International Journal of Applied Engineering Research
ISSN 0973-4562 Volume 7, Number 4 (2012) pp. 383-396

http://www.ripublication.com/ijaer.htm

Development of a Hydraulic Brake Control System for 100kW Horizontal Axis Wind Turbines using Pressure Relief and Directional Valves

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Abstract

In order to satisfy specification required by standard a hydraulic brake control system is developed to be implemented to a 100 kW horizontal axis wind turbine under construction. The hydraulic brake control system consists of mechanical brake, hydraulic system and controller. The brake controller has two control modes i.e. normal braking and emergency braking. The hydraulic brake was designed to be able to stop the wind turbine from critical condition i.e. wind speed 21 (m/s) and turbine rotor speed 45 rpm without any help of other means. From calculation it was obtained that torque at the brake be 1.49 kNm. The brake has brake disc whose diameter is 61 cm. The hydraulic system is constructed by a hydraulic pump, a relief valve, a directional valve and three calipers fitted with pads each having area of 37.2 cm^2. A brake controller sends on/off signal to activate/deactivate pump and directional valve. From the experimental results using test bed in the laboratory it is concluded that when initial rotational speed is 600 rpm the mechanical brake could stop the rotating mass in 1.2 second under hydraulic pressure setting 18 bar and 1 second under pressure setting 30 bar. From numerical analysis result it is estimated that when the hydraulic brake is installed in the real wind turbine it can stop the turbine from the critical condition in 1,1 second using hydraulic pressure setting 18 bar with deceleration of 283.8 rpm/sec just before stopping.

Keywords: Horizontal Axis Wind Turbine, Hydraulic Brake, Brake Controller, Relief Valve, Directional Valve.

Introduction

The IEC 61400-1 international standard requires specification for a braking system used in a wind turbine as follows [1]: “The braking system shall be able to bring the rotor to idling mode or complete stop from any operation condition. Means shall be provided for bringing the rotor to a complete stop from a hazardous idling stage in any wind speed less than the wind speed limit defined for maintenance and repair”. The standard also states [1]: “It is recommended that at least one braking system operates on an aerodynamic principle, as such acting directly on the rotor. If this recommendation is not met, at least one braking system shall act on the rotor shaft or on the rotor of the wind turbine”. A braking system must be able to maintain the turbine stop completely for the defined wind conditions for at least one hour after it is applied, and for longer periods of grid loss, it can be applied by an auxiliary power supply or by manual operation [1].

Many commercial wind turbines available in the international market provide some mains of braking systems. Some of them incorporate mechanical brake on the shaft while some others do not. It is describe in [2] “The Northern Power 100® uses a main shaft braking system consisting of two caliper brakes which can be motor applied for normal braking and are fail safe in emergency conditions. In addition to the two mechanical brakes, the turbine includes an electro dynamic brake that is incorporated into the power converter. The turbine may be stopped under any circumstance by using any two of the three brakes”. The Enercon’s wind turbine E33, whose rated power is 330kW, is equipped with a brake system for maximum turbine reliability using three independently operated pitch mechanisms [3]. This pitch mechanism brake system is supported by a standby power supply (batteries) in case of supply failure. The Nordex S77/1500 (1.5MV) has three independent electromotive blade pitch brakes as its operational brake and a disc brake as its secondary brake [4]. The FD20-100/12 wind turbine whose rated power is 100 kW has a mechanical brake, an electromagnetic brake, and a yaw regulating for fulfilling protection specification [5]. The WES30 (80kW) is equipped with mechanical passive blade pitch for first safety system and active yawing out of the wind for second safety system but it is not equipped with any mechanical brake [6].

In this research, a mechanical-electrical control system for a 100 kW horizontal axis wind turbine electrical power generation plant has been designed. This wind turbine has 3 blades without any blade pitch control mechanism. The mechanical-electrical control system is constructed of a mechanical brake controller, a yaw controller and an electrical torque controller. All these 3 controllers are coordinated by a supervisory controller. This paper described the development of the brake control system used in the wind turbine.

During a grid failure when the energy cannot be fed into the grid, the rotating turbine rotor must be stopped by a mechanical brake. The braking torque must be higher than the aerodynamic torque regardless of wind speed and the brake must be able to absorb the energy which is stored in the rotating masses as well as the energy which is taken from the wind [7]. To reject input equivalent pressure disturbances and to cancel disturbance due to disc irregularities, a pressure controller with disturbance estimator can be used [8]. Such a controller has been implemented on a laboratory-sized test bench using a proportional valve.

Designing a mechanical brake working on the shaft of a three blades horizontal axis wind turbine requires considerations of some mechanical parameters values. Turbine and generator inertia time constants, turbine and generator self damping, torsional stiffness of the flexible coupling between turbine rotor and generator, and mutual damping of the changes in the twist angle of the coupling can be estimated [9].

In this research, a hydraulic brake having a pressure relief valve and a solenoid operated directional valve has been built. The maximum operating pressure of the brake can be set by rotating the knob of the relief valve while the acting duration time of the directional valve is controlled by on/off signal. A brake controller was designed to control the pressure and the acting duration of the brake. Before installation in the real wind turbine power plant, the braking control system was tested in the laboratory. This paper reports design method and experimental results to assess the function of the hydraulic control system.

Development of Hydraulic Brake & Its Controller

In this research a hydraulic brake control system for a 100 kW wind turbine electrical power generation plant is built. This hydraulic brake is designed to act both as a parking brake and an emergency brake. The parking brake stops the rotation of the turbine for maintenance purpose according to normal shut down procedure. In the normal shut down procedure, initially the yaw controller rotates the nacelle to decelerate the turbine rotor and then the brake stops the rotor. The emergency brake works when an over speed is detected due to electrical load trip, failure of the power converter, malfunction of the yawing system, and other failures. The hydraulic brake was designed based on emergency brake specification.

The key parameter in brake design is the brake design torque. After considering some factors including variation of coefficient of friction, possible loss of caliper spring force, aerodynamic load factor and thermal rise margin, the minimum design braking moment is 1.87 times the maximum aerodynamic torque [10]. Figure 1 shows the braking procedure used in this paper. The brake is fixed in the high speed shaft (HSS) and gear ration between HSS and low speed shaft (LSS) is 31.5. ω_1 represents the rated rotational speed of the HSS, ω_2 denotes the rotational speed of the HSS after a grid loss when the brake is just activated, and ω_3 is the rotational speed when the brake start giving its effect. There is assumed brake application delay time. The cut out wind speed (21 (m/s)) is chosen as the reference wind speed for torque calculation. The HSS rotational speed is set as follows: ω_1=40 rpm x 31.5=1260 rpm, ω_2 =(1,1 x 40 rpm) x 31.5=1386 rpm, and ω_3 =(44 +1)*31.5=45 x 31.5=1418 rpm. The aerodynamic torque at 21 (m/s) wind speed and 45 rpm rotational speed of LSS is 25kNm. Hence the brake design torque is 46.8 kNm referred to the LSS, or equivalently 1.49 kNm at the brake.

The maximum rotor speed corresponds to the HSS is 1418 rpm = 148 rad/s. So the maximum permissible brake-disc radius as regards centrifugal stresses is about 90/148 =0,61 m =61 cm [10]. Therefore, 61cm diameter is selected.

The total brake-pad area is governed by the need to keep the maximum power dissipation per unit pad area below 11.6MW/m^2 [10]. The power dissipation is equal to the product of the braking torque and the rotational speed, so it is at maximum at the onset of braking i.e. 1.49 x 148 = 220.5 kW, giving a required total area of the brake pads of 220,4/11600 = 0,019 m^2. This area can be provided by three calipers fitted with pads each having area of 37.2 cm^2, giving total area of 0.0223 m^2.

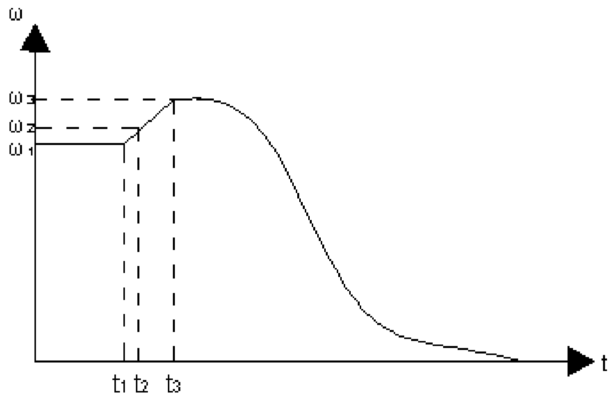


Figure 1: Brake Application Procedure used in this Paper.

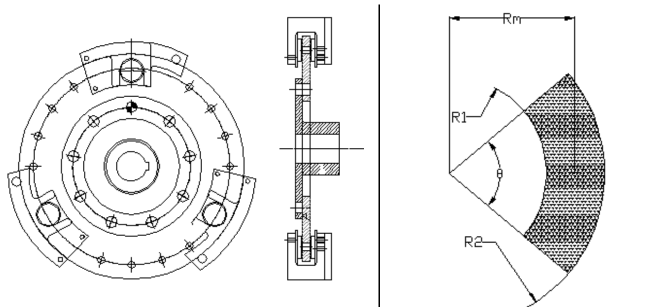
Figure 2 shows technical figure of the designed brake. Figure 2 (a) is assembly composed by brake disc, brake hub, caliper and brake pad. The outer diameter of the brake disc is 61 cm, and its thickness is 2 cm. The brake disc is coupled to the rotating shaft through the hub. At steady state condition, total brake moment τ_{BT} is given by the following equations.

$$\tau_{BT} = \mu F_{BT} R_B K_B \qquad (1)$$

where:

$$R_B = \frac{(R_1 - R_2)}{2}; \; K_B = \frac{2\theta}{3Sin\,(\theta/2)}\left[1 - \frac{R_1\,R_2}{(R_1 + R_2)^2}\right]$$

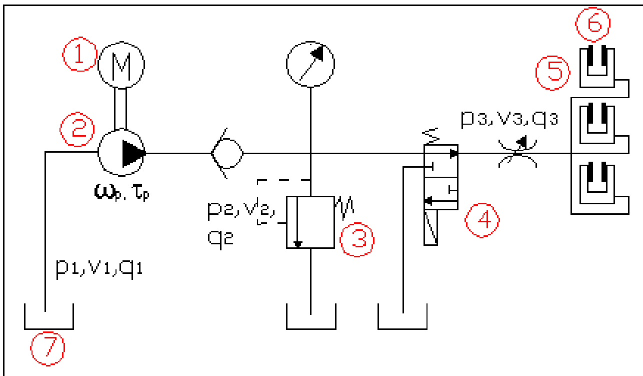
μ, F_{BT} and R_B denotes friction coefficient of the brake pad, total brake pressing force and effective radius, respectively. Parameters $R_{1,}R_2$ and θ are illustrated in figure 2 (b). In this paper by defining material and dimension, the above parameters are set as follows: $\mu = 0.45$, $R_1 = 23.6$cm, $R_2 = 29.2$ cm, $R_B = 26.38$ cm, $\theta = 16.8$ and $K_B = 0.5$. Hence, the designed total brake pressing force is 12.55kN.



Brake Disc and Hub Brake Pad Notation

Figure 2: Design Of Brake Disc, Hub and Brake Pad

The brake pressing force is produced by a hydraulic system. Figure 3 shows diagram of the hydraulic system used in this research. A braking cycle includes pressing stage, holding stage and decompressing stage. In the pressing stage the hydraulic pump is activated to force the oil to the relief valve. An excess pressure will cause part of the oil be delivered back from the relief valve into the reservoir tank. The directional control valve is switched to parallel mode in order to pass the oil from the relief valve to flow into the caliper and then pushes the piston in the caliper to grip the brake disc by the pads. The holding stage starts soon after the shaft stops rotating and the pressure at the caliper reaches its maximum value set by the relief valve. The holding stage is maintained until the brake is released. In the decompressing stage, the hydraulic pump is deactivated and at the same time the directional valve is switched to cross mode. The piston in the caliper is kicked back to its normal position by the caliper spring and oil flows both from the caliper and the pump to the reservoir tank.



Sub system	Components	
Pressure source	1	Electrical motor
	2	Hydraulic pump
	3	Pressure control valve (relief valve)
Control	4	Directional control valve
Load	5	Caliper
	6	Brake pad
Reservoir	7	Oil tank

Figure 3: Design of Hydraulic System for Mechanical Brake.

A three phase asynchronous electrical motor MX motori is used as prime mover of the hydraulic system. Its specification is: 1.1kW, 1415 rpm, 380VAC, 50 Hz. The motor shaft is coupled by an elastic coupling to the shaft of hydraulic pump. The hydraulic gear pump JTEKT HPI AAN 0200 is used. It has the specification as follows: capacity 2 cc/rev, peak pressure 125 Bar, maximum working pressure 105 Bar, maximum speed 5000 rpm, nominal flow at maximum speed 10 L/min, nominal flow at 1500 rpm 3 L/min, input power at 1000 rpm and 100 bar is 0.3 kW, and input torque at 100 bar is 0.32 Nm. At the inlet of the pump, the minimum pressure is 0.8 bar (0. 2 bar under the air pressure) and the maximum pressure is 1.2 bar (0.2 bar over the air pressure) [11].

A relief modular valve MBA-01-C0.3-14-30 with maximum operating pressure 21 MPa and maximum flow 35 L/min is used as pressure control valve [12]. Its pressure difference lays between 0 MPa and 0.45 MPa. When the pressure difference is 0 MPa the flow becomes minimum (0 L/min) and when the pressure difference is 0.45 MPa the flow becomes maximum (35 L/min). The spool of the valve is positioned by a spring. Pressure adjustment is done by loosening the lock nut and turning the pressure adjustment screw. The lock nut must be re-tighten after making pressure adjustment. The adjustable minimum pressure is 18 bar.

A "two-positions spring-offset solenoid operated directional valve" DSG-01-2B2-A220-50 with maximum operating pressure 35 MPa, maximum flow 85 L/min and maximum T-line back pressure 21 MPa is used as directional control valve [13]. It can be opened and closed by connecting and disconnecting electrical source 220 VAC, 50 Hz, 1.01 A inrush current and 0.21 A holding current. Its spool position change over time from OFF to ON is estimated less than 15 ms and from ON to OFF is estimated less than 23 ms. The piston diameter is 5.5cm. At each caliper, two spiral springs are used each has spring coefficient of 21.77778 N/mm or 21777.78 N/m.

Two pressure transmitters of type Best NR 0610-481-03-0-001 are used to measure hydraulic pressure at the outlet of the directional valve and at one of the caliper inlet. Its specification is as follows: output signal 0-10 VDC, supply voltage 12-31VDC, pressure range 0-100 bar, maximum pressure 200 bar and response time (10-90%) is 2 ms or less [14].

The block diagram of the hydraulic brake control system is illustrated in figure 4. The brake controller C_B receives wind speed signalV_W, yaw angle signal θ_Y, rotational speed C_r, and control mode signal C_M . It sends on/off signal to the hydraulic pump and the directional valve. When using a proportional pressure control valve, it also sends pressure command θ_V.

This paper focuses on dynamical characteristics of the hydraulic brake system using on/off control. Analysis is carried out to observe experiment results using laboratory scale test bed. Dynamical response of the real wind turbine is then estimated based motion equations derived in the following.

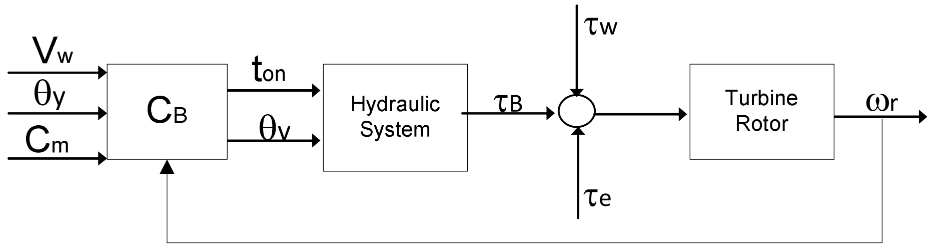


Figure 4: Block Diagram of the Hydraulic Brake Control System.

By assuming that the fluid in the hydraulic system is incompressible and there are no losses inside the hydraulic pump, from energy balance principle the following equation holds.

$$\tau_p = \frac{V_p}{2\pi} \cdot (p_2 - p_1) \tag{2}$$

$$q_p = \frac{V_p}{2\pi} \cdot \omega \tag{3}$$

p_1 and p_2 represents pressure just before the pump and after the pump, respectively. τ_p is the torque exerted by the motor to the pump, V_p is the ideal volume per one rotation, ω_p is the rotational speed of the pump and q_p is the flow rate of the pump.

The maximum pressure p_{2max} is set by the relief valve. Some fluid is passed by the relief valve into the directional valve and part of the fluid is delivered back to the reservoir tank.

$$q_2 = q_p - q_{rv} \tag{4}$$

q_2 and q_{rv} represents flow rate entering the directional valve and flow rate returned back to the tank.

By assuming that the flow is non-viscous, steady and incompressible the Bernoulli equation holds at the orifice in the directional valve. Since average speed leaving the orifice in the directional valve v_3 is much larger than the average speed entering the orifice the following relation holds.

$$v_3 = \sqrt{\frac{2}{\rho}(p_2 - p_3)} \xrightarrow{yields} q_3 = A_{eff}\sqrt{\frac{2}{\rho}(p_2 - p_3)} \tag{5}$$

Where: p_2 and p_3 denotes pressure before the orifice and after the orifice, respectively. A_{eff} is effective cross sectional area of the orifice, ρ is the mass density of the hydraulic fluid, and q_3 is the flow rate of the fluid.

After leaving the directional valve, the fluid enters into the cylinder chamber at the caliper. Deriving from the law of conservation of mass, dynamics of the pressure in the cylinder chamber p_3 is governed by the following equation.

$$\dot{p}_3 = \frac{\beta}{A_{pc}x}\left(q_3 - A_{pc}\dot{x}\right) \tag{6}$$

A_{pc} is the cross sectional area of the piston in the cylinder chamber, x is the position of the piston, and β is the isothermal bulk modulus of the fluid. The motion equation of the piston is given by the following equation.

$$\left.\begin{array}{r} m_p\ddot{x} + d_p\dot{x} + k_p x = A_{pc}p_3; \forall x < L_p \\ k_p L_p = A_{pc}p_3 - F_B; \forall x = L_p \end{array}\right\} \tag{7}$$

m_p, d_p, k_p denotes the mass of the piston, the damping coefficient, and the spring coefficient, respectively. L_p is the maximum stroke of the piston and F_B is the force pressing the brake pad to the brake disc in the caliper.

Eventually the pressing force gives effect to the brake rotating mass. Assuming there exists elastic coupling between brake and turbine having spring coefficient K_g, the following motion equations hold.

$$J_B\ddot{\theta}_B + D_B\dot{\theta}_B + K_g(\theta_B - \theta_T) = -\tau_B \tag{8}$$

$$J_T\ddot{\theta}_T + D_T\dot{\theta}_T + K_g(\theta_T - \theta_B) = \tau_T \tag{9}$$

J_* and D_* denote moment of inertia and damping coefficient, respectively. θ_* and τ_* represent rotational angle and torque. Note that all parameters and variables are referred to the high speed shaft.

Transfer function from input voltage u to the position of the spool in the solenoid operated directional valve x_s can be expressed as follows.

$$\frac{X_s(s)}{U(s)} = \frac{G_s}{(\tau_m + s)(s^2 + 2\zeta\omega_n s + \omega_n^2)} \tag{10}$$

where:τ_m is magnetic time constant, G_s is gain constant, ω_n is the natural frequency, and ζ is damping coefficient. From the datasheet of the directional valve used in this paper, it can be estimated that change over time from solenoid OFF to solenoid ON is less than 15 ms and from solenoid ON to OFF is less than 23 ms [13].

From datasheet and measurement results of components of the hydraulic system some parameters are known and some other parameters and variables can be calculated. In the followings pressure is in Pa and flow rate is in L/min. Equation of the pump becomes as follows.

$$\tau_p = 0.032 \times 10^{-5} \times p_2 \text{ (Nm)} \tag{11}$$

$$q_p = 0.002 \cdot \omega \text{ (L/min)} \tag{12}$$

where: ω is in rpm.

Concerning the orifice and cylinder chamber, the following equations are obtained.

$$q_3 = 0.068\sqrt{(p_2 - p_3)} \leq 10 \text{ (L/min)} \tag{13}$$

$$(p_2 - p_3) \leq 21626.30 \text{ (Pa)} \tag{14}$$

$$\dot{p}_3 = \frac{343.2\times10^5}{x}(16.67q_3 - 23.746\dot{x}) \text{ (Pa/sec)} \tag{15}$$

Position of the piston x is in cm and velocity of the piston $\dot{x}$ is in cm/sec. When the pad starts touching the brake disc it is assumed $x = 1\text{cm}$ and $\dot{x} = 0$. Thus the following equations are obtained.

$$\dot{p}_3 = 5719{,}71 \times 10^5 \times q_3 \text{ (Pa/sec)} \tag{16}$$

$$F_B = 0.002375p_3 - 353 \text{ (N)} \tag{17}$$

For simplicity, it is assumed that the coupling between the brake shaft and the turbine emulator in the testing apparatus is rigid so that the rotating mass motion equation is simplified as follows.

$$J_T\ddot{\theta}_B + D_T\dot{\theta}_B = \tau_T - 0.712F_B \tag{18}$$

p_3 and $\dot{\theta}_B$ were measured against time t (sec) in the experiment so that $\dot{p}_3$, $\ddot{\theta}_B$ and F_B can be directly calculated. In turn, q_3, q_2 and τ_p can be estimated.

From the above derivation, theoretically it is clear that dynamical characteristics of the hydraulic pressure at the caliper (p_3), is affected by torque produced by the electrical motor of the pump (τ_p), rotating mass properties, and external torque (τ_T). All these variables also have effect on rotational speed of the brake disc (ω_B).

Experimental Results and Analysis

To evaluate performance of the hydraulic brake made in this research, a Functional Analysis Test (FAT) has been conducted in the laboratory. Figure 5 shows photos of the experimental set up in the laboratory. This set up is constructed by main elements such as: mechanical brake, electronic brake controller, wind turbine emulator, and monitoring computer. The wind emulator is realized using an induction motor driven by a speed control inverter. The mechanical brake shaft is coupled to the wind turbine emulator shaft. The mechanical brake consists of brake disc, calipers, hydraulic power pack, and electrical control panel. The electronic controller includes a rotary encoder, 2 pressure transducers, a digital signal processor (DSP), and control valve. The monitoring computer is connected to the electronic controller and it is used to carry out data logging.

The induction generator used as turbine emulator has the specification as follows: 11kW, 50 Hz, 380 VAC, 21.5 A, 2910 rpm, and PF=0,91. The rotational speed of the emulator is controlled by the inverter. When the brake takes action, the inverter will manipulate current to maintain the speed yielding rise in the current. To protect the induction motor from over current an MCB MY320E C20 is used to limit the maximum current be 20 A.

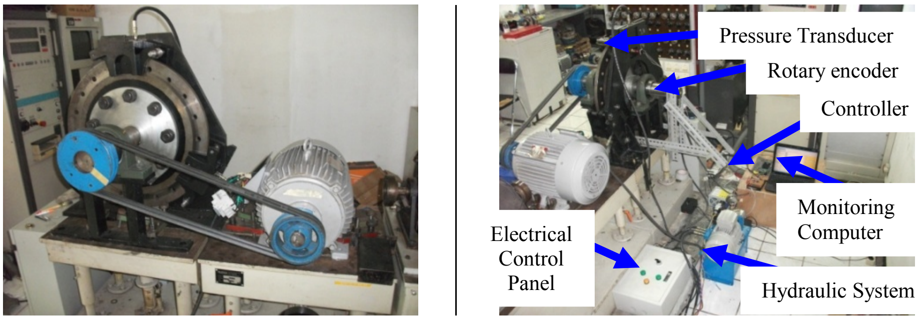


Figure 5: Photos of the Experimental Set up In the Laboratory.

Rotational speed of the brake shaft and hydraulic pressure are measured against time for different settings of initial rotational speed and maximum pressure setting. Since the wind turbine emulator has capacity of 11kW, the maximum rotational speed which can be realized in the experimental set up is limited only 600 rpm. 9 combinations of experiments were conducted as shown in table 1. Only experiment results with the maximum pressure setting of 18 bar and 30 bar are reported in this paper.

Table 1: Combination of experiment

Parameter	Experiment								
	1	2	3	4	5	6	7	8	9
Maximum pressure setting (Bar)	18	18	18	30	30	30	40	40	40
Initial speed (rpm)	200	400	600	200	400	600	200	400	600

Figure 6 and figure 7 show experimental results when the maximum hydraulic pressure is set to 18 bar and 30 bar, respectively. Horizontal axis indicates time in second, left hand side vertical axis expresses rotational speed of the brake shaft in rpm, and right hand side vertical axis denotes pressure in bar. Number in the legend implies initial value of rotational speed of the brake shaft (200 rpm, 400 rpm, 600 rpm), while the following letters implies value expressed by the curve weather it is real measured rotational speed (RW), real measured pressure at the outlet of the directional valve (RP1), or real measured pressure at the caliper before the piston chamber (RP2).

From these results the following qualitative information is obtained:

1. Higher value of initial rotational speed when the brake is activated results in larger delay time of the response of the caliper pressure. It can be explained using equation (18),
2. Larger value setting of maximum pressure yields faster response of the caliper pressure.

3. It is clear that longer time is required to completely stop the rotating shaft when initial rotational speed is higher. This delay time depends on both the hydraulic system of the brake and the inertia of the rotating body.

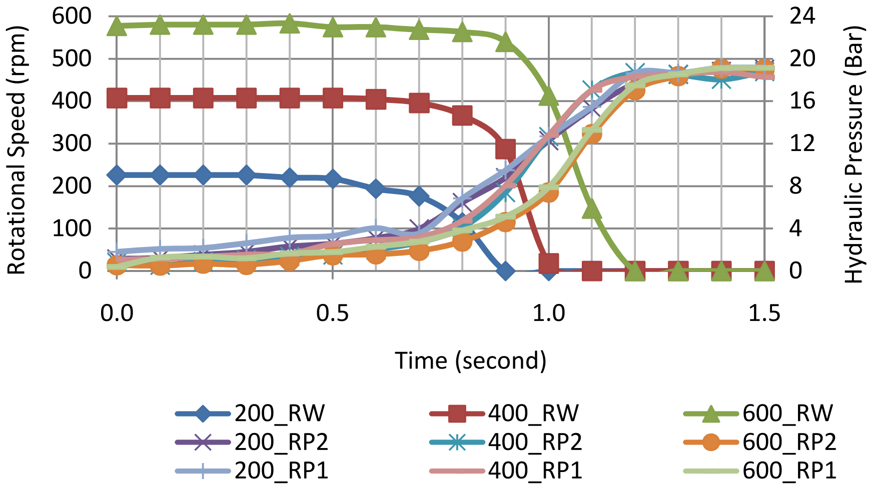


Figure 6: Experimental Results when Maximum Pressure is 18 Bar.

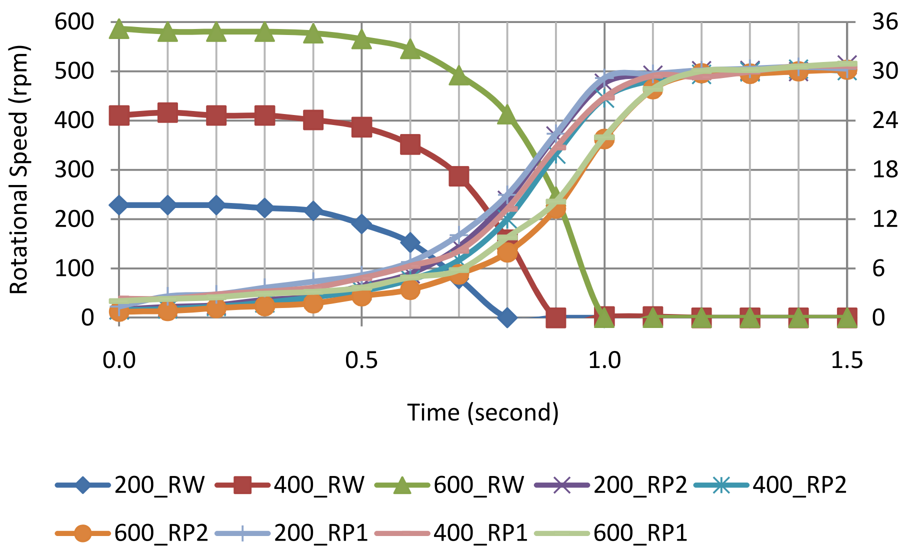


Figure 7: Experimental Results when Maximum Pressure is 30 Bar.

From figure 6 and 7, it can be estimated that the pad starts touching the brake disc at time $t = 0.5$ sec for figure 6 and $t = 0.4$ for figure 7. Before the pad touching the disc the energy of the fluid is used to move the piston in the caliper.

Figure 8 shows flow rate and torque when initial rotational speed is 600 rpm. The flow rate q_3 is calculated using equation (16). It is obvious that after the pump is activated the flow rate raises from 0 cc/sec to its maximum value then it decreases to 0 cc/sec when the pressure reaches its maximum value. The brake torque is calculated using equation (17).

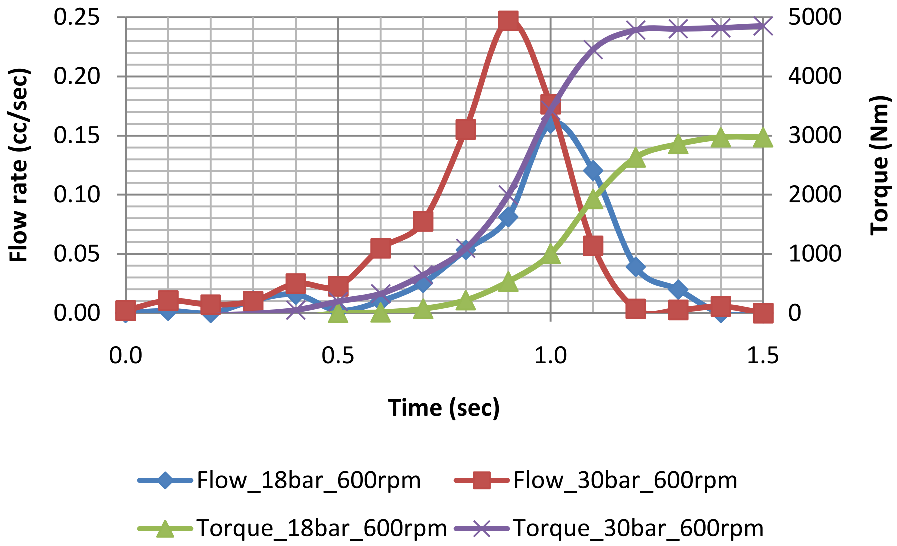


Figure 8: Flow rate and Brake Torque.

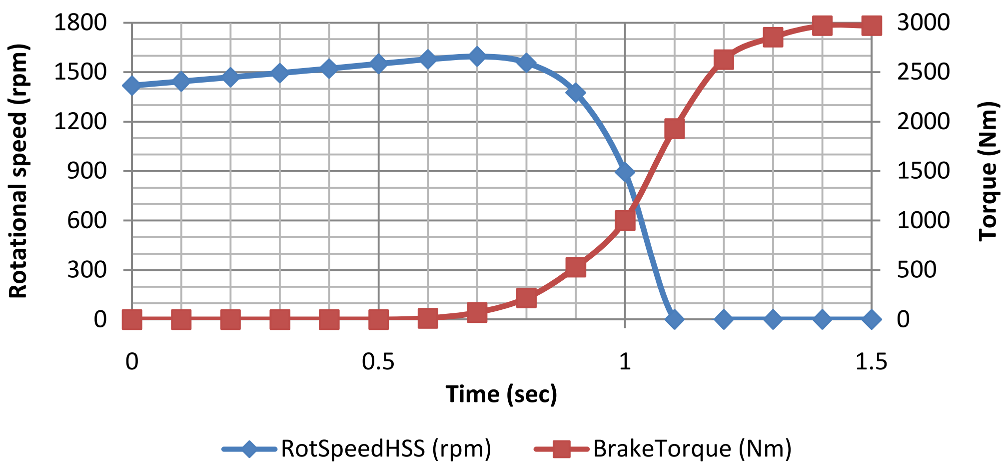


Figure 9: Simulation Result using the Real Turbine Parameters under Emergency Condition.

Figure 9 shows estimated rotational speed of the real wind turbine when the torque in the same figure is applied to the wind turbine. This result is obtained through numerical analysis using motion equation (18). In practice, this can be realized using a proportional pressure control valve to control the brake torque. From this simulation result the following information is obtained.

1. The brake can stop the turbine in 1.1 second after it is activated.
2. During the first 0.5 second before the brake takes effect, the turbine experiences acceleration of 8.4 rpm/sec.
3. In 0.2 sec after the pad touches the disc, the turbine starts decelerating.
4. The turbine experiences deceleration of 283.8 rpm/sec just before it stops rotating.

Conclusion

From the experimental and simulation results the following conclusion can be drawn.

1. The hydraulic brake control system developed in this research worked well during the functional analysis test using the test bed in the laboratory.
2. When initial rotational speed of the brake disc is 600 rpm, it can be stopped in 1.2 second using maximum pressure setting of 18 bar.
3. When the above pressure curve is applied to the real wind turbine under emergency condition, from numerical analysis it can be estimated that the turbine can be stopped in 1.1 second with deceleration of 283.8 rpm/sec just before stopping.

Acknowledgment

The authors would like to deliver their gratitude to the Indonesian Institute of Science (LIPI) for providing financial support through the competitive research grant. They also thank to all members of the Mechatronics Division at the Research Centre for Electrical Power and Mechatronics (P2Telimek) for their support, especially to Roni S. Permana. The research on development of a 100 kW horizontal axis wind power plant was conducted as a collaboration research between Indonesian Institute of Science (LIPI) and the Research & Development Agency of the Ministry of Energy and Mineral Resources of the Republic of Indonesia.

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