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MODELING A PNEUMATIC SPEED GOVERNOR USING ELETRONIC PRESSURE VALVES AND DIRECTIONAL VALVES

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Abstract. *In this paper, a mathematical model of a pneumatic system is presented as an alternative for the hydraulic systems used to control the guide vanes mechanism in francis turbines of small hydropower plants. The pneumatic technology has lower acquisition cost, good power-to-weight ratio, easy to use, low maintenance and cleaner operation (does not uses oil). The system was designed to operate with components that are widespread in industrial applications, which have a good quantity of manufacturers, reliability, and availability of replacement parts. The main components of the system are an asymmetric pneumatic actuator, an electronic pressure control valve, a standard directional control valve and a pressure reducing valve. The control of the system is made by a P.I.D. Controller that controls the electronic pressure valve and an algorithm that controls the directional control valve, commutating the last one according to the load profile. The simulation is carried out in the software Matlab Simulink, using the parameters provided by the company Reivax S/A Automation and Control of a Francis turbine that has 438 kW of generating capacity and net height of 15.5 m. To achieve the precision required by the international standards of the application, the main state variables and nonlinearities of the pneumatic system are modeled, obtaining the maximum error of 1.05 mm or 0.8 % of the actuator's stroke.*

Keywords: *pneumatic drives, position control, speed governor, Francis turbines and dynamic modeling.*

1. INTRODUCTION

Hydroelectric powerplants are a fundamental part of the Brazilian energy matrix, corresponding to 12% of the country's energy matrix and 60% of the electrical energy of Brazil (Aneel, 2018). The hydroelectric powerplants are classified by their power generating capacity, according to Aneel (2018), in three main groups: Hydropower Generating Plant (HGP or in Portuguese: "Central Geradora Hidrelétrica") with generating capacity up to 1 MW, Small Hydropower Plant (SHP) with generating capacity up to 30 MW and Hydroelectric Power Plant with generating capacity above 30 MW. The scope of the research project is to design pneumatic governor systems focused on powerplants with generating capacity up to 3MW, thus corresponding to HGP and SHP. Currently, in Brazil, there are 701 HGP and 426 SHP operating units, along with 7 HGP and 133 SHP units in construction or to be constructed.

This relevant energy source has as its main components a penstock, hydroelectric turbine and an electric generator (Itaipu, 2018). The turbine is responsible for converting the potential energy stored in the water column into mechanical energy, which is converted to electrical energy by the generator. These hydroelectric turbines can be of several types and models, however, only the control mechanism and algorithms of Francis turbines will be analyzed in this paper.

In Francis turbines, the water flow and the frequency of the generated energy depend on the guide vanes position of the turbine. The energy frequency can vary according to the demand of the electrical grid and the guide vanes are responsible to maintain a constant frequency. The system that controls the position of the guide vanes is called Speed Governor and it is traditionally actuated by hydraulic systems, as can be seen in Fig. 1. For smaller applications, such as the scope of this paper, the hydraulic system does not have the distributor valve and the proportional valve is connected directly on the hydraulic actuator.

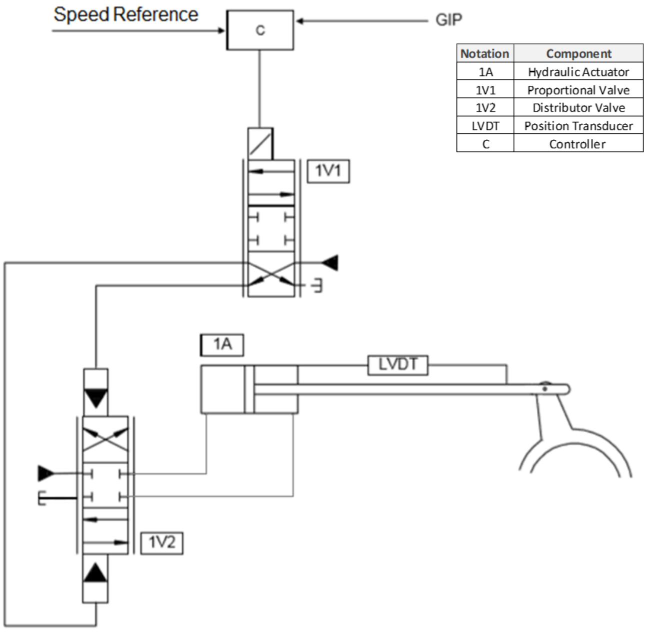

Figure 1. Hydraulic Actuation System. Adapted from Furst (2011).

The advantages of using pneumatic actuators in this application are relative to the low cost of acquisition, easy to install (plug and play), widespread technology in the industry with high reliability and availability of components and manufacturers, and do not use oil (environmentally friendly).

The control of the proposed pneumatic system is made through the control of the pressure in one chamber of the actuator along with constant pressure on the other chamber. The controller used in this configuration was a Proportional, Integrative and Derivative (P.I.D) controller and the technical requirements of the system are based on the IEEE 125 (2011), IEC 61362 (1998), IEC 60308 (2005) standards and specialized bibliography.

The pneumatic system modeled in this paper has the goal of replacing the hydraulic system used to control the guide vanes position. To properly evaluate this new approach, the parameters used in this model comes from a real Francis Turbine, which has a generation capacity of 438 kW and a net height of 15.5 m.

2. TECHNICAL REQUISITES IN THE GOVERNING SYSTEM OF FRANCIS TURBINES

The technical requisites considered in this paper are based on the international standards of the application and specialized bibliography. These technical recommendations regard:

- Maximum closing time

The maximum closing time is specified regarding the over speed allowed on the turbine and is given for this application as 2.5 seconds.

- Minimum opening time

The minimum opening time is specified regarding the water hammer on the penstock and is given for this application as 4 seconds.

- Emergency closing

In emergency situations, the operators and/or the control system has to have the possibility to shut down the system.

- Servomotor's Time constant (T_y)

The time constant recommended by the IEC 61362 standard represents the time that the actuator takes to vary 63 % of the positioning step given as reference. The recommended values for this constant vary from 0.1 to 0.25 seconds for the main servomotors of francis turbines. For very small displacements, higher values of T_y are acceptable.

Due to the non-linearity of the system modeled in this paper and to properly compare this dynamic requisite, it will be converted into settling time, which can vary from 0.5 to 1.25 seconds ($5.T_y$).

- Control accuracy

The control accuracy for the position of the servomotor is given by the IEC 1207 (2004) as 1 % of the actuator's stroke and since the stroke of this application is 131 mm, the maximum error of the system is ± 1, 31 mm.

- Load force

The load force that is applied to the pneumatic actuator by the water flow and friction on a Francis turbine is shown in Figure 1. According to Voith (1974), Apud Mendoza (2006), the maximum load on the system occurs on around 80-90% of the actuators' stroke and is zero around 15% of the total stroke. In Figure 2, the opening load is the orange curve, the static load is the blue curve and the closing load is the gray curve. The maximum value of the load curve used is according to Schreiber (1978) and is shown in Equation 1, where k is a constant of proportionality, L is the stroke of the actuator, P_{max} is the maximum generation potency installed and H is the net height of the turbine.

$$F_{max} = \frac{1}{75} \cdot \frac{k}{L} \cdot \frac{P_{max}}{\sqrt{H}} \tag{1}$$

Understanding the load profile of which the pneumatic system is subjected is extremely important when sizing the system. By knowing this profile, the oversizing of the system can be avoided and consequently the overall efficiency of the system and the range of applicable components can be increased.

Since the transition between the curves is yet unknown, it is used for simulation purposes an Adjusted Load Profile (yellow curve), which still evaluates the maximum load values and the region where the load is zero.

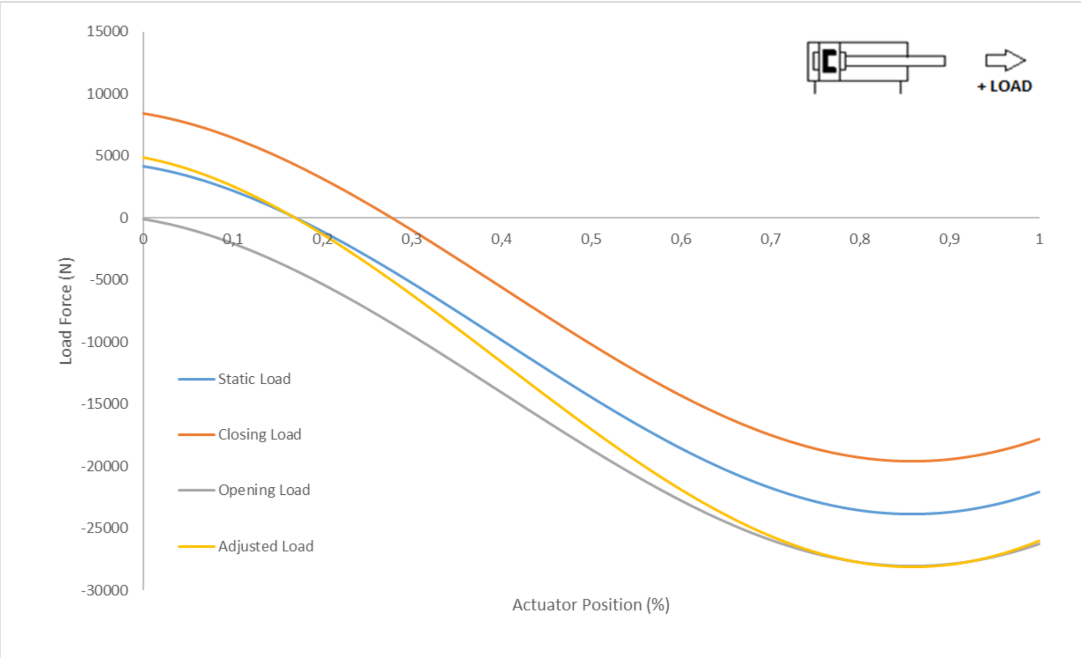

Figure 2. The load applied in the actuator by the water (Static Load) and friction in the mechanism. Adapted from Voith (1974), Apud Mendoza (2006).

3. PNEUMATIC GOVERNING SYSTEM MODELLING

According to Saravanakumar (2017), a pneumatic system for position control has as the following basic structure: Actuator, a flow control mechanism, airpower source, sensors for data acquisition and a control system. In this paper, only the relations between the airflow control mechanism, actuator, and control system will be analyzed.

The system was designed in a way that the position control can be done without using proportional valves (5x3) or on-off valves with fast switching, due to the fact that these kinds of valves have a very limited number of manufacturers, being more expensive (5x3 proportional valves) and are not suitable for applications that require long lifecycle (fast switching valves).

The circuit of the pneumatic system analyzed in this paper was sized based on the methodology presented in Mendoza (2006) and is shown in Figure 2, which is considered to be connected directly to the turbine's guide vanes mechanism. The model used for the dynamic analysis can be found in Vigolo (2018) and is used due to the fact that considers nonlinearities like friction and temperature dynamic behavior.

The flow control mechanism for chamber A is an electronic pressure regulator valve (V2) and, for chamber B is a directional control valve (V3) along with a pressure reducing valve (V5). The emergency valves (V4 and V1) were designed to be commutated on normal operating conditions, upon emergency (when the system needs to be closed immediately) V4 supplies the maximum pressure available to the chamber B and V1 exhaust the air in chamber A. The reason of using on-off valves in the designed system is to a set constant pressure on chamber B and the position control is made through the electronic pressure regulator valve (EPR).

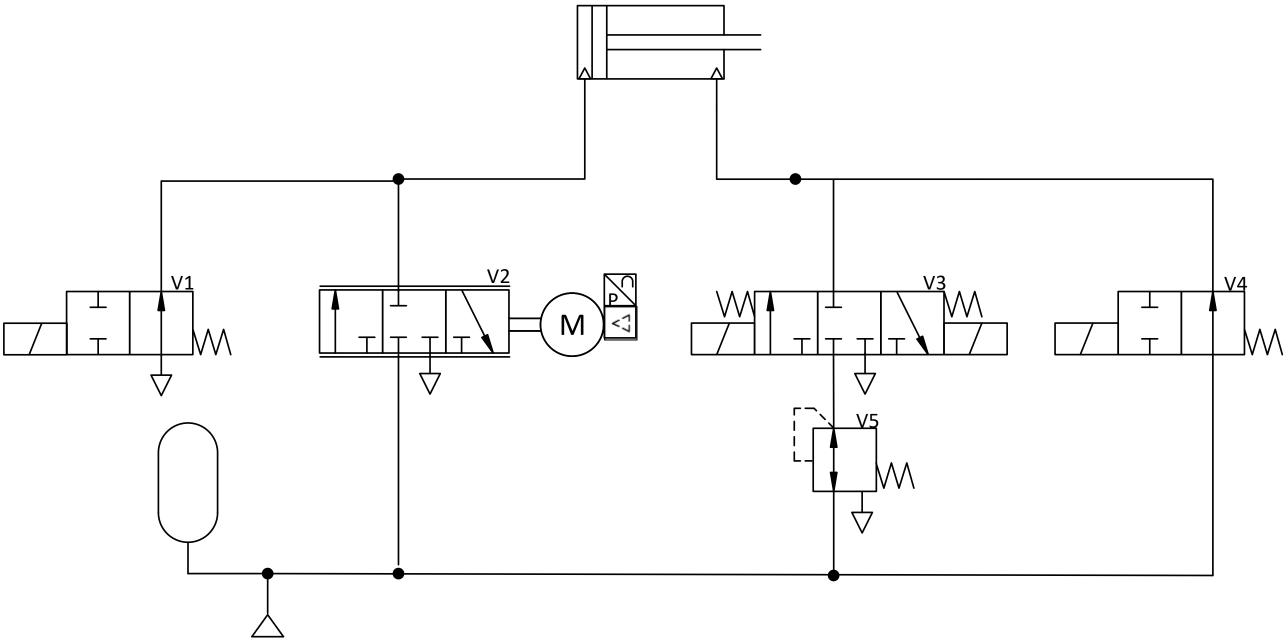

Figure 3. Proposed pneumatic circuit.

The parameters used for the components in the simulation were taken from datasheets of standard industrial components, the supply pressure considered was a 0.9 MPa and the room temperature set to 293,15 K. The pneumatic actuator considered in the simulations have d_{piston} = 250 mm, d_{rod} = 40 mm and stroke of 131 mm.

The sonic conductance of the valves is (C) 3.34e-8 m^5/N.s and pressure ratio (b) is 0.3 which can be found in the datasheet of Camozzi's electronic pressure regulator (EPR) LRPA-34-2-2-00. The on-off valves and pressure regulator (V1, V4, V5) shown in Figure 2 are considered to have the sonic conductance 1.5 times bigger than the ERP valve. The other parameters used in the dynamic model can be found in Vigolo (2018).

3.1 Directional and Eletronic Pressure valve model

The model used to represent the mass flow through the directional valve (V2) and electronic pressure valve (V1) is the equation of the ISO Standard 6358 (2005), shown in Equation 2, where p_1 is the upstream pressure, p_2 is the downstream pressure, T_1 is the upstream temperature and T_0 is the atmospheric temperature, ρ_0 is the air density at atmospheric pressure and temperature, and is considered that the parameters C and b are the same for each pair of ports of all valves. The dynamic of the solenoid and spool movement of the valves are neglected because the dynamic response of the system is much slower than the valve's dynamics.

$$q_m = \begin{cases} p_1.C.\rho_0.\sqrt{\frac{T_1}{T_0}}.\sqrt{1-\left(\frac{\frac{p_2}{p_1}-b}{1-b}\right)^2}, & para\ \frac{p_2}{p_1} > b \\ p_1.C.\rho_0.\sqrt{\frac{T_1}{T_0}}, & para\ \frac{p_2}{p_1} < b \end{cases} \tag{2}$$

The directional control valve (V3) sets the pressure on the chamber B according to the position of the actuator (x), in steady-state operation, which can be 0.45 MPa (for x ≤ 25%) or 0.1 MPa (for 25% < x ≤ 100%). The switching frequency of this valve is much lower than 5 Hz, which is the maximum switching frequency for a standard industrial directional control valve. This valve can be used for this function because the normal operating conditions of small turbines are to be near the maximum opening of the actuator and then excessive switching is avoided.

The electronic pressure valve is modeled in a way that it receives a signal for pressure reference from 0 to 10 V, corresponding to the regulating range of 0.1 – 0.9 MPa of the LRPA-36-2-2-00 valve. The valve will then open or close its control orifice in order to control the pressure on the chamber A and match the reference. This valve model has an embedded closed loop controller, which gains are not given in the valve's datasheet and because of that, the valve control algorithm used in the simulation is a proportional controller with gain equal to one. The effect of this assumption is that the gains used in the P.I.D. Controller will be higher than those used when operating a real ERP valve.

3.2 Pneumatic actuator modeling

The actuator is modelled trough the evaluation of Newton's Second Law (Equation 3), which gives as outputs the acceleration, speed and position of the actuator as a function of the pressure on the chambers (p_a and p_b), area of the piston on the chamber A (A_A) and chamber B (A_B), atmospheric pressure (p_0), the piston rod area (A_0), friction (F_{at}) and load force (F_{load}), which are shown in Equation 03.

$$p_A A_A - p_B A_B - p_0 A_0 - F_{load} - F_{at} = M.\frac{d^2x}{dt^2} \tag{3}$$

The friction model considered is the LuGree model, along with the parameters used by Vigolo (2018) and adapted due to the diameter of this actuator being 2 times bigger than the one used in the reference. This adaptation consists of multiplying the friction force by the ratio of perimeters, which is $K_{friction} = 2$. This approach is used because the LuGre's parameters or the static friction map for this size of the actuator are not available and for precise positioning systems, the friction dynamic effects on the system's dynamics are a very important behavior to be analyzed.

In order to evaluate the dynamic behavior of the pressure and temperature in each chamber of the actuator, the continuity equation, the first law of thermodynamics and the ideal gas state equation are used. The dynamic behavior of the pressure is shown in Equation 4, where T is the temperature in the actuators chambers, R is the ideal gas constant, $q_{m_{in}}$ and $q_{m_{out}}$ are the mass flow in the chambers.

$$\frac{dp}{dt} = \frac{p}{T}.\frac{dT}{dt} + \frac{1}{V}\left(q_{m_{in}}.T.R - q_{m_{out}}.T.R - p.A.\frac{dx}{dt}\right) \tag{4}$$

$$\frac{dT}{dt} = \frac{\frac{\delta Q}{\delta t} \pm p.A.\frac{dx}{dt} + q_{m_{in}}.\left(C_p.T_{in} - C_v.T\right) - q_{m_{out}}.T.R}{C_v.m} \tag{13}$$

The temperature behavior was described by the energy equation where Q is the heat flow of the chamber, C_p and C_v are the specific heat at constant pressure and volume, respectively, and m is the air quantity inside the chamber in a specific time. The ± indicates the possibility of the system performing work to the ambient or the ambient performing work to the system.

The heat exchange model is done according to newton's law of cooling. Neglecting the radiation heat transfer, the heat transfer between the ambient and the interior walls can be expressed as:

$$\frac{\delta Q}{\delta t} = \lambda.A_{sur}.(T_{amb} - T_{ch}) \tag{14}$$

Where A_{sur} is the surface area that is exchanging heat to the ambient, T_{amb} and T_{ch} are the ambient and chamber temperatures, respectively, and λ is the constant global coefficient of heat exchange. Since λ is an experimental parameter, the mean value of the ones presented in Beater (2007), Apud Pasieka (1991) is used. Other parameters used for the dynamic temperature analysis are the same as used in Vigolo (2018).

4. CONTROL STRATEGY

Analyzing the load profile from Figure 1 one can see that from 0% up to 15% of the actuator's stroke, the load force is positive and after 15-20% of the total stroke, the load direction is negative. This attribute of the load profile allows the system to control the position of the actuator only with precise control of the pressure in one chamber. The control of the system is made by two algorithms, which are shown in Fig. 4, being one algorithm for the directional control valves and a P.I.D. Controller for the ERP Valve.

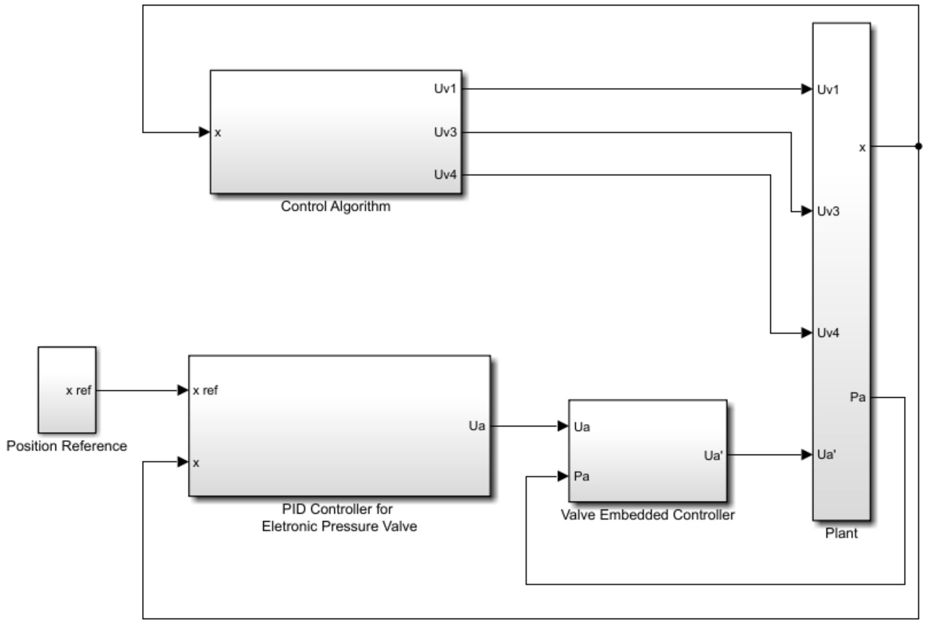

Figure 4. Proposed Control algorithms.

The control algorithm (CA) that controls the directional valves is a function of the piston position. If the position of the actuator is below 25% of its stroke, the CA will switch on the valve V2, which will set the pressure of 0.45 MPa (set by the manual pressure regulator V3) on chamber B. If the position of the actuator is above 25% of its stroke, the CA will switch off the valve V2 and set atmospheric pressure on chamber B. For the case that fast closing of the distributor is required, the valve V3, V1, and V4 will be switched off (until 25% of the actuator's stroke, when V3 is switched on and V2 is switched off) and will set intermediate pressure (0.45 MPa) on the chamber B, the valve V1 will set the chamber A to atmospheric pressure, closing the distributor under the specified time. For the case that fast opening is required, the valve V4 and V1 are switched on and the valve V3 is switched to set atmospheric pressure to chamber B. This opening and closing logics aim to avoid both water hammer and overspeed effects on the turbine's equipment.

The P.I.D. controller used for the ERP sends a control signal as the pressure reference, which goes to the valve's embedded controller (proportional controller with gain equal to 1) and is compared to the pressure on the chamber A. Due to the nonlinearities of the designed pneumatic system, such as the varying load, the use of two different constant pressures in chamber B and the fast and precise response required for the application, the gains of the controller needed to be relatively high. The proportional gain used was equal to 3100, which resulted in oscillations around the reference and to solve this, high values of derivative gain were used, which is equal to 600. The integral gain set was 30.

The load profile may change for different francis turbines, according to the intrinsic characteristics of the equipment, as well as the wear in the components, however, the load profile should be similar. For the case which the positive parcel of the load has higher magnitude, setting the constant pressure (controlled by the valve V5) on higher values tend to maintain the systems dynamic behavior and the control characteristics.

5. RESULTS AND DISCUSSIONS

The simulations were carried out in the software Matlab® Simulink® and the results obtained, considering the full opening and full closing of the actuator, are shown in Fi. 4. Both movements are in agreement with the opening and closing times specification, which were around 4 seconds for the opening and 2.5 seconds for the closing movement.

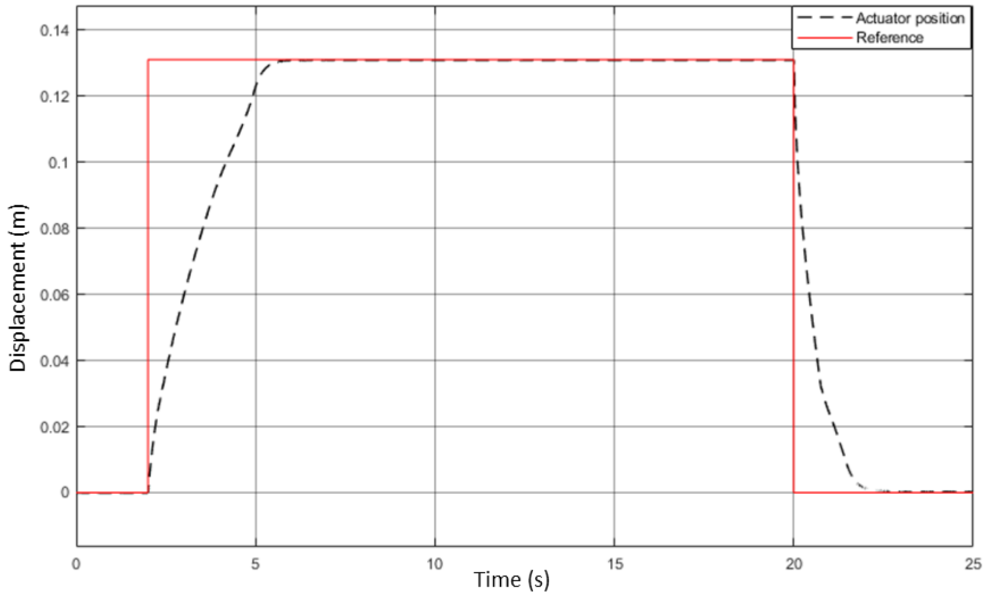

Figure 4. Opening and closing movements of the actuator.

Fig. 5 shows the step response of the actuator positioning. Fig. 6 presents the positioning error. The position steps vary from 0 to 100% of the actuator's stroke. The system was able to control the position of the actuator throughout all its stroke with a steady-state error being under the tolerance limits (± 1%). The maximum steady-state error obtained was 0.8% of the actuator's stroke, which corresponds to 1.05 mm.

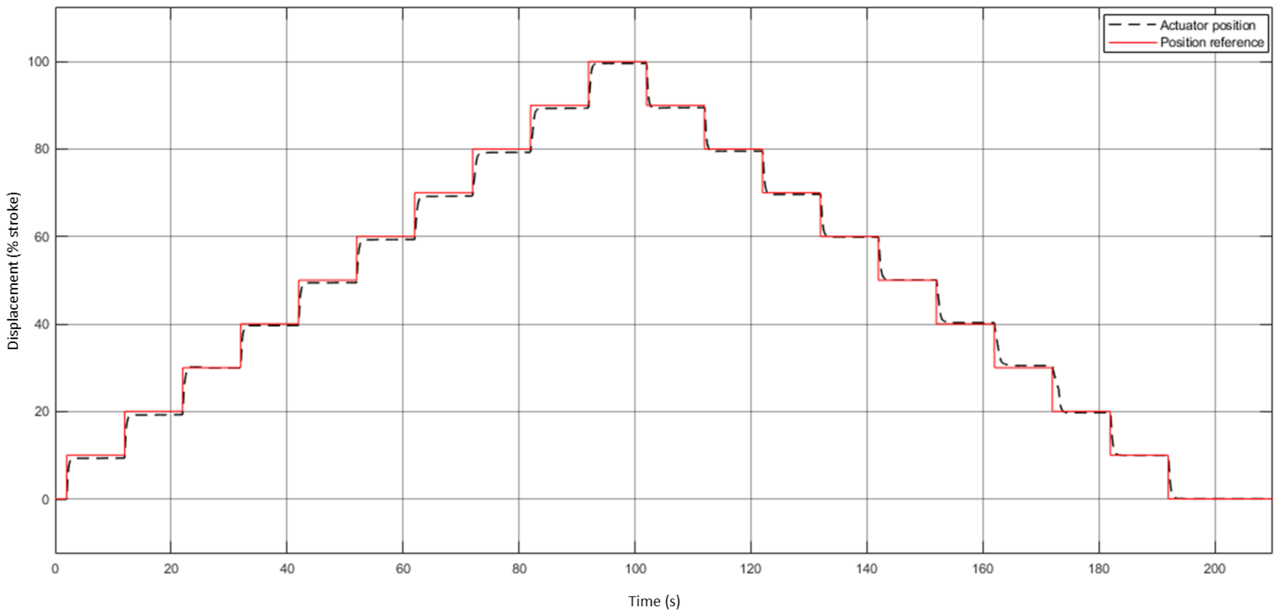

Figure 5. Position step response of the actuator.

From Fig. 7 the settling times of the actuator in two different conditions is presented. Fig.7 (b) shows that in the region of the maximum load the settling time of the system is approximately 1.5 seconds. In Fig. 7 (a) is shown the dynamic of the system in a condition which the negative load force is relatively low and the pressure in chamber B is still atmospheric. The settling time was approximately 1.5 seconds. This value is higher than the one recommended by the IEC 61362 (1998) standard, however, in small steps, a higher value is accepted.

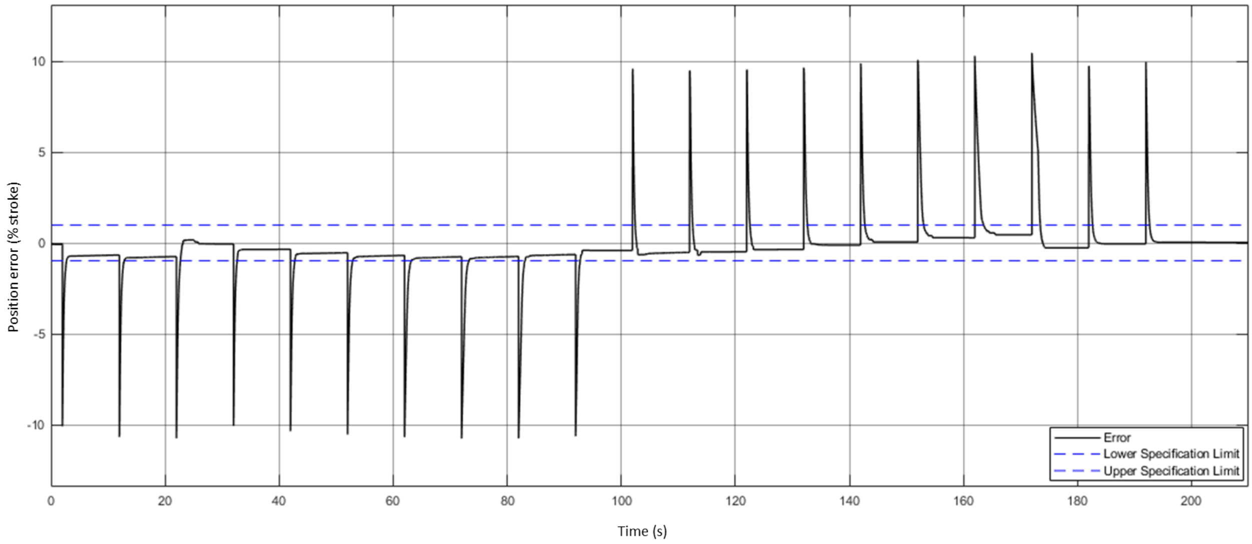

Figure 6. Positioning error and the specification limits.

Figure 7 also shows the moment when the 0.45 MPa is connected to chamber B, which occurs at 173 seconds. This change notably affects the actuator speed, changing the system's dynamic. This behavior is the main reason why the proportional gain of the system has to be high to control both situations alongside a varying load.

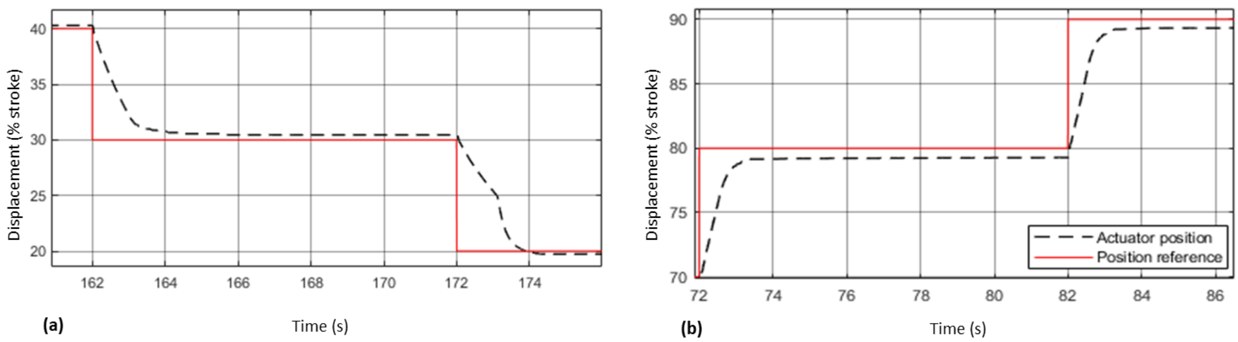

Figure 7. Settling times in two different positions.

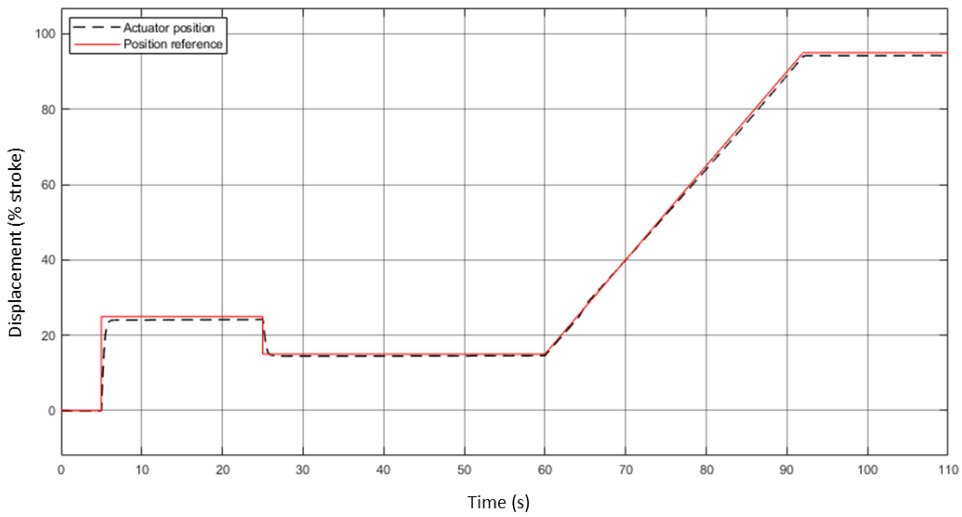

Figure 8. The positioning of the actuator simulating a synchronizing operation.

In Fig. 8 is presented the position response of the actuator when the position reference is given as the turbine is synchronizing with the electric generator. In this movement, the actuator is held at a low opening for a period of time and then the reference goes slowly to the operational point, which for small turbines tends to be near the maximum opening of the actuator.

It is relevant to state that the load profile used in this paper is an approximation and both magnitude and profile can change depending on the turbine's design. In order to apply this pneumatic approach effectively, the profile must be evaluated for each application.

6. CONCLUSION

This paper presented an alternative solution for the position control of the guide vanes of francis hydraulic turbines. The results showed that the system dynamic characteristics attended to the required technical requisites. The analysis performed in this paper also shows that the range of industrial pneumatic components and different arrangements that can be applied for pneumatic systems can increase, which allows the developing of a wide range of systems for small hydroelectric power plants.

Although the pneumatic solution for this application tends to have a lower cost and other advantages when compared with the traditionally used hydraulic systems, its economic viability is bonded to having all the automation systems needed to operate a turbine (not only the speed governor), such as the brake system and actuation of the inlet valve with a pneumatic power source. The reason why all the automation systems must have the same power source is that the cost and the amount of oil used in a hydraulic power unit are very similar when it actuates one or all the systems, which makes the economic viability of using both power sources unfeasible to be achieved.

In future studies, another technical requirement that will be evaluated is related to the control valve and actuator dynamic aspects, such as dead zone, friction force and experimental temperature parameters, which are relevant for system dynamics behavior and precise positioning applications such as the system modeled in this paper.

It is also relevant to develop a control algorithm that controls the on-off valves based on different strategy, such as actuate the directional valve of chamber B based on the pressure differential of the chambers instead of the actuator's position. This upgrade in the algorithm will help the control of the system when applying the pneumatic solution in a real turbine, in which the load profile might change. For the EPR valve, an adaptive P.I.D. controller might be useful in order to obtain different gains for each part of the system, which tends to reduce the proportional gain and thus reduce the oscillations in the real system.

7. ACKNOWLEDGMENTS

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