ИСПИТУВАЊЕ НА ВОДЕНИОТ УДАР ВО ХЕЦ ГЛОБОЧИЦА

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ABSTRACT

Water hammer should be one of the key elements of the feasibility and design studies in order to ensure safe operation of the hydro-electric powerplant (HPP). The main objective of this paper is to identify critical flow regimes which may cause unacceptable water hammer in a Francis turbine HPP. Water hammer is described by the set of hyperbolic partial differential equations, the continuity equation and the equation of motion. The method of characteristics is used for solving these equations. The water turbine is treated as a boundary condition within the method of characteristics. The paper concludes with water hammer analysis in Globocica HPP, Republic of Macedonia. The system under consideration is fitted with two 23 MW Francis turbines. There is a reasonable agreement between the computational and field test results.

1. INTRODUCTION

Feasibility and design studies of hydropower systems include water hammer analysis in order to ensure safe operation of the system. The main objective of this paper is to identify critical flow regimes which may cause unacceptable water hammer in a Francis turbine hydro-electric powerplant (HPP). Transient regimes may cause excessive water hammer and possible column separation in the system. These include the turbine load acceptance, load reduction or sudden load rejection, turbine runaway, shutoff valve closure, and a combined operation of the turbine and valve. Water hammer may disturb operation of the hydropower system and damage the system components (pipe collapse or rupture). High pressure fluctuations in the flow-passage system and the turbine rotational speed are traditionally controlled by appropriate operational means (guide vanes closing and opening). Additional protective measures against unacceptable water hammer include installation of surge control devices (flywheel, pressure relief valve, surge tank, air cushion surge chamber, aeration pipe, air valve) and redesign of the pipeline layout [1], [2], [3],

[4]. Operational, safety and economic factors are decisive for optimal selection of the method for controlling transients.

Water hammer is described by a set of hyperbolic partial differential equations, the continuity equation and the equation of motion [1], [3]. The method of characteristics is used for solving the water hammer equations. The water turbine is treated as a boundary condition within the numerical grid of the method of characteristics. The paper concludes with water hammer analysis in Globocica HPP, Republic of Macedonia. The system under consideration is fitted with two 23 MW Francis turbines. Comparison between the results of measurement and calculation for the case of sudden full-load rejection of one turbine unit and results from the parametric numerical analysis for sudden full-load rejection of one and two turbine units are presented and discussed. There is a reasonable agreement between the computational and field test results.

2. WATER HAMMER MODEL

Water hammer is the transmission of pressure waves along the pipeline resulting from a change in flow velocity. Unsteady flow in closed conduits is described by two one-dimensional equations; the continuity equation and the equation of motion [1], [3]:

$$\frac{\partial H}{\partial t} + v\frac{\partial H}{\partial x} - v\sin\theta_p + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \qquad (1)$$

and

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + v\frac{\partial v}{\partial x} + \frac{\lambda v|v|}{2D} = 0 \qquad (2)$$

in which H = piezometric head (head), t = time, v = pipe flow velocity, x = distance, θ_p = pipe slope, a = wave speed, g = gravitational acceleration, λ = Darcy-Weisbach friction factor, and D = pipe diameter. The symbols are defined as they appear in the paper. For most engineering applications, the pipe slope and convective acceleration terms in equations (1) and (2) are small and neglected [1], [3]. Usually, the discharge Q = vA replaces the flow velocity v; A = pipe area. Equations (1) and (2) are a set of quasi-linear hyperbolic partial differential equations. The common method of solving these equations is by the method of characteristics (MOC) transformation [1], [3]. The resulting water hammer compatibility equations, written in a finite-difference form, are (small terms are neglected) [3]:

- along the C+ characteristic line ($\Delta x/\Delta t = \alpha$):

$$H_{j,t} - \left[H_{j-1,t-\Delta t} + \frac{a}{gA}(Q_u)_{j,t} - (Q_d)_{j-1,t-\Delta t}\right] + \frac{\lambda\Delta x}{2gDA^2}(Q_u)_{j,t}\left|(Q_d)_{j-1,t-\Delta t}\right| = 0 \qquad (3)$$

- along the C- characteristic line ($\Delta x/\Delta t = \alpha$):

$$H_{j,t} - \left[H_{j+1,t-\Delta t} - \frac{a}{gA} (Q_d)_{j,t} - (Q_u)_{j+1,t-\Delta t} - \frac{l x}{2gDA^2} (Q_d)_{j,t} \left| (Q_u)_{j+1,t-\Delta t} \right| = 0 \qquad (4)$$

in which j = computational section index, Q_u = discharge at the upstream side of the computational section, Q_d = discharge at the downstream side of the computational section, Δx = reach length, and Δt = time step. At a boundary, the boundary equation replaces one of the water hammer compatibility equations.

A constant value of the Darcy-Weisbach friction factor l is used in equations (3) and (4). This assumption may be corrected for the case of rapid transients by introducing unsteady friction term in the above equations [5]. Discharge at the upstream side of the computational section Qu and the discharge at the downstream side of the section Qd are identical for the water hammer case ($Q_u \equiv Q_d$) i.e. the pressure at a section is greater than the liquid vapour pressure. Transient cavitating flow in a pipeline system occurs when the pressure drops to the liquid vapour pressure. The standard water hammer solution is no longer valid. The discrete vapour cavity model (DVCM) is used in most engineering water hammer software packages [3], [6], [7].

2.1 Water turbine boundary condition

The water turbine boundary condition simulates a number of operating regimes including a turbine start-up, load acceptance, load reduction and stoppage of the unit, load rejection and runaway. The two most important governing parameters, the maximum pressure head rise and the maximum turbine rotational speed rise, are normally defined by the full-load rejection of the turbine. For this case, the water turbine in-line boundary condition is described by the following set of equations:

- water hammer compatibility equations (3) and (4)
- head balance equation:

$$H_u - H_r\left(\alpha^2 + q^2\right) W_H\left(y(t), x\right) - H = 0 \qquad (5)$$

- dynamic equation of the turbine unit rotating masses:

$$\alpha^2 + q^2 \; W_T\left(y(t), x\right) + \beta_{t-2\Delta t} - I \frac{\pi}{30} \frac{n_r}{T_r} \frac{1}{\Delta t}\left(\alpha - \alpha_{t-2\Delta t}\right) = 0 \qquad (6)$$

in which Hu = head at the upstream side of the turbine, H_r = rated turbine head, a = n/nr = dimensionless turbine rotational speed, n = turbine rotational speed, r = rated conditions, $q = Q/Qr$ = dimensionless discharge, $WH(y(t), x)$ = dimensionless head characteristic, $y(t)$ = dimensionless guide vanes position (and runner blades position for double regulated machines), $x = \pi + tg^{-1}(q/a)$ = angular position in turbine characteristic curve, H = head at the downstream side of the turbine, W_T(y(t), x) = dimensionless torque characteristic, $\beta = T/Tr$ = dimensionless torque, T = torque, I = polar moment of inertia of rotating parts. The unknowns in this system of equations are the heads *Hu* and *H*, discharge *Q* (*Qu* ≡ *Qd*) and turbine rotational speed *n*. The system of non-linear equations is solved by the Newton-Raphson method [8].

3. WATER HAMMER ANALYSIS IN GLOBOCICA HPP

Globocica HPP, Republic of Macedonia is comprised of the following system components - see Figure 1:

- upstream reservoir,
- headrace tunnel of diameter $D = 4.4$ m and length $L = 7709.6$ m,
- orifice surge tank with lower gallery and overflow (6 orifices of diameter $d = 0.85$ m; shaft diameter $D = 12.15$ m),
- penstock of equivalent diameter $D = 3.524$ m and total length $L = 121.44$ m,
- two branch pipes of diameter $D = 2.2$ m and length $L = 47.85$ m,
- two vertical shaft 23 MW Francis turbines of net head $Hn = 101.8$ m and discharge $Qn = 25$ m^3/s,
- downstream reservoir.

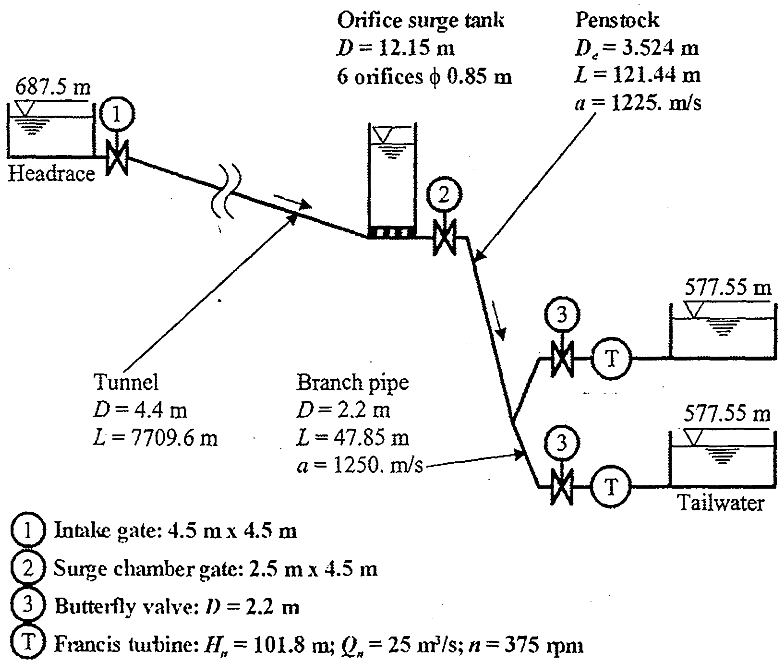

Figure 1. Layout of Globocica HPP, Republic of Macedonia.

The water level in the upstream reservoir (headrace water level) zu can be in range from 682.5 m to 692.0 m, however, the practical maximum level has never exceeded 687.5 m; the level in the downstream reservoir (tailwater level) zd is in range from 577.3 m to 577.55 m. The steady-state speed of the turbine is n = 375.0 rpm and the polar moment of inertia of the unit rotating parts I = 100.0 × 103 kgm³.

Recently new digital turbine governors have been installed in the powerplant. The owner of the plant and contractors have performed various operating regimes during trial tests including sudden load rejections. The main objective of measurements was to set guide vanes closing and opening times for each turbine and to check resulting maximum pressure in the scroll-case at the turbine inlet and maximum turbine rotational speed. This paper presents (1) comparison between the results of measurement and calculation for the case of sudden full-load rejection of one turbine unit and (2) results from the parametric numerical analysis for sudden full-load rejection of one and two turbine units at different net heads. Calculations have been performed with the aid of water hammer computer packages in Litostroj [9], [10], [11].

3.1 Comparison between the results of measurement and calculation for the case of sudden full-load rejection of one turbine unit

Comparison between the results of measurement and calculation for the case of sudden full-load rejection of one turbine unit from turbine output of 23.6 MW is presented; the second turbine was stationary in this case. The turbine was disconnected from the electrical grid followed by the full closure of the guide vanes. **Figure 2** depicts a two-speed guide vane servomotor stroke for the case considered.

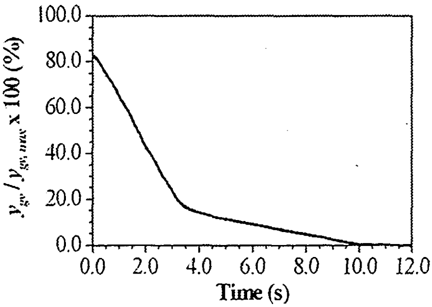

Figure 2. Guide vane (y_{gv}) servomotor stroke.

Pressure rise at the turbine inlet is one of the most significant design parameters, in particular in HPP with long conveyance systems. **Figure 3** compares the pressure rise in the turbine scroll-case. In the case of the unit shutdown there is a reasonable agreement between the computed pressure head rise of 28.9 % and measured one of 28.0 %. The computed maximum pressure head in the scroll-case at the turbine inlet hsc, max = 127.m (datum level z = 576.75 m) is lower than the maximum permissible pressure head hmax = 135. m. The turbine rotational speed n is depicted in **Figure 4**. Again, there is a reasonable agreement between the results of computation and measurement. The computed maximum turbine rotational speed rise of 29.4 % is slightly higher than the measured one of 27.6 %. The computed maximum speed rise is lower than permissible one of 32. %. Surge tank water level oscillations were not recorded at this test - these measurements are planned in the stage of final governor settings. Surge tank oscillations will be addressed in **Section 3.2.**

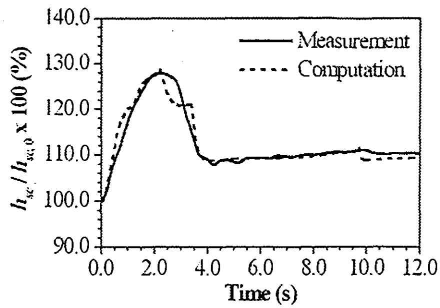

Figure 3. Computed and measured scroll-case pressure head (*hsc*) after sudden load rejection of one turbine unit (*hsc*, 0 = 98.5 m).

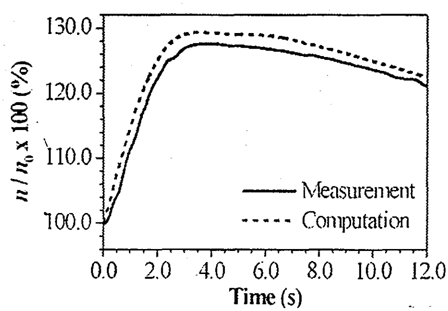

Figure 4. Computed and measured turbine rotational speed (n) after sudden load rejection of one turbine unit (n_0 = 375 rpm).

3.2 Parametric numerical analysis for sudden full-load rejection of one and two turbine units

Parametric analysis of sudden full-load rejection of one and two turbines at minimum, normal and maximum headrace and tailrace water levels i.e. z_u = {682.5, 687.5, 692.} m and z_d = {577.3, 577.55} m has been performed for a number of guide vane servomotor two-speed strokes. Sudden full-load rejection is the most severe transient regime which occurs at normal operating conditions [1]. The turbine(s) is (are simultaneously) disconnected from the electrical grid followed by the full closure of the guide vanes. The analysis has been performed for minimum servomotor closing times T_f = {5., 6., 6.5, 7.} s with cushioning time T_h = 7. s starting at guide vane servomotor stroke ygv, h = 16.7 % (IEC 60308). A guide vane closing time with T_f = 6.5 s and T_h = 7. s has been selected as optimal closing time.

Figure 5 shows the maximum scroll-case pressure head $h_{sc,max}$ (datum level z = 576.75 m) for the considered transient regimes at the optimal guide vanes closing time (T_f = 6.5 s, T_h = 7. s). The maximum pressure head is lower than the maximum permissible pressure head (h_{max} = 135. m) for all considered cases. The maximum turbine rotational speed rise Dn_{max} is depicted in Figure 6. The computed maximum speed rise is lower than the permissible speed rise (32. %) for all computational runs.

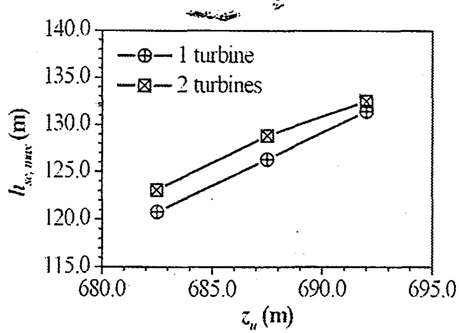

Figure 5. Maximum scroll-case pressure head ($h_{sc,max}$) after sudden full-load rejection of one and two turbines (z_u = headrace water level).

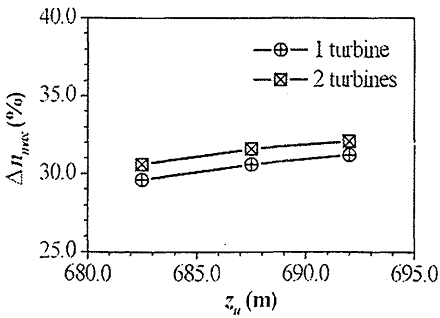

Figure 6. Maximum turbine rotational speed rise (Dn_{max}) after sudden full-load rejection of one and two turbines (z_u = headrace water level).

Figure 7 shows the maximum and minimum water level oscillations during considered transient regimes. The maximum water level oscillations are lower than the upper ceiling level of the surge shaft of 700.5 m and the minimum water level oscillations are well above the bottom level of the surge tank of 659. m.

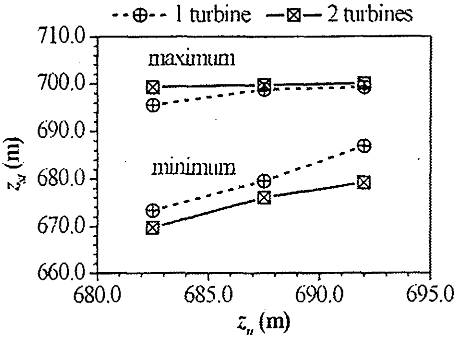

Figure 7. Maximum and minimum surge tank water level oscillations (z_{st}) after sudden full-load rejection of one and two turbines (z_{u} = headrace water level).

Envelopes of the maximum and minimum piezometric heads along the tunnel and the penstock for the considered transient regimes at the normal headrace water level z_{u} = 687.5 m are shown in **Figure 8**. This diagram is important for design engineers to construct safe and economic pipeline system. The computed minimum head is well above the pipeline profile. The same conclusion holds for other four considered cases.

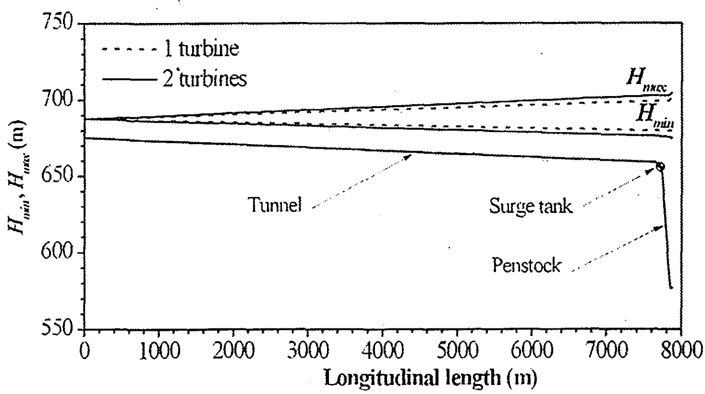

Figure 8. Envelopes of maximum (H_{max}) and minimum (H_{min}) piezometric heads along the tunnel and the penstock after sudden full-load rejection of one and two turbines at the headrace water level z_{u} = 687.5 m.

4. CONCLUSIONS

Transient analysis in hydro-electric powerplants should include critical operating conditions such that the loads induced by water hammer are kept within the prescribed limits. The method of characteristics computational results reasonably agree with the results of measurements for the case of sudden load rejection of one turbine in Globocica HPP. The method is recommended for engineering transient analysis in hydraulic systems. Sudden full-load rejection of two Francis turbines at maximum gross head has been found to be the most severe transient regime in Globocica HPP at normal operating conditions.

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