

Article

System Identification of a Servo-Valve Controlled Hydraulic Cylinder Operating Under Variable Load

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Abstract: This work presents an in-depth study on the system identification of a servo-valve controlled hydraulic cylinder operating under variable load. This research addresses the growing demand for improved control systems (enhancing time response, settling time, and precision) in variable load hydraulic actuators, such as those used in blade pitching systems of wind turbines. The paper begins by detailing the experimental setup, followed by the development of the system's mathematical model, a fourth-order transfer function (TF). The experimental data collected by a proposed data acquisition system are used for the dynamic identification of the hydraulic setup using periodical signals as commands. All possible combinations of TFs up to order 8 are identified. After an initial visual preselection of the 15 most accurate ones, analyses comparing quality indicators between the measured (experimental) and the TF (simulated) step and sinusoidal responses are conducted to determine the most accurate TF. The paper concludes with the presentation and analysis of the dynamic model, identified as being a fourth-order TF, which replicates the system dynamics with the greatest fidelity. It provides an identification methodology with significant potential for industry practitioners aiming to improve, optimize, and enhance control strategies for variable load hydraulic actuators.



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Keywords: experimental identification; mathematical model; hydraulic system; servo-valve; hydraulic cylinder; variable load; reliability

MSC: 37M10

1. Introduction

Renewable energy sources produced the most electrical energy within the European Union in 2023 (44.7%), with wind accounting for 19% [1]. From all the new renewable power installations in the European Union, 69 GW in 2023, the total capacity of the new wind power installations represents 25% corresponding to 17 GW. From the total installed wind power, 14 GW (82%) was installed onshore and 3 GW (18%) offshore [1]. Taking into account the European Union 2030 energy strategy which aims to reduce its greenhouse gas emissions by at least 55%, increase the share of renewable energy to at least 45% (27%) of consumption, achieve energy savings of 11.7% (27%) or more until 2020 (2050) [2] and considering the enormous potential of offshore wind, which could meet Europe's energy demand seven times [3], it is expected that the offshore wind will become one of the main electricity generators in Europe [4].

The offshore wind turbines have many advantages compared to other sources of energy, such as low greenhouse gas emissions compared to gas and coal-powered energy and such as allowing the energy to be produced while saving land and reducing to almost zero the visual, noise, and shadow impacts compared to onshore wind turbines. The main advantage of offshore wind turbines consists of a better output due to larger turbines (up to 15–25 MW [5]), compared to the onshore wind turbines (6.03 MW [6]) and due to running with more qualitative wind (higher speed with a steady performance).

The main disadvantages of offshore wind turbines compared to the onshore ones consist of high OPEX (operating expenses) due to accessibility conditioned by the weather, increase in installation and transmission expenses as well as new issues concerning saltwater corrosion [7], and the substructure inspection in an offshore environment [4]. Further reductions in OPEX can be achieved by increasing the operational performance standards on each turbine. To optimize offshore wind energy production and reduce costs, unscheduled maintenance must be minimized and continuous operation must be ensured.

Reference [8] provides a detailed analysis of wind turbine subsystems, identifying the hydraulic pitch system as the primary contributor to overall failure rates, accounting for 13% of all failures. Auxiliary components such as lifts, ladders, and nacelle seals are grouped under ‘Other Components’, contributing 12.2% to the failure rate, while the generator, gearbox, and blades account for 12.1%, 7.6%, and 6.2%, respectively. Given the significant impact of hydraulic pitch system failures on wind turbine reliability from the above research background, this system is selected as the focus of our investigation.

The primary function of the hydraulic pitch system is to precisely control the pitch angle of the turbine blades, ensuring both safe operation and optimal power output. An inaccurate model of the hydraulic pitch system can lead to poor control performance, without allowing the wind turbine to reach its maximum power, representing the research motivation behind this work.

The objective of this paper is to identify experimentally the mathematical model of a servo-valve controlled hydraulic cylinder operating under a variable load, necessary for the proper development, analysis, and synthesis of an optimized, robust, reliable, and efficient pitch control system.

Even though many mathematical models that were developed analytically for such systems can be found in the relevant research [9–11], due to the complexity of pitch hydraulic systems involving resistance, delays, and nonlinear behaviors, deriving analytically their dynamic characteristics [12] from physical and structural data prove to be highly challenging and inaccurate. For example, in reference [9], although the proposed model seems to faithfully replicate the dynamic characteristics of the considered system, since it does not present an experimental validation, the accuracy of these models cannot be fully determined. In reference [10], the inaccuracy of the mathematical model parameters is solved by combining the system dynamic analysis and component technology samples. In reference [11], the inaccuracies of analytical modeling are attempted to be resolved by the control strategy (sliding mode variable structure controller and dynamic soft variable structure controller). The main contribution of the present work consists of solving the inaccuracy of the mathematical (analytical) model of the hydraulic pitching system of a wind turbine blade by obtaining it through experimental identification because the most reliable approach to obtaining the characteristics of such a system is through experimental methods.

A hydraulic stand (test rig) is used as an experimental stand to determine the dynamic properties of a wind turbine blade pitching system. The stand includes two hydraulic circuits, each one having a double rod, and a double-acting hydraulic cylinder: one controlled

by a servo-valve (emulating the hydraulic actuator) and the other loaded via a pressure relief valve (emulating the variable load).

The dynamic properties were obtained using deterministic (periodic) signals as input commands for the servo-valve control (sinusoidal and rectangular signal with various amplitudes and frequencies), facilitating the derivation of frequency characteristics. These responses provide critical insights into the system's transient and steady-state behaviors [13].

For complex systems, such as a servo-valve controlled hydraulic cylinder, dynamic identification must consider the interactions between the physical variables of each system element (servo-valve and hydraulic cylinder). The overall system is treated as an interconnected assembly, with each component exhibiting independent transient behaviors. Additionally, the inherent nonlinearity of many industrial systems must be taken into account, as parameters like amplification coefficients and time constants often vary with the operating conditions. These systems are typically described by nonlinear differential equations, where the coefficients depend on variable values [14].

The proposed data acquisition and logging system was used to determine experimentally the mathematical model (the transfer function) of the system considered. The experimental data obtained were used for the dynamic identification of all the possible combinations of TFs up to 8th order. After an initial visual preselection of the 15 most accurate TFs, three analyses were conducted to identify the most accurate one based on comparisons between the measured (experimental) and transfer function (simulated) responses. The first analysis involved comparing their fitting accuracy, the second focused on quality indicators (overshoot, rise time, response time, and settling time) derived from their step responses, and the third compared additional quality indicators (maximum and minimum amplitude and phase shift) derived from their sinusoidal responses.

The paper concludes by presenting and analyzing the dynamic model identified as a fourth-order TF (being the same order as the one developed analytically), which replicates the system dynamics with the greatest fidelity.

2. Hydraulic Stand (Test Rig)

The hydraulic stand presented in Figure 1 has been used in the current work in determining experimentally the dynamic model of a hydraulic wind turbine blade pitching system. This test rig can be found in the Hydraulic Machinery Laboratory “Aurel Barghazan” of the Mechanical Faculty from Politehnica University of Timisoara [15].

The test rig consists of two main systems, one which is the actuating system, emulating the wind turbine blade pitching system, and one emulating the loading on the wind turbine blade.

Figure 2 presents the hydraulic diagram of the test rig consisting of two main circuits, an actuating one and a loading one.

The actuating system consists of the hydraulic pump (labeled with H3), actuated by the electrical motor (labeled with H1), the pressure filter (labeled with H4), the main line pressure sensor (labeled with T5), the main pressure manometer (labeled with H6.1), the servo-valve (labeled with H7), the actuating cylinder pressure sensors (labeled with T8.1 and T8.2), the actuating cylinder (labeled with H9.1), the LVDT (Linear Variable Differential Transformer) position transducer (labeled with T10), the force transducer (labeled with T11), the flow meter (labeled with T14) and the pressure relief valve (labeled with H 15).

The pressure relief valve is set at 100 bar and when it is actuated, the pressure system can be easily checked either with the manometer (labeled with H6.1), or with the main line pressure sensor (labeled with T5) in Figure 2.



Figure 1. The test rig emulating the wind turbine blade pitching system.

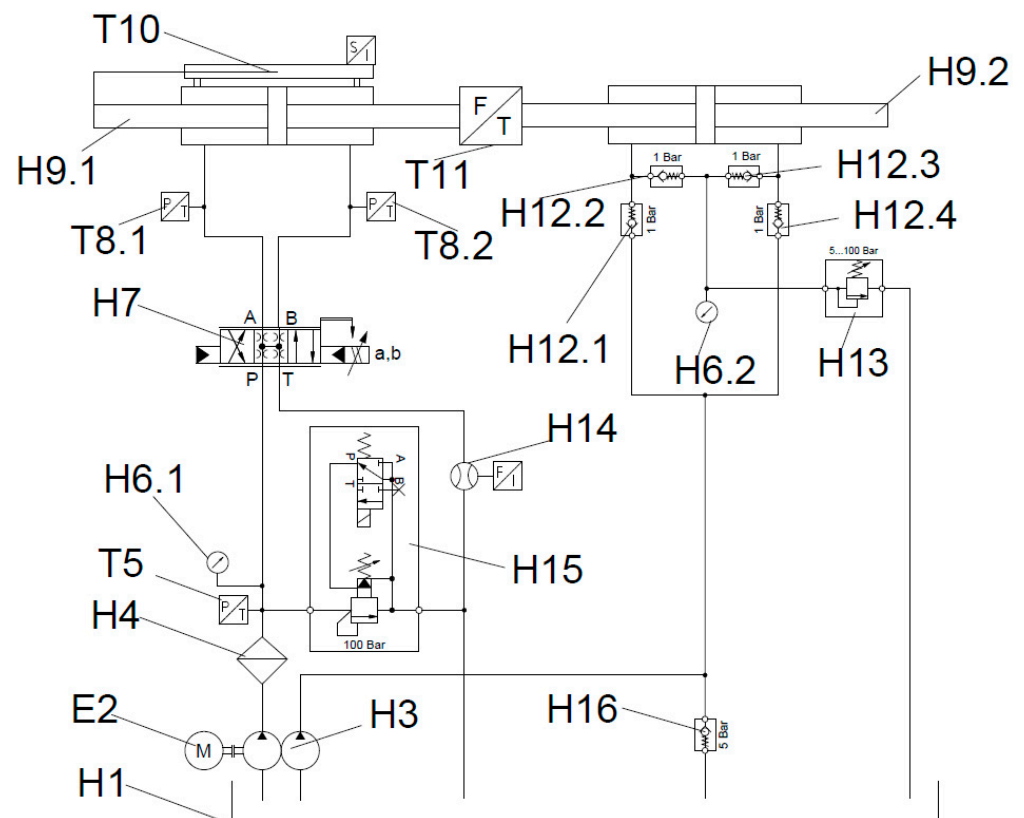


Figure 2. Hydraulic diagram of the test rig.

The servo-valve (labeled with H7) feeds oil to the actuating cylinder, which can be moved either to the left or right. The electrical command for the servo-valve is given from a signal generator which can generate step, sine, or triangular signals with different amplitudes and frequencies.

The pressure on either side of the actuating cylinder is measured with the pressure sensors (labeled with T8.1 and T8.2).

The flow in the actuating system is measured with the turbine flow meter (labeled with T14).

The loading system consists of the following parts: the oil feeding pump, the twin pump of the one from the actuating system, which feeds oil to the loading cylinder (labeled with H9.2), the check valves (labeled with H12.1 to H12.4) ensuring a specific oil flow path, the load pressure setting valve (labeled with H13) with the manometer (labeled with H6.2) and the check valve (labeled with H16) ensuring a pressure of 5 bar in the loading system protecting it for cavitation occurrence [16,17].

The force given by the loading system emulating the load on the blade is measured with the force transducer (labeled with T11).

The hydraulic elements of the testing rig are the following:

H1—Reservoir, which has a volume of 300 L and is equipped with a breather and oil level indicator. This has the role of storing and cooling the oil;

E2—Electrical drive motor type ATF 32M 4A with a maximum of 7.5 kW electrical power. The electric motor drives the hydraulic pumps converting the electrical power into mechanical power;

H3—Twin hydraulic gear pump with fixed displacement produced by Uzina Mecanica Plopeni type PRD22-2188D [18].

The twin hydraulic pump transforms the mechanical power at the electrical motor shaft into hydraulic power feeding both systems with oil. The main pump ensures the oil flow at a maximum pressure set by the relief valve and the second pump feeds the part that is not pressurized of the loading cylinder through the check valve system;

H4—Pressure line filter [19].

The pressure line filter has the role of feeding the system with clean oil without any contamination in it. The filter housing and all the connecting elements are designed so that pressure peaks caused by accelerated fluid volumes when the servo-valve suddenly opens can be safely handled. The filter element is made of an organic fiber that has a high filtration degree, with a 5 μm mesh, suitable for the used servo-valve;

H6.1, H6.2—Manometer.

The manometers are HansaFlex GMM 100–160 H, Class 1, with a maximum indicating pressure of 160 bar;

H7—Rexroth Servo-valve type 4WS2EM10-51/45B11ET210K31EV [20].

Rexroth servo-valve 4WS2EM10-51/45B11ET210K31EV [20] is an electrically operated, 2-stage directional servo-valve and it is mainly used in applications to control position, force, and velocity. Figure 3 presents the main components of this servo-valve: an electro-mechanical converter (torque motor) (labeled with 1 in Figure 3), a hydraulic amplifier (nozzle flapper plate principle) (labeled with 2), and a control spool (labeled with 3) in a sleeve (2nd stage), which is connected to the torque motor via a mechanical return [20].

The torque motor coils (labeled with 4) receive an electrical input signal, which generates a force acting on the armature (labeled with 5), which is connected with a torque tube (labeled with 6) creating a torque. The flapper plate (labeled with 7) which is connected to the torque tube (labeled with 6) through a pin, moves from the central position between the two control nozzles (labeled with 8), creating a pressure differential over the front faces of the control spool. This pressure differential makes the control spool

move, connecting the pressure line port to one of the ports connected to the hydraulic motor (actuator); at the same time, the other hydraulic motor port is connected to the return port [20].

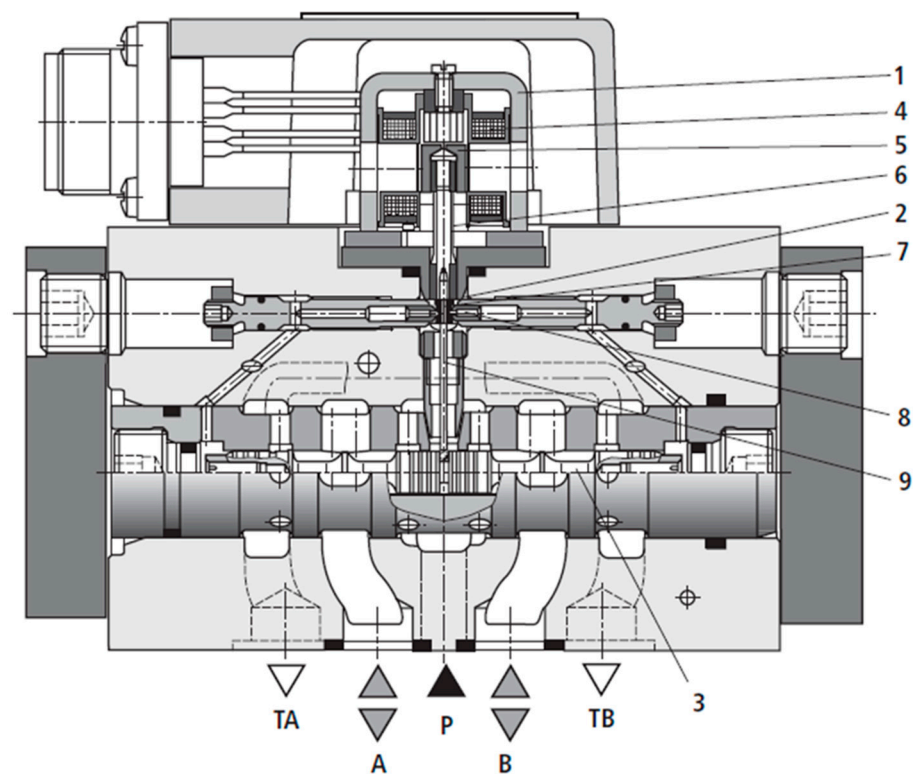


Figure 3. Rexroth servo-valve type 4WS2EM10-51/45B11ET210K31EV elements [20].

The control spool is connected to the flapper plate using a bending spring (mechanical return and feedback) (labeled with 9). The control spool moves until the feedback torque across the bending spring and the torque created by the torque motor are balanced and the pressure differential at the nozzle flapper plate system becomes zero [20].

The stroke of the control spool and implicitly, the flow of the servo-valve are controlled proportionally by the input signal which is given electrically. The analog input signal (command) is amplified so that the output signal actuates the servo-valve controlling the flow through it [20].

The zero flow of the valve can be calculated with Equation (1):

$$Q_{zero} = \sqrt{\frac{\text{inlet pressure [bar]}}{70 \text{ bar}}} \cdot 1.2 = \sqrt{\frac{100 \text{ bar}}{70 \text{ bar}}} \cdot 1.2 = 1.43 \left[\frac{1}{\text{min}} \right] \quad (1)$$

the characteristic curves (measured with HLP 32, Toil = 40 °C ± 5 °C) are presented in Figures 4 and 5. For more details, please check [20].

H9.1, H9.2—Double rod double acting hydraulic cylinder, CGT3MS2/40/28/300F11/B11HEUTAWF15845 [21].

Both hydraulic cylinders, the actuating cylinder and the loading one are identical and they are mechanically connected through the force transducer;

H15—Pressure relief valve, pilot operated DBW 10 BG2-52/200S6EG24N9K4R12 [22].

The pilot-operated pressure relief valve is used to set the maximum pressure in the system. Based on an analysis of the characteristic curves provided in the technical specifications [22], among other operating characteristics, the pressure variation with the flow remains nearly constant within the operating range;

H13—Pressure relief valve DBDH 10 G1A/100 [23].

The pressure relief valve 13 is used to simulate the force on the blade. It can be adjusted from 5 to 100 bars, creating a corresponding force at the cylinder. The pressure set can be seen with the manometer 6.2. Based on an analysis of the characteristic curves provided in the technical specifications [23], among other operating characteristics, the pressure variation with the flow remains nearly constant within the operating range;

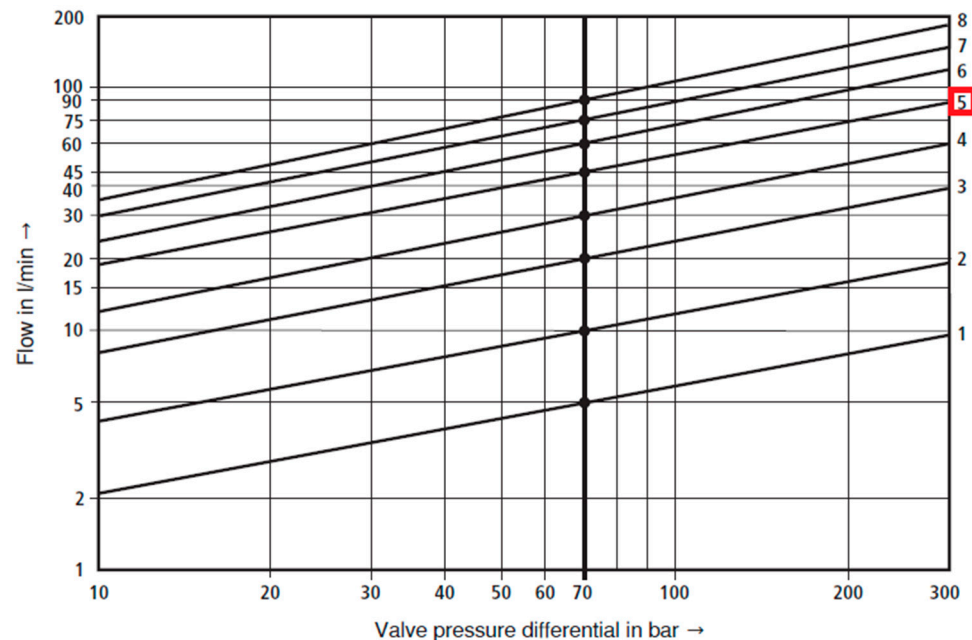


Figure 4. Flow/load function with 100% command value signal (tolerance $\pm 10\%$) [20], the characteristic of the servo-valve type used in the stand is labeled with 5 (highlighted in red).

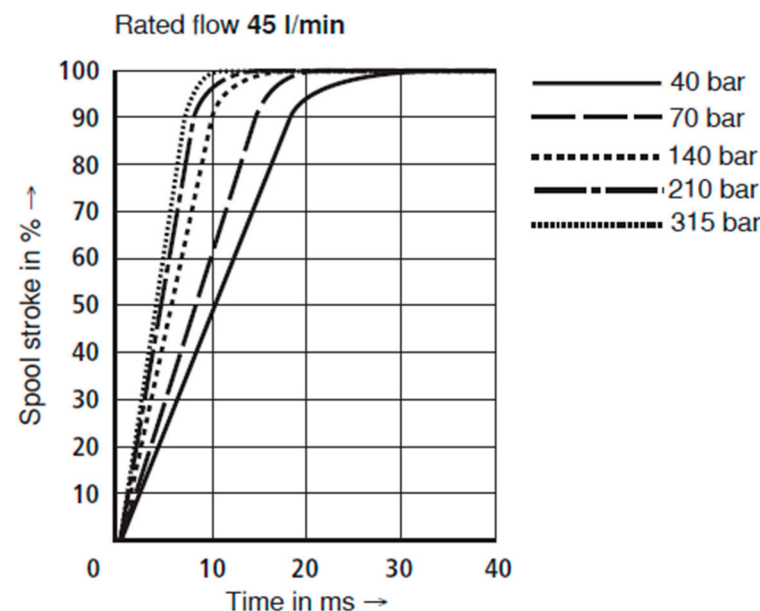


Figure 5. Transition function with step response without flow [20].

3. Data Acquisition and Logging System

The block diagram from Figure 6 illustrates the main components of the data acquisition and logging system used for determining experimentally the mathematical model of the servo-valve controlled hydraulic cylinder operating under variable load, representing the blade pitch actuator of a wind turbine.

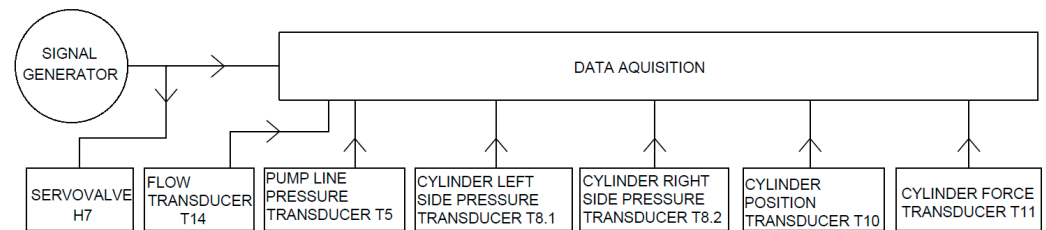


Figure 6. Block diagram of the data acquisition and logging system.

The data acquisition and logging system consists of the following parts and transducers, which are illustrated in Figure 6:

- the servo-valve H7—Rexroth 4WS2EM10-51/45B11ET210K31EV [20];
- the flow transducer T14—Omni RT [24];
- the pump line pressure transducer T5—Mannesman Rexroth HM-15-1X/600 [25];
- the cylinder left side pressure transducer T8.1—PS-200-D-G2-1;
- the cylinder right side pressure transducer T8.2—PS-200-D-G2-1;
- the cylinder position transducer T10—Monitran MTN/EICR150 [26];
- the cylinder force transducer T11—Lorenz K-25 20kN [27];
- the signal generator Tektronix CFG253 [28];
- the data acquisition board DT9804 [29].

T14—The flow transducer Omni RT [24].

The technical specifications of the Omni RT [24] flow transducer used for measuring the system flow rate are presented in Table A1 from Appendix A.

T5—The pump line pressure transducer Mannesman Rexroth HM-15-1X/600 [25].

The technical specifications of the pump line pressure transducer Mannesman Rexroth HM-15-1X/600 (VFD) [25] are presented in Table A2 from Appendix A.

T8.1 and T8.2—Cylinder pressure transducer PS-200-D-G2-1.

The technical specifications of the pressure transducer PS-200-D-G2-1 are presented in Table A3 from Appendix A.

T10—LVDT position transducer Monitran MTN/EICR150 [26].

Table A4, of Appendix A, presents the technical specifications of the position transducer Monitran MTN/EICR150 [26].

T11—Cylinder force transducer Lorenz K-25 20kN [27].

The technical specifications of the force transducer Lorenz K-25 20kN [27] are presented in Table A5 from Appendix A.

The signal generator Tektronix CFG253 [28].

The Tektronix CFG253 [29] can produce sine, square, and sawtooth waves and TTL (Transistor–Transistor Logic) signals in a frequency range from 0.03 Hz up to 3 MHz. The function generator has a symmetry function to control the rise and fall times of sine or sawtooth waves and the duty cycles of square waves. It also has a sweep function that makes the output signal traverse a range of frequencies.

Table A6, from Appendix A, presents the technical specifications of the signal generator Tektronix CFG253.

The data acquisition board DT9804 [29].

The DT9804 [29] from Figure A1 of Appendix A, is a multifunction USB data acquisition module featuring 16 single-ended (SE)/8 differential (DIFF) inputs with 12- or 16-bit resolution, up to 100 kS/s aggregate sample rate, 16 digital I/O lines, 2 counter/timers, and optional 12- or 16-bit analog outputs, connected to a HP Gaming Pavilion—15-dk0022ns 8PK68EA.

Several experiments have been conducted in order to cover the entire operating range in terms of servo-valve signal shape (sinusoidal or rectangular), frequency (0.3 Hz, 0.45 Hz,

0.6 Hz, 0.75 Hz, and 0.9 Hz), amplitude (± 2 V, ± 5 V, and ± 10 V) and load (pressure variation on the hydraulic cylinder acting as a load for 5 bars, 30 bars, 50 bars, and 70 bars).

The raw experimental data have been acquired using seven of the available input channels of the DT9804 and logged on MATLAB (version R2024b), the position of cylinder rod on Analog Input Channel 0, p31 on Analog Input Channel 1, p32 on Analog Input Channel 2, flow on Analog Input Channel 3, force on Analog Input Channel 4, p33 on Analog Input Channel 5 and the servo-valve command, Analog ± 10 V, given by the signal generator on Analog Input Channel 6 with a sampling rate of 1000 values/second (one value/1 millisecond).

The raw value given by the position transducer (Analog ± 10 V) the data logged on MATLAB is converted to corresponding physical units according to Equation (2):

$$Pos = a \cdot Pos_A + b \quad (2)$$

where Pos represents the extension of the hydraulic cylinder rod in mm, Pos_A represents the raw analog value (± 10 V), $a = 15.676$ represents the calibration slope, and $b = 0.748$ represents the calibration offset (a and b coefficients being obtained by calibrating the transducer so that the value measured by it, to be the same as the cylinder piston stroke (± 150 mm) and the value 0 to represents the cylinder in the middle position).

The experimental data stored in MATLAB have been filtered using a lowpass filter designed in MATLAB in order to attenuate the frequencies above the specified passband frequency (F_{pass}).

The upper limit of the passband frequency of the filter was chosen to be higher than the maximum frequency at which the servo-valve can make the cylinder oscillate. The maximum oscillation frequency of the cylinder is calculated considering the maximum flow rate given by the servo-valve $Q = 12$ L/min in the system and the maximum stroke of the cylinder ($L = 0.15$ m), using Equation (3):

$$F_{pass} = \frac{1}{T} = \frac{1}{L/v} = \frac{1}{L/(Q/S)} = 2.1 \text{ Hz} \approx 3 \text{ Hz} \quad (3)$$

with

$$S = \frac{\pi \cdot (D^2 - d^2)}{4} = 0.000641 \text{ m}^2 \quad (4)$$

where T represents the time interval during which the cylinder makes an entire stroke, v represents the cylinder's maximum velocity, S represents the active surface of the hydraulic cylinder ($S = 0.000641 \text{ m}^2$), and D represents the diameter of the cylinder ($D = 40$ mm) and d representing the diameter of the rod ($d = 28$ mm).

4. Mathematical Model

In the current section, the mathematical model of the servo-valve controlled hydraulic cylinder operating under variable load is presented.

4.1. Servo-Valve Model

The servo-valve dynamics can be approximated with a first-order equivalent system when they operate at frequencies up to 50 Hz ($\omega_n \leq 50$ Hz), according to Equation (5) from [30,31]:

$$H(s) = \frac{Q}{\Delta U_c} = \frac{K}{1 + \tau \cdot s} \quad (5)$$

where Q [m^3/s] is the servo-valve volumetric flow rate, ΔU_c [V] is the input voltage (command) of the servo-valve, K proportional coefficient in the stationary operating regime, and τ is the servo-valve time constant.

4.2. Double Rod, Double Acting Hydraulic Cylinder

The mathematical model of the double rod, double-acting hydraulic cylinder starts from the continuity equation of the hydraulic cylinder [32]:

$$Q = S_p \cdot \frac{dX_p}{dt} + C_{tp} \cdot P_L + \frac{V_t}{4 \cdot E_u} \cdot \frac{dP_L}{dt} \quad (6)$$

where S_p represents the cylinder active surface [m²]; X_p represents the hydraulic cylinder stroke [m]; V_t represents the active volume of the cylinder [m³]; C_{tp} represents the total leakage of the cylinder [m³/s]; E_u the bulk modulus of the fluid [Pa]; and P_L is the differential (load) pressure between the cylinder chambers [Pa].

Applying the Laplace transform results in Equation (7):

$$Q = S_p \cdot X_p \cdot s + \left(C_{tp} + \frac{V_t}{4 \cdot E_u} \cdot s \right) \cdot P_L \quad (7)$$

The piston force is expressed as [32]:

$$F_p = S_p \cdot P_L = M_t \cdot \frac{d^2 X_p}{dt^2} + B_L \cdot \frac{dX_p}{dt} + K_a \cdot X_p + F_z \quad (8)$$

where F_p is the force exerted by the piston [N]; M_t is the total mass (load + piston mass + road mass) [kg]; B_L is the total damping factor (load + piston + road) [N s/m]; K_a is the total stiffness [N/m]; and F_z is the perturbation force [N] being so small that can be neglected.

Applying the Laplace transform results in Equation (9):

$$P_L = \frac{1}{S_p} \cdot \left(M_t \cdot X_p \cdot s^2 + B_L \cdot X_p \cdot s + K_a \cdot X_p + F_z \right) \quad (9)$$

Combining Equations (5)–(9) results in the following equivalent transfer function (Equation (10) with its parameters presented in Equation (11)) of the entire system (the servo-valve controlled hydraulic cylinder operating under variable load):

$$\frac{X_p}{\Delta U_c} = \frac{4 \cdot E_u \cdot S_p \cdot K}{A \cdot s^4 + B \cdot s^3 + C \cdot s^2 + D \cdot s + 4 \cdot E_u \cdot C_{tp} \cdot K_a} \quad (10)$$

with

$$\begin{aligned} A &= V_t \cdot M_t \\ B &= V_t \cdot M_t + 4 \cdot E_u \cdot C_{tp} \cdot \tau + V_t \cdot B_L \cdot \tau \\ C &= 4 \cdot E_u \cdot C_{tp} + V_t \cdot B_L + 4 \cdot E_u \cdot \tau \cdot (S_p)^2 + 4 \cdot E_u \cdot C_{tp} \cdot B_L \cdot \tau \\ D &= 4 \cdot E_u \cdot (S_p)^2 + 4 \cdot E_u \cdot C_{tp} \cdot B_L + 4 \cdot E_u \cdot C_{tp} \cdot K_a \cdot \tau \end{aligned} \quad (11)$$

5. System Identification

The system identification of the hydraulic test rig has been conducted entirely in MATLAB. Its transfer functions were identified from measured data (input–output data) by using PEM (prediction error minimization with Levenberg–Marquardt) as numerical optimization techniques to fit the mathematical model to the data with the help of the system identification toolbox from MATLAB. Even though the hydraulic test rig is a SIMO (Single Input Multiple Output) in the current paper the TF of a SISO (Single Input Single Output) system will be identified, considering the servo-valve command as input and the position (extension) of the hydraulic cylinder rod as output as depicted in the block diagram from Figure 7.

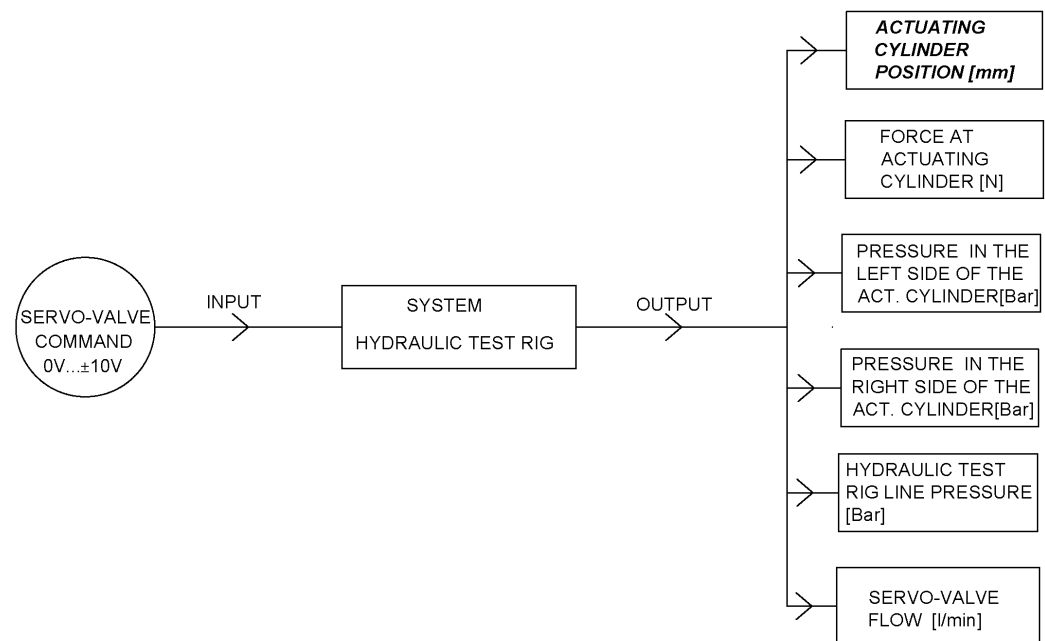


Figure 7. Block diagram of the generated input and measured output data.

From the entire experimental data sets the TF of the SISO system is conducted on three data sets selected as the most representative (only rectangular responses were considered, due to their advantages in terms of broad frequency content, simpler time-domain, providing a clearer picture of nonlinearities in the system and more suitable for this system, where sudden changes like disturbances or commands are common):

- A. A rectangular shape signal with a frequency of 0.3 Hz, an amplitude of ± 10 V, and a constant load exerted by the second hydraulic cylinder being held at a constant pressure of 50 bars (labeled as R_0.3Hz_10A_50bars);
- B. A rectangular shape signal with a frequency of 0.45 Hz, an amplitude of ± 5 V, and a constant load exerted by the second hydraulic cylinder being held at a constant pressure of 50 bars (labeled as R_0.45Hz_5A_50bars);
- C. A rectangular shape signal with a frequency of 0.6 Hz, an amplitude of ± 5 V, and almost no load exerted by the second hydraulic cylinder; the chamber pressure being 5 bars (labeled as R_0.6Hz_10A_5bars).

From each of the experimental data sets mentioned above, all the possible combinations of TFs with the number of zeros between 0 and 7 and the number of poles between one and eight are identified (ensuring that for each TF the number of poles is greater than the number of zeros). The TFs with the numbers of zeros and/or poles greater than seven, respectively, eight, are considered by the authors to be unusable in the industrial context.

An initial preselection of the most accurate TFs is performed based on the visual comparison between the experimentally measured response with the simulated response of the TFs (identified according to the previous paragraph). The TFs that offered significant differences in time response, delay, overshoot, settling time, settling amplitude, and damping were discarded, keeping under consideration only the following 15 TFs (Equations (12)–(26)):

$$TF_1 = \frac{983.6}{s^2 + 14.9 \cdot s + 63.5} \quad (12)$$

where the TF_1 is a transfer function with 2 poles and 0 zero identified on the A experimental data set.

$$TF_2 = \frac{15.76 \cdot s + 741.6}{s^2 + 12.03 \cdot s + 48} \quad (13)$$

where the TF_2 is a transfer function with 2 poles and 1 zero identified on the A experimental data set.

$$TF_3 = \frac{6366 \cdot s + 2.567 \cdot 10^5}{s^3 + 361.6 \cdot s^2 + 4226 \cdot s + 1.662 \cdot 10^4} \quad (14)$$

where the TF_3 is a transfer function with 3 poles and 1 zero identified on the A experimental data set.

$$TF_4 = \frac{2.595 \cdot 10^5 \cdot s + 1.068 \cdot 10^6}{s^4 + 245.5 \cdot s^3 + 5202 \cdot s^2 + 3.271 \cdot 10^4 \cdot s + 6.976 \cdot 10^4} \quad (15)$$

where the TF_4 is a transfer function with 4 poles and 1 zero identified on the A experimental data set.

$$TF_5 = \frac{4960 \cdot s^2 + 1.405 \cdot 10^5 \cdot s + 4.091 \cdot 10^5}{s^4 + 216.3 \cdot s^3 + 2750 \cdot s^2 + 1.521 \cdot 10^4 \cdot s + 2.622 \cdot 10^4} \quad (16)$$

where the TF_5 is a transfer function with 4 poles and 2 zeros identified on the A experimental data set.

$$TF_6 = \frac{10 \cdot s^3 + 3263 \cdot s^2 + 8.266 \cdot 10^4 \cdot s + 3.701 \cdot 10^5}{s^4 + 131.3 \cdot s^3 + 1765 \cdot s^2 + 1.088 \cdot 10^4 \cdot s + 2.377 \cdot 10^4} \quad (17)$$

where the TF_6 is a transfer function with 4 poles and 3 zeros identified on the A experimental data set.

$$TF_7 = \frac{1452 \cdot s^3 + 4.191 \cdot 10^4 \cdot s^2 + 6.656 \cdot 10^5 \cdot s + 1.045 \cdot 10^7}{s^5 + 49.83 \cdot s^4 + 1313 \cdot s^3 + 2.3 \cdot 10^4 \cdot s^2 + 2 \cdot 10^5 \cdot s + 6.749 \cdot 10^5} \quad (18)$$

where the TF_7 is a transfer function with 5 poles and 3 zeros identified on the A experimental data set.

$$TF_8 = \frac{5483 \cdot s^2 + 1.329 \cdot 10^5 \cdot s + 5408}{s^4 + 100.9 \cdot s^3 + 1813 \cdot s^2 + 8576 \cdot s + 273.9} \quad (19)$$

where the TF_8 is a transfer function with 4 poles and 2 zeros identified on the B experimental data set.

$$TF_9 = \frac{-7.629 \cdot s^3 + 6185 \cdot s^2 + 1.666 \cdot 10^5 \cdot s + 5634}{s^4 + 118.4 \cdot s^3 + 2230 \cdot s^2 + 1.073 \cdot 10^4 \cdot s + 288} \quad (20)$$

where the TF_9 is a transfer function with 4 poles and 3 zeros identified on the B experimental data set.

$$TF_{10} = \frac{5008 \cdot s^3 + 4.037 \cdot 10^4 \cdot s^2 + 9.129 \cdot 10^4 \cdot s + 3.532 \cdot 10^4}{s^5 + 67.01 \cdot s^4 + 764.3 \cdot s^3 + 3535 \cdot s^2 + 6085 \cdot s + 1726} \quad (21)$$

where the TF_{10} is a transfer function with 5 poles and 3 zeros identified on the B experimental data set.

$$TF_{11} = \frac{25.6 \cdot s^4 + 3008 \cdot s^3 + 4135 \cdot s^2 + 6.017 \cdot 10^4 \cdot s + 3002}{s^5 + 35.97 \cdot s^4 + 244.4 \cdot s^3 + 952.6 \cdot s^2 + 3811 \cdot s + 153.5} \quad (22)$$

where the TF_{11} is a transfer function with 5 poles and 4 zeros identified on the B experimental data set.

$$TF_{12} = \frac{26.63 \cdot s^5 + 2750 \cdot s^4 + 9344 \cdot s^3 + 9.235 \cdot 10^4 \cdot s^2 + 2.852 \cdot 10^5 \cdot s + 3.168 \cdot 10^4}{s^6 + 34.51 \cdot s^5 + 303.4 \cdot s^4 + 1698 \cdot s^3 + 8981 \cdot s^2 + 1.842 \cdot 10^4 \cdot s + 1504} \quad (23)$$

where the TF_{12} is a transfer function with 6 poles and 5 zeros identified on the B experimental data set.

$$TF_{13} = \frac{1.815 \cdot 10^7 \cdot s^3 + 4.971 \cdot 10^6 \cdot s^2 + 7.415 \cdot 10^8 \cdot s + 1.733 \cdot 10^8}{s^7 + 54.76 \cdot s^6 + 8547 \cdot s^5 + 2.254 \cdot 10^5 \cdot s^4 + 1.567 \cdot 10^6 \cdot s^3 + 9.314 \cdot 10^6 \cdot s^2 + 4.916 \cdot 10^7 \cdot s + 8.861 \cdot 10^6} \quad (24)$$

where the TF_{13} is a transfer function with 7 poles and 3 zeros identified on the B experimental data set.

$$TF_{14} = \frac{5486 \cdot s^6 + 3.316 \cdot 10^4 \cdot s^5 + 9.952 \cdot 10^5 \cdot s^4 + 1.463 \cdot 10^6 \cdot s^3 + 2.993 \cdot 10^7 \cdot s^2 + 4.787 \cdot 10^6 \cdot s + 8.446 \cdot 10^5}{s^8 + 72.54 \cdot s^7 + 883.5 \cdot s^6 + 1.394 \cdot 10^4 \cdot s^5 + 8.256 \cdot 10^4 \cdot s^4 + 4.396 \cdot 10^5 \cdot s^3 + 1.1929 \cdot 10^6 \cdot s^2 + 3.3625 \cdot 10^5 \cdot s + 4.654 \cdot 10^4} \quad (25)$$

where the TF_{14} is a transfer function with 8 poles and 6 zeros identified on the B experimental data set.

$$TF_{15} = \frac{4607 \cdot s + 39.93}{s^3 + 51.14 \cdot s^2 + 290.5 \cdot s + 8.814 \cdot 10^{-5}} \quad (26)$$

where the TF_{15} is a transfer function with 3 poles and 1 zero identified on the C experimental data set.

6. Analysis and Results

A first analysis of the selected transfer functions was made by comparing the fitting between the experimental and the TFs responses to the same inputs (the generated sinusoidal commands of the servo-valve) measured during the experiments.

Table 1 presents the average fitting between the experimental and the TFs responses to all the sinusoidal experiments.

Table 1. Results of the fitting analysis on the transfer functions' response to the sinusoidal input signals data sets.

Transfer Function	Fitting
TF ₁	70.43%
TF ₂	70.50%
TF ₃	70.46%
TF ₄	70.08%
TF ₅	69.43%
TF ₆	69.50%
TF ₇	69.09%
TF ₈	83.67%
TF ₉	83.66%
TF ₁₀	83.20%
TF ₁₁	83.57%
TF ₁₂	82.04%
TF ₁₃	80.48%
TF ₁₄	84.85%
TF ₁₅	83.48%

From Table 1, it can be seen that the response of TF_{14} fits most all the measured responses to the sinusoidal generated inputs (84.85%), followed by TF_8 (83.67%) and TF_9 (83.66%). TF_8 and TF_9 are in the same order as the one from the mathematical model obtained in Section 4.

A second analysis of the selected TFs was made by comparing the quality indicators between the measured step response with the TF response. The quality indicators considered are the overshoot, rise time, response time, and settling time.

Table 2 presents the error between the considered quality indicators of the measured step response with the TF response.

From Table 2, it can be seen that in terms of overshoot, only TF_{15} , TF_8 , TF_9 , and TF_{10} do not present any overshoot, the same as the measured response, while the other TFs present a bit of overshoot. By analyzing the error of each TF rise time and settling time, it can be seen that TF_9 approaches better the measured step response in most cases (1.21%, respectively, 1.22%), followed by TF_8 (1.64%, respectively, 1.65%), even if their rise time (an error of 7.05% for TF_9 and an error of 7.7% TF_8) is slightly higher than the measured one,

compared to the response of other TFs such as TF_2 (−0.49%) and TF_3 (−0.62%). TF_8 and TF_9 are in the same order as the one from the mathematical model obtained in Section 4.

Table 2. Error between the quality indicators of the measured step response with the transfer function response.

Transfer Function	Overshoot	Rise Time	Response Time	Settling Time
TF ₁	0.03353186260500330%	12.37%	4.86%	12.40%
TF ₂	0.62393249155571400%	8.78%	−0.49%	8.81%
TF ₃	0.65276996707479100%	8.70%	−0.62%	8.73%
TF ₄	1.33961470903823000%	10.19%	1.78%	10.22%
TF ₅	0.07777378234981290%	7.87%	−1.65%	7.89%
TF ₆	0.04167246906108630%	7.85%	−1.80%	7.87%
TF ₇	0.03815403136799310%	11.99%	2.43%	12.02%
TF ₈	0.00000000000000000%	1.64%	7.70%	1.65%
TF ₉	0.00000000000000000%	1.21%	7.05%	1.22%
TF ₁₀	0.00000000000000000%	73.80%	114.88%	73.94%
TF ₁₁	0.00089273709754022%	4.56%	60.21%	33.70%
TF ₁₂	0.21519780470244100%	11.31%	129.95%	86.26%
TF ₁₃	0.00002230307479680%	38.13%	54.84%	38.21%
TF ₁₄	2.37380512845963000%	8.23%	16.27%	18.69%
TF ₁₅	0.00000000000000000%	11.77%	22.96%	11.80%

Figure 8 depicts an example of a comparison among the many comparisons made (based on which Table 2 was obtained) between the step responses of the TFs vs. the step response of the hydraulic test rig obtained experimentally.

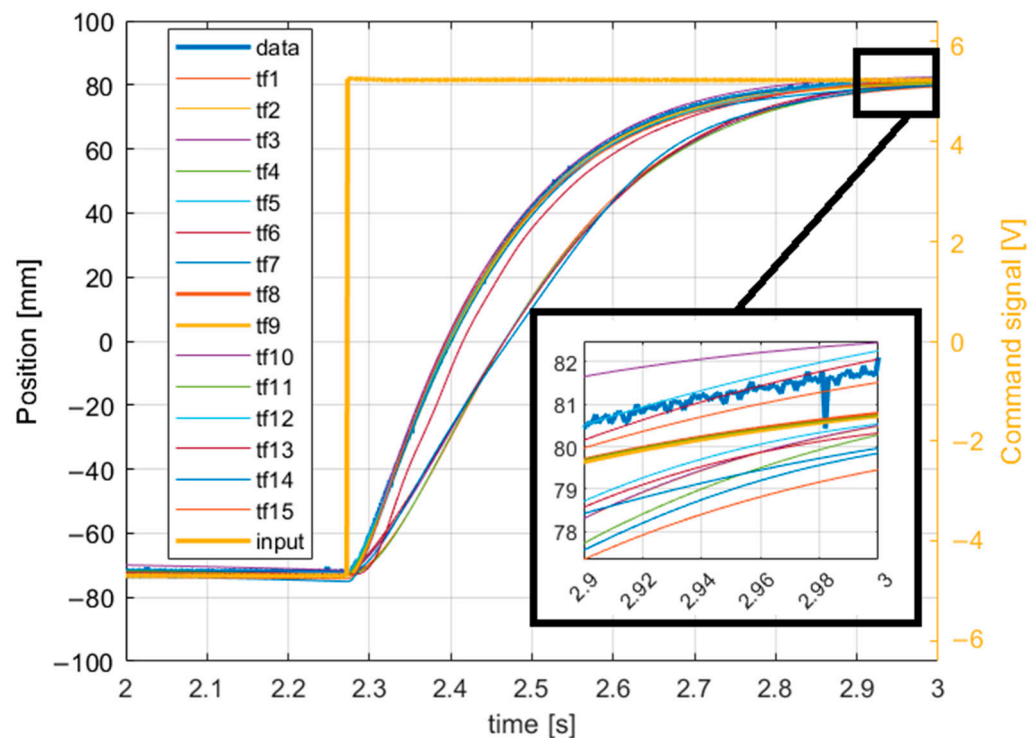


Figure 8. The transfer functions step responses vs. the step response of the hydraulic test rig obtained experimentally.

The third analysis of the selected TFs was made by comparing other quality indicators between all the measured sinusoidal responses with the TFs' sinusoidal response, consider-

ing the same input data. The other quality indicators considered are the maximum and minimum amplitude and the phase shift.

Table 3 presents the error between the quality indicators of the measured sinusoidal response with the TF's sinusoidal responses.

Table 3. Error between the quality indicators of the measured sinusoidal response with the transfer functions sinusoidal responses.

Transfer Function	Maximum Amplitude	Minimum Amplitude	Phase Shift
TF ₁	−3.526%	−0.887%	2.420%
TF ₂	−3.800%	−1.158%	2.411%
TF ₃	−3.782%	−1.138%	2.416%
TF ₄	−3.883%	−1.229%	2.488%
TF ₅	−3.230%	−0.582%	2.359%
TF ₆	−3.371%	−0.727%	2.356%
TF ₇	−3.647%	−0.998%	2.402%
TF ₈	0.646%	3.284%	0.577%
TF ₉	0.623%	3.278%	0.575%
TF ₁₀	1.916%	4.238%	0.641%
TF ₁₁	2.611%	5.313%	0.492%
TF ₁₂	0.959%	3.366%	0.705%
TF ₁₃	1.446%	3.899%	1.215%
TF ₁₄	−1.599%	1.172%	0.341%
TF ₁₅	0.635%	3.237%	0.597%

From Table 3, it can be seen that in the average case, the transfer functions TF_{15} , TF_9 and TF_8 present the smallest errors between their maximum and minimum amplitudes compared to the experimental ones in the given order: an error of 0.635% between the maximum amplitudes of TF_{15} and the experimental one, respectively, an error of 3.237% for the minimum amplitudes; an error of 0.623% between the maximum amplitudes of TF_9 and the experimental one, respectively; an error of 3.278% for the minimum amplitudes; an error of 0.646% between the maximum amplitudes of TF_8 and the experimental one, respectively; and an error of 3.284% for the minimum amplitudes. By analyzing the error between each TF response phase shift, it can be seen that TF_{14} , TF_{11} , TF_9 , and TF_8 present the smallest error compared to the other TFs (an error of 0.341% for TF_{14} , 0.492% for TF_{11} , 0.575% for TF_9 and 0.577% for TF_8).

According to the results of the analysis undertaken, the TFs with the most precise, accurate, and consistent response are the transfer functions TF_9 and TF_8 , being the same order as the one from the mathematical model obtained in Section 4.

Figure 9 depicts a comparison between the TFs' sinusoidal responses vs. the sinusoidal response of the hydraulic test rig obtained experimentally.

The upcoming paragraphs present a comparative analysis of the characteristics of the TFs selected as the most accurate TF_8 and TF_9 .

In Figure 10, the Bode diagrams of TF_8 (a) and TF_9 (b) are presented.

From Figure 10a, it can be seen that the gain margin is infinite, meaning there is no crossover frequency where the magnitude reaches 0 dB before the phase hits -180° . Suggesting that the system is robust in terms of gain variations and has no risk of instability due to excessive gain. Also, from the magnitude phase, it can be observed that at low frequencies, the magnitude is relatively constant, indicating that the system has a steady-state gain in this range and as the frequency increases, there is a clear roll-off at approximately -20 dB/decade. This behavior is typical of a system dominated by a single pole at higher

frequencies. The magnitude continues to decrease as frequency increases, which is desired for rejecting high-frequency noise or disturbances.

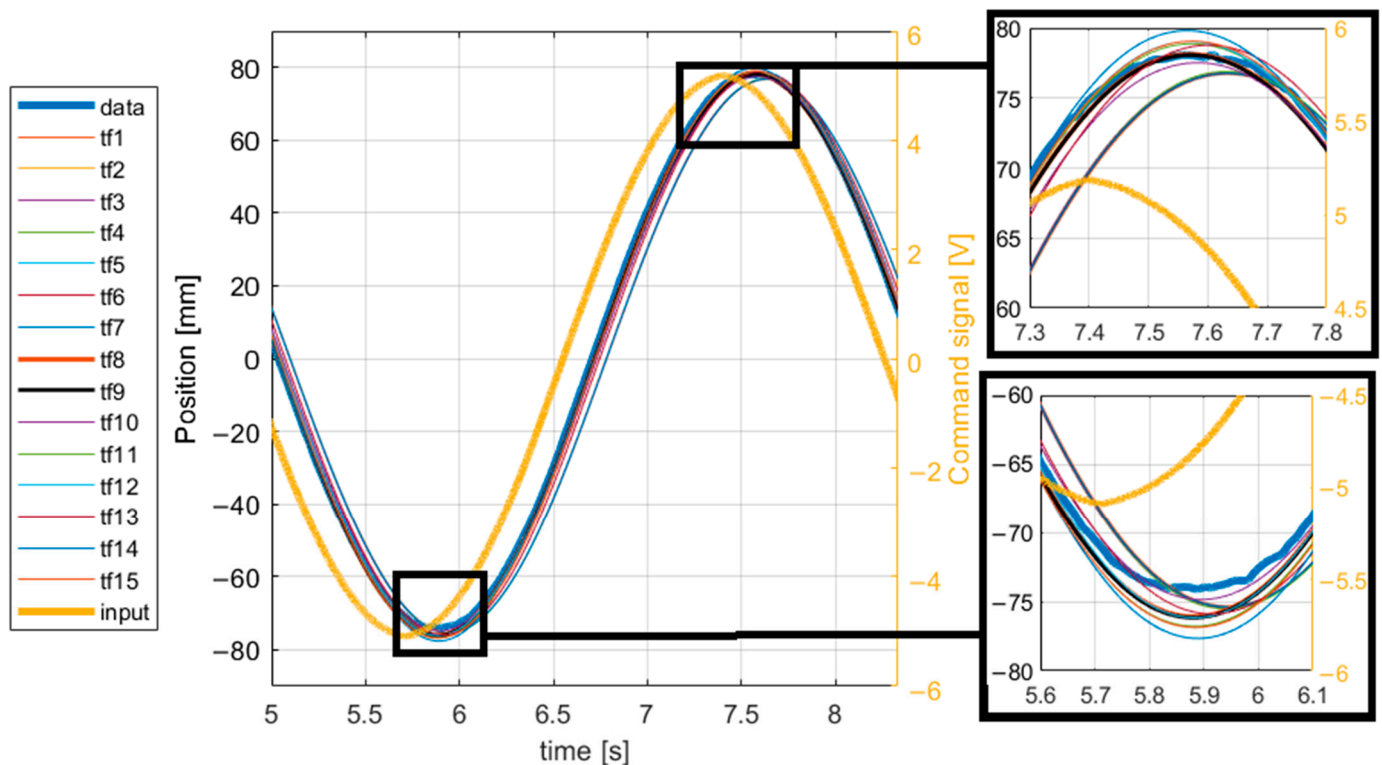


Figure 9. The transfer functions sinusoidal responses vs. the sinusoidal response of the hydraulic test rig obtained experimentally.

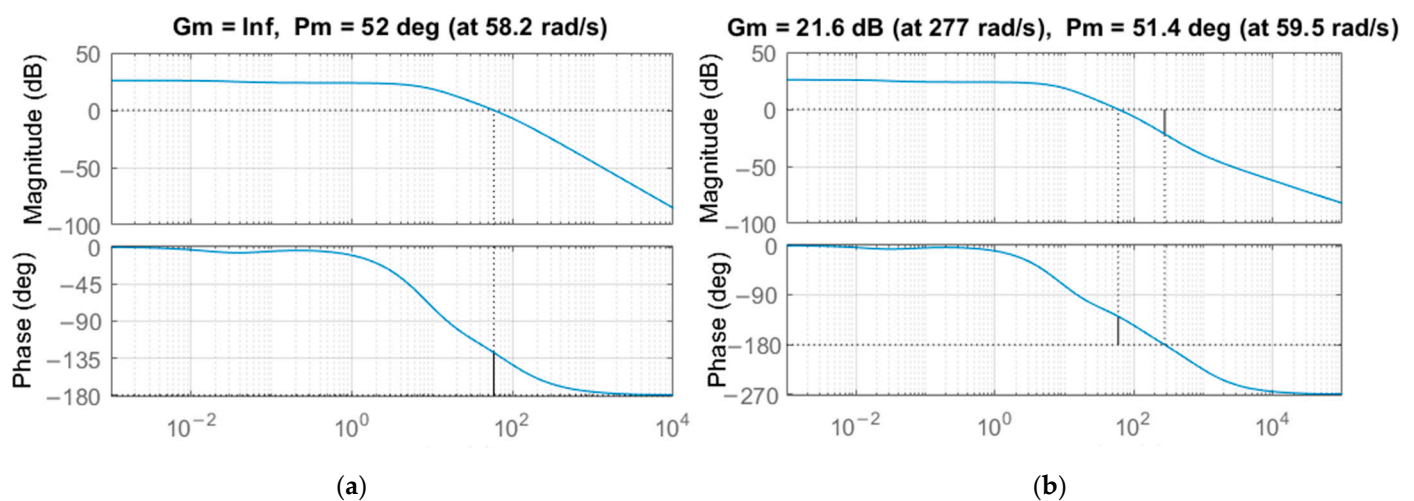


Figure 10. Bode diagrams of (a) TF_8 ; (b) TF_9 .

A phase margin of 52 degrees for TF_8 indicates a well-damped system with good stability characteristics. Generally, a phase margin between 45 and 60 degrees is considered acceptable for stability and reasonable transient performance. At very low frequencies, the phase starts close to 0° , indicating minimal phase lag in the system's response, and as the frequency increases, the phase begins to drop gradually. Around the crossover frequency of 58.2 rad/s, the phase reaches approximately -128° , which aligns with the phase margin of 52° (since $-180^\circ + 52^\circ = -128^\circ$). The phase continues to decrease beyond this point,

and the system maintains a relatively smooth transition, characteristic of a system without excessive poles or zeros in the right half-plane.

Based on the Bode diagram of TF_8 , it can be concluded that the system is stable and has a robust phase margin, meaning it can handle variations in gain without becoming unstable. The phase margin of 52 degrees ensures that the system is not prone to oscillations and can maintain stability even if there are minor changes in system parameters.

Regarding the transient response of TF_8 , because of the 52-degree phase margin, the system should have a moderate transient response, with minimal overshoot and good damping. This suggests that the system can handle input changes smoothly without excessive oscillations or delay.

The roll-off in the magnitude plot implies good attenuation of high-frequency signals, making the system effective at filtering out high-frequency noise.

From Figure 10b, it can be seen that TF_9 has a gain margin is 21.6 dB at a frequency of 277 rad/s. This indicates that the system can tolerate a 21.6 dB increase in gain before becoming unstable. A positive gain margin like this suggests that the system has a reasonable buffer against gain variations. At low frequencies, the magnitude remains relatively constant, indicating steady-state gain behavior in this region, and as the frequency increases, the plot shows a roll-off that begins at a rate of -20 dB/decade, suggesting a system dominated by a single pole. This behavior continues until it transitions into higher frequencies, where the magnitude decreases more sharply.

The phase margin is 51.4 degrees at a frequency of 59.5 rad/s. This value is very close to the generally acceptable range of 45 to 60 degrees, indicating good stability and a well-damped system response. The phase starts around 0° at very low frequencies, indicating minimal phase lag for low-frequency inputs and the phase drops progressively, and around 59.5 rad/s, it reaches approximately -128.6° (consistent with a phase margin of 51.4° since $-180^\circ + 51.4^\circ = -128.6^\circ$), as the frequency continues to increase, the phase further decreases, approaching -180° but never quite reaching that point within the frequency range shown.

Based on the Bode diagram of TF_9 , it can be concluded that with a gain margin of 21.6 dB and a phase margin of 51.4 degrees, the system is stable and has a reasonable level of robustness. It can withstand some variations in system parameters without becoming unstable.

Regarding the transient response of TF_9 , a phase margin of 51.4 degrees suggests that the system will have moderate overshoot and good damping. The system should respond to inputs without excessive oscillations and should settle relatively quickly.

The roll-off characteristics are beneficial for filtering out high-frequency disturbances.

Finally, it can be concluded that both TFs represent stable and well-damped systems, with phase margins around 51–52 degrees, providing adequate transient performance. TF_8 is more robust in terms of gain variations due to its infinite gain margin while TF_9 being also robust, has a finite gain margin of 21.6 dB, making it less tolerant to gain changes. Due to this reason, from the analysis of the Bode diagrams of both TFs (TF_8 and TF_9), TF_8 is preferable because it has higher robustness matching the robustness of the hydraulic setup and a simpler control structure.

In Figure 11, the pole-zero maps of TF_8 (a) and TF_9 (b) are presented.

From Figure 11a, it can be seen that all four poles (marked with “x” at the following values $p_1 = -79.3952$, $p_2 = -13.4756$; $p_3 = -7.9618$; $p_4 = -0.0322$) are located in the LHP (left half plane), indicating a stable system with some placed relatively far from the origin and others closer to it. The poles closest to the origin are the dominant poles, governing the overall transient response of the system representing slower dynamics, while the poles further from the origin represent faster dynamics that decay quickly and do not significantly

influence the long-term behavior of the system. The two zeros (marked with “0” at the following values $01 = -24.2071$ and $02 = -0.0407$) indicate frequencies where the system output will have a reduced or no response. Zeros have a significant impact on shaping the frequency response, potentially adding phase lead or lag, depending on their location. Because the zeros are close to the origin or on the left half-plane, they improve response speed or reduce steady-state errors.

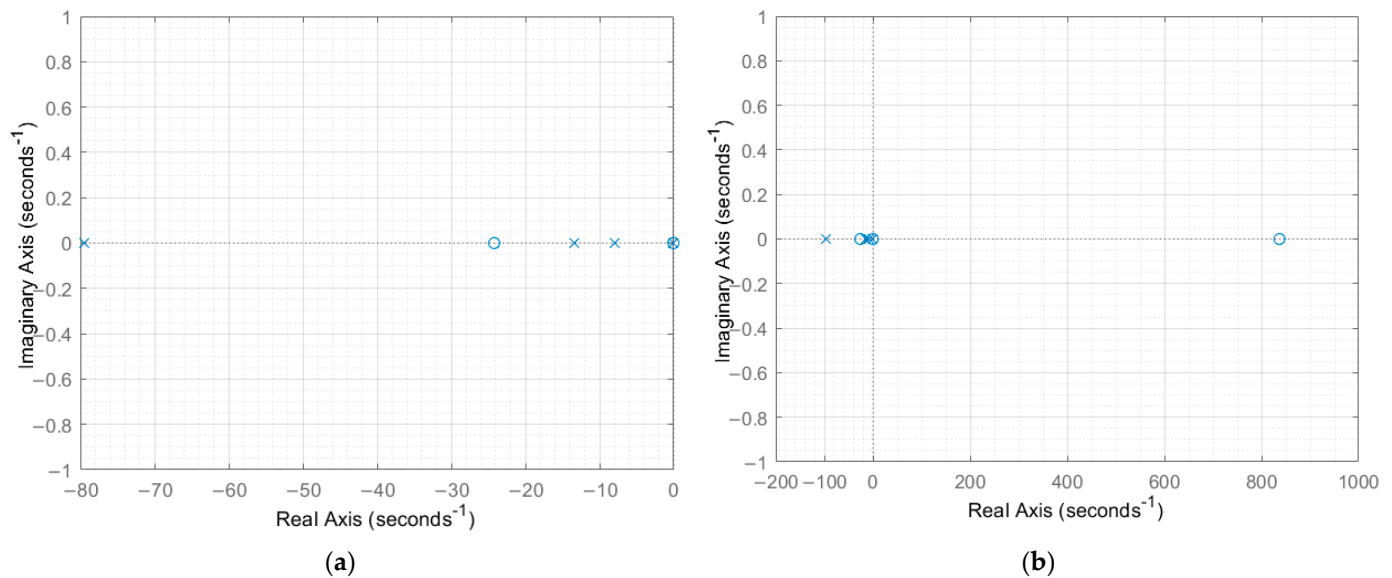


Figure 11. Pole-zero maps of (a) TF_8 ; (b) TF_9 .

From Figure 11b, it can be seen that the poles of TF_9 ($p1 = -96.3721$, $p2 = -14.0994$; $p3 = -7.8553$; $p4 = -0.0270$) are almost the same as the ones of TF_8 , leading to the same dynamics of the system. Regarding the zeros of TF_9 , the first two are almost the same ($01 = -26.0582$ and $02 = -0.0339$), apart from the third zero ($03 = 836.8597$) located in the RHP (right half plane, far away from the origin) that affects the transient response and steady-state performance, introducing phase shifts or affecting the overall system dynamics. The zero from RHP far away from the origin may not significantly impact the system's behavior at low frequencies but could influence high-frequency response or phase characteristics.

Finally, introducing the third zero (located in the RHP far away from the origin) makes the response of the system slower and affects the system's steady-state behavior. This zero has a more pronounced effect on the overall system dynamics, particularly at higher frequencies. Overall, the dynamics of TF_8 fit better than the dynamics of the hydraulic setup.

Figure 12 illustrates a comparison between the Nyquist diagrams of TF_8 (depicted by the blue line) and TF_9 (depicted by the blue line), with arrowheads indicating the direction of the two branches: negative ($-\infty$ to 0) and positive (0 to ∞). The red marker (“+”) indicates the point $(-1, 0)$.

From Figure 12, it can be concluded that both TFs are stable and behave similarly; TF_8 may be a little more sensitive to external disturbances due to its slightly larger Nyquist contour while TF_9 seems to offer a more conservative response, potentially providing better robustness in practical scenarios.

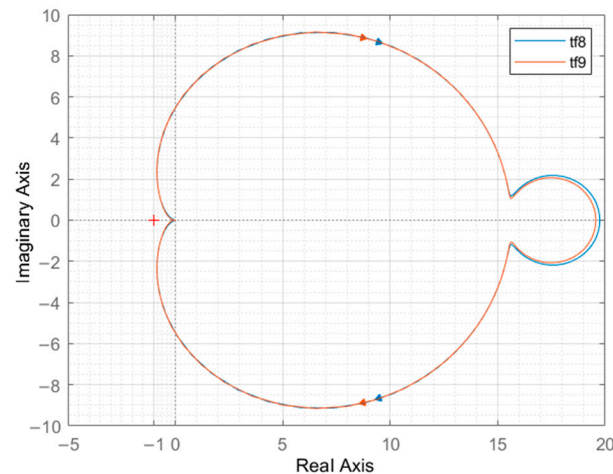


Figure 12. Nyquist diagram of TF_8 and TF_9 .

7. Conclusions

In conclusion, the results obtained in this study recommend that the hydraulic stand described in Section 2, together with the data acquisition and logging system developed and proposed in Section 3, be used for the experimental identification, testing, or validation of models and control strategies for the wind turbine blade pitching system.

The hydraulic system used in this paper replicates with remarkable fidelity the operational characteristics of the blade pitching systems currently used in industrial wind turbines, considering also the variable load on it, performed through the loading system. The loading system is protected against cavitation occurrence [16,17] through a pressurized system composed of a feeding pump and a check valve system.

The data acquisition and logging system proved versatile and efficient in experimental identification and determining the performance of control strategies. Its adaptability allows for the inclusion of additional transducers or sensors [33], extending its applicability to a wide range of dynamic systems.

The main contribution of the current research consists of the experimental identification methodology presented in Section 5, which offers improved precision, reliability, and efficiency over traditional analytical approaches. This method identified 15 transfer functions of varying complexity (various poles and zeros), with transfer functions TF_8 and TF_9 being the most accurate based on quality indicators such as overshoot, rise time, settling time, phase shift, and amplitude. The mathematical model proposed in Section 4 is confirmed by the experimental identification, being in the same order as the one experimentally identified (same poles, different zeros), meaning that the mathematical (analytical) model is recommended when identification is not possible.

Both TF_8 and TF_9 exhibit stable and well-damped responses, as confirmed by Bode, pole-zero, and Nyquist analyses, but based on system dynamics, TF_8 aligns more closely with the actual wind turbine blade pitching system behavior.

This experimental identification method is recommended for relatively low-order systems (≤ 8), as the identification time increases exponentially with the increase in the model order. Furthermore, the quality of the data used (particularly how effectively it is filtered) has a significant impact on the accuracy of the identified model.

In future research, for further enhancing the hydraulic stand capability for dynamic testing and validation, the pressure relief valve (H13—DBDH 10 G1A/100) will be replaced with a proportional pressure relief valve, in order to emulate a real-time loading profile exerted by the wind on the wind turbine blades.

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Conflicts of Interest: The authors declare no conflicts of interest.

Appendix A

Table A1. The flow transducer Omni RT technical specifications [24].

Technical Properties	Values	Unit
Supply voltage	+18 to +30	VDC
Power consumption	<1	W
Measuring range	0.8...8	m ³ /h
Measurement accuracy	+/-1	%
Medium temperature	-20...+85	°C
Ambient temperature	-20...+70	°C
Storage temperature	-20...+80	°C
Max. particle size	0.5	mm
Pressure loss at Qmax	0.3	Bar
Analog output	4...20	mA
Protection to DIN 40 050	IP 67	-

Table A2. The pump line pressure transducer Mannesman Rexroth HM-15-1X/600 technical specifications [25].

Technical Properties	Values	Unit
Operating voltage	+10 to +30	V
Current consumption I max	10	mA
Measuring range	0...600	Bar
Over-load protection	900	Bar
Burst pressure	2000	Bar
Dead volume V	approx. 450	mm ³
Output signal	4 to 20 (2 conductors)	mA
Temperature compensation		
Zero point type	≤0.08%/10 K; max. ≤ 0.15%/10 K	K
Range type	≤0.08%/10 K; max. ≤ 0.15%/10 K	K
Linearity tolerance type	≤0.1% 1; max. ≤ 0.3% 1 (from 100 bar max. ≤ 0.2%)	%
Hysteresis type	≤0.05% 1; max. ≤ 0.1% 1	%
Repeatability	≤0.05% 1	%
Rise time	t ≤ 0.5 ms	ms
Long-term drift (6 months):		
Zero signal	≤0.1% 1	%
Range	≤0.1% 1	%
Ambient conditions		
Nominal temperature range	ϑ -25 to +85 °C	°C
Operating temperature range	ϑ -40 to +85 °C	°C
Storage temperature range	ϑ -40 to +100 °C	°C
Medium temperature range	ϑ -40 to +100 °C	°C
EMC compatibility to IEC 801-4 Shock	severity class 3	-
	500	g/ms
	Vibration resistance to IEC 68-2-6 (at 10 to 500 Hz) 20 g	
	Protection to DIN 40 050 IP 65	

Table A3. The cylinder pressure transducer PS-200-D-G2-1.

Technical Properties	Values	Unit
Supply voltage	+18 to +36	VDC
Measuring range	0...200	Bar
Operating temperature	−10...+85	°C
Accuracy	+/-5	%

Table A4. The LVDT position transducer Monitran MTN/EICR150 [26].

Technical Properties	Values	Unit
Linearity	<5% of full stroke	mm
Measuring range	+/-150	mm
Operating temperature	−30...+85	°C
Accuracy	+/-5	mm
Output	4...20	mA

Table A5. The force transducer Lorenz K-25 20kN [27].

Technical Properties	Values	Unit
Nominal force F_{nom}	20	kN
Accuracy class compression force or tension force	0.1	% F_{nom}
Accuracy class compression force and tension force	0.2	% F_{nom}
Rel. repeatability error in unchanged mounting Position b_{rg}	0.08	% F_{nom}
Relative creep	<±0.06	% $F_{nom}/30$ min
Rated characteristic value C_{nom}	$2.00 \pm 0.1\%$	mV/V
Input/output resistance R_e/R_a	350	Ω
Insulation resistance R_{is}	$>2 \times 10^9$	Ω
Rated range of excitation voltage $B_{U,nom}$	nom 2...12	VDC
Reference temperature T_{ref}	23	°C
Rated temperature range $B_{T,nom}$	−10...70	°C
Operating temperature range $B_{T,G}$	−30...80	°C
Storage temperature range $B_{T,S}$	−50...95	°C
Temperature effect on zero signal TK_0	±0.04	% $F_{nom}/10$ K
Temperature effect on characteristic value TK_C	±0.12	% $F_{nom}/10$ K
Maximum operating force F_G	130	% F_{nom}
Force limit F_L	150	% F_{nom}
Breaking force F_B	>300	% F_{nom}
Permissible oscillation stress F_{rb}	70	% F_{nom}
Lateral force resistance	50	% F_{nom}
Rated displacement S_{nom}	<0.25	mm
Preferential direction	Tension direction	
Material	Stainless steel	

Table A6. The signal generator Tektronix CFG253 [28].

Characteristics	Measurement
Outputs	Square wave, sine wave, sawtooth wave, TTL pulse, and sweep functions for all outputs
Line Voltage at 50–60 Hz	Range 90 to 110, 108 to 132, 198 to 242, and 216 to 250 VAC
Frequency ranges, nonskewed waveform (Freq/1)	0.3–3000 Hz
Frequency ranges, skewed waveform (Freq/10)	0.03–300,000 Hz
Frequency multiplier	Variable 0.3 to 3.0 times the selected frequency range
Frequency/1 dial accuracy	±5% of full scale of frequency/1
Frequency/10 dial accuracy	±5% of full scale of frequency/10
Sine wave distortion	<1% from 10 Hz to 100 kHz

Table A6. Cont.

Characteristics	Measurement
Sawtooth linearity	20 Hz to 200 kHz_99%
Square response	200 kHz to 3 MHz_97%
Main output amplitude	<100 ns rise/fall time maximum output into 50 Ω load Two ranges: 0...20 V peak to peak; 0...2 V peak to peak.
Impedance	50 $\Omega \pm 10\%$
DC offset	<−10 V to >+10 V (open circuit), and <−5 V to >+ 5 V (into 50 Ω load)
Duty cycle	5:1 minimum duty cycle change (50% at Center:Cal position), with symmetry button (Freq 10) pushed in
Sweep rate	Continuously variable from 0.5 to 50 Hz
Sweep width	Variable from 1:1 to 100:1

Figure A1 presents the block diagram of the data acquisition board DT9804 [29].

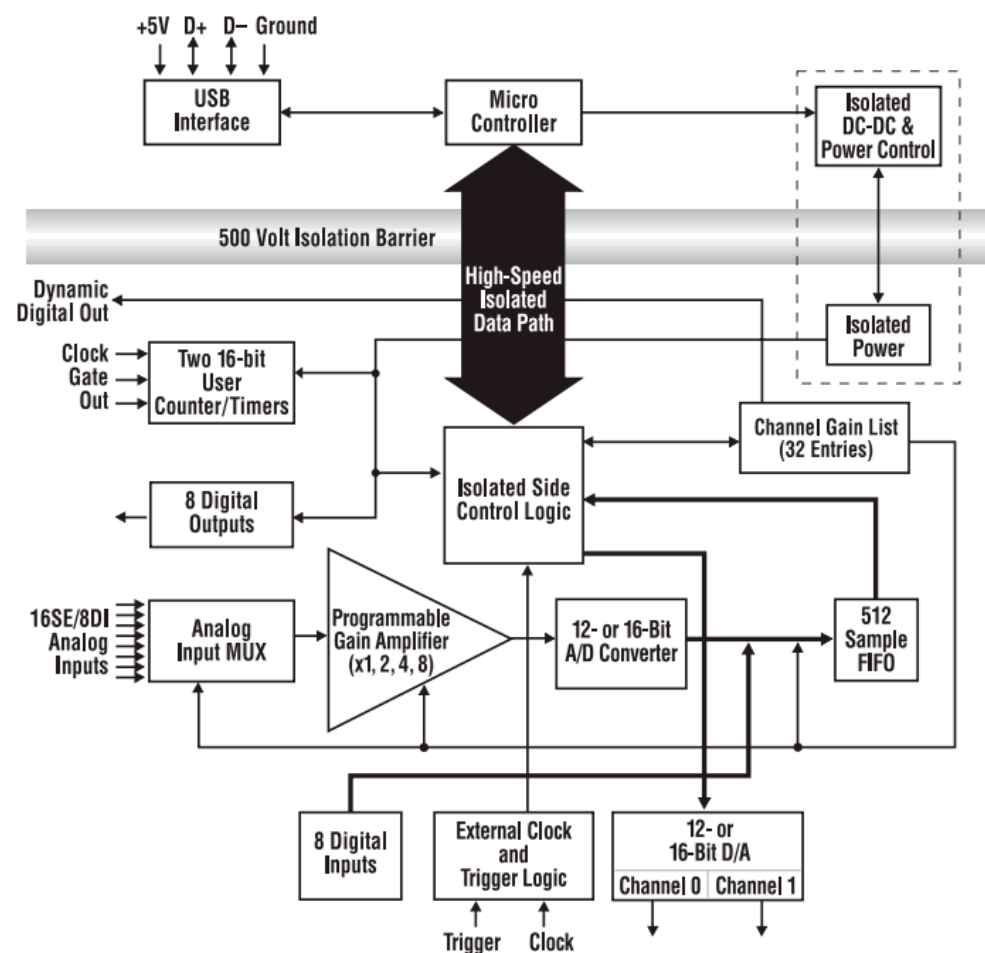


Figure A1. Data acquisition board DT9804 block diagram [29].

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