit, there are two sets of main and auxiliary generators. The total fluid generated from the multiple turbines is regulated and sent to the main generators. Excess power is transferred to the auxiliary generator. Fig. 4 illustrates the system configuration of the multiple wind turbines, the energy collection, and the central energy generation unit.

In this configuration, the fluids of multiple mechanically decoupled wind turbines are combined. The speed of each hydraulic pump determines the fluid generation from each wind tower. The fluid combiner collects the fluid power and the proportional valve regulates the output fluid to the main generator to deliver the power determined by the load. The auxiliary motor transfers the excess power to the auxiliary generator.

VI. EXPERIMENTAL VERIFICATION AND DISCUSSION

In this section, the mathematical model behavior is compared with the experimental results obtained from a prototype. The prototype parameters and system values are listed in Table I. Fig. 4 demonstrates an overlay of the experimental setup and hydraulic circuitry. A pulley-belt mechanism is used in decreasing the speed and increasing the dc motor torque needed to run the pump. The system operating conditions, such as angular velocity, flows, and pressures, were precisely measured by fast prototyping dSPACE 1104 hardware.

To demonstrate the accuracy of the mathematical model of the hydraulic wind energy transfer prototype, two distinct test configurations were considered as 1) forced flow distribution and 2) natural flow distribution between the hydraulic motors. These configurations were used to analyze the dynamic system behavior and evaluate the mathematical model performance. In the first configuration, the valve was completely open to direct the flow toward the auxiliary motor (Motor B). Hence, the primary motor (Motor A) was excluded from the hydraulic circuit. In the second configuration, which is also known as the natural flow split, a directional valve was used to distribute the flow generated by the
dc pump between both hydraulic motors, based on the hydraulic circuitry and geometric characteristics of the prototype.

A. Case I: Mathematical Model for Forced Flow Distribution

Fig. 5 shows the schematic diagram of the hydraulic circuit configuration for the forced flow distribution. A pulsed width modulation (PWM) signal of 100 Hz with 10% duty cycle was used to control the proportional valve to direct the flow toward the auxiliary motor. The speed step response of the system was generated by applying a step voltage to the dc motor to accelerate the hydraulic pump from 0 to 300 rpm. After reaching a steady state, a second step voltage was applied to speed up the system from 300 to 400 rpm, followed by a step down back to 300 rpm to analyze the undershoots.

Fig. 6 illustrates the angular velocity profile applied to both the prototype and the mathematical model. Fig. 7 shows the flows of the hydraulic pump obtained from the mathematical model and measured from the prototype. This figure demonstrates the close transient profile and steady-state values obtained from the model and the experiment.

Fig. 8 illustrates the auxiliary motor flows obtained from the mathematical model and the experimental setup. This figure demonstrates a close agreement in the auxiliary motor velocity as a result of step-up and step-down pump velocities and proves the accuracy of the mathematical model. Fig. 9 shows the angular velocity of the auxiliary motor obtained from the mathematical model and the experimental setup. As the figure shows, the
velocities are close. The mechanical system imperfections and dependency on operating pressure resulted in a slight deviation at lower angular velocities. A slight difference between the measured and calculated values was the result of geometrical differences in the prototype and the mathematical model. The experimental results demonstrate the accuracy and performance of the mathematical model of the hydraulic wind energy transfer system when the auxiliary motor is in the circuit.

B. Case II: Mathematical Model of Natural Flow Distribution

In the second configuration, the natural flow split, a directional valve distributed the flow generated by the dc pump (wind turbine) between both hydraulic motors. The flow was naturally distributed between the hydraulic motors through the hydraulic circuitry considering the geometrical characteristics of the prototype, the displacement of the motors, and the position of the flow-control valves, such as check valves and the pressure relief valves. Fig. 10 displays the schematic diagram of the second hydraulic circuit configuration.

In this configuration, a step input voltage is applied to the dc motor to simulate a speed-step input to the hydraulic pump from 0 to 300 rpm. As the system reached the steady-state condition, in about 10 s, another step function was applied to increase the speed from 300 to 400 rpm. A speed step down was also scheduled from 400 to 300 rpm to investigate the accuracy of the mathematical model in determining undershoots. Fig. 11 illustrates the supplied pump angular velocity profile, which is the input to the system. This speed was applied to the mathematical model to investigate the modeling performance. Fig. 12 illustrates both the pump flow measured from the prototype and that of the mathematical model. This figure demonstrates the close flow profiles generated from the mathematical model and the experimental setup both in transient and steady-state conditions.

Fig. 13 illustrates an accurate calculation of the primary motor flow from the mathematical model, which was in good agreement with the prototype. Due to the asymmetries in the hydraulic
circuit and the unmodeled dynamics of the transmission lines, the flow recorded from the auxiliary motor showed a slight shift from what the mathematical model calculated, at 300 rpm. Some of the contributing factors to this dissimilarity are, namely, the effect of the hydraulic circuit, the effect of directional valve dynamics, and the slight displacement and leakage factor variation in the experimental setup. The flow profile of the auxiliary motor is shown in Fig. 14.

Figs. 15 and 16 illustrate the angular velocity profiles of the primary and the auxiliary motors obtained from the mathematical model and the experimental setup. As the figures demonstrate, the speed profiles of the mathematical model and the experimental setup are in close agreement both in transients and in steady-state values. A slight deviation was observed at the starting point transients of the auxiliary motor, which was due to the effects described for flow calculations (Fig. 17). The speed of the auxiliary motor was also influenced by motor inertia and the damping factor.

In summary, the experimental results prove the accuracy and performance of the mathematical model of this innovative hydraulic wind energy transfer system. The transient dynamics of the system and steady-state values were in good agreement with what was obtained from the prototype.

VII. Power Transfer Scaling Challenges

Scaling of the hydraulic wind power transfer system has certain limitations and challenges associated with them. Some of these factors are identified as follows:

A. Power Transfer Distance

Distance between the towers on which the hydraulic pump is located and the motor generator of the central unit is a limiting factor. As the transmission line length is increased, the pressure drop due to frictional losses will increase. Appropriate sizing of the transmission lines is critical to improving the overall efficiency of the power transfer system. Additionally, the number of pumps integrated in the centralized generation unit needs to be optimized based on the amount of frictional losses.

B. Hydraulic Motor/Pump Considerations

1) Power Rating: High power rating hydraulic pumps in the range of several kilowatts operate at low revolutions per minute. For instance, a 700-kW bent axis hydraulic pump operating at full load rotates at 1200 rpm under a load torque of 5.6 kN·m. The pump has a displacement of 60 (in³/rev) which is not capable of running under higher load torques. As the torque capacity of pump is increased, a special radial piston pump is needed, whose operating speed is much less than that of the bent axis pump. For instance, the pump delivering 20−kN·m load can reach top speed 300 rpm when generating 500 kW of power.

In other words, increasing torque and power simultaneously is not possible for all ranges of the power delivery spectrum. Hence, selecting a hydraulic pump with desired speed and torque needs careful optimization.

2) Pump Displacement and Operating Pressure: Since the generator is synchronous, the delivered power by the hydraulic pump is a function of the load torque on the hydraulic pump shaft. The torque is the product of displacement and the operating pressure. Therefore, for the power rating in the order of megawatts, either the required displacement would be very
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