

Dynamic Closing Modelling of the Penstock Protection Valve for Pipe Burst Simulations

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Abstract. The penstock protection valve equipped with downstream air valve is a safety component aiming to cut off the discharge in case of pipe burst. The hydraulic transient simulation of pipe burst scenario is challenging since the closing time results from the system dynamics, air is admitted through the downstream air valve and cavitation may occur in the penstock. To perform such simulation, a dynamic model of the penstock protection valve is developed and presented in this paper. Closing time, maximum flow rate reached during the pipe burst or maximum oil pressure in the cylinder of servomotor during the non-linear closing law are results derived with this model. Moreover, influence of cavitation in the valve and in the penstock are considered in the model. It is showed that simulation of pipe burst should consider i) cavitation model in the penstock protection valve, ii) water column separation in the penstock and iii) air admission by the air valves located downstream to the valve to obtain realistic results.

1. Introduction

The sequence of the protection components of a hydraulic power plant during normal, exceptional or accidental events plays a fundamental role for the safety of the installation. The operating times of these components are a key parameter in the safety chain. The penstock protection valve is aiming to cut off the discharge in case of pipe burst and closing time will depend on the forces balance on the valve obturator.

The Montbovon hydroelectric power plant located in Canton Fribourg in Switzerland, which is operated by Groupe E SA, comprises an upper reservoir, a headrace tunnel, a headrace surge tank, two penstocks and two Francis turbines generating units of 16 MW each, operated under a maximal gross head of 90 mWC. In the framework of the review of the safety chain of the Montbovon power plant, a dynamic model of the butterfly penstock protection valve has been developed for hydraulic transient simulations of pipe burst. This model is based on the momentum equation which takes into account the different torques acting on the obturator. Consequently, the valve closing time results from the simulation and is not imposed a priori into the simulation. Similar model was developed to simulate unit valve closing for normal scenario such as emergency shutdown or quick shutdown [1]. The model developed in this paper goes further by taking into account the cavitation influence which occurs in case of pipe burst. First, the dynamic penstock protection valve model is presented. Then, measurements of closing valve with water discharge at different output power of the unit are used to calibrate and validate the developed valve model. Finally, simulation of pipe burst is performed with different modelling approach of the system which highlights the influence of cavitation modelling.

2. Modelling for Pipe Burst Simulations

2.1. Dynamic Penstock Protection Valve Model (DPPV)

The Montbovon hydroelectric power plant has been modelled with the SIMSEN software to perform hydraulic transient simulations. The model includes the headrace tunnel, the headrace surge tank, two penstocks in parallel each feeding one Francis turbines generating units of 16 MW. The hydraulic layout and main data characteristics of the powerplant are showed in Figure 1.

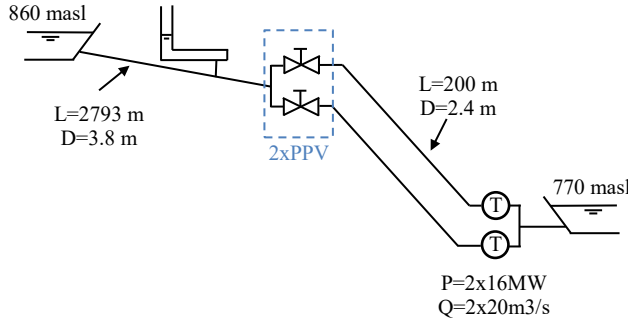


Figure 1. Hydraulic layout of the Montbovon hydroelectric power plant.

Figure 2. Penstock protection valve of DN1800.

Two penstock protection valves of diameter 1.8 m equipped with a downstream air valve are located at the top of each penstock, as showed in Figure 2. The pipe burst at the bottom of the penstock induces a low pressure wave in the penstock with occurrence of cavitation. Simulation of such scenario with closing of penstock protection valve requires advanced modelling of :

- dynamic closing of penstock protection valve depending on the flow conditions on the obturator, see Section 2.2.
- air admission by air valve located downstream the penstock protection valve, see Section 2.3.
- water column separation in the penstock, see Section 2.4.

The theoretical models of each of the above-mentioned points are presented in this section.

2.2. Modelling of Penstock Protection Valve

The motion of the penstock protection valve is described by the rotating mass inertia equation of the obturator, see equation (1).

$$J \frac{d\omega}{dt} = M_{tot} \quad (1)$$

Where J and ω are respectively the moment of inertia and the rotational speed of valve rotating parts and M_{tot} is the sum of all torques which are applied on the valve obturator.

2.2.1. Geometrical parametrization of the hydraulic servomotor A parametrization of the hydraulic servomotor geometry is presented in Figure 3. The servomotor position and the obturator angle are defined respectively by parameters z_v and α . The angle of the connecting rod with vertical line is β .

Derived from geometrical properties, the relation between the angle of the connecting rod β and the obturator angle α is given by equation (2).

$$\begin{cases} \sin \beta = \frac{l_{1v} \cos(\alpha) - \Delta_{v0}}{l_{2v}} \\ \cos \beta = \frac{z_0 - z_v - l_{1v} \sin(\alpha)}{l_{2v}} \end{cases} \quad (2)$$

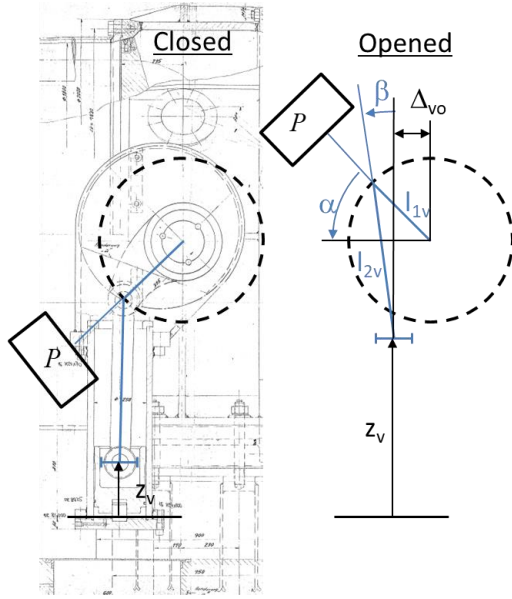


Figure 3. Parametrization of the penstock protection valve geometry.

2.2.2. *Servomotor kinematic* The piston displacement speed dz_v/dt can be expressed from the rotational speed ω of the obturator and defines the oil discharge Q_{oil} feeding the piston chamber, see equation (3).

$$\begin{cases} \frac{dz_v}{dt} = l_{1v}\omega \cos(\alpha) \\ Q_{oil} = -\frac{dV_{oil}}{dt} = -f_k \frac{dz_v}{dt} \end{cases} \quad (3)$$

With f_k the cross sectional area of the piston chamber.

2.2.3. *Oil diaphragm driving closing time.* The closing of the penstock protection valve is performed by discharging the oil volume of the piston chamber at the atmospheric pressure through a diaphragm component. The singular head losses of this diaphragm are expressed according to equation (4).

$$\frac{p_{oil} - p_{ext}}{\rho_{oil}g} = \frac{K_d}{2gA_{ref}^2} Q_{oil}^2 \quad (4)$$

From this equation, the oil pressure in the piston chamber p_{oil} can be computed and the external pressure p_{ext} corresponds to the atmospheric pressure. This modelling assumes that the head losses of the complete auxiliary oil circuit are lumped in a single diaphragm. The head loss coefficient K_d is function of the orifice diameter which drives the closing time of the penstock protection valve. Usually, this parameter is calibrated from measurements of the closing time in dead water conditions and is kept constant for all simulations with discharge through the valve.

2.2.4. *Balance of torques.* The total torque applied on the obturator M_{tot} , considered in the rotating mass inertia equation (1), is constituted of :

- motor torques such as:
 - the hydraulic torque M_h induced by the hydraulic forces on the obturator;
 - the counter weight torque M_{cw} which initiates the valve closing;
 - the eccentricity torque M_{ecc} which is not included in the hydraulic torque;
- resistive torques such as:
 - the friction torque M_{fr} of the trunnions under the hydraulic thrust;
 - the oil torque of the cylinder M_{oil} .

This total torque is a function of obturator angle α , connecting rod angle β and oil pressure p_{oil} , see equation (5).

$$M_{tot} = M_h + M_{cw} + M_{ecc} - M_{fr} - M_{oil} = f(\alpha, \beta, p_{oil}) \quad (5)$$

The hydraulic torque M_h is a function of the obturator angle α and the discharge Q . It is expressed from the head drop through the valve $H_k = (1 + \xi) C^2 / 2g$ with ξ the head loss coefficient and the torque coefficient K , see equation (6). The two coefficients $\{K, \xi\} = f(\alpha)$ are function of the obturator angle and are usually given by the valve manufacturer.

$$M_h = g(\alpha, Q) = K \frac{D^3}{12} \gamma (1 + \xi) \frac{C^2}{2g} \quad (6)$$

The counter weight induces a motor torque which is defined by equation (7):

$$M_{cw} = l_{cw} P \cos(\alpha) \quad (7)$$

The eccentricity torque is induced by the hydraulic force F_h which features an angle δ with the obturator, see equation (8). Moreover, this angle depends on the obturator angle: it is almost constant and close to 90° over the complete range of the obturator angle and tends towards 0° in the open position. This hydraulic force is computed from the axial thrust F_{hx} derived from the projected surface of the obturator A_v^* and the pressure difference between inlet and outlet valve, as showed by equation (9).

$$M_{ecc} = F_h \cdot ecc \cdot \sin(\delta(\alpha)) \quad (8)$$

$$F_h = F_{hx} \cos(\delta - \alpha) = \Delta p \cdot A_v^* \cos(\delta - \alpha) \quad (9)$$

The friction torque M_{fr} is given by equation (10) with μ the friction coefficient, $d_{trunnion}$ the diameter of trunnions and F_{hx} the axial thrust.

$$M_{fr} = \mu \frac{d_{trunnion}}{2} F_{hx} = \mu \cdot \frac{d_{trunnion}}{2} \cdot \Delta p \cdot A_v^* \quad (10)$$

The oil torque of the cylinder is given by equation (11).

$$M_{oil} = p_{oil} f_k \cos(\beta) \cos(\alpha - \beta) l_{1v} \quad (11)$$

2.2.5. Quasi-static assumption. It is assumed that the derivative of the rotational speed is small and a quasi-static approach is chosen. Hence, according to equation (1), the sum of the torques is equal to zero and the oil pressure can be derived, see equation (12).

$$p_{oil} = \frac{M_{dyn} + M_{cp} + M_{ecc} - M_{fr}}{f_k l_{1v} \cos(\beta) \cos(\alpha - \beta)} \quad (12)$$

Equation (4) is used to derive the expression of the rotational speed of the obturator closing which is given by equation (13).

$$\omega = \sqrt{\frac{p_{oil} - p_{ext}}{\rho_{oil} g} \frac{2g A_{ref}^2}{K_d (f_k l_{1v} \cos(\alpha))^2}} \quad (13)$$

2.2.6. Cavitation influence. It is known that cavitation can occur during valve closing and consequently, the hydraulic torque is modified. To model such influence, a Thoma number is defined and computed from the downstream static pressure p_2 and the head drop through the valve H_k , see equations from (14) to (17). The torque and the head loss coefficients $\{K, \xi\} = f(\alpha, \sigma)$ are function of the obturator angle and this Thoma number. This dependency to the Thoma number is usually given from data measurements on test rig performed by the valve manufacturer. Typical data can be found in the literature [2], [3].

$$\sigma = \frac{B^* + H p_2}{H_k} \quad (14)$$

$$B^* = \frac{p_{atm} - p_v}{\rho g} \quad (15)$$

$$H_k = (1 + \xi) \frac{C^2}{2g} \quad (16)$$

$$Hp_2 = \frac{p_2}{\rho g} \quad (17)$$

2.3. Air Valve Model (AV)

The modelling of the air valve located downstream the penstock protection valve is based on the assumption that the air volume admitted is a function of local pressure p and mass m , see equation (18). The difference of upstream and downstream discharges is defined by the derivative of air volume which is developed by a first order Taylor development and introduces the derivative of the air mass, see equation (19) which is a non-linear function of the local pressure according to Wylie and Streeter [4].

$$V = f(p, m) \quad (18)$$

$$Q_1 - Q_2 = -\frac{dV}{dt} = -\frac{\partial V}{\partial p} \frac{dp}{dt} - \frac{\partial V}{\partial m} \frac{dm}{dt} \quad (19)$$

2.4. Water Column Separation Model in the Penstock (FGM)

The cavitation occurrence in water reduces significantly the wave speed a_0 in pipes. Wylie and Streeter [4] derived the wave speed a in homogenous liquid free gas mixture characterized by an initial void fraction α_0 defined for a reference absolute pressure p_0 and leads to the following equation:

$$a = \frac{a_0}{\sqrt{1 + \frac{p_0 \alpha_0 a_0^2}{\rho g^2 (H - Z - H_v)^2}}} \quad (20)$$

Thus, the wave speed in liquid gas mixture is function of the local piezometric head and reach very low values during simulations of water column separation [5].

3. Validation of Penstock Protection Valve Model

3.1. Model Calibration

The orifice diameter of the oil cylinder diaphragm is the parameter of the penstock protection valve model which must be calibrated. To reach this purpose, on-site tests have been performed with closing valve of unit U1 under three flow conditions: i) dead water condition, unit output power of ii) 7.6MW and iii) 3.9MW respectively named Test017 and Test014. Unit U2 was at standstill and Unit 1 was kept connected to the grid with constant guide vane opening during the valve closing. The closing induced opening of the air valve without cavitation occurrence, which was confirmed afterwards by simulation. The top of the penstock filled up with air and behaved like a surge shaft with stabilization of water level at a height corresponding to the machine's head with discharge equal to zero. Few seconds after stabilization, a quick shutdown of the unit was ordered. To obtain simulation results in agreement with measurements for the three tests, an orifice diameter of 3.5mm is required. Table 1 compares the closing time between measurements and simulations and Figure 4 shows the non-linear time evolution of the obturator position simulated with the model.

3.2. Validation

To validate the model of the penstock protection valve, a comparison of the time evolution of the obturator position and the oil pressure in the cylinder chamber is presented respectively in Figure 5 and Figure 6 for Test017 and in Figure 7 and Figure 8 for Test014.

Table 1. Comparison of closing time of penstock protection valve between measurement and simulation.

	P [MW]	tf_VT [s]	
		Measurement	Simulation
Dead water	-	155	156
Test017	7.6	75	75
Test014	3.9	96	94

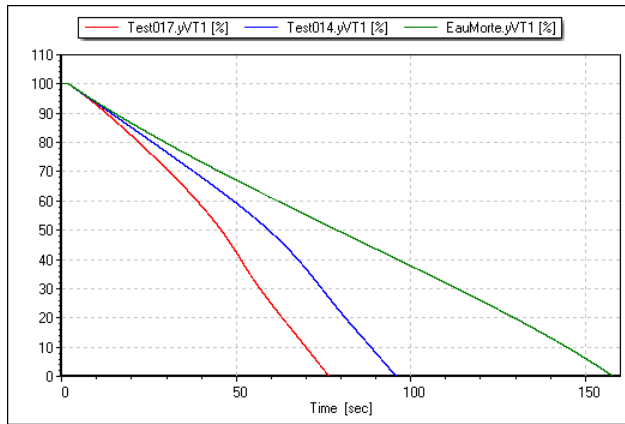


Figure 4. Time evolution of the obturator position during closing of penstock protection valve under three flow conditions: i) dead water, unit output power of ii) 7.6MW and iii) 3.9MW.

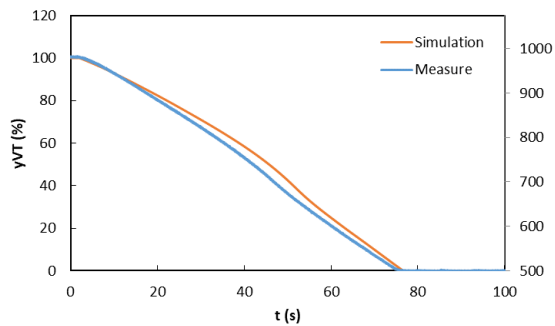


Figure 5. Comparison of time evolution of the obturator position between measurement and simulation for Test017 with unit output power of 7.6MW.

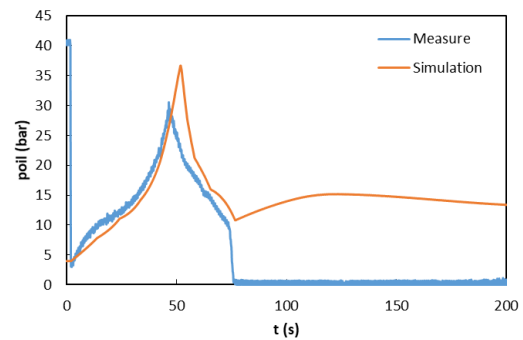


Figure 6. Comparison of time evolution of the oil pressure in cylinder between measurement and simulation for Test017 with unit output power of 7.6MW.

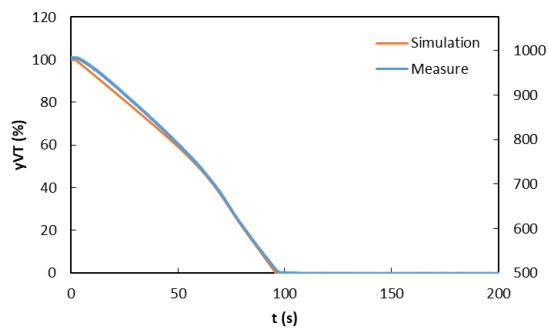


Figure 7. Comparison of time evolution of the obturator position between measurement and simulation for Test014 with unit output power of 3.9MW.

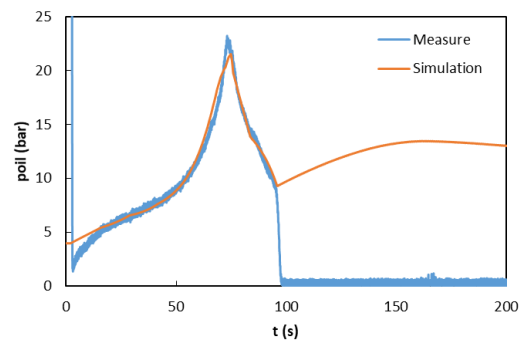


Figure 8. Comparison of time evolution of the oil pressure in cylinder between measurement and simulation for Test014 with unit output power of 3.9MW.

The non-linearity of the closing law is well reproduced by the model and both time evolution and maximum oil pressure reached during the closing is in good agreement with measurements. Influence of end position on oil pressure is not modelled, which explains differences after closure. Better results are obtained with the lowest unit output power of 3.9MW. This maximum value strongly depends on the characteristic curve of the hydraulic torque coefficient K .

4. Simulation Results of Pipe Burst

Based on this calibrated and validated model of the penstock protection valve, a simulation of pipe burst at the bottom of the penstock with protection valve closing is performed. The different physics modelling required for such scenario, presented in Section 2, have been progressively introduced in a reference load case named Case1, as showed in Table 2.

Table 2. List of simulated load cases with description of physics modelling involved.

	Physics modelling		
	DPPV	Cavitation in DPPV	FGM+AV
Case1	✗	✗	✗
Case2	✓	✗	✗
Case3	✓	✓	✗
Case4	✓	✓	✓

DPPV = Dynamic Penstock Protection Valve model
FGM = Free Gas Mixture model
AV = Air Valve

This reference load case, named Case1, simulates an imposed valve closing time which is not a result of the simulation contrary to the DPPV model. Moreover, despite of pressure below the vapour pressure along the penstock, cavitation modelling is not taken into account. Based on measurements of closing time at different unit output power, a linear extrapolation of the closing time was guessed for the pipe burst discharge. An imposed value of 12s is used in this reference load case. However, with such approach, the non-linearity of the system is not taken into account. A comparison of time evolution of obturator position, pipe burst discharge and oil pressure in cylinder are compared between the four load cases of Table 2 in Figure 9 and Figure 10. The main results are summarized in Table 3. The following conclusions on physics modelling approach can be done:

- Extrapolation of closing time from measurements at different unit output power for pipe burst simulations is not accurate. With the DPPV model, the simulated closing time is longer between +137% and +262% depending on the modelling;
- If cavitation modelling in the valve or in the penstock are not considered, the maximum oil pressure in cylinder servomotor is much higher;
- Modelling cavitation occurrence in the valve like in Case3, decreases the prediction of maximum oil pressure in cylinder of -48% compared to Case2 whereas the closing time is much less influenced;
- Modelling of column separation in the penstock with air admission through the air valve like in Case24, decreases the prediction of maximum oil pressure in cylinder of -44% compared to Case3.

Table 3. Comparison of valve closing time, pipe burst discharge and maximum oil pressure in cylinder for the different load cases.

	tf_VT [s]	QVT [m3/s]	Poil [bar]
Case1	12	47	-
Case2	28.5	71	417
Case3	30.8	67	274
Case4	43.5	61	152

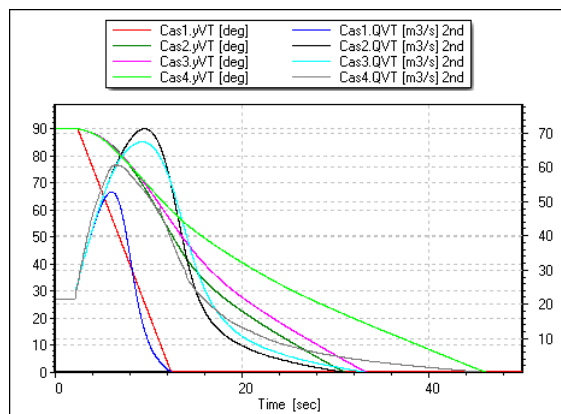


Figure 9. Comparison of valve closing time and pipe burst discharge simulated with the different physics modelling.

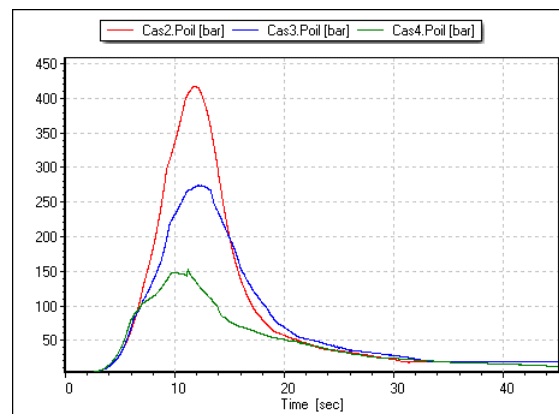


Figure 10. Comparison of oil pressure in cylinder simulated with the different physics modelling.

5. Conclusions

To perform simulations of pipe burst, a dynamic model of penstock protection valve has been developed. Considering a balance of the different torques acting on the obturator, the valve closing time results from the simulation and is not imposed a priori as a boundary condition. More than the closing time, maximum oil pressure in the cylinder of the servomotor can be computed. This model has been calibrated for the Montbovon hydroelectric power plant from measurements of valve closing with water discharge at different output power of units. Validation was performed by comparing time evolution of oil pressure during valve closing. Based on this calibrated and validated model, simulation of pipe burst at the bottom of the penstock has been performed. The physics modelling has been progressively increased to simulate such scenario: i) use of the dynamic penstock protection valve (DPPV), ii) cavitation modelling in the valve (CavDPPV); iii) cavitation in the penstock and air admission by air valve (CavDPPV+FGM+AV). It has been showed that modelling of valve cavitation, penstock cavitation and air admission by air valve are necessary to obtain realistic results. Indeed, basic modelling without cavitation influence leads to extreme results which would suggest that the servomotor or the entire valve should be replaced. In case of renovation option, an advanced modelling must be considered to obtain realistic dimensioning of servomotor.

References

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