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# MODELING A HYDRO-PNEUMATIC SYSTEM TO ACTUATE A TURBINE INLET BUTTERFLY VALVE

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***Abstract.*** *The objective of this paper is to develop a mathematical model to analyze the dynamic behavior of a hydro-pneumatic system applied to actuate turbine inlet butterfly valves at small hydropower plants. It is composed of commercial components, which provide good access to various manufacturers and replacement part suppliers. The pneumatic components have a good power-to-weight ratio, lower acquisition cost and clear operation, whereas hydraulic components are robust and work with high pressures and loads. The parameters of the analyzed system are from a commercial equipment designed to supply 30 tons, enough to overcome the torque requirements of the current turbine inlet butterfly valve from a hydropower plant with 438 kW of generation capability, based on parameters provided by REIVAX S/A Automation and Control and requirements from American Water Works Association standards. The aim of the proposed system is to replace the conventional hydraulic actuating systems that use a large amount of oil by this compact system powered by compressed air that uses a considerably smaller oil quantity. The mathematical model was implemented in Matlab Simulink and the results show a smooth displacement of the main actuator, despite the intermittent fluid pumping. Moreover, the hydraulic pressure followed the resistive forces profile accordingly, indicating that with the correct design, the system is suitable for a wide range of hydropower plants.*

**Keywords:** Hydro-pneumatic, turbine inlet butterfly valve, hydropower plant, modeling.

## 1. INTRODUCTION

Hydraulic sources represent about 60.7 % of the total Brazilian production of electrical energy. Just considering small hydropower plants, there is more than 5.2 GW of installed power divided among 426 facilities. In addition, there are another 30 small hydropower plants under construction and 105 being designed (ANEEL, 2019).

Every hydropower plant is composed of a minimum set of essential elements for operation. Among these components, there is the turbine inlet valve that controls the water flow from the reservoir to the turbine. Due to the high force required to actuate it, it is convenient to use a mechanical lever which usually is actuated by a hydraulic system.

Another possible solution that is being investigated in this paper is the use of a pneumatic power source. According to Bollmann (1994), the main advantages of these systems are ease of transportation, energy storing, good power-to-weight ratio, low environmental impact, durability, safety, and easy operation.

This paper proposes the analysis of a conceptual system to amplify the force using a hydro-pneumatic system, in order to make feasible the use of systems with a pneumatic power source to actuate the turbine inlet butterfly valves at hydropower plants, with an acceptable level of speed control. Therefore, a mathematical model of the system was developed using parameters of commercial components and force requirements provided by REIVAX S/A Automation and Control, a project partner company.

## 2. HYDROPOWER PLANTS

Using natural or artificial falling water, hydropower plants transform potential energy into electric energy through a hydraulic turbine powered by the water flow. They are basically composed of eight components. The dam is responsible to increase the level of water at the reservoir, enabling the water to enter the penstock. The control gate and the turbine inlet valve control the flow through the penstock. The turbine transforms the kinetic energy of the water into rotational mechanical energy, which is converted to electrical energy by the generator. Afterward, the turbine, the water returns to its natural course through the outflow (Henn, 2006). Figure 1 represents these components in a hydropower plant structure.

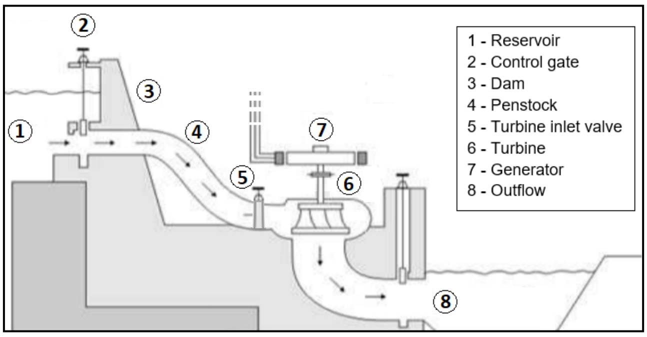


Figure 1. Components of a hydropower plant. Adapted from Furnas (2016).

### 2.1 Turbine inlet valve

Turbine inlet valves are an essential component of hydropower plants. Besides the function already mentioned, it is the dispositive responsible for shutting down immediately the water flow when an over speed or any other problem is identified in the turbine. ABNT NBR 8609 (2015) lists five different valve models for large applications: ring valve, butterfly valve, ball valve, diffuser valve, and gate valve. According to Huang (1996), butterfly valves (Fig. 2) are mainly used for both on-off and throttling services involving large flows of liquids. Some advantages are lightweight and the ease of bringing the valve from fully opened to fully closed very quickly. Pereira (2015) affirms that these valves are often used as turbine inlet valve.


Figure 2. Turbine inlet butterfly valve. Adapted from Corroco (2019).

Butterfly valves are composed of three main parts. The body, where the components are assembled, the valve disk that blocks the water flow, and the shaft that supports the disk (Huang, 1996). Other important components for turbine inlet valves are the actuating system that rotates the shaft to open the valve, the bypass valve used to balance upstream and downstream pressures and the counterweight, responsible for shutting down the valve in any operational condition.

During operation, the turbine inlet valve is subjected to many different torques. The actuating system is designed to overcome the torques that oppose the opening movement. The Manual M49 from American Water Works Association (AWWA, 2017) presents a methodology to evaluate these torques. In this paper, a butterfly valve with a counterweight able to close the valve without help from the actuating system was considered. Therefore, the main operational torques evolved are the total unseating torque, $T_{tus}$, and the total opening torque, $T_{to\theta}$, calculated by the following equations:

$$T_{tus} = T_{b0} + T_{cg0} + T_h + T_{us} + T_p + T_{ecc} \quad (1)$$

$$T_{to\theta} = T_{b\theta} + T_{cg\theta} + T_{d\theta} + T_p \quad (2)$$

where $T_{b0}$ and $T_{b\theta}$ are the bearing torque at valve angle 0° and $\theta$ respectively, $T_{cg0}$ and $T_{cg\theta}$ are the center of gravity torque at valve angle 0° and $\theta$ respectively, $T_h$ is the hydrostatic torque, $T_{us}$ is the unseating torque, $T_p$ is the packing and hub torque, $T_{ecc}$ is the eccentricity torque, and $T_{d\theta}$ is the dynamic torque at the valve angle $\theta$. These torques can be estimated according to AWWA (2017) by:

$$T_{b\theta} = \frac{\left(\pi . {D_d}^2 . \Delta P_\theta + W_{d\&s}\right). D_s . C_f}{8} \tag{3}$$

$$T_{cg\theta} = W_d . C_g . \cos(\theta + \gamma) \tag{4}$$

$$T_h = \frac{\rho . \pi}{5{,}333} . \left(\frac{D_d}{12}\right)^2 . \left(1 + \frac{8 . \varepsilon_2}{D_d}\right) \tag{5}$$

$$T_{us} = \left(C_{usc} + C_{usp} . \mathit{\Delta} P_{max}\right). {D_d}^2 \tag{6}$$

$$T_p = C_{pck} . D_s \tag{7}$$

$$T_{ecc} = \frac{\pi . {D_d}^2 . \varepsilon_2 . \Delta P_\theta}{4} \tag{8}$$

$$T_{d\theta} = C_{t\theta} . {D_d}^3 . \Delta P_\theta \tag{9}$$

where $D_d$ is the disc diameter, $\Delta P_\theta$ is the pressure drop at valve angle $\theta$, $W_{d\&s}$ is the weight of the disc and shaft, $D_s$ is the shaft diameter, $C_f$ is the coefficient of friction between shaft and bushing, $W_d$ is the disc weight, $C_g$ is the distance of the center of gravity of the disc from shaft centerline, $\gamma$ is the center of gravity offset angle, $\varepsilon_2$ is the disc offset, $C_{usc}$ and $C_{usp}$ are the constant or pressure independent coefficient and the pressure dependent coefficient of unseating torque respectively, $\mathit{\Delta} P_{max}$ is the maximum pressure drop across the closed valve, $C_{pck}$ is the packing coefficient, and $C_{t\theta}$ is the coefficient of dynamic torque at valve angle $\theta$. Figure 3 helps to identify some of these parameters.

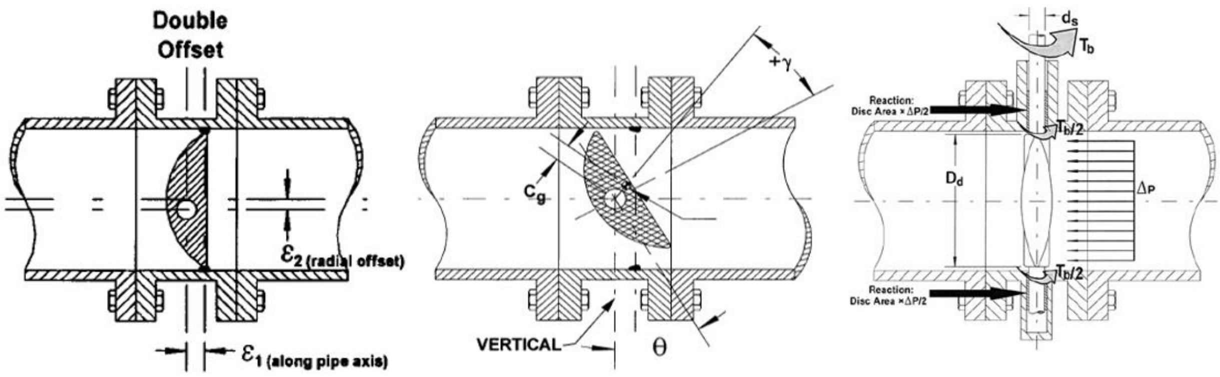


Figure 3. Butterfly valve variables identification. Adapted from AWWA (2017).

Using the data from REIVAX S/A Automation and Control, these torques were calculated for a hydropower plant with 438 kW of installed power. The turbine butterfly inlet valve has 900 mm of diameter, a counterweight of 300 kilograms installed in a lever arm of 600 mm and a hydraulic actuating system with a lever arm of 250 mm. The current valve is not equipped with a bypass valve, which means that it operates with unbalanced upstream and downstream pressures, with flow through the valve.

After the resistive torque evaluation, C504-15 Standard (AWWA, 2015) recommends the minimum design factor of 1.25 for pneumatic actuating systems, operating in open-shut service. To attend the standard and considering some oxidation, a design factor of 1.5 was adopted for the mathematical model.

The unseating and the total opening torque during the valve displacement are indicated in Fig. 4. It already includes the torque generated by the counterweight and the design factor correction. A sixth-degree polynomial (Eq. 10) was estimated to reproduce the calculated torque into the mathematical model, being $x_v$ the valve displacement.

$$T_{to} = (1.239e5). x_v^6 - (5.522e5). x_v^5 + (9.445e5). x_v^4 - (7.783e5). x_v^3 + (3.101e5). x_v^2 - (5.580e4). x_v + 9071.3 \tag{10}$$

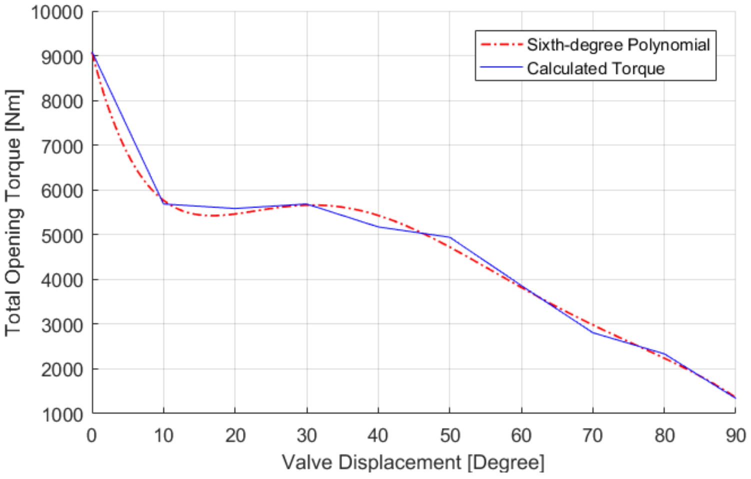


Figure 4. Torque profile of the current turbine inlet valve.

## 3. HYDRO-PNEUMATIC SYSTEM

The proposed hydro-pneumatic system uses compressed air as the power source and hydraulic oil as the working fluid, excluding the necessity of a hydraulic unit power and large amounts of mineral oil, which are used in conventional hydraulic systems commonly applied for turbine inlet valves. Combining pneumatic and hydraulic technologies, this system is capable to amplify the working pressure by the ratio of the areas of pneumatic and hydraulic pistons, becoming able to overcome the force requirements from small hydropower plants.

Figure 5 represents the proposed system. It is a hydro-pneumatic jack with a few adaptations. The directional valve V1 is used to turn on and off the system, V2 represents the internal mechanism responsible to coordinate the operation of actuator A1, generating the necessary oscillation to pump hydraulic fluid. The check valves V3 and V4 control the flow orientation, toggling between the reservoir and the main actuator A2. The directional valve V6 is responsible for the retracting of the actuator A2 and the flow control valve V5 is used to adjust returning speed. However, V1, V5, and V6 were not modeled since it does not influence significantly the system operation during the turbine inlet valve opening.

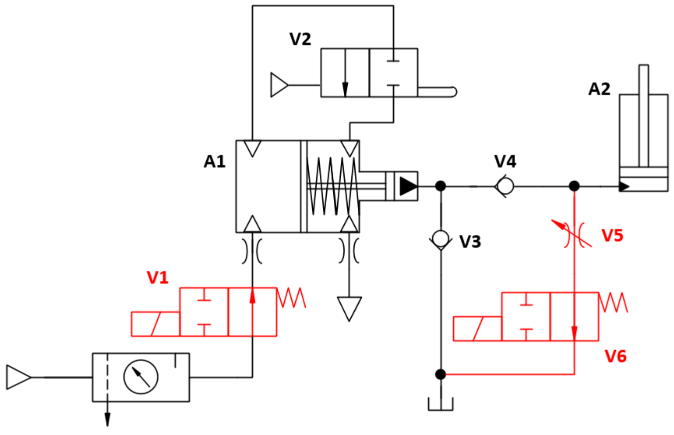


Figure 5. Proposed hydro-pneumatic system representation.

Dimensions of the modeled components were measured in a commercial hydro-pneumatic jack designed to work up to 30 tons of load. Table 1 shown some characteristics of the components.

Table 1. Hydro-pneumatic system parameters.

| Parameter | Value | Parameter | Value |
|---|---|---|---|
| A1 areas ratio | ~ 136 | A2 piston diameter | 75 mm |
| A1 stroke | ~ 33 mm | A2 stroke | 350 mm |
| A1 spring elastic constant | ~ 6000 Nm | V3 and V4 inlet diameter | 5 mm |
| Pneumatic pressure | $6.5 \times 10^5$ Pa | Oil reservoir pressure | $1 \times 10^5$ Pa |

### 3.1 Mathematical modeling

A mathematical model of the system was developed using Matlab Simulink to analyze how it behaves when loaded by the torque from the turbine inlet butterfly valve. The mass flow ($qm$) through the valve V2 and the orifices of actuator A1 are estimated according to Mendoza (2006), by the following equation:

$$qm = Cd.A_o.P_1.\sqrt{\frac{2}{R.T_1}}.\sqrt{\frac{\gamma}{(\gamma-1)}.\left[\left(\left.\frac{P_2}{P_1}\right|_{cr}\right)^{\frac{2}{\gamma}} - \left(\left.\frac{P_2}{P_1}\right|_{cr}\right)^{\frac{\gamma+1}{\gamma}}\right]}.w(a) \tag{11}$$

$$w(a)\begin{cases} = \sqrt{1-\frac{\left(\frac{P_2}{P_1}-\left.\frac{P_2}{P_1}\right|_{cr}\right)^2}{\left(1-\left.\frac{P_2}{P_1}\right|_{cr}\right)^2}} \\ = 1 \end{cases} \tag{12}$$

where $Cd$ is the discharge coefficient, $A_o$ is the orifice area, $P_1$ and $P_2$ are the upstream and downstream pressures, respectively, $P_1/P_2|_{cr}$ is the critical pressure ratio, $T_1$ is the upstream temperature, $R$ is the gas constant and $\gamma$ is the specific heat ratio.

To describe the state variables of the pneumatic chambers the approach presented in Vigolo (2018) was used. The pressure behavior was described by the continuity equation, which applied to the cylinder chamber states that

$$\frac{dp}{dt} = \frac{p}{T}.\frac{dT}{dt} + \frac{1}{V}.\left(q_{m_{in}}.T.R - q_{m_{out}}.T.R - p.A.\frac{dx}{dt}\right) \tag{13}$$

where $p$, $T$ and $A$ are the pressure, the temperature and the area of the chamber, respectively. $x$ is cylinder position and $q_m$ is the mass flowing in and out from the chamber. The temperature behavior was described by the energy equation

$$\frac{dT}{dt} = \frac{\frac{\delta Q}{\delta t} \pm p.A.\frac{dx}{dt} + q_{m_{in}}.\left(C_p.T_{in} - C_v.T\right) - q_{m_{out}}.T.R}{C_v.m} \tag{14}$$

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 with the external environment was calculated using Newton's Law of cooling. Neglecting the radiation heat transfer, the heat transfer between the ambient and the interior walls can be expressed as:

$$\frac{\delta Q}{\delta t} = \lambda.A_{sur}.(T_{amb} - T_{ch}) \tag{15}$$

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 $\lambda$ is the overall heat transfer between the chamber and the ambient.
For the actuator motion, Newton's second law was used:

$$M.\frac{d^2x}{dt^2} = p_A.A_A - p_B.A_B - p_{hA}.A_h - \left(x_{pl_{cyl}} + x\right).k - F_{friction} + F_{contact\_cyl} \tag{16}$$

where $M$ is the mass of the cylinder and rod, $k$ is the spring elastic constant, $x_{pl\_cyl}$ is the preload displacement of the spring and $p_{hA}$ is the hydraulic chamber pressure of the actuator A1. $F_{friction}$ is the friction force and $F_{contact\_cyl}$ is the end stroke force, defined by three expressions:

$$F_{contact\_cyl} = \begin{cases} -k_r.x - c_r.v_{cyl}, & if\ x < 0 \\ 0, & if\ 0 \le x \le L \\ -k_c.(x-L) - c_c.v_{cyl}, & if\ x > L \end{cases} \tag{17}$$

where $k_r$ and $c_r$ are the spring stiffness and damping coefficient of the rear end cap, $k_c$ and $c_c$ are the spring stiffness and damping coefficient of the rod end cap, $v_{cyl}$ is the cylinder velocity, and $L$ is the maximum cylinder displacement.

The methodology presented by Knutson and Ven (2016) was adopted to model the check valves. In the same way, Newton's second law was applied to determine the motion of the check valve sphere. The sum of forces considered is composed by pressure force, spring force and contact force:

$$F = \Delta P_{val}.A_{port} - k_{sring_{val}}.(x_{val} + x_{pl\_val}) + F_{contact\_val} \quad (18)$$

where $\Delta P_{val}$ is the pressure differential across the check valve and $A_{port}$ is the port area. The spring force is given by the spring stiffness, $k_{spring\_val}$, multiplied by the ball position, $x_{val}$, plus the spring preload displacement, $x_{pl\_val}$. The contact force, $F_{contact\_val}$, is given by Eq. (16) with the parameters adjusted for the check valve.

The flow rate through the check valves are calculated by modeling it as an orifice with turbulent flow:

$$Q = C_d.A_0.\sqrt{\frac{2.|P_{h2} - P_{h1}|}{\rho}}.sign(P_{h2} - P_{h1}) + A_{valve}.v_{ball} \quad (19)$$

where $C_d$ is the discharge coefficient, $\rho$ is the fluid density, $v_{ball}$ is the ball velocity, $A_{valve}$ is the projected circle area of the ball, and $A_o$ is the passing orifice area. For V3, $P_{h1}$ is the reservoir pressure and $P_{h2}$ is the pressure of the actuator A1 hydraulic chamber ($P_{ha}$). For V4, $P_{h1}$ is $P_{ha}$ and $P_{h2}$ is the pressure of actuator A2. In this system, ball check valves were considered, therefore the passing area is different from the disc check valve modeled by Knutson and Ven (2016). According to Andersen (1967), it can be estimated by the following equation:

$$A_o = \frac{\pi.D_{port}^{\ 2}}{4}.\frac{\left(\frac{2.x_{val}}{D_{port}} + \sqrt{\frac{D_{ball}^{\ 2}}{D_{port}^{\ 2}} - 1}\right)^2 + 1 - \frac{D_{ball}^{\ 2}}{D_{port}^{\ 2}}}{\sqrt{\left(\frac{2.x_{val}}{D_{port}} + \sqrt{\frac{D_{ball}^{\ 2}}{D_{port}^{\ 2}} - 1}\right)^2 + 1}} \quad (20)$$

As illustrated in Fig. 6, $D_{port}$ is the inlet orifice diameter, and $D_{ball}$ is the ball diameter.

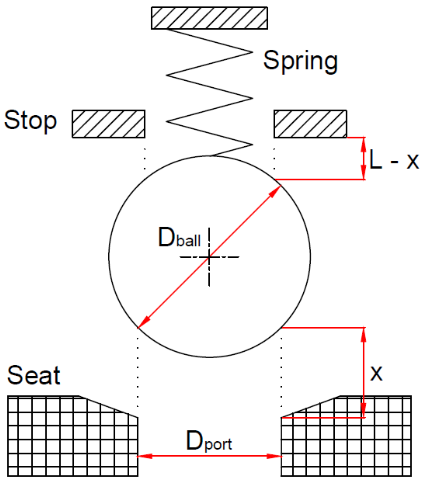


Figure 6. Ball check valve diagram.

The continuity equation is also used to define the pressure behavior inside the hydraulic chambers.

$$\frac{dP}{dt} = \frac{\beta}{(V_o + x.A_p)}.\left[(q_{vin} - q_{vout}) - \frac{dx}{dt}.A_p\right] \quad (20)$$

where $q_{vin}$ and $q_{vout}$ are the inlet and outlet flow rate of the cylinder chamber, $\beta$ is the Bulk modulus, $V_o$ is the initial volume of the chamber, and $A_p$ is the piston area. The piston motion of the cylinder A2 is also defined by Newton's second law.

## 4. RESULTS

Simulations of the modeled circuit were carried out with loading conditions given by Eq. (10), which represents the operational conditions from a small hydropower plant during a complete opening movement of the turbine inlet valve. Due to the oscillating movement of actuator A1 (Fig. 7), the hydraulic pressure varies between low pressure, to drain oil from the reservoir, and high pressure, necessary to overcome the resistive forces imposed by butterfly valve, as seen in Fig. 8.

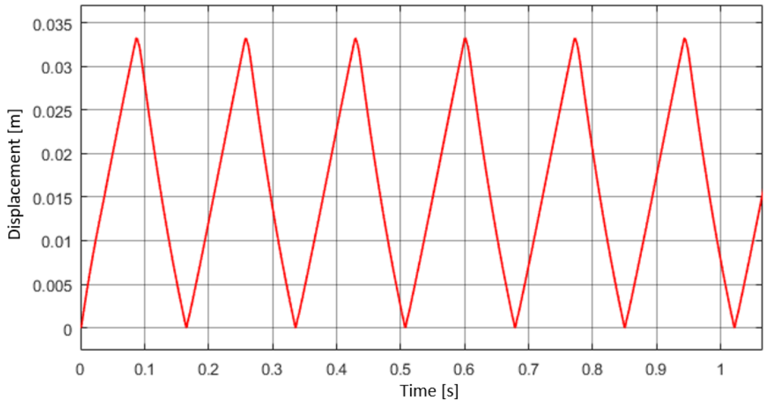


Figure 7. Actuator A1 displacement.

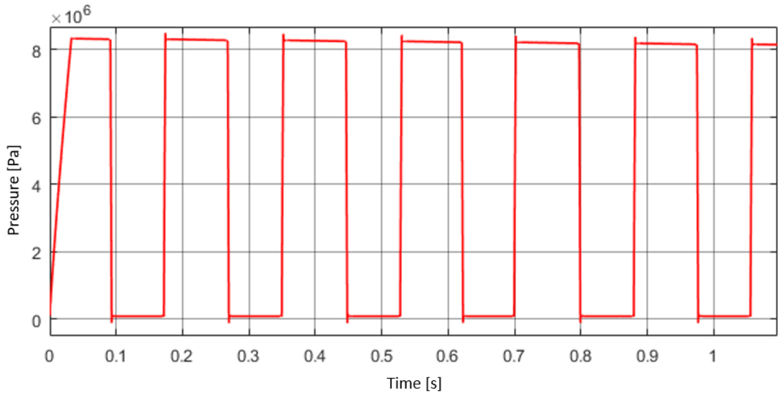


Figure 8. Pressure behavior of actuator A1 hydraulic chamber.

The pressure of the actuator A2 (Fig. 9) follows the load force profile from the turbine inlet valve. The beginning and the end of each pumping cycle produces a short oscillation, which is caused by the unseating and seating time of the check valve V4. Besides that, the maximum pressure reached during the simulation was 83 bar, which is equivalent to 12.5% of the system pressure when working at the maximum supported load.

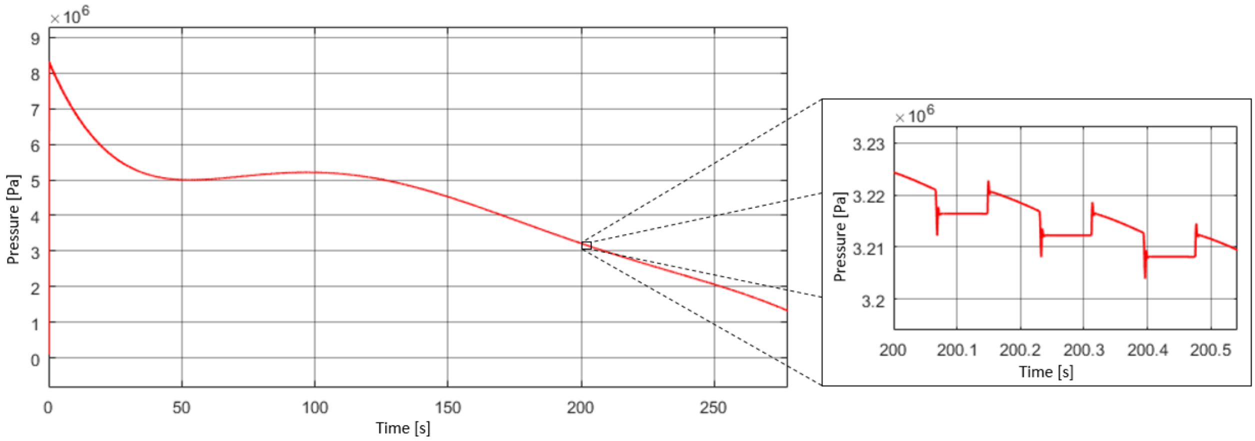


Figure 9. Pressure behavior of actuator A2.

The displacement of the actuator A2 occurs in a sequence of little steps of about 0.2 mm. According to the simulation, it takes 276 seconds to reach the end of the stroke. Figure 9 shows the displacement behavior.

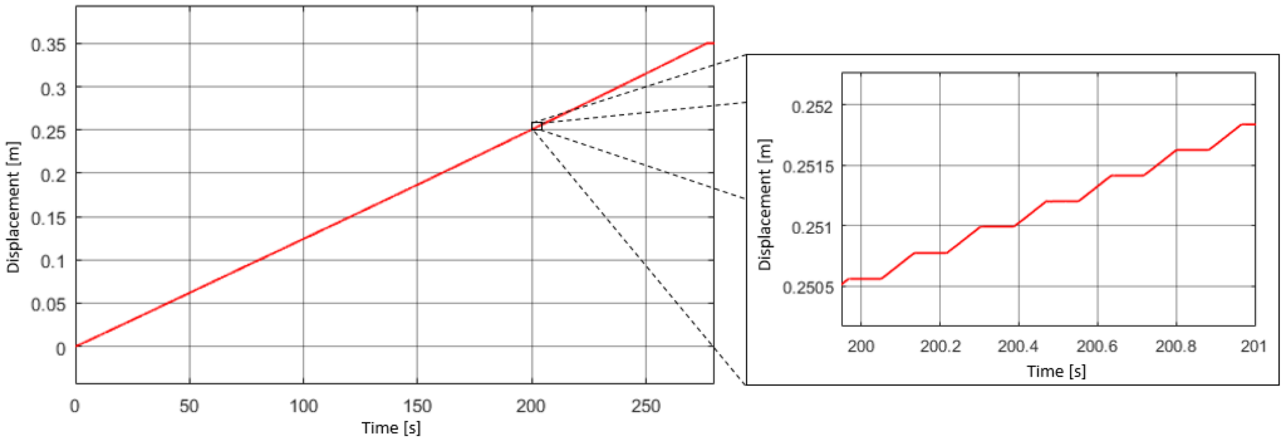


Figure 9. Actuator A2 displacement.

After each step, the valve V4 holds the actuator A2 position, seating itself and blocking the reverse flow. Also, even with the load variation of almost 31 kN during the movement, the opening velocity had a negligible change.

## 5. CONCLUSIONS

Is this paper the development of a mathematical model of a hydro-pneumatic system is described. The model is used to perform a dynamic analysis of the system under the loading conditions of the turbine inlet valve. The results demonstrated that the hydraulic pressure of the main actuator (A2) remained stable during its displacement, having just small oscillations which are caused by the intermittent flow from the pneumatic pump.

The system attended the minimum opening time of 72 seconds, which is stated by C504-15 (AWWA, 2015), however, the achieved opening time is 96 seconds higher than the conventional maximum opening time of 180s from REIVAX data. The main point is that the system parameters are from a standard commercial hydro-pneumatic jack. In future studies, the developed mathematical model will be used to optimize the system parameters, aiming the reduction of the opening time.

The proposed hydro-pneumatic system showed capable to replace the conventional hydraulic systems. The economic viability becomes more attractive when a full pneumatic automation and control system of hydropower plants are considered, a situation where the hydraulic unit power and the accumulator could be completely replaced by a unique air compressor and an appropriated reservoir. Moreover, the system is capable to overcome loads up to 8 times greater than the simulated, which means that it could be suitable for a wide range of hydropower plants.

## 6. ACKNOWLEDGEMENTS

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