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Variable speed and constant frequency control of hydraulic wind turbine with energy storage system

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Abstract
To eliminate the adverse effect of the fluctuation and intermittence of wind power on the quality and stability of electrical power system, an energy storage system is introduced into the closed-loop hydraulic system of hydraulic wind turbine for the first time. The whole hydraulic system consists of a fixed displacement pump, a variable displacement motor, two proportional control valves and an energy storage system. The energy storage system absorbs or releases oil as the wind fluctuates. When the wind suddenly disappears, the generator can continue to produce electrical energy by means of the discharge of the hydraulic energy stored in the energy storage system. On the basis of modelling all hydraulic components, the simulation model of the hydraulic system is established. A new control method is presented within this article, which keeps the motor speed constant to generate constant frequency electrical power when the rotational speed of the wind wheel changes. Ultimately, simulations under the two conditions of step and sine wind wheel speeds are done. The simulation results demonstrate how the motor, the proportional valves and the energy storage system work together when the wind wheel speed varies and also prove validity of the control method we designed.

Keywords
Hydraulic wind turbine, energy storage system, constant frequency, variable speed, mathematical modelling

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Introduction

Wind power resources as renewable and alternative energy resources have been received attention and concern. The size of wind turbine has been steadily increasing over the last few decades because of the advance on system design and component technology.[1–3] In a conventional wind turbine, the mechanical transmission system passes on the wind energy absorbed by the wind wheel to the generator; the gearbox and generator are installed in the nacelle on the top of the tower. There are following disadvantages to this way: first, the speed-increase gearbox is a key component of wind turbine, which has higher failure rate. With the development of large-scale wind turbine, the gearbox is more apt to fail. Regular repairs are required, which can considerably increase operation and maintenance costs.[4] Second, the structural strength of the tower and nacelle is high enough to support the weight of the wind wheel, the gearbox and the generator. With the increase of wind turbine capacity, the weight sustained by the tower and nacelle greatly increases, which also brings a very large rise in foundation engineering costs.[5]

In order to overcome these shortages, the improvements of traditional wind turbine have been carried out. A newer wind turbine generally eliminates the

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gearbox by replacing it with the direct-drive permanent magnet generator (PMG), which is more expensive and heavier.[6] Recently, the research on replacing the mechanical transmission in wind turbine with hydraulic transmission is becoming the most popular topic in the world. The wind turbine equipped with a hydraulic transmission system is also known as hydraulic wind turbine. In this wind turbine, there is only a hydraulic pump in the nacelle, which is directly connected to the shaft of the wind wheel. The gearbox is removed and the generator is put on the ground. The hydraulic motor on the ground is an execution component of the hydraulic transmission system. The output oil of the pump flows into the hydraulic motor that drives the generator. Thus, the drive component in the nacelle is more compact and light compared to the conventional wind turbine, which reduces tower weight and maintenance costs.

Silva et al. has established the mathematical models of the wind turbine with different drive trains, including the conventional gearbox, direct-drive layout and advanced hydrostatic transmission, to analyse the performance of multi-MW wind turbine and evaluate the application of advanced hydrostatic transmission to the conventional wind turbine. They concluded that hydrostatic transmission can improve system reliability, reduce nacelle weight and cut maintenance costs.[7] C Qin et al.[5] from University of Virginia yielded similar results that hydraulic transmission can significantly reduce the head weight that the tower needs to support. Thus, 33%–50% of the tower mass may be saved through decreasing the shell thickness of the tower. Z Jiang et al.[8] has solved the differential equations of hydraulic wind turbine using the Runge–Kutta–Fehlberg method with step size and error control. They found that hydraulic wind turbine has decent performance under constant and turbulent wind conditions. Y Lin et al.[9] has constructed the joint simulation model of hydraulic wind turbine using the AMESim and MATLAB software. A control algorithm was designed and verified by a 20 kW prototype. The mathematical model of hydraulic wind turbine was verified by Y Zhang and colleagues, using MATLAB and AMESim. A PID control was introduced into the displacement and speed control of the motor to maintain constant motor speed. The simulations and experiments showed that the control system has high-performance speed regulation and great effectiveness.[10]

We all know that wind energy in nature is random and intermittent, which contributes to the unstable output of wind turbine. The fluctuating power from wind farms would debase the quality of the power system. Y Fan et al. has studied that an energy storage system is used in offshore wind turbine with hydraulic transmission system (see Figure 1(a)). They concluded that the combined hydraulic transmission system including an accumulator has good performance and can deliver constant demand power.[11] However, a large number of wind turbines away from the ocean are installed on the land; this kind of wind turbine is called land wind turbine. The closed-loop hydraulic system can be selected to transmit wind energy (see Figure 1(b)). By comparison, the two hydraulic systems are completely different because of working medium (seawater and oil), components (variable and fixed pump, pelton turbine and motor) and working principle (open and closed system). The closed-loop hydraulic system of land wind turbine is more complex, which increases the difficulty of the control system. It seems to be blank that an energy storage system is added to land wind turbine to eliminate the fluctuation and intermittence of wind power from the existing documents. This is the

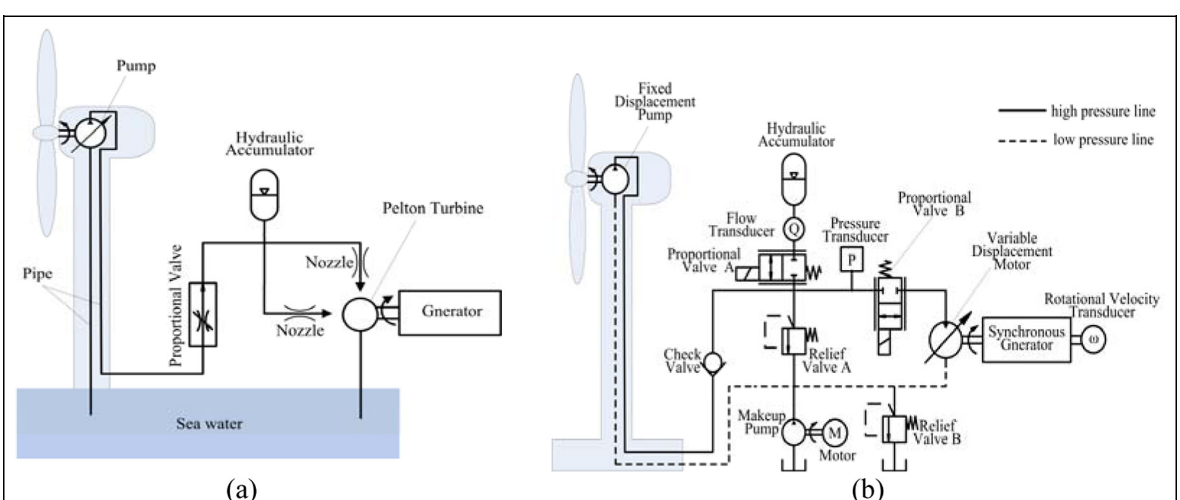

Figure 1. (a) Open system of offshore wind turbine and (b) closed system of land wind turbine.

innovation point in this thesis, which is skilfully combining the closed pump-controlled motor system with a large capacity energy storage system and proposing a novel solution for the constant motor speed.

In the present study, the land wind turbine with closed-loop hydraulic transmission system is used as case studies. An energy storage system is employed to smooth the fluctuation of wind power and provide energy to the motor without wind. To validate the control method of constant motor speed and show the working process of the energy storage system, modelling the whole hydraulic system and simulating under two conditions had been completed. The schematic diagram of the proposed system is shown in Figure 1(b). In section 'System overview', its configuration and working principle are explained. In section 'Mathematical model', the mathematical model of every component is described and the control scheme and strategy are presented in section 'Control scheme'. Time-domain simulations of Micon 600 kW are performed in section 'Simulation and discussion'. Finally, the conclusions are drawn in section 'Conclusion'.

System overview

As shown in Figure 1(b), with the addition of the hydraulic accumulator and proportional valves, the whole system is not a typical pump-control-motor speed regulating system. A fixed pump in the nacelle is directly driven by the wind wheel, through which the hydraulic oil in the low pressure line is sucked and squeezed into the high pressure line, the variable motor and the hydraulic accumulator on the ground. The proportional valve A is used to control the oil flow of the accumulator, avoiding the damage to the pump caused by the shortage of the oil in the low pressure line.

Through comparing the actual speed of the motor with the rated speed of the synchronous generator, the oil flowing into the variable motor is regulated by the proportional valve B. The main functions of relief valve A are adjusting and limiting the maximum operating pressure of the hydraulic system and ensuring the system safety. The makeup pump and relief valve B not only keep the pressure of the low pressure line constant but also complete the heat exchange of the system. The rated flow of the makeup pump is 10%–20% of the maximum flow of the pump.

Mathematical model

In this section, the mathematical models of all parts within this system are established. For the convenience of modelling and analysing the hydraulic system, the dynamic characteristics of the proportional valve, the relief valve, the displacement-varying mechanism of

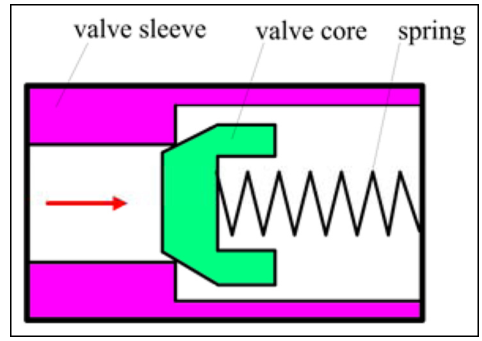

Figure 2. Diagram of the check valve.

the motor be ignored. The pressure-flow characteristics of the check valve, the relief valve are linear.

Fixed pump

In this hydraulic wind turbine, a fixed displacement pump installed in the nacelle is used to transform wind power into hydraulic energy. The pump shaft is directly connected to the wind rotor, which means the rotational speeds of the pump and the wind wheel are equal. The pump flow Q_{pump} and the torque T_{pump} are described as follows[12]

$$Q_{pump} = V_{pump}\omega_{pump}\eta_{pv} \tag{1}$$

$$T_{pump} = V_{pump}(P_{puint} - P_{puout})\eta_{pm} \tag{2}$$

where ω_{pump} and V_{pump} represent the rotational speed and the volumetric displacement of the pump, respectively. P_{puint} and P_{puout} represent the inlet and outlet pressure of the pump, respectively. η_{pv} and η_{pm} represent the volumetric and mechanical efficiencies of the pump, which are generally considered to be constant.

Check valve

The check valve can automatically prevent oil from flowing backwards into the pump, without which the hydraulic system would be very unsafe. The check valve comprises a valve sleeve, a valve core and a spring (as shown in Figure 2). When the difference between P_{cvin} and P_{cvout} is greater than P_{cr}, the valve core moves right and the oil flows through the check valve. If the difference is less than P_{cr}, the oil is turned off. The flow Q_{cv} is computed by the following equation[13]

$$Q_{cv} = \begin{cases} (P_{cvin} - P_{cvout})\, K_{cv} & P_{cvin} - P_{cvout} \geq P_{cr} \\ 0 & P_{cvin} - P_{cvout} < P_{cr} \end{cases} \tag{3}$$

where Q_{cv} is the flow through the check valve. P_{cvin} and P_{cvout} are the inlet and outlet pressure of the check valve, respectively. P_{cr} represents the cracking pressure of the check valve. K_{cr} is the pressure-flow coefficient

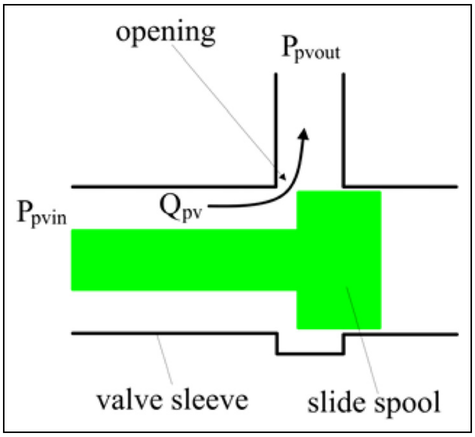

Figure 3. Schematic diagram of the throttle orifice.

of the check valve, which is obtained from the static characteristic curve of the valve that we select.

Proportional valve

The proportional valve plays an important role in electro-hydraulic proportional control system, because of combining fast response, high precision of controlling, easy transmission of electric energy and large density in hydraulic power. Two proportional valves are used in this hydraulic system. One is installed near the hydraulic accumulator, aiming at controlling the charge and discharge flow of the accumulator and avoiding the cavitation of the pump. The other is placed near the motor, in order to maintain the motor speed at constant by adjusting the oil flowing into the motor.

The proportional valve is mainly composed of a slide spool, a valve sleeve and a proportional solenoid. The proportional solenoid is an electrical–mechanical converter, which drives the slide spool by producing mechanical force. The annular section in the valve sleeve and the sharp edge on the slide spool form a typical variable throttle orifice (as shown in Figure 3); its opening area changes with the movement of the slide spool. The relationship among flow Q_{pv}, opening area A, pressure P_{pvin} and pressure P_{pvout} is described in following equation[14]

$$Q_{pv} = C_d A \sqrt{\frac{2(P_{pvin} - P_{pvout})}{\rho}} \tag{4}$$

$$A = \begin{cases} A_{max} I_{pv} & 0 \le I_{pv} \le 1 \\ A_{max} & I_{pv} > 1 \end{cases} \tag{5}$$

where Q_{pv} is the flow through the throttle orifice. C_d is the flow coefficient. It is set to 0.7. A is the actual opening area of the throttle orifice, which is proportional to the input current I_{pv} of the proportional solenoid. A_{max} is the maximum value of A. P_{pvin} and P_{pvout} are the inlet and outlet pressure of the proportional valve, respectively.

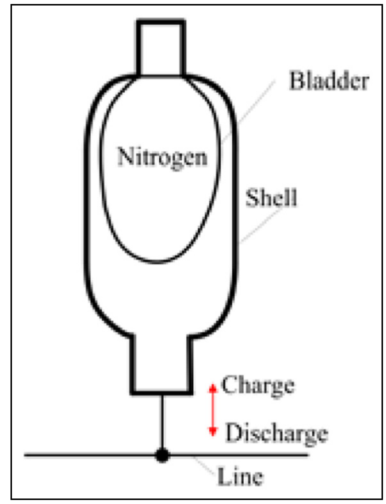

Figure 4. Sketch map of the bladder accumulator.

Hydraulic accumulator

The hydraulic accumulator is utilized in this hydraulic system for storing the surplus oil from the pump. In order to absorb quickly the excess oil and reduce the costs of the experimental prototype, the bladder accumulator is selected. This kind of accumulator has a cylindrical metal shell of which a bladder is fixed at the inner top, as shown in Figure 4.

When the pressure of the oil in the line is greater than the pressure of nitrogen in the bladder, oil is forced into the accumulator. As a result, the pressure of nitrogen rises for the decrease in the volume. Finally, the pressure of nitrogen in bladder is equal to the pressure of the oil outside the bladder. Contrarily, the oil is squeezed into the line by the bladder. The gas in bladder is assumed to obey the ideal gas law; the relationship between variables P and V is expressed by the ideal gas equation[15]

$$PV^n = P_0 V_0^n \tag{6}$$

$$P = \frac{P_0 {V_0}^n}{V^n} = \frac{P_0 {V_0}^n}{(V + \int q dt)^n} \tag{7}$$

where P_0 and V_0 represent the precharge pressure and the nitrogen volume, respectively. P and V are the pressure and volume of nitrogen in the process of compression, respectively. q is the accumulator flow. n is the polytropic index. In an isothermal process, n equals one. In an adiabatic process, n equals one point four.

Relief valve

The relief valve is an important pressure control component in hydraulic system. Its main functions are adjusting and limiting the maximum working pressure or keeping the working pressure constant through putting the excess oil to the tank. The relief valve is consisted of a cone valve core, a valve seat and a control spring. When the force on the valve core is greater than the spring force, the valve core moves away from the valve seat; the oil flows into the tank through the orifice formed by the valve core and valve seat. The flow Q_{rv} is computed by the following equation[16]

$$Q_{rv} = \begin{cases} (P_{rvin} - P_{rvout} - P_{op})\,K_{rv} & P_{rvin} - P_{rvout} \geq P_{op} \\ 0 & P_{rvin} - P_{rvout} < P_{op} \end{cases} \tag{8}$$

where Q_{rv} is the overflow flow through a relief valve. P_{rvin} and P_{rvout} represent the inlet pressure and outlet pressure of the relief valve, respectively. P_{op} is the set value of opening pressure. K_{rv} is the pressure-flow coefficient, which is got through the static characteristic curve of the relief valve.

Variable displacement motor

In this system, for less nacelle weight and convenient maintenance, the variable motor is placed on the ground. The motor shaft is connected to the generator shaft by coupling. In this way, the hydraulic energy from the pump can be converted into the mechanical energy required by the generator. The speed ω_m and torque are described as follows[17]

$$\omega_m = \frac{Q_m \eta_{mv}}{V_m I_m} \tag{9}$$

$$T_m = \frac{(P_{min} - P_{mout}) V_m I_m}{\pi\ 20} \tag{10}$$

where ω_m and T_m are the rotational speed and torque of the motor, respectively. Q_m is the motor flow. V_m is the maximum volumetric displacement of the motor. I_m is the input control current of the displacement-varying mechanism, which is between 0 and 1. P_{min} and P_{mout} represent the inlet pressure and outlet pressure of the motor, respectively. η_{mv} is the volumetric efficiency of the motor, which is regarded as a constant.

Oil line

The pressurized oil is delivered to the motor and the hydraulic accumulator by the oil line. Because the length of the oil line is very large, the line loss is considered in this article. Figure 5 provides an illustration to understand the parameters of the oil line. The flow of the oil in pipeline is designed to be laminar. The line loss is computed by the following equation[18]

$$\Delta P_{op} = \frac{128 \rho v L}{\pi D^4} Q_{op} \tag{11}$$

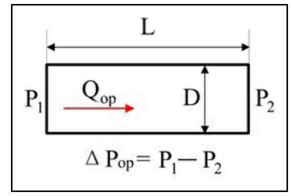

Figure 5. Parameter diagram of the oil line.

where ΔP_{op} is the pressure loss for the line friction and the oil viscosity. ρ and v are the density and velocity of the hydraulic oil, respectively. D and L are the internal diameter and length of the line. Q_{op} is the flow through the oil line.

Pressure transducer and rotational speed transducer

A pressure transducer is used to convert the working pressure to the electrical signal. A rotational speed transducer plays an important role in the constant speed control of the motor, which converts the rotational speed of the motor to the electrical signal

$$U_{pt} = PK_{pt} \tag{12}$$

$$U_{rt} = \omega K_{rt} \tag{13}$$

Here P and U_{pt} are the working pressure and the corresponding voltage value, respectively. K_{pt} is the gain coefficient of the pressure transducer. ω and U_{rt} are the rotational speed of the motor and the corresponding voltage value, respectively. K_{rt} is the gain coefficient of the rotational speed transducer.

Control scheme

To take advantage of the energy storage system, the double closed-loop control including the working pressure of the accumulator and the rotational speed of the motor is proposed and used to keep the motor speed constant. As illustrated in Figure 6, two differences are calculated in this control system: one is the speed difference between the actual speed of the motor and the synchronous speed of 1500 r/min, on which the input current I_{pv} of the proportional valve depends. Another is the pressure difference between the working pressure of the accumulator detected by the pressure transducer P and the designed operation pressure of 200 bar, on which the displacement adjustment of the motor relies. The motor displacement D_1 is calculated according to the function relation between the rotational speed ω and the pump displacement $V_{pump.}$ The displacement D_2 is D_1 plus C_1 that is proportional to the pressure difference. When the switching signal S is greater than 1500, the closed-loop control of the accumulator pressure begins to take effect; D is equal to D_2.

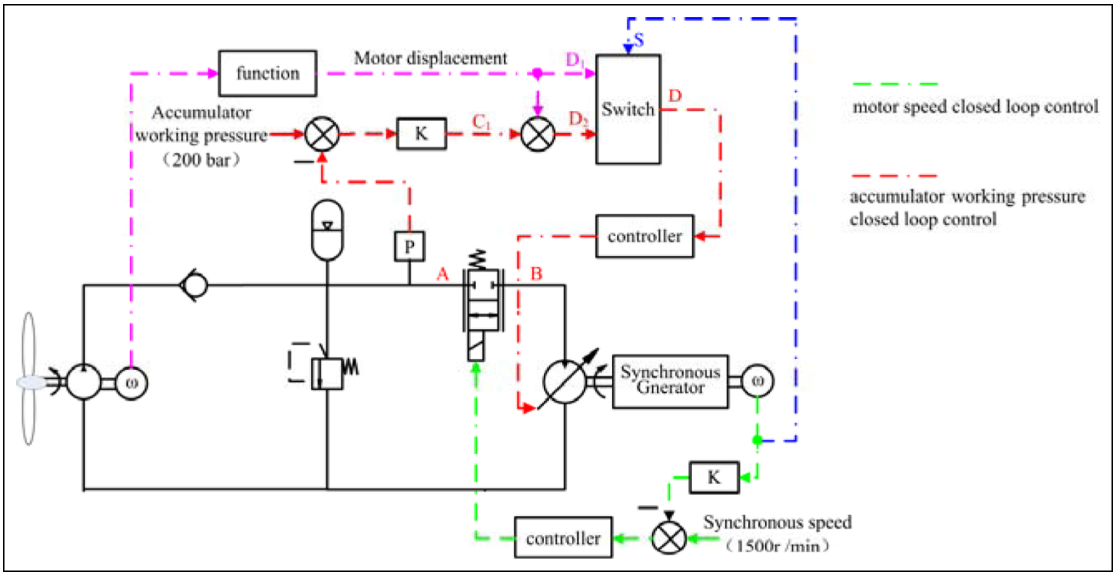

Figure 6. Control schematic of the constant motor speed.

The constant speed control strategy of the motor is shown in Figure 7. If the pump flow is greater than the flow required by the motor, the speed and pressure differences are negative at the same time. The proportional valve opening decreases under the negative speed difference, which raises the inlet pressure of the proportional valve (see point A). The part of the oil from the pump flows into the accumulator because the rise of the pressure at point A. The motor displacement increases by the effects of the negative pressure difference, which reduces the outlet pressure of the proportional valve (see point B). The pressure drop becomes large as a result of the increase of the inlet pressure and the decrease of the output pressure, which brings a corresponding increase in the motor flow. When the two differences are positive simultaneously, the motor displacement and the proportional valve opening change in the opposite direction. Thus, the rotational speed of the motor is always kept at 1500 r/min because the displacement and flow of the motor change in the same proportion.

Simulation and discussion

The design of the experimental prototype

Wind wheel. The wind wheel of Micon 600 kW wind turbine is utilized directly for absorbing wind energy, which is designed by NEG MICON. The gearbox and the generator in the nacelle are replaced by a fixed hydraulic pump. The main parameters of Micon 600 kW are shown in Table 1.

Table 1. Parameters of the Micon 600 kW wind turbine.

Parameter	Symbol	Value	Unit
Hub height	H	40	m
Rotor diameter	D	43	m
Rated wind speed	υ_{rated}	11	m/s
Swept area	A	1452	m^2
Cut-in wind speed	υ_{in}	3.5	m/s
Cut-out wind speed	υ_{out}	25	m/s

Hydraulic pump. Hägglunds CBM 2000 is selected for converting wind energy into hydraulic power, which is especially suitable for wind turbine because of the characteristics of low speed and high torque. It is a radial piston hydraulic motor, which is used as a hydraulic pump. The maximum working pressure of CBM2000 in this hydraulic system is 210 bar. The parameters of CBM2000 are listed in Table 2.

Table 2. Parameters of the CBM2000 in the nacelle.

Parameter	Symbol	Value	Unit
Displacement	v_{pump}	88,279	cm^3/rev
Rated speed	n_{rated}	44	r/min
Specific torque	T_s	1400	N m/bar
Max pressure	P_{max}	350	bar
Max torque	T_{max}	460	kN m

Hydraulic motor. Rexroth A6VM1000 is used to transform hydraulic power into mechanical power. The rotational speed of A6VM1000 is kept at the synchronous speed (1500 r/min) of the generator. The parameters of A6VM1000 are demonstrated in Table 3.

Hydraulic accumulator. We choose HYDAC SB330-200 bladder accumulator as the energy device. Its main

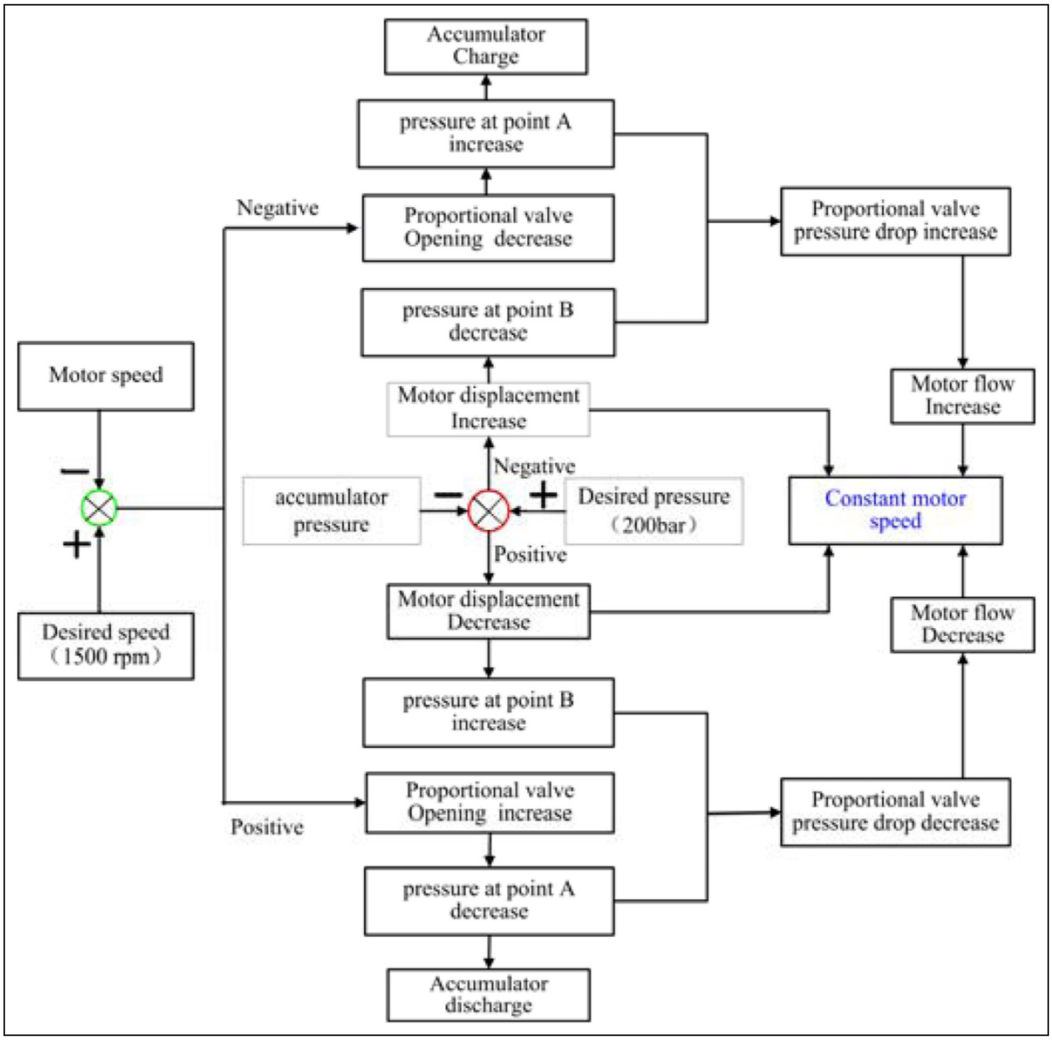

Figure 7. Control strategy of the constant motor speed.

Table 3. Parameters of the A6VM1000 on the ground.

Parameter	Symbol	Value	Unit
Max displacement	V_m	1000	cm^3/rev
Max speed	n_{max}	1600	r/min
Max input flow	Q_{max}	1600	L/min
Nominal pressure	P_{nom}	350	bar
Max torque	T_{max}	5571	N m

Table 4. Parameters of the SB330-200 on the ground.

Parameter	Symbol	Value	Unit
Nominal volume	V	200	L
Gas precharge pressure	P_0	120	bar
Max operation pressure	P_2	210	bar

parameters are described in Table 4. There are 60 SB330-200 bladder accumulators in this experimental prototype because of the cost. When the wind speed is too small to drive the pump, the generator can continue to produce electric energy for a minute with the hydraulic energy stored in the entire accumulators.

Simulation results

In order to verify the effectiveness of the constant speed control and demonstrate the working process of the storage system with the change of the wind wheel speed, the mathematical models in MATLAB software are established and the simulations under two different wind wheel speeds are achieved.

Step wind wheel speed. In this section, a step wind wheel speed (from 14 to 17 r/min) is taken as the input of the hydraulic system, which obtains the performance of constant motor speed control when the wind suddenly increases. The wind wheel speed and the pump output flow are shown in Figure 8. The speed mutation occurs

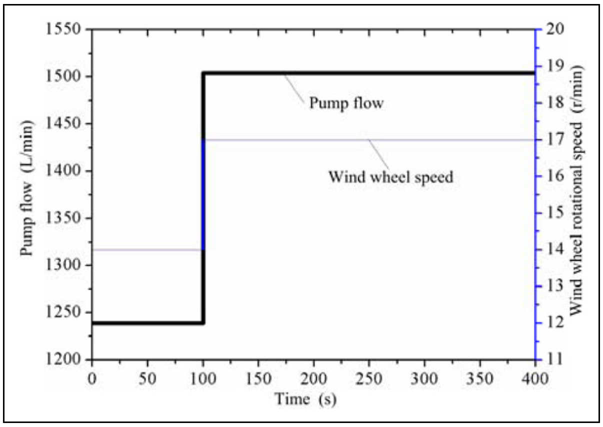

Figure 8. Step wind wheel speed and pump flow.

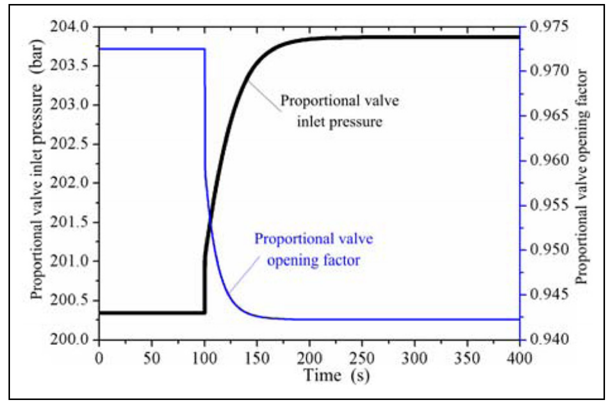

Figure 10. Proportional valve opening factor and inlet pressure.

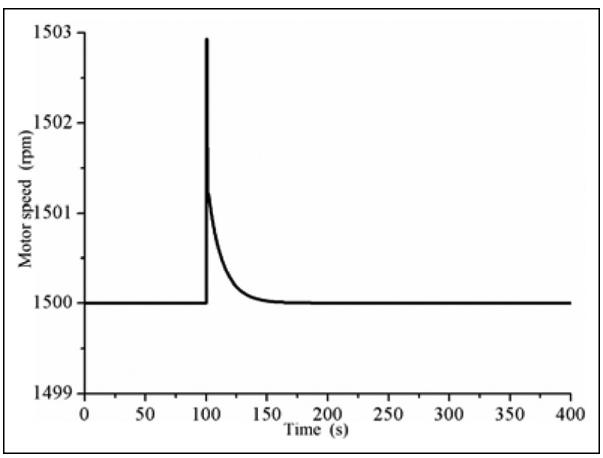

Figure 9. Motor speed.

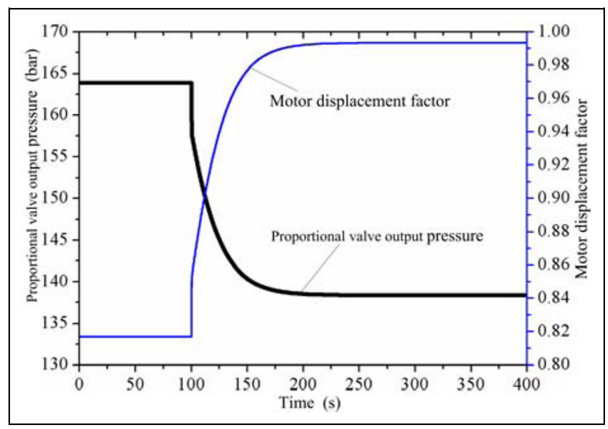

Figure 11. Proportional valve output pressure and motor displacement factor.

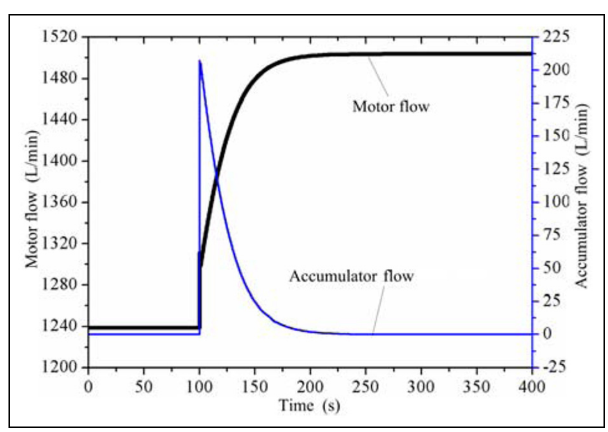

Figure 12. Motor flow and accumulator flow.

at Sec.100. The pump flow increases from 1238 to 1504 L/min accordingly.

Figure 9 shows that the motor speed is transitorily greater than the desired value of 1500 r/min for the sudden increase of the pump flow. The maximum motor speed almost reaches to 1503 r/min, and then it quickly returned to the original value.

From Figures 10–12, it has been shown that the proportional valve opening diminishes from 0.973 to 0.942 times the maximum opening as a result of the negative speed difference. The inlet pressure of the proportional value goes up due to the decrease of the opening and the increase of the pump flow, which squeezes the oil into the accumulator.

From Figure 11, it has been shown that the motor displacement becomes greater on account of the negative pressure difference, from 0.82 to 0.99 times the maximum motor displacement. When the motor output power is constant, the motor operating pressure becomes smaller with the increase of the motor displacement. The accumulator flow and the motor flow are shown in Figure 12.

Sine wind wheel speed. Here, a sine wind wheel speed is applied to simulate the periodic variation of wind speed that frequently occurs. The sine amplitude varies from 14 to 17 r/min and its frequency is 0.1 Hz. Figure 13

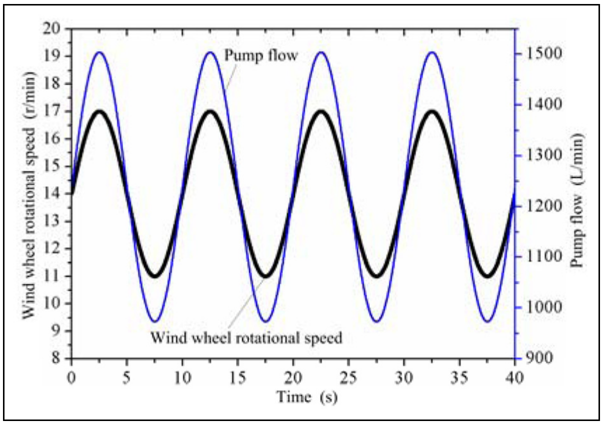

Figure 13. Wind wheel speed and pump flow.

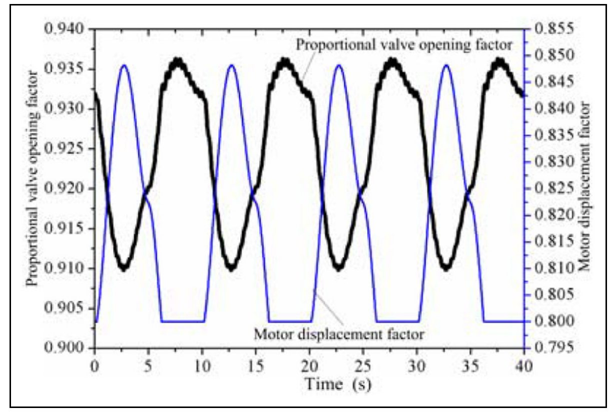

Figure 15. Proportional valve opening factor and motor displacement factor.

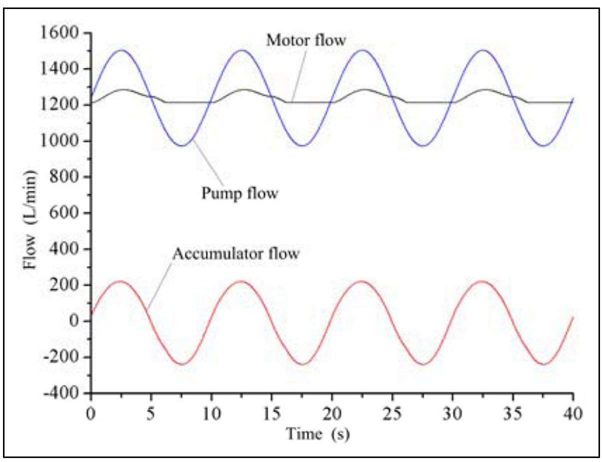

Figure 14. Pump flow, motor flow and accumulator flow.

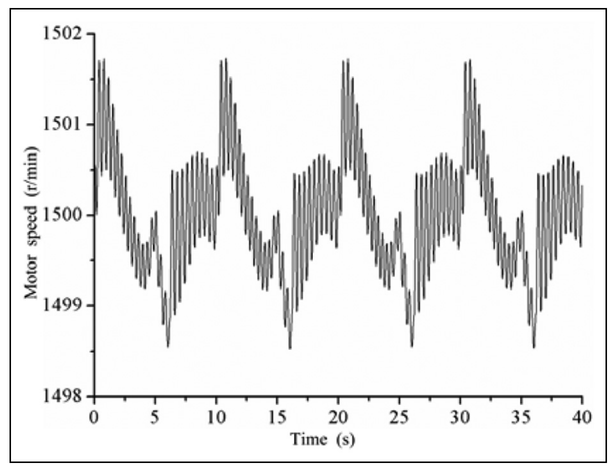

Figure 16. Motor speed.

shows that the pump output flow has the same variety frequency as the wind wheel speed, and the pump flow is 88,279 times the wind wheel speed.

As shown in Figures 14 and 15, during the peak of the pump flow, the displacement and flow of the motor are at their peak simultaneously. The accumulator charge flow also reaches the maximum that is equal to the maximum pump flow minus the maximum motor flow. During the trough of the pump flow, the motor flow is held at 1200 L/min because the motor displacement factor is restricted to the minimum value of 0.8. The accumulator discharge flow supplies the lack of the pump flow due to the lower wind speed.

From Figure 16, it has been shown that the motor speed slightly fluctuates up and down around 1500 r/min, and the range of the speed error is less than 2 r/min, which meets the demand of the constant frequency power produced by the generator.

Conclusion

An energy storage system is introduced to land hydraulic wind turbine, which is a closed pump-control-motor hydraulic system. With Micon 600 kW wind turbine as a test object, the hydraulic principle diagram is designed, and the calculation and selection of the main components are completed. The MATLAB mathematical model of the whole hydraulic system is constructed. A double closed-loop control method of the motor speed and the accumulator pressure is employed to satisfy the demand on the constant speed of the synchronous generator regardless of the fluctuation of wind speed. The responses under step and sine wind wheel speeds are simulated and analysed. The simulation results demonstrate that the constant speed control system of land hydraulic wind turbine with an energy storage system has fast response and very little steady-state error. Meanwhile, the accumulator working pressure is kept at around 200 bar throughout by adjusting the proportional valve opening and the motor displacement. In this way, the energy storage system is always ready to provide hydraulic energy to the motor without wind, which significantly improves the quality and stability of electric energy from wind turbine.

Declaration of conflicting interests

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References

1. GWE Council. *Global wind energy outlook*, 2014, http://www.gwec.net/wp-content/uploads/2014/10/GWEO2014_WEB.pdf
2. Ackermann T. *Wind power in power systems*. Chichester: John Wiley & Sons, 2005.
3. Musial W and Ram B. *Large-scale offshore wind power in the United States: assessment of opportunities and barriers*, 2010, http://www.nrel.gov/docs/fy10osti/40745.pdf
4. Ragheb A and Ragheb M. Wind turbine gearbox technologies. In: *Proceedings of the 1st international nuclear and renewable energy conference (INREC10)*, Amman, Jordan, 21–24 March 2010. New York: IEEE.
5. Qin C, Innes-Wimsatt E and Loth E. Hydraulic-electric hybrid wind turbines: tower mass saving and energy storage capacity. *Renew Energ* 2016; 99: 69–79.
6. Polinder H, van der Pijil FFA, de Vilder GJ, et al. Comparison of direct-drive and geared generator concepts for wind turbines. *IEEE T Energy Conver* 2006; 21: 725–733.
7. Silva P, Giuffrida A, Fergnani N, et al. Performance prediction of a multi-MW wind turbine adopting an advanced hydrostatic transmission. *Energy* 2014; 64: 450–461.
8. Jiang Z, Yang L, Gao Z, et al. Numerical simulation of a wind turbine with a hydraulic transmission system. *Enrgy Proced* 2014; 53: 44–55.
9. Lin Y, Tu L, Liu H, et al. Hybrid power transmission technology in a wind turbine generation system. *IEEE/ASME T Mech* 2015; 20: 1218–1225.
10. Chang GA, Zhang Y, Kong X, et al. Controls of hydraulic wind turbine. *MATEC Web Conf* 2016; 40: 08002.
11. Fan Y, Mu A and Ma T. Study on the application of energy storage system in offshore wind turbine with hydraulic transmission. *Energ Convers Manage* 2016; 110: 338–346.
12. Merrit H. *Hydraulic control systems*. Chichester: John Wiley & Sons, 1967.
13. Izadian A, Hamzehlouia S, Deldar M, et al. A hydraulic wind power transfer system: operation and modeling. *IEEE T Sustain Energ* 2014; 5: 457–465.
14. Valdés JR, Miana MJ, Núñez JL, et al. Reduced order model for estimation of fluid flow and flow forces in hydraulic proportional valves. *Energ Convers Manage* 2008; 49: 1517–1529.
15. Moran MJ, Shapiro HN, Boettner DD, et al. *Fundamentals of engineering thermodynamics*. New Delhi, Wiley India Pvt Ltd, 2010.
16. Akkaya AV. Effect of bulk modulus on performance of a hydrostatic transmission control system. *Sadhana* 2006; 31: 543–556.
17. Esposito A. *Fluid power with applications*. New Jersey: Prentice Hall International, 2000.
18. Hunt BW. *Fluid mechanics for civil engineers*. Christchurch, New Zealand: Department of Civil Engineering, University of Canterbury, 1995.