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Investigation of Valve-Closing Law on the Maximum Head Rise of a Hydropower Plant

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Abstract. *Piping systems commonly experience the transient-state situation as the result of changes to flow conditions during pump failures, valve closures or turbine load rejection. This paper addresses transients as a consequence of the load rejection of a Francis hydropower plant (Karun 4, Ahwaz, Iran). To control the turbine system and related equipment during load rejection, the valve closing law of wicket gates is of paramount importance. The pressure rise at the end of the pressure shaft, the pressure drop in the draft tube and the speed rise while the electromagnetic braking torque disappears are solely dependent on the closing curve. Thus, an optimum closing law can eliminate the probable risk of damage to the units. This paper develops a computational model to calculate water-hammer system components, such as pressure rise, speed rise, discharge variations and pressure fluctuations. Results obtained from the present model are compared and validated with those obtained by a consultant at the Karun project. The effects of different valve-closing laws on the maximum head rise at the end of the pressure shaft and other components are also investigated.*

Keywords: *Method of characteristics; Hydropower plant; Transient flow; Valve-closing law; Karun 4 hydropower plant.*

INTRODUCTION

The power output of a hydropower plant varies constantly due to the grid demand generating variations of the turbine's rotating speed and involving a reaction of the control system. If this system is shut down suddenly, there is no load over the turbine. From this point to the complete cutting off of fluid flow through the system, the rotation of the turbine speeds up. Such situations are referred to as turbine load rejection, which can be one of the most destructive transient situations if necessary foresight is not taken into consideration. Following the turbine load rejection, peculiar events may be observed such as a pressure rise at the end of the pressure shaft, a pressure drop in the draft tube and a runner speed rise. After the electromagnetic braking torque disappears, the hydraulic torque tends to reach the runner to its runaway speed if the distributor fails to close. The wicket gates are responsible for controlling the flow rate passing through the turbine. Thus, the valve-closing law plays an important role in the evolution of pressure and rotational speed. There is always a break point in the closing-law because of the limited closing speed. A fast closure at the start of the load rejection reduces the speed rise, but causes a sharper pressure drop in the draft tube. By changing the break point, a compromise between the allowable speed rise and draft tube pressure may be obtainable [1].

The writers are not aware of any comprehensive study of the effect of a valve-closure schedule on a water hammer. Azoury et al. [2] studied this effect on the water hammer of a single pipeline. However, there has been no investigation into the effect of the valve-closing law on turbine characteristics. This paper is concerned with the pressure rise at the end of the pressure shaft of the Karun 4 hydropower plant by investigating the effects of the valve-closing law. A computational model is developed and validated by comparing the results with those obtained by a Karun project consultant [1]. In addition, different closing laws are examined to reduce the maximum head rise.

The governing equations describing unsteady pipe flow with the assumptions of fluid compressibility, linear elasticity of walls of conduits and the steady-state formula of the head-loss during the transient state can be expressed in terms of two dependent

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variables: centerline pressure and average velocity [3]. The steady or quasi-steady friction approximation has been used to determine shear stress and the peculiar events occurring. One-dimensional models have been developed for predicting transient flow behavior [4]. In two-dimensional models, some simplifications are made due to the complexity of the equations [5,6].

The plan of the paper is as follows: Governing equations, continuity and momentum equations in one-dimensional flow are presented in the next section. The method of characteristics is then discussed to reduce partial differential equations into ordinary ones. Subsequently, turbine boundary conditions are set to close the system of equations based on Chuadhry's modeling [7]. Finally, validation of the present model is addressed, followed by conclusions on the Karun 4 hydropower plant closing-law.

GOVERNING EQUATIONS

The governing one-dimensional equations describing unsteady pipe flow are presented in terms of two dependent variables of centerline pressure, $P(x,t)$ and average velocity, $V(x;t)$ and two independent variables of the coordinate distance along the pipe, x, and time t [8]. The assumptions of fluid compressibility, linear elasticity of walls of conduits and the steady-state foumula of the head-loss during the transient state are also considered.

$$L_1 = V_t + VV_x + \frac{1}{\rho}P_x + g\sin\alpha + \frac{f}{2D}V|V| = 0, \quad (1)$$

$$L_2 = \frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x} + \rho a^2\frac{\partial V}{\partial x} = 0, \quad (2)$$

in which a = wave speed; g = gravitational acceleration; ρ = mass density of fluid; α = pipe angle from the horizontal line; and D = pipe diameter.

METHOD OF CHARACTERISTICS

Perhaps the most common solution to unsteady pipe flow is obtained by the Method Of Characteristics (MOC). The latter reduces Equations 1 and 2 into two algebraic Equations expressed in terms of discharge Q and head H, called compatibility equations, which are valid along their characteristic lines (Figure 1). These compatibility equations are of the form [9,10].

$$H_{i,k} = C_{c^+} - B_{c^+}Q_{i,k}, \quad (3)$$

$$H_{i,k} = C_{c^-} - B_{c^-}Q_{i,k}, \quad (4)$$

in which the constants of integration are:

$$C_{c^+} = H_{i-1,t-\Delta t} + Q_{i-1,t-\Delta t}\left[B - R|Q_{i-1,t-\Delta t}|(1-\varepsilon)\right], \quad (5)$$

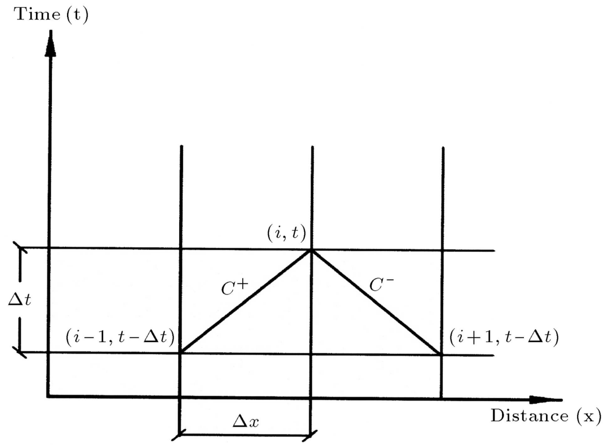

Figure 1. The $x-t$ grid showing characteristics.

$$B_{c^+} = B + \varepsilon R|Q_{i-1,t-\Delta t}|, \quad (6)$$

$$C_{c^-} = H_{i+1,t-\Delta t} - Q_{i+1,t-\Delta t}\left[B - R|Q_{i+1,t-\Delta t}|(1-\varepsilon)\right], \quad (7)$$

$$B_{c^-} = B + \varepsilon R|Q_{i+1,t-\Delta t}|, \quad (8)$$

where i = ith cross-section; Δt = time step; and ε is the linearization constant. Equations 4 and 5 are simultaneously solved at any grid intersection point to obtain the following equations for the unknowns $H_{i,k}, Q_{i,k}$:

$$H_{i,k} = \frac{C_{c^+}B_{c^-} + C_{c^-}B_{c^+}}{B_{c^+} + B_{c^-}}, \quad (9)$$

$$Q_{i,k} = \frac{C_{c^+} - C_{c^-}}{B_{c^+} + B_{c^-}}. \quad (10)$$

At the end of a pipe segment, only one of the compatibility equations is present. Hence, to progress with the solution to the desired time step, it is necessary to introduce appropriate boundary conditions. The next section deals with the Francis turbine boundary condition presented in [11].

MODELING OF A FRANCIS TURBINE

Figure 2 shows the schematic diagram of a hydropower plant. The flow through its Francis turbine depends upon the net head, the rotational speed of the unit and the wicket-gate opening. The data for the turbine flow and power output are presented in a graphical form known as Hill charts. The values in these charts are based on the reference parameters, unit flow q, unit speed ϕ and unit power p (Table 1).

The computational procedure starts by determining the value of the wicket-gate opening at the end of time step τ_P. To enter the Hill chart, the values of N_e

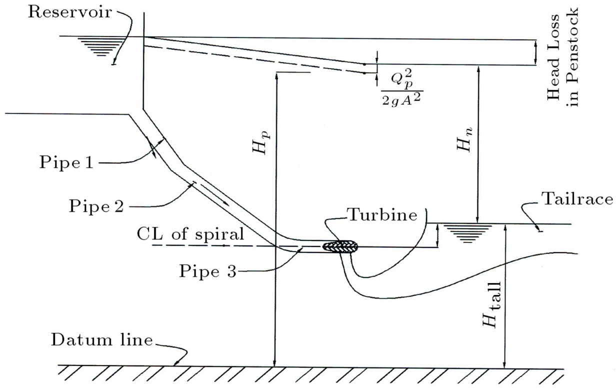

Figure 2. Schematic diagram of the power plant.

Table 1. Definition of reference parameters and unit values.

Parameters	Definition	Units
Q	$\frac{DN}{84.45\sqrt{H_n}}$	$\frac{\text{m.RPM}}{\sqrt{\text{m}}}$
q	$\frac{Q}{D^2\sqrt{H_n}}$	$\frac{\text{m}^3/s}{\text{m}^2\sqrt{\text{m}}}$
p	$\frac{P}{D^2 H_n^{\frac{3}{2}}}$	$\frac{\text{kW}}{\text{m}^2\ \text{m}^{\frac{3}{2}}}$

and H_{ne} (extrapolated rotational speed and net head) may be determined by a parabolic extrapolation. The turbine characteristics may be written in a linear form:

$$q = a_0 + a_1\phi, \qquad \text{or} \qquad a_2 H_n^{\frac{1}{2}} = Q_P - a_3. \tag{11}$$

The speed of the turbo-generator changes according to the following equation:

$$P_{\text{tur}} - \frac{P_{\text{gen}}}{\eta_{\text{gen}}} = WR^2 \left(\frac{2\pi}{60}\right)^2 N\frac{dN}{dt}, \tag{12}$$

in which P_{gen} = generator load; P_{tur} = power developed by the turbine both in kW; N = speed in rpm; WR^2 = total moment of inertia of the turbine and generator in kgm^2, and η_g = generator efficiency. Integrating both sides of Equation 12 and solving for N_P yields:

$$N_P = \left\{ N_1^2 + 182.38\frac{\Delta t}{WR^2}\left[0.5(P_{\text{tur}1} + P_{\text{tur}P}) - \frac{0.5}{\eta_{\text{gen}}}(P_{\text{gen}1} + P_{\text{gen}P})\right]\right\}^{0.5}, \tag{13}$$

where subscripts 1 and P indicate the values of the variables at the beginning and end of the time step.

By simultaneously solving the energy equation, the C^+ compatibility equation and the turbine characteristics (Equation 11), the instantaneous flow rate at the entrance to the scroll case, Q_P, yields:

$$Q_P = Q_P(H_{\text{tail}}, C_{c^+}, B_{c^+}, D_r, A, \tau_P, q, \phi), \tag{14}$$

in which A = cross-sectional area of the pressure conduit at the turbine inlet; H_{tail} = tail water level above datum and D_r = diameter of the runner. Now, Equation 11 is used to determine the instantaneous net head, H_n. The value of ϕ_e is computed using the estimated value of N_e and the computed value of H_n; the turbine output, $P_{\text{tur}P}$, is then determined form the turbine characteristic data for ϕ_e and τ_P [12]. Up to this point, the procedure started with assumed values (extrapolated values) for N_e and H_n. It was only an initialization for the procedure which are not necessarily the correct (converged) values for either rotational speed or head rise at the next time step. Thus, if the assumed rotational speed and the calculated one (i.e. Equation 13) are within the given tolerance, the converged velocity at the next step is found, otherwise the calculated values would be taken as an initialization until convergence. The convergence criteria we have taken here is (Nnew − Ninitial)/Nnew < 0.2. After finding a converged value for rotational speed, the time step can be incremented to analyze the system in later time steps.

KARUN 4 TEST CASE AND RESULTS

Karun 4 is one of the newly constructed hydropower plants in Iran (Ahwaz). Figure 2 shows a schematic diagram of this power plant. Pipe physical data and specifications of its Francis turbine are given in Tables 2 and 3, respectively. Table 4 contains steady-state data of the power plant.

To validate the present model, the rotational speed of the runner, the flow at the entrance to the scroll case and the pressure rise at the end of the pressure shaft are compared with the Voith results [1] and shown in Figure 3. Figure 3d shows Voith's closing law. The first stroke is 5 s $< t <$ 12 s and the second stroke takes place when 12 s $< t \leq$ 19 s. Similarly, the third stroke is when 19 s $< t <$ 26 s. The wicket gates are closed in 26 s during three steps.

Table 2. Pipe physical data.

Pipe Number	Pipe Length (m)	Pipe Diameter (m)
1	157.55	6.5
2	261.70	6
3	40.68	4.368

Table 3. Turbine characteristics.

Runner Diameter (m)	Moment of Inertia (tm^2)	C.L of Spiral (m.a.s.l.)
4.7	6250	834

*: m.a.s.l.: Meters above sea level.

Table 4. Steady-state power plant data.

Power (W)	**Head Water Level HWL (m.a.s.l.)**	**Tailrace Water Level TWL (m.a.s.l.)**	H_n **(m)**	Q **(m^3/s)**	**Opening Gate (%)**
295×10^6	1025	843.2	179.6	178.5	96.6

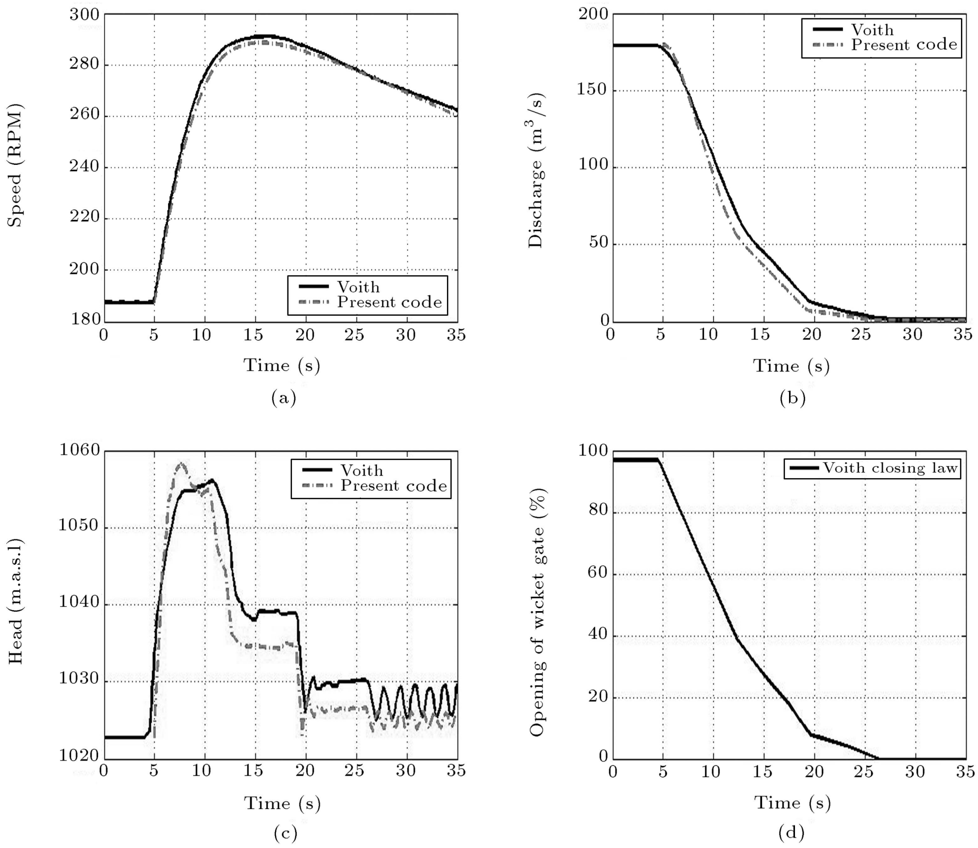

Figure 3. Comparison between the present code and Voith's results.

Upon closing the wicket gates during the first stroke, the head at the end of the pressure shaft increases severely. At this stage, the head rise is accompanied by an increase in the rotational speed and a decrease in the discharge through the turbine. It can be seen that during the first stroke of the closing law, the maximum head rise occurs. At the end of the first stroke, when the slope of the closing law changes (t = 12 s) rise slumps and after a short period of time, the maximum runner speed takes place (t = 16 s) and keeps decreasing afterwards. Thus, the maximum rotational speed has its maximum at the second stroke of closing (12 s $< t <$ 19 s). Based on Voith's closing law, there is a 5-second delay in maximum head rise and in the maximum of rotational speed, thus, they do not occur simultaneously. During the second stroke, the head rise is nearly constant with little change in its value. It is abvious that the discharge through the turbine decreases as the wicket gates are closing and the trend of its curve is the same as the valve closing law by a scaling in its values. After the end of the second stroke (t = 19 s), the slope of the closing law changes to reach zero (Figure 3 shows another fall in the value of the head rise) and when the valve is completely closed, head rise fluctuations around the steady-state head exist until the wall shear stress reduces them to zero. In the last stage of closing (Figure 3), the rotational speed is decreasing because the flow through the turbine is decreased enough to produce no more hydraulic torque to make the runner run and after a period of time, the rotational speed will reach its steady-state value. When the wicket gates are set to be fully closed ($t >$ 26 s) the fluctuations in the head rise subject to wall shear stress will be damped and the system returns to another steady-state condition.

To investigate the effects of the valve-closing law and the change in the break point, different closing laws are examined. Vakil and Firoozabadi [13,14] showed that if the valve could be closed as rapidly as possible in its first closing phase, the pressure peaks will be

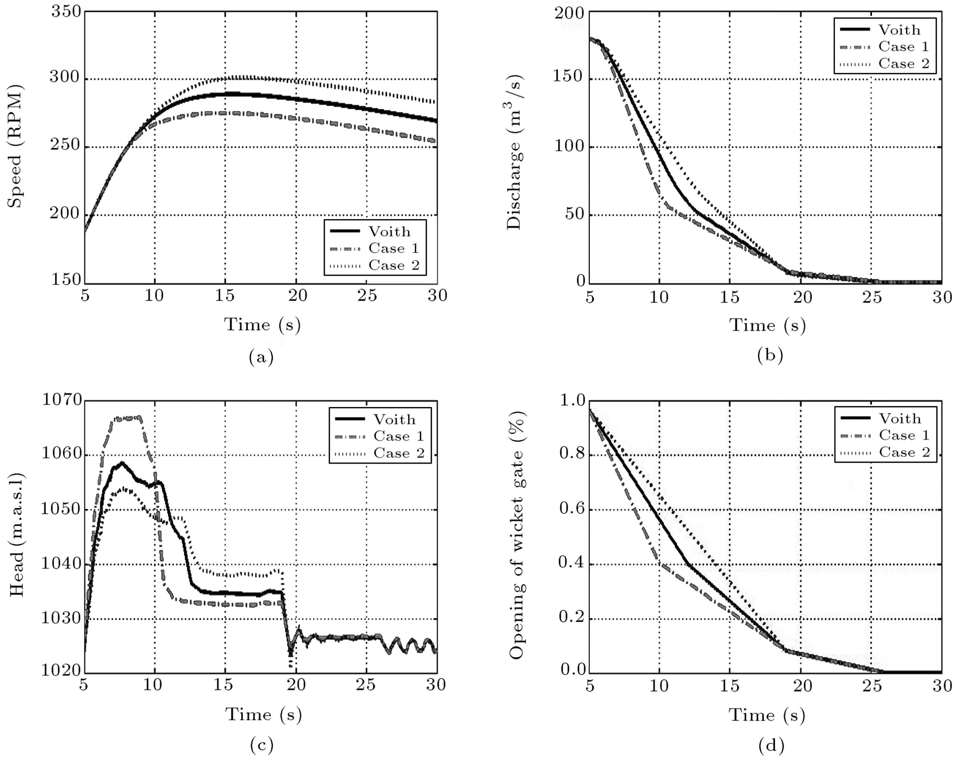

Figure 4. Effects of valve closing law on discharge rotational speed and head rise curves.

effectively reduced. Thus, it was expected here that by increasing the speed of the closing-law by changing the break point, the maximum head rise at the end of the pressure shaft reduces, but the reverse occurred. As shown in Figure 4, the faster the valve can be closed, the further the head rise takes place, whereas the maximum rotational speed decreases by rapidly closing. As we expected, it can be seen from Figure 4 that faster closing would reduce the discharge through the turbine. This is because the head-discharge relationship in a real turbine cannot be simply presented by the valve model. The flow through a Francis turbine depends upon the net head, rotational speed of unit and wicket-gate opening. Therefore, a suitable valve model should contain all these factors.

As mentioned earlier, it seems that the maximum head rise would be reduced if the valve could be closed slower at its first stroke. However, it was shown that the rotational speed would infringe its runaway speed, which has destructive effects on the system and if not controlled, would damage the unit. Thus, it can be deduced from what is said that if the closing-law could be kept near Case 2 in Figure 4, the result is in favor of maximum head rise and if it can be kept near Voith's closing-law, the results would be in favor of the rotational speed. This motivates one to check other cases that come about between these two. In other words, the first stroke should be short enough so that Voith's closing law approaches Case 1 in Figure 4. In this case, in order to close the valve faster, one changes the break point along Voith's closing-law. Figure 5 shows different valve closing laws having the same slope (the same speed) in the first stroke but with different break point along the line AB. Point B moves along the line AB to examine the effects of the shorter movement in the first stroke. As Point B moves towards Point A (Figure 5), the results are more acceptable, e.g. Case AB$''$. So, on the one hand, the maximum head rise decreases (Figure 6) since the line AB$''$ is close enough to Case 2 to benefit from its properties and, on the other hand, it gains the advantage of

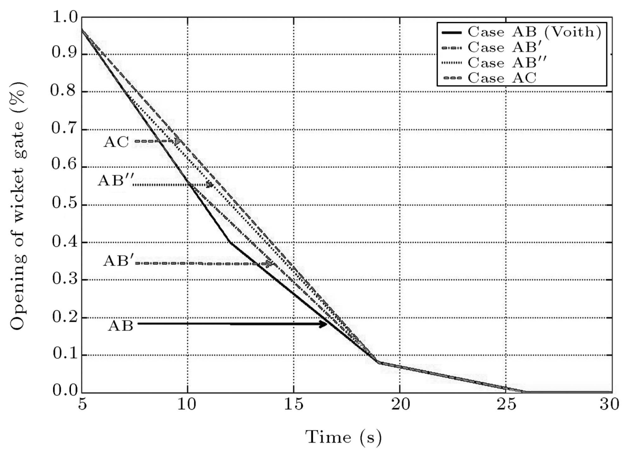

Figure 5. Valve closing laws.

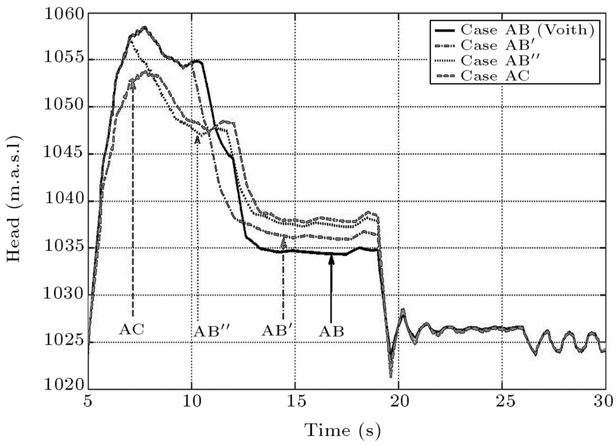

Figure 6. Effect of valve closing law on the head rise.

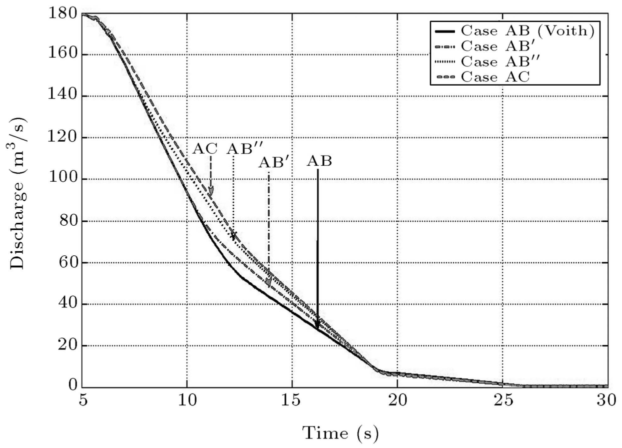

Figure 8. Effect of valve closing law on discharge through the turbine.

being along the line AB (Voith's closing law). The latter means that the rotational speed remains in an acceptable range and does not violate the runaway speed (Figure 7). Therefore, Case AB″ can be a reasonable choice for closing the wicket gates to meet all the required conditions in addition to maintaining the discharge through the turbine as close to the Voith closing law (Figure 8).

It should be noted that what we have done in this article is to first validate the written code with Voith's results [1] (Voith closing law) and then, examine more closing laws for the system in the hope of getting a better compromise for maximum head rise and maximum rotational velocity decrease. So, it was attempted to use the advantages of all the closing laws possible.

SUMMARY AND CONCLUSIONS

Based on the Francis turbine modeling, the Karun 4 power plant was modeled to investigate the effect of the valve closing-law on the head rise at the end of the pressure shaft and other water hammer components. A simple model of a valve revealed that the more rapid closing of the valve at the first stroke of closing could reduce the maximum head rise. This result, however, was not applicable to a real turbine. Results showed that rapid closing would result in an increase in head rise and a decrease in the maximum rotational speed rise. These results revealed the complicated nature of the head-discharge relationship described in a graphical form called a Hill chart, which needs to be modeled more accurately than a simple valve model. Thus, the more rapidly the servomotors could close the wicket gate, the further decrease could result in maximum head rise and, in return, the maximum rotational speed rise would increase. To reach a compromise value between these values, other design parameters should be taken into consideration. However, simultaneously reducing the maximum increase in the head rise and rotational speed upon closing the valve, especially at its first stroke, would meet fairly desirable conditions.

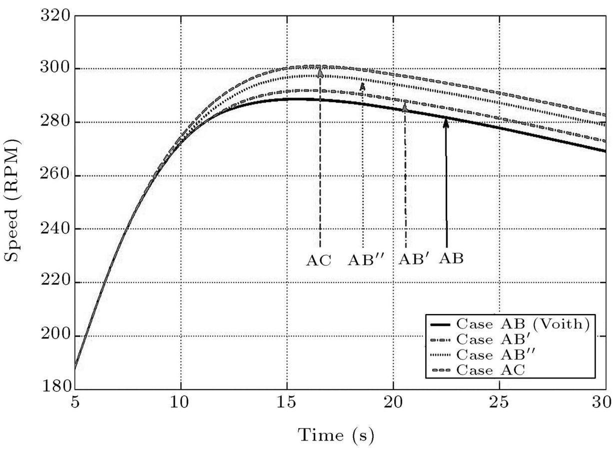

Figure 7. Effect of valve closing law on rotational speed.

NOMENCLATURE

A	cross-sectional area of the pipe
a	wave speed
$a_0,\ a_1$	hill chart constants
B_{c^+}	constant
B_{c^-}	onstant
B	constant
C^-	negative characteristic line
C^+	positive characteristic line
C_{c^+}	constant
C_{c^-}	constant
D	pipe diameter
D_r	runner diameter
f	Darcy-Weisbach friction factor
g	gravitational acceleration

H	piezometric head
H_n	turbine net head
H_{ne}	extrapolated net head
H_{tail}	tail water level above datum
HWL	head water level
i	ith cross-section
k	cnstant
L	pipe length
N	rotational speed
N_P	next time step rotational speed
N_e	extrapolated rotational speed turbine power
P_{tut}	turbine power
P_{gen}	generator load
p	unit power
Q	discharge
Q_P	next time step discharge
q	unit flow
R	constant
t	time
TWL	tail water level
Δt	time step
V	fluid velocity
WR^2	total moment of inertia of the turbine and generator
x	distance along pipe

Greek Symbols

α	pipe angle due to horizontal line
ϕ	unit speed
λ	unknown multiplier
η_g	generator efficiency
ρ	fluid density
ε	linearization constant
τ_p	opening of wicket gate

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