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MODELING AND CONTROL OF A HYDRAULIC DYNAMOMETER

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Abstract. *This paper describes the implementation of a digital control system on an old hydraulic dynamometer that originally had analog control board. The dynamometer´s break is commanded by the internal pressure generated by the water's vortex in the hydraulic circuit. The internal pressure is commanded by a butterfly valve coupled to a DC servomotor controlled by an analog PID board. The aim of this study is to replace this analog system by a digital control system. Therefore, the model of the dynamometer is characterized using the unidimensional integrated theory and the modeling of flux vortex. The digital control of the dynamometer is implemented in two parts: one will use an Arduino microcontroller to control the DC servomotor. The other part, implemented in ScadaBR, will permit the performance of several tests strategies of the dynamometer.*
Keywords: *Hydraulic Dynamometer, Control, modeling.*

1. INTRODUCTION

In order to verify the performance of automotive internal combustion motors it is necessary to measure its torque and its velocity. In general there are three forms to test these motors (Killerdar, 2012):

1. Testing the motor on a dynamometer;
2. Testing the vehicle on a dynamometer roller;
3. Testing the vehicle on a test road.

The dynamometer can reproduce the diverse environment to which the motor is submitted, and diagnose possible problems quickly and securely without the inconvenient of having to test the vehicle on the road at each modification. Besides power measurement, performance failure, leaks and the origin of noises can be tracked in real time.

The dynamometer is equipment that works as a brake on the motor and collects load and motor rotation data, though obtaining the power of the motor under test. The dynamometer test bench permits it to obtain calibration maps, performance tests, fault diagnostics, measuring of pollution emissions gases, testing new fuels and lubricants and materials.

This paper presents the hydraulic dynamometer DYNABAR of SCHENCK of Brazil. Hydraulic dynamometers are used in a wide range of motors, from kart motors to ship motors. The have a low inertia related to its torque capacity, but their non-linearity complicates its modeling and control (Passenbrunner, 2011). The DYNABAR shown in figure 1, has a hydraulic brake of the kind D2101E and was fabricated in 1980 and is used in combustion laboratory of the University of Brasilia for didactic purposes and research on internal combustion engines. Its instrumentation measures torque and rotation speed of the examined engine.

Figure 1. Dynamometer DYNABAR and a diesel engine (left).

It has an analogic control board (PID control) as shown in figure 2 that controls the hydraulic water flow through the dynamometer by commanding a butterfly valve coupled to a direct current position servomotor.

Figure 2. Analogic Board Control.

The analog control board also collects rotation and torque data of the engine under test. Due to its time of use, the board has showed some problems as repeatability and maintenance. Formerly it was possible to make tests of constant speed, proportional speed to torque, constant momentum torque, constant pressure, proportional momentum torque to squared speed and outer entrance (SHENCK do Brrasil, 1979). Nonetheless, the board suffered some damages and so nowadays it only permits constant speed test.

2. HYDRAULIC MODEL

In order to model the dynamometer the one-dimensional theory will be used. Two factors have to be considered to obtain the dynamometer's dynamic behavior. One is the fact that it works with the working fluid partially filled within the hydraulic chamber; another is the fact that the valves are considerably more ticker than others torque converters (Hodgson, 1991). Figure 3 shows an internal view of an hydraulic dynamometer.

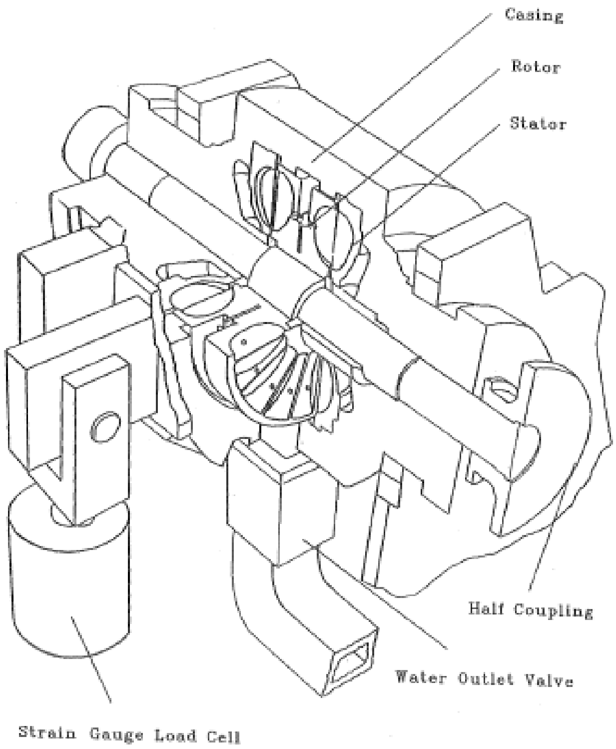

Figura 3. Shematic vision of the internal parte of a general hydraulic dynamometer. Fonte: Hodgson, 1991.

The conservation mass theory is applied to the stator with inclusion of the valve's thickness and non-axial valves. The conservation of angular momentum of water flows through the rotor gives torque. When the conservation of mass theory is used for the stator, its gives the center of vortex and is considered invariant with fluid fill. To quantify the mass flow, a expression of the stream tube flow was derived and then integrated. Rearranging the mass flow at inlet and outlet and using the method of Newton-Raphson gives the vortex flow radius (R_m) showed at equation 1 (Hodgson, 1991).

$$R_M = \frac{\frac{2}{3} * \pi \sin\alpha_{SO} * (R_O{}^3 - R_I{}^3) - \frac{1}{2} * Z_S * t * (R_O{}^2 - R_I{}^2)}{\pi \sin\alpha_{SO} * (R_O{}^2 - R_I{}^2) - \frac{1}{2} * Z_S * t * (R_O - R_I)} \quad (1)$$

α_{SO} = vane stator angle; R_O= Cup outer radius; R_I= cup inner radius; R_{CO} =air-water interface outer radius; R_{CI} =air-water interface inner radius; Z_S= stator vane number ; t= vane thickness;

The power inlet to the rotor must transfer the work fluid to keep the energy conservation. Both torque and energy dissipation are functions of percentage fill (PF). The rate that water fills the machine must quantify where the boundaries of integration are given by air-water radii interaction. The equation 2 represents the percentage fill that considers mass flow of the fluid and of the air too (Hodgson, 1991).

$$PF = 1 - \frac{(R_{CO} - R_{CI})^2 * (\pi * (R_{CO} - R_{CI}) - \frac{Z*t}{\sin\alpha}}{(R_O - R_I)^2 * (\pi * (R_O - R_I) - \frac{Z*t}{\sin\alpha}} \quad (2)$$

$\propto$=Vane angle; Z=vane number; t=vane thickness;

The rotor controls volume to conserve the angular momentum, the torque shaft (Tshaft) inlet is equal to the rate of change of angular momentum of fluid throw the rotor. The strength surfaces are zero if ignore the viscosity and symmetry allowing the strengths in the body to be neglected, thus obtain eq. 3 that represents shaft torque (Hodgson, 1991).

$$T_{Shaft} = \frac{\partial}{\partial t} \iiint_{CV} r \times V * \rho * dV + \iint_{CS} r \times V * \rho * V_{xyz} * dA \quad (3)$$

V=absolute velocity of the flow; ρ=fluid's density; r=radius from machine axis; Vxyz=relative velocity of the fluid.

The first term of the equation is an angular momentum on the volume control, while the second is a net quantity that pass through the surface control. For steady and unsteady flows this integrals must be accomplishment.

Water lose a parcel of power inlet by three ways: incidence loss, changing speed between stator and rotor; friction loss, contact of valves with toroid surface; and secondary circulation loss, water flow vortex.

The work fluid flows through a helicoidal way and then has a variation in its pressure that has effect in self-emptying behavior of machine. The stator pressure is lower than in the rotor. The pressure is generated by action of fluid vortex in work machine. Water flows in a rotate vortex at reference plan that rotate with the same speed of machine element where the fluid is analyzed. Applying Newton's second law to the fluid, rearranging and integrating, the net pressure is given by eq. 4 (Hodgson, 1991).

$$\begin{aligned} P = \rho * [&\left(\omega^2 + 2 * \omega_p * \omega * \sin\gamma_V * C_{WV}\right) * \frac{1}{2} * r_V{}^2 + \left(\dot{\omega_p} * \sin\gamma_A * C_{WR} + \dot{\omega_p} * \sin\gamma_R * C_{WWR}\right) \\ &* \left\{\frac{1}{2} * (r_V - R_M * \cos\theta) * \left(r_V{}^2 + R_M{}^2 - 2 * r_V * R_M * \cos\theta\right)^{\frac{1}{2}} - \frac{1}{2} * (\cos^2\theta - 1) * R_M{}^2 \right. \\ &\left. * \ln\left|(r_V - R_M * \cos\theta) + \left(r_V{}^2 + R_M{}^2 - 2 * r_V * R_M * \cos\theta\right)^{\frac{1}{2}}\right|\right\}] + C \end{aligned} \quad (4)$$

Where ρ=fluid's density; ω=angular velocityof the fluid; ω_P=angular velocity of the rotor; γ_V,γ_A,γ_R=angles between vectors; Cwv, Cwr, Cwwr=cofactors; r_V=distance from vortex centre; θ=cup angle; C=integration constant;

The dynamometers' model can be simplified by two rotors linked by a transmission as shown in figure 4.

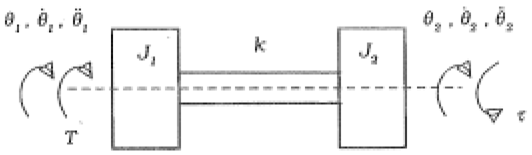

Figure 4. Dynamometer motor system representation. Source: Hodgson, 1991

The sum of the torques in the motor and dynamometer is:

$$J_1 * \dot{\omega}_1 = T - k * \delta \tag{5}$$
$$J_2 * \dot{\omega}_2 = k * \delta - \tau \tag{6}$$

Where J_1 and J_2 =polar inertia of the motor and dynamometer; T=motor torque; τ= dynamomter torque ; ω_1 e ω_2=rotor velocities;

The difference between the rotors positions is defined as relative angle displacement (δ). So the difference between velocities is:

$$\dot{\delta} = \omega_1 - \omega_2 \tag{7}$$

Input power to rotor should be transferred to the working fluid to maintain power conservation. Both torque and power are fill percentage functions (PF). The rate which the water fills the compartment should be quantified. Mass flow is compared at inlet and at outlet of the working compartment making possible to obtain the fill percentage. Taking in account constant density water (ρ) during the process the fill percentage is:

$$\frac{\partial}{\partial t}(PF) = \frac{1}{V_F} * (Q_i - Q_o) \tag{8}$$

Q_i e Q_o=Flow at inlet and outlet; V_F=maximum volume on fluid compartment.

Using the eq. 3 static and dynamic terms are defined that together represent torque of the dynamometer eq. 9.

$$\tau = K_1 * \omega * \omega_p + K_2 * \omega^2 + K_3 * \omega_p + K_4 * \omega + K_5 * \dot{\omega}_p + K_6 * \dot{\omega} \tag{9}$$

K_1, K_2, K_3, K_4, K_5 and K_6 = geometric coeficients.

Eq. 5, 6, 7 and 8 represent the model of the system dynamometer-motor according to Hodgson.

3. CONTROL STRUCTURE

The analog controller has all the functions to command and regulate the hydraulic brake. It has also protection devices to permit the controllers functioning in all service modes. The diagram for the constant velocity mode is presented in figure 5 and has three control loops.

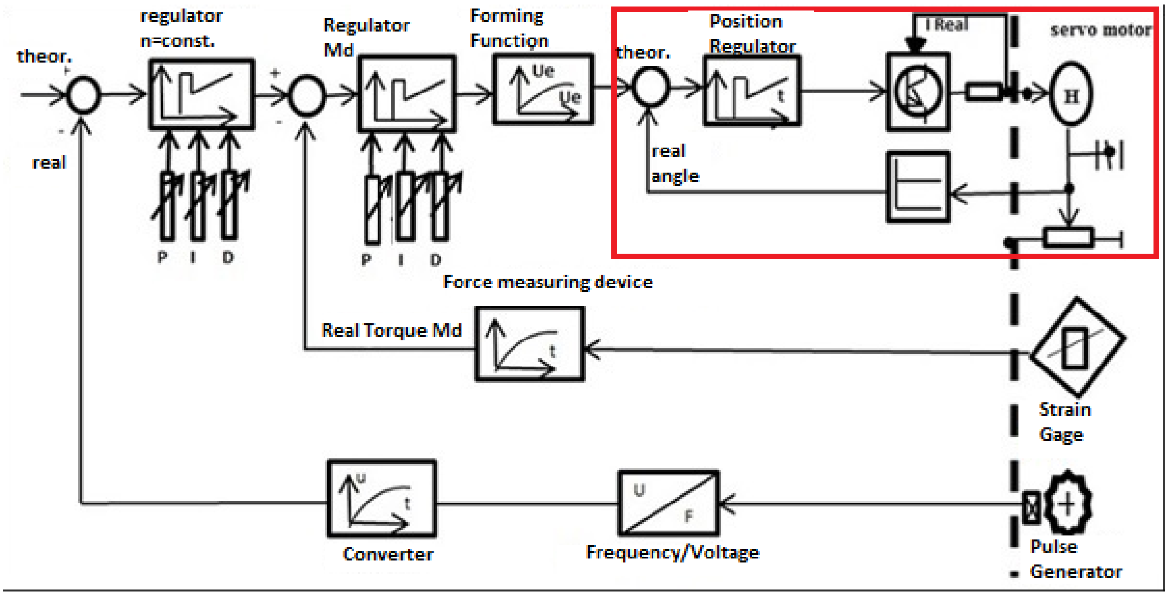

Figure 5. Analogic Board Control. Fonte: Shenck do Brasil, 1979.

At the upper right side of the figure the DC servomotor coupled to a potentiometer is represented. The potentiometer indicates the position of the butterfly valve. The servomotor is commanded by a position control loop marked in red in figure 5 where the potentiometers signal is fed backed to position controller that will command the butterfly valve.

Beneath the strain gauge to measure the motor's torque is represented and it is part of a torque control feedback loop with a programmable PID controller. The motor velocity sensor is implemented by a magnetic pulse transducer and is also part of a feedback control loop. The velocity control loop has also a programmable PID controller and its reference is at the left side of figure 5.

The analog controller is substituted by a digital control system implemented with a supervisory computer and an Arduino microcontroller board. The Arduino control board is implementing the servomotor position controller while the supervisory computer will implement the other control loops.

3.1 Arduino DC servo controller

The supervisor system is implemented in the ScadaBR platform and in figure 6 the complete configuration is presented. The positioning of the servomotor is implemented with a PWM power source controlled by the Arduino board. The real position of the butterfly valve is read with the Arduino's Analog to digital converter. The ScadaBR software in the supervisor computer is connected to the Arduino control board through a serial RS232 communication link. The strain gauge sensor is connected to a measurement board that is linked to the ScadaBR through another RS232 communication line with the modbus protocol. The velocity of the motor under test is measured with a special indicator board that is linked to the ScadaBR through an RS485 communication line.

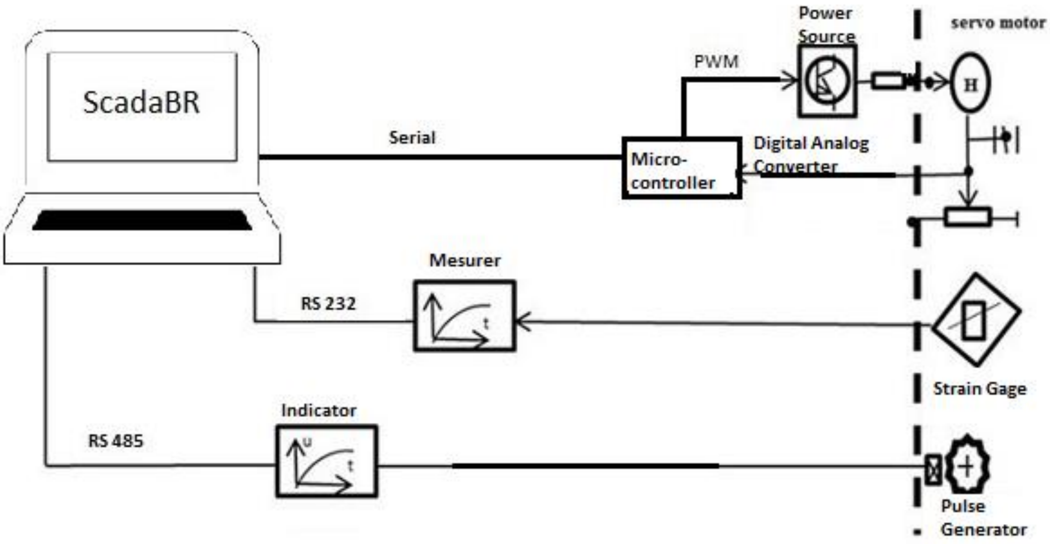

Figure 6. New Control and Supervision of Position.

As said before, The Arduino will control the position of butterfly valve that will follow the reference indicated by the ScadaBR software. The control algorithm in the Arduino is a PID controller implemented by the 2° method of Ziegler-Nichels.

The 2° Method is applied with a closed loop where *error=reference-position*. When applies a critical gain where the system behavior starts to oscillate. The oscillation was considered near unitary gain (Kcr=1). This oscillation has a critic period equal to the reverse of grid frequency (Pcr= 0,0175).

Table 1. Values of PID tune 2° method.

Controller	K_p	T_i	T_d
P	0,5	∞	0
PI	0,45	0,014583	0
PID	0,6	0,00875	0,002188

These values are replaced in the discrete PID formula (Eq. 15), found by backwards, forwards differences methods and trapezoidal approximation (Alves, 2010).

$$u(k) = K_{p(Digital)} * e(k) + K_{i(Digital)} * \sum_0^{j=k} e(j) + K_{d(Digital)} * [e(k) - e(k-1)] \quad (15)$$

Rewriting the values from table 1 to implement the controller we obtain table 2.

Table 2. Values of the digital PID controller.

Digital controller	K_p	K_i	K_d
P	0,5	-	-
PI	0,411429	0,077143	-
PID	0,6	0,171429	0,525
PD	0,6	-	0,525

The parameters where implemented in the control algorithm and figure 7 represents the results to a step response of the DC servomotor with P, PI, PD and PID control. The data was measured with the ScadaBR system.

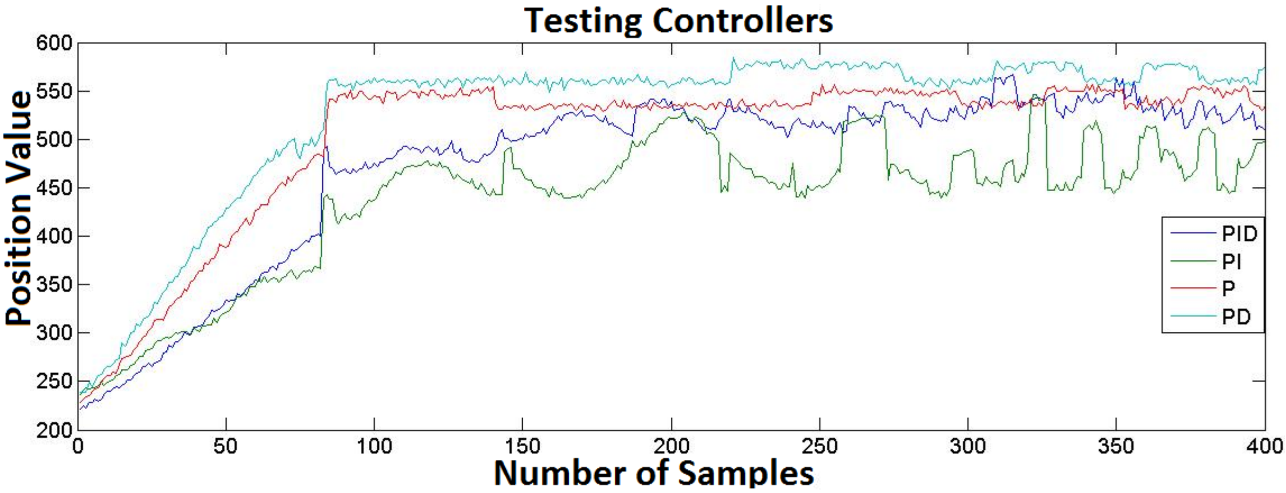

Figure 7. Servomotor response to step input for P,PI,PD and PID.

As noticed in figure 7 the P and PD algorithms are more stable. Nonetheless, PID control just shows good control in smalls changes in references, while P or PI controller are more sturdy. The PD algorithm was chosen and it was fine tuned for several parameters.

4. RESULTS

The values used in the controller were the same as the ones used by Roos and Montijo (2005). Figure 8 shows the serveral parameters tested in the system, showing that the values of Kp=3 e Kd=1 had the best performance.

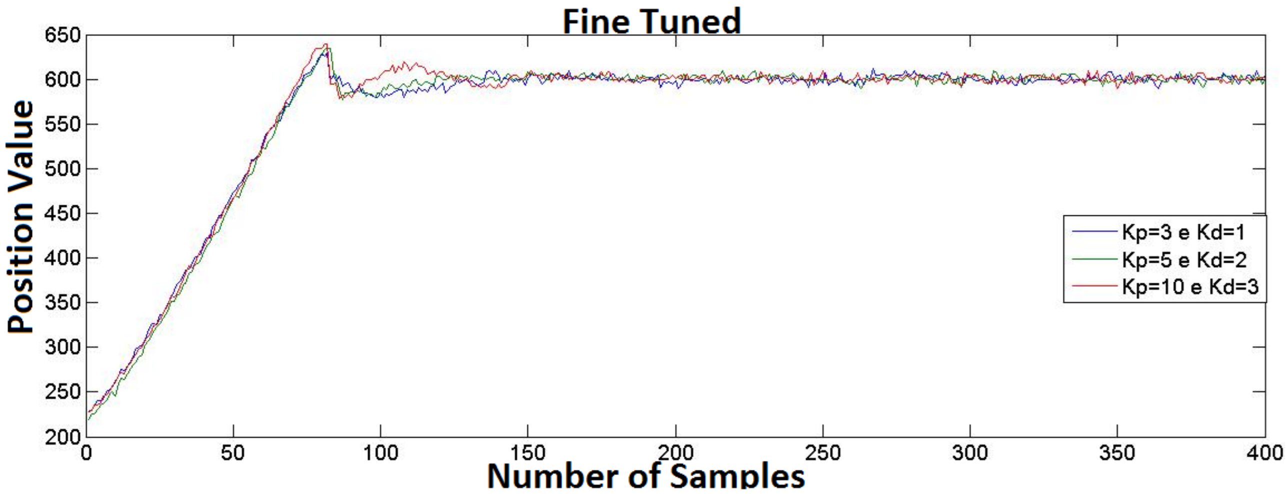

Figura 8. Fine tune of PD controller.

The analog controller of the Dynamometer was substituted by and Scada system implemented by ScadaBR and an Arduino digital servo controller. The three separate analog control loop can be implemented by this system. The first servo loop to control the position servo of the hydraulic butterfly valve is implemented by the Arduino controller board. The two outer loops to control respectively the torque and motor velocity can be implemented by the supervisory computer. In order to obtains the control parameters it is necessary to derive the hydraulic dynamometer model. With the supervisory system it is possible to obtain the K1, K2, K3, K4, K5 and K6 constant of the simplified dynamometer model. With this proposed controle structure using ScadaBR and Arduino it will be possible to performance the several tests strategies of the dynamometer.

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