

Numerical and field tests of hydraulic transients at Piva power plant

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Abstract. In 2009, a sophisticated field investigation was undertaken and later, in 2011, numerical tests were completed, on all three turbine units at the Piva hydroelectric power plant. These tests were made in order to assist in making decisions about the necessary scope of the reconstruction and modernisation of the Piva hydroelectric power plant, a plant originally constructed in the mid-1970s. More specifically, the investigation included several hydraulic conditions including both the start-up and stopping of each unit, load rejection under governor control from different initial powers, as well as emergency shut-down. Numerical results were obtained using the method of characteristics in a representation that included the full flow system and the characteristics of each associated Francis turbine. The impact of load rejection and emergency shut-down on the penstock pressure and turbine speed changes are reported and numerical and experimental results are compared, showing close agreement.

1. Introduction

This paper summarises and compares the numerical and field-measured results associated with hydraulic transient conditions at the Piva hydroelectric power plant in Montenegro [1]. The partial differential equations were solved via the well-known method of characteristics (MOC). Extreme pressure values in the inflow and outflow system are determined using numerical simulation of various transient regimes including the behaviour of the protection devices. Modelling of the Francis turbine characteristics was given particular attention. Since the numerical and experimental results agreed closely, the numerical model could be assumed to be a valid tool for the modernisation and reconstruction of the plant's hydro mechanical transient processes. In 2009, the measurements and research of unsteady phenomena in the Piva hydroelectric power plant were performed. Various transient processes were analysed and significant changes in operating parameters of the system (net head and discharge) were made. These changes were sufficiently significant to affect the safety and reliability of the plant and to require the control of transient processes both upstream and downstream of the turbine. With a too rapid closure of the wicket gates, water hammer would exceed the critical pressure of the penstock, and event that could destroy the entire system [2].

Historically there have been a lot of experimental and numerical studies of hydraulic transients in hydropower systems. For small hydropower plants, rupture disks or safety membranes have sometimes been proposed [3]. Satisfactory agreement of measured and computational results of transient pressures have been obtained for Catalan Power Plant in Turkey [4]. The impulse method was used to analyse hydraulic resonance in hydropower systems [5]. A variety of specialized numerical methods



have also been used to simulate hydraulic transients in hydropower systems, such as nonlinear analogue-digital simulation method [6], finite volume method (FV) [7], implicit method of characteristics (IMOC) [8] and stochastic method [9].

The low values of pressure in the region downstream of the turbine can cause cavitation, formation of draft tube vortex and water column separation. The conditions of regulation are often selected on the basis of the results obtained through the transient analyses and the pressure values need to be confirmed to remain within prescribed limits.

The governing equations take into account water and penstock elasticity, assuming the waterways are slightly elastic, with the defined wave speed [10]. The aim here, as in all modelling, is to keep the models simple and yet to achieve acceptable engineering predictions.

Transient operating regimes in hydroelectric power plants are extremely important. Results obtained during the analyses of transition regimes are the basis for designing flow-passage system, determining protection measures and making selection for optimal regimes during exploitation of the hydroelectric power plant. The final conclusion is that transient calculations played a crucial role in the improvement of the Piva hydroelectric power plant.

2. General description of the Piva hydroelectric power plant

The Piva hydroelectric power plant, is the peak hydroelectric power plant in the Montenegrin system with an installed capacity of 360 MW. It is located in the north-west of Montenegro, at a distance of 10 + 600 km from the location where the Tara River and the Piva River join to form the Drina River. The nearest town is Plužine, some 15 km from the power plant. The basin has a length of about 40 km. The useable capacity of the basin is about 790 million cubic metres. The dam is a concrete-arch in its construction, 220 m high and a crest length of 268 m (the crest elevation is 678 m a.s.l). The dam is shown in Figure 1 and has been in operation since 1976.

As a part of the 2009 reconstruction and modernisation project, extensive tests and measurements of important parameters of the plant were conducted [11]. The results of the numerical models that have been developed for transient analyses were compared with experimental results obtained at the Piva site.



Figure 1. Piva hydroelectric power plant

Basic technical characteristics of the Piva hydroelectric power plant are:

- Installed power - 360 MVA,
- Total volume of the basin - $880 \times 10^6 \text{ m}^3$,

- Useful volume of the basin - $790 \times 10^6 \text{ m}^3$,
- The projected annual electricity output - 860 GWh,
- The average annual electricity output (1976 – 2007) – 737 GWh,
- Three Francis turbines with a vertical shaft $3 \times 120 \text{ MW}$ (250 rpm),
- Three - three phase generator with a vertical shaft (250 rpm) of 120 MVA,
- Three - three phase transformers (15.7 / 220 kV) of 120 MVA,
- Concrete arch dam with its hydraulic height of 190 m,
- Elevation of the water level in the basin - 675.0 m above sea level,
- Minimum operating water level - 595.0 m above sea level,
- Maximum net head - 181.9 m,
- Minimum net head - 99.9 m,
- Rated discharge - $3 \times 80 \text{ m}^3/\text{s}$,
- The catchment area of the Piva hydroelectric power plant amounts to 1,760 km^2 .

2.1. Hydraulic components and transient analyses

The waterway components are shown in Figure 2. Input slide gates are part of the intake structure, followed by three inlet penstocks whose diameters vary from 5.0 m at the entrance to 3.4 m in front of the turbine, with the three downstream penstocks lined with concrete. Butterfly valves are at the end of the penstock in front of each Francis turbine. In the tailrace system, three draft tubes are connected to the common lower surge tank. At the end of each draft tube there are draft tube gates. The tailrace tunnel extends from the lower surge tank to the outlet channel.

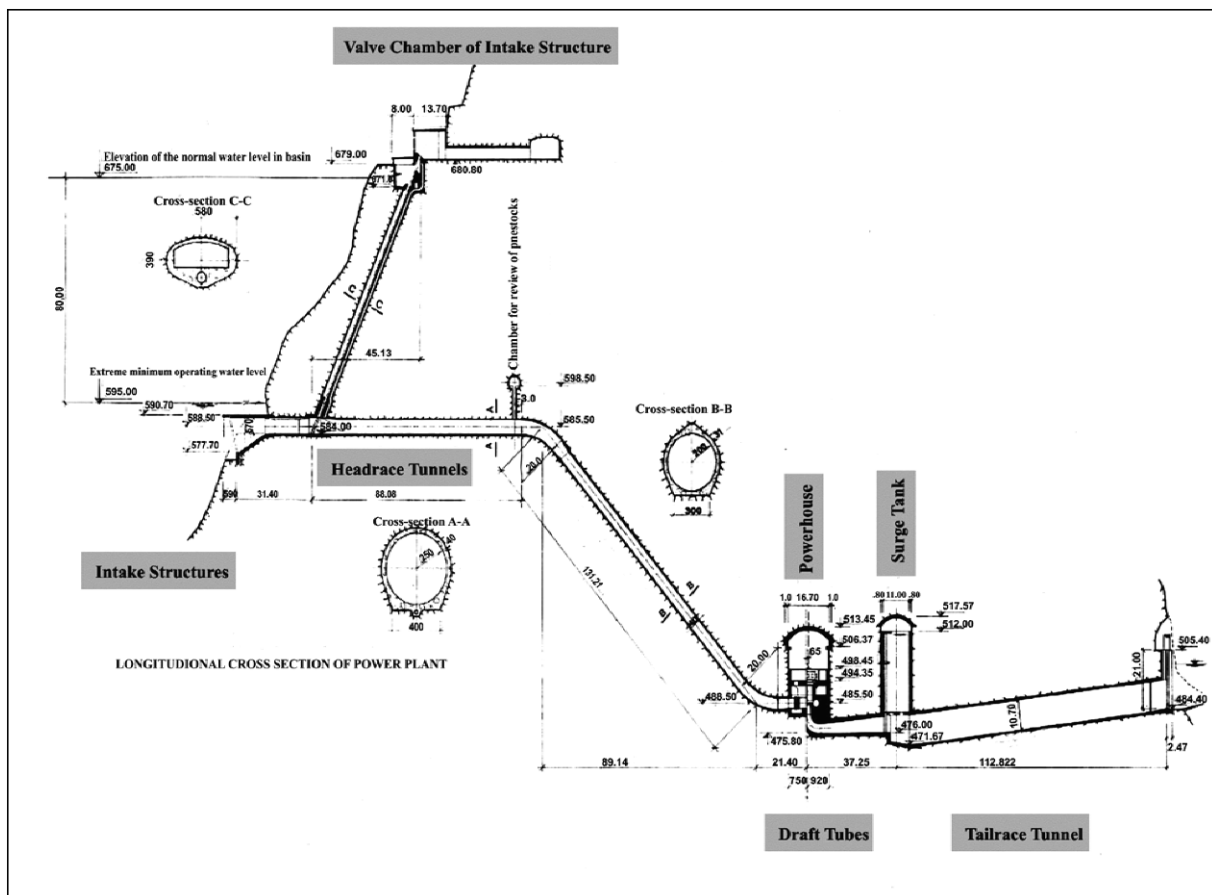


Figure 2. Piva hydroelectric power plant flow – passage system

The behaviour of the turbines depends on the electrical grid and the system frequency. If the turbines operate in an isolated electrical grid, power consumption and production automatically change the turbines to frequency control [12]. When they operate within an interconnected grid, the turbines work in parallel with other generators. In this case, all units jointly control the frequency of the system which is practically constant. As long as the generator is connected to the system, the electricity grid controls the speed. From the above mentioned, the conclusion is that while the discharge on the turbine can change, turbine rotational speed remains constant.

3. Mathematical model

The mathematical model of the physical system of the Piva HPP consists of the following phases:

- Determining the basic equations which describe the behaviour of the system;
- Solving the basic equations through computer simulation;
- Calibration of mathematical models and running predictions of new operating conditions.

3.1. Basic equations for transient calculations and their numerical solutions

One-dimensional equations of unsteady flow in penstocks are a continuity equation (1) and an equation of motion (2), [13].

$$\frac{\partial H}{\partial t} + V \frac{\partial H}{\partial x} - V \sin \theta + \frac{a^2}{g} \frac{\partial V}{\partial x} = 0 \quad (1)$$

$$g \frac{\partial H}{\partial x} + \frac{\partial V}{\partial t} + V \frac{\partial V}{\partial x} + f \frac{V|V|}{2D} = 0 \quad (2)$$

in which H = piezometric head, V = flow velocity, a = propagation velocity of circulation resulting wave, g = acceleration due to gravity, f = Darcy-Weisbach friction factor, θ = angle of the penstock with the horizontal, x = spatial coordinate, D = diameter of the penstock and t = time. These equations are well-known and can be solved within either the time or the frequency domain by applying many different mathematical methods [14].

3.2. Solving the basic equation with - Method of characteristics (MOC)

Using the method of characteristics (MOC) these partial differential equations are transformed into ordinary differential equation, determining a family of curves that are valid in space-time plane along characteristic lines. The direction of propagation of disturbing waves through the pipe in time and space is defined by characteristic lines [15].

3.3. Boundary conditions

In this paper, two boundary conditions are used, one at each end of the penstock. The first boundary condition represents the - Francis turbine and the second the - reservoir of the Piva HPP.

The characteristics of the Francis turbine is quite difficult to model mathematically, because the flow in the turbine is very complex. Herein, the characteristics of Francis turbine of the Piva HPP, were extracted from data measured on both the model and dynamically operating turbine. The second boundary condition it is much simpler, namely where the penstock enters the reservoir the piezometric pressure is effectively constant.

3.4. Analytical and numerical modelling

The results obtained from the developed numerical model describe the transients in the Piva hydroelectric power plant. Unit 3 results are presented in this paper, as it has the most complex configuration. Two models are analysed:

- The model (1) is represented by the equations: $Y_V = f(t)$, $a_V = f(Y_V)$, $Q = f(a_V)$ determined by the measurement defining the turbine discharge as a function of time $Q = f(t)$, after which other variables were calculated. This is called here the “Empirical Method” and the method gives direct calculations.
- The Francis turbine is represented in model (2) by its hill chart characteristics. This method is named here as the “Theoretical Method”.

With the corresponding equations, which are compatible with the equations obtained by applying the method of characteristics, each of the components of the flow–passage system is included.

3.4.1. Empirical Method

The Empirical Method is based on the results of measurements represented by $Q = f(t)$ corresponding to the 8s closing time. The pressure of the turbine spiral casing inlet was also calculated.

3.4.2. Theoretical Method

The Theoretical Method is based on the model test turbine hill chart and dynamic equation related to the unit (3), [16],

$$n_p = \left\{ n_1^2 + 730.46 \frac{\Delta t}{GD^2} \left[0.5 \cdot |P_{tur1} + P_{turP}| - \frac{P_{genF}}{\eta_{gen}} \right] \right\}^{0.5} \quad (3)$$

with: unit power - P_{1l} , unit discharge - Q_{1l} , unit rotational speed - n_{1l} lines in the hill chart represent the relationship between these parameters. In equation (3): n_p = rotation speed aggregate at the end interval, n_1 = rotation speed aggregate at the beginning interval, Δt = time step of integration, $GD^2/4$ = total moment of inertia of turbine and generator, P_{tur1} = power turbine during the initial interval, P_{turp} = power turbine at the end interval, P_{genF} = power generator at time $t + \Delta t$, η_{gen} = efficiency generator. Model tests provide data for the turbine hill chart. Model efficiency, is usually less than the efficiency of the prototype and could be calculated by applying scale effect formulas. Guide vanes closing is done in two stages.

4. Comparison of calculated and measured data for the emergency closure at 85.5 MW.

Figure 3 shows the changes of pressure (calculated and measured) in the spiral case during the emergency shutdown of unit 3 at $P = 85.5$ MW. The input parameters for the Empirical Method are listed in Table 1 and input data for the Theoretical Method are listed in Table 2.

Table 1. Input parameters for the Empirical Method, the power $P = 85.5$ MW.

	$P(MW)$	$H_{rez}(m)$	$Q(m^3/s)$	$L(m)$	$D(m)$	$a(m/s)$	$t_c(s)$	$t_p(s)$	$a_{vo}(mm)$
EM	85.5	672.3	53.9	304	4.3	1200	7.5	3.5	153.9

Table 2. Input parameters for the Theoretical Method, the power $P = 85.5$ MW.

	$GD^2/4$ (kgm^2)	$H(m)$	$Q(m^3/s)$	$D_{rk}(m)$	$L_{1c}(m)$	$D_{1c}(m)$	$a_{1c}(m/s)$	$L_{2c}(m)$	$t_c(s)$
TM	1575000	181.5	53.9	2.9	152	5	1174	132.7	7.4
	$D_{2c}(m)$	$a_{2c}(m/s)$	$L_{3c}(m)$	$D_{3c}(m)$	$a_{3c}(m/s)$	$L_{4c}(m)$	$D_{4c}(m)$	$a_{4c}(m/s)$	$t_p(s)$
TM	4	1222	5	3.6	1220	17.7	3.19	1300	3.15

In the Theoretical Method, in addition to the data in Table 2 for the unit power, unit discharge and unit rotational speed from the hill chart described the turbine. The closure time of the guide vanes is much greater than the time reflection of the waves in the penstock, $2L/a = 0.51$ s.

The maximum value of the pressure that was measured is 711.7 m a.s.l and occurs 1.3 s after the initiated closure; the pressure increase was $\Delta H = 42.7$ m. The maximum pressure obtained by the Empirical Method was 715.1 m a.s.l, and corresponding increase of pressure $\Delta H = 46.1$ m. The difference obtained by measurement and calculation have approximately the same value of 3.4 m. In the Empirical Method guide vanes closure was performed in two steps with a time of 7.5 s.

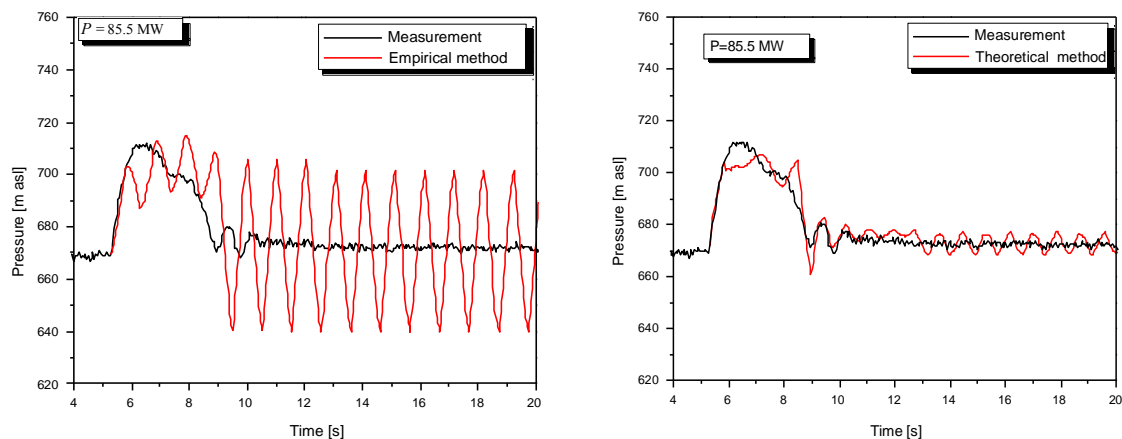


Figure 3. Change in pressure on the entrance of the spiral case of Francis turbine during the emergency shutdown of unit 3 with power $P = 85.5$ MW

The maximum pressure obtained by the Theoretical Method was 707.1 m a.s.l, with the pressure increase of $\Delta H = 38.1$ m. The difference between the measured and calculated values is 4.5 m. The closure time was the same and it was 7.5 s. The calculated values of the maximum pressure, were lower than the maximum allowed pressure of 740.5 m a.s.l.

Calculations with the Empirical Method, compared to the measurements, showed significant deviation. These discrepancies are especially notable in the second part of the closure with huge unrealistic pressure oscillations. On the other hand, the curve of pressure change which was obtained by the Theoretical Method was close to the pressure obtained by the measurements. From the text above it can be concluded that the results obtained by the Theoretical Method are significantly better.

Figure 4 shows measured and calculated speed of rotation. The maximum measured speed was 290 rpm, 3.3 s after the closure of the guide vanes. The corresponding speed calculated by the Theoretical Method was 284.7 rpm. The difference was 5.2 rpm or 1.8-%. The maximum permitted rotational speed is 342.5 rpm.

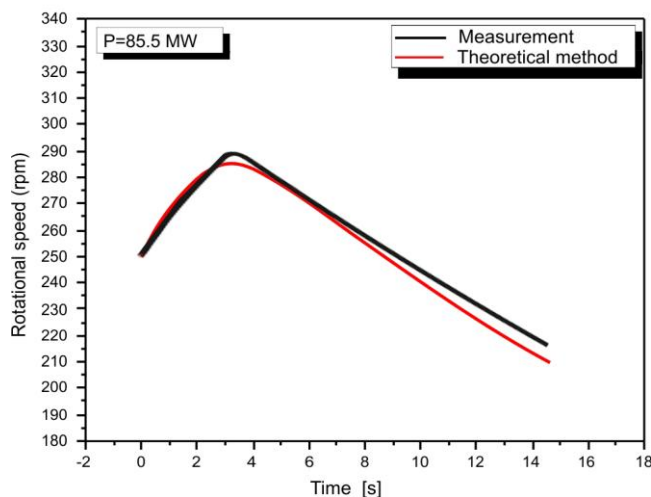


Figure 4. Speed of the turbine measured and obtained by the Theoretical Method, during the emergency shutdown at $P = 85.5$ MW.

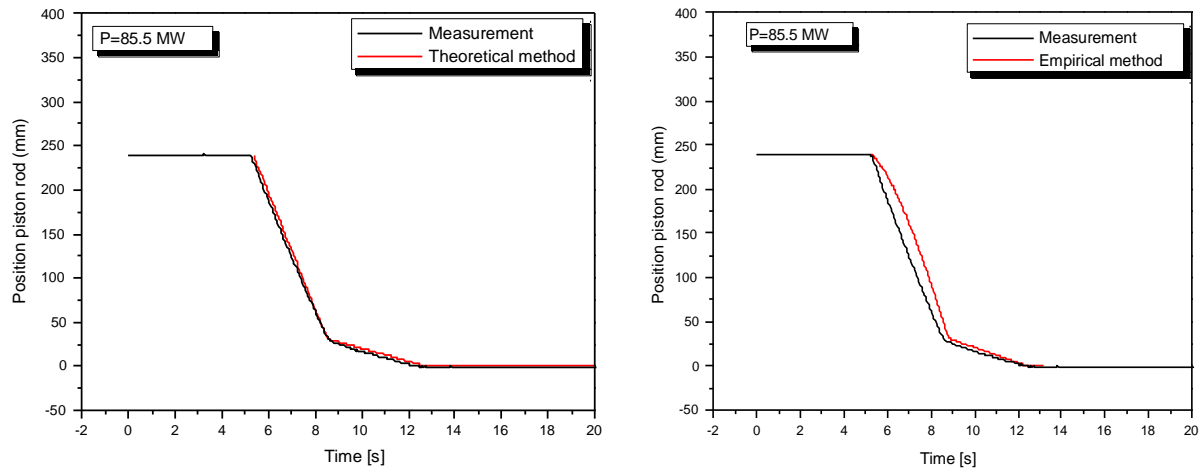


Figure 5. Servomotor stroke during the emergency shutdown at $P = 85.5$ MW.

The analysis of the numerical results shows that the Theoretical Method matches the measured results, while the Empirical Method has some discrepancies during the first stage of the closure, but no such discrepancies occurred in the second stage. Figure 6 diagram shows the discharge through the Francis turbine. Figure 6 shows: penstock cross-section, as well as the maximum and minimum pressure. In the first 3.6 s of the turbine closure, the discharge is decreased to the value of $3.9 \text{ m}^3/\text{s}$ or 92.6-%. During the second part of the closure, the discharge is reduced by only $3.9 \text{ m}^3/\text{s}$ or 7.3-%. The second stage of closure prevents the pulsations of pressure and creation of vacuum under the runner (water column separation).

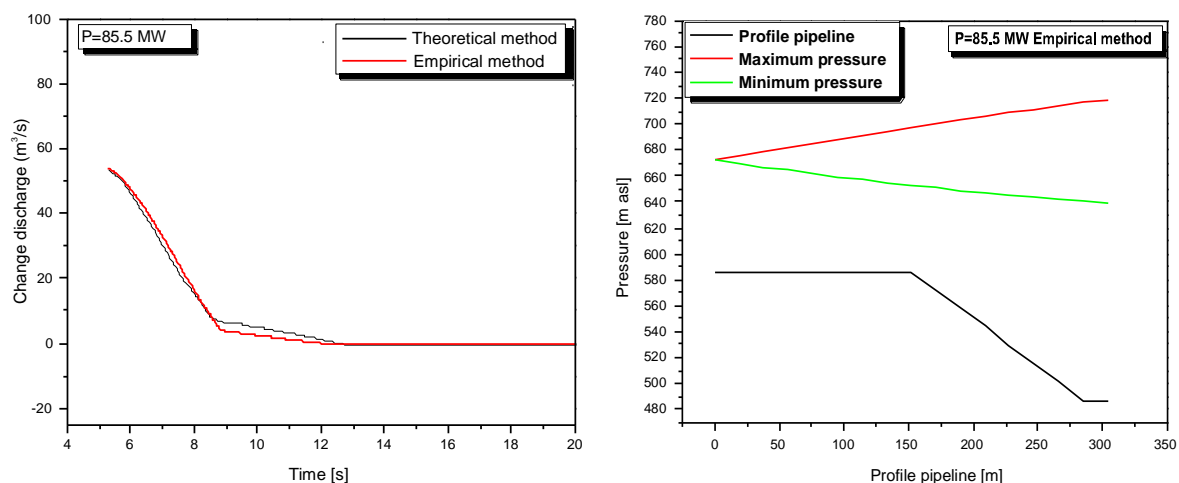


Figure 6. The left diagram shows change of discharge through the Francis turbine. The right diagram shows the profile of the unit 3 penstock - maximum and minimal pressure along the profile of the penstock, during the emergency shutdown of unit 3 with power $P = 85.5$ MW

5. Conclusion

The two numerical models of the Theoretical Method and the Empirical Method have been developed to simulated transient processes in the hydraulic systems with the Francis turbines. Comparison of the

measurements showed that the Theoretical Method, based on a representation of the turbine hill chart is better at reproducing measured discharge and rotational speed. The Numerical Empirical Method, based on measured discharge through the turbine $Q_P = f(t)$, does not directly take into consideration the turbine behaviour in transient operation such as the dynamic equation of turbine and the parameters form the hill chart of the turbine.

6. Nomenclature

a	wave speed	P	power
$a_{1c}, a_{2c}, a_{3c}, a_{4c}$	wave speed in parts of the penstock	P_{turI}	power turbine during the initial interval
a_V	distance between guide vanes	P_{turp}	power turbine at the end interval
$GD^2/4$	total moment of inertia of turbine and generator	P_{genF}	power generator at time $t + \Delta t$
g	acceleration due to gravity	P_{II}	unit power
D	diameter of the penstock	Q	discharge
$D_{1c}, D_{2c}, D_{3c}, D_{4c}$	diameters of penstock parts	Q_P	instantaneous discharge at the entrance to the spiral case turbine
D_{rk}	output diameter runner	Q_{II}	unit discharge
unit 3	third unit Piva hydro power plant	t	time
f	Darcy-Weisbach friction factor	t_c	time closed guide vanes
H	piezometric head	t_P	time transition from first degree at the second degree closing guide vanes
H_{rez}	fluid level in the basin	V	flow velocity
L	length of penstock	x	spatial coordinate
$L_{1c}, L_{2c}, L_{3c}, L_{4c}$	length of penstock parts	Y_V	stroke piston rod servomotor
n_p	rotation speed unit 3 at the end interval	ΔH	changes piezometric head
n_I	rotation speed unit 3 at the beginning interval	Δt	time step of integration
η_{gen}	efficiency generator		
θ	angle of the penstock with the horizontal		
n_{II}	unit rotational speed		

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