Analysis of Pump-Turbine “S” Instability and Reverse Waterhammer Incidents in Hydropower Systems

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Abstract

Hydraulic systems continually experience dynamic transients or oscillations which threaten the hydroelectric plant from extreme water hammer pressures or resonance. In particular, the minimum pressure variations downstream of the turbine runner during the load rejection or other events may cause dangerous water column separation and subsequent rejoinder. Water column separation can be easily observed from the measurements of site transient tests, and has indeed caused serious historical damages to the machine and water conveyance system. Several technical issues regarding water column separation in draft tubes, including “S” instability of turbine characteristic curves, numerical instability and uncertainty of computer programs, are discussed here through case studies and available model and site test data. Catastrophic accidents experienced at a Kaplan turbine and in a long tailrace tunnel project, as well as other troubles detected in a more timely fashion, are revisited in order to demonstrate the severity of reverse water hammer. However, as there is no simple design solutions for such complex systems, this paper emphasizes that the design of hydraulic systems is always difficult, difficulties that are compounded when the phenomena in question are non-linear (water hammer), dynamic (involving wave interaction and complex devices of turbines, controls, and electrical systems), and non-monotonic (severity of response is seldom simply connected to severity of load as with vibrations and resonance, and the complexity of transient loads), and thus may lead to high economic and safety challenges and consequences.

Keywords: Pump-turbine; Load rejection; Transients; Reverse waterhammer; “S” instability; Water column separation.

1. Introduction

The design, construction and operation of a hydropower plant involve numerous challenges in environmental and hydrologic assessments, engineering planning and design, financing with far-sighted political perspective, demanding construction and supervision, painstaking commissioning and troubleshooting, and meticulous operation and control. Tens of thousands of details must be well conceived, accurately executed, and carefully coordinated for a project to achieve economic, social, technical and environmental success. Overlooking or poorly integrating such details can cause great complications. One such vital detail is hydraulic design of tailrace tunnels, for which the paper overviews a few particular hydro systems and provides background on the observed water column separation and rejoinder phenomena. There tends to be a bias favoring the analysis of upstream hydraulic components including the turbine, while less attention is paid to the downstream water passage, often treating the design of draft tube and tailrace as mere afterthoughts. Yet, many important, and potentially destructive, hydraulic phenomena occur in the draft tube, its extension, and/or the pressurized tailrace tunnel, especially if the tailrace is long. The term “tailrace tunnel” in this context refers to a pressurized conduit. Reflections on actual events in such systems lead to vital lessons for future safe and effective design. In hydropower plants with long tailrace tunnels, the minimum pressure at the downstream of turbine runner, resulted from load rejection or other transient events, can seriously damage or impair both the turbine assembly and the water conveyance system.

Reliable knowledge of all loads acting upon a system is the key to safe hydraulic system design. The most dangerous stresses are those provoked by pressure surge and vibrations; and the worst case is the occurrence of resonances. Maximum pressure during transient events, such as those associated with rapid closing and opening of wicket gates, can break equipment while events like earthquakes can destroy pipelines, valves or other components and may even result in the loss of life. Sound design of a new plant is impossible without a complete analysis of transient events. Transient analysis also helps to prevent resonance in existing
plants and thus increases the reliability of plants and reduces their operating and maintenance costs.

The costs associated with cavitation, transient conditions, vibrations and stress analysis for mini, small and big hydroelectric plants are often quite comparable in absolute values; thus the relative cost, as a fraction of the total investment, is obviously much greater for smaller plants. Such analysis costs in large plants usually entail less than one percent of the total, whereas these costs can be a sizable fraction of the overall expenses in some small plants. Though suffering from the same problems as the large ones, there is a dangerous though understandable tendency to simplify analysis and review process for reducing the design costs of small plants. The net result is that there tends to be higher risks in smaller plants [26].

Pumped-storages (and hydro storages) are of paramount importance as they are the most reliable and affordable energy storages for accommodating intermittent renewable generators in the power grid, also providing ancillary services as readily adjustable stand-by and running reserve for nuclear and other thermal generators. However, large hydro facilities require a large amount of capital investment, so their design, construction and operation needs to be well conceived, executed and coordinated, if they are to achieve the safe and economical operation. The challenges of high-head typical of pump-storage further complicate system design. This fact, together with cost stress on the construction, equipment and labor, justifies the need for more rational design of new pumped storage plants. That is, the system components should be strained as close to the allowable limits as possible, without endangering the safety.

For a low-specific-speed (high-head) reversible pump-turbine, owing to their “S” shaped turbine characteristics, the operations are unstable near or at runaway conditions, leading to oscillations characterized by large pressure fluctuations and transients as the system passes through the unstable dangerous zone [10, 11, 13, 32, 33]. The analysis of transients is rather a complex and time consuming task, each case introducing some new problems. Our experience on “S” form instability and hydraulic resonance, gathered from pumped-storage plant Bajina Basta in Serbia (previously, Yugoslavia), the plant endangered by hydraulic transients, is reviewed in this paper. Transient analyses were exercised at all design levels of Bajina Basta plant by both the design team (led by the first author) and the manufacturer (Toshiba). The “S” type instability was discovered and first publicly reported during 1974-1976 [32], while the numerical instability of computer program in the “S” zone is discussed for the first time in this paper.

In order to avoid the danger of water column separation and consequent reverse waterhammer, the minimum pressure in any conduit (tunnel, penstock, pipeline), as a design criterion, should not fall below 50 kPa (0.5 bar), even temporarily [1, 17, 23, 24]. This minimum pressure must be the instantaneous value during a transient event and at the highest position of waterway, typically at the runner outlet or at the upper limit of draft tube lining. Since pressure transducers only can be attached to the wall of draft tube, and since the water in the draft tube is usually in rotary motion, the measured pressure is actually the maximum value for a given cross-section; the pressure in the vortex core, by contrast, is likely at absolute vapor pressure or at the air release pressure of the dissolved air, or slightly above this if air injected [7, 14, 31]. Therefore, the measured pressures cannot be used directly as a design criterion. Some manufacturers have empirical pressure and velocity distribution data for their machines [12], and others have measured data from model turbine draft tubes [7]. All the data for each machine must be carefully evaluated because the similarity laws do not strictly apply to two-phase flows [8, 2121].

In practice, the lowest transient pressures are typically approximated using one-dimensional waterhammer theory and transient modeling, which only represents the “cross-sectional average” pressures which are usually higher than those found in the vortex core and lower than those measured at the wall of draft tube cone. In addition, the numerical analysis results could be uncertain and inaccurate around the unstable zones of turbine characteristics, so the numerical analysis must be thoroughly performed and experimentally confirmed. A larger safety margin is prudent for the minimum pressures in the draft tube, particularly given the complexity and uncertainty of these transient flows, as well the lack of an applicable similarity law. Moreover, the speed increment of the runner, in runaway condition, is crucial since the voids formed by the centrifugal force of the high-speed rotating water can be large; the pressure rise caused by even slow accelerations/decelerations of the tailrace water may have a strong influence on void collapse.

2. Transient analyses of Bajina Basta Pump-Turbine System

2.1 Four quadrant characteristic curves and “S” instability of reversible pump-turbines

High-head reversible pump-turbines with low-specific-speed have an unusual characteristic in turbine operating modes: unlike typical behavior, beyond the runaway zone, any decrease in speed reduces the discharge. In an $n'_1$-$Q'_1$ diagram partial load curves for a pump-turbine are markedly “S” shaped, as shown in Error! Reference source not found.. In this runaway zone the flow through the runner is highly complex: the water near the bend separates, flowing toward the turbine, while the water near the crown goes opposite way - towards the wicket gates. The net discharge might be either negative (turbine) or positive (pump). These operating modes are, naturally accompanied with violent vibrations and highly developed cavitation [3, 20 and 33]. Nevertheless, such machines are needed and rapidly developed towards higher head and greater unit power. The pump-storage power plant ”Bajina Basta” ( ) has such low specific speed pump-turbines. Is specific speed $n_h = 27 \left( n_h = n\sqrt{Q/H^{3/4}} \right)$, having a typical “S” type unstable characteristic as all other high-head pumps and pump-turbines. The numerical simulations exhibited intensive instability and resonance, though the accuracy of analysis is reasonably low.

To discuss the pump-turbine model characteristics, the unit speed, unit flow and unit hydraulic torque are defined as: $n = \frac{N}{\sqrt{H}}$, $Q = \frac{Q}{D^2H}$, $m = \frac{M}{D^2H}$, $M$ is the hydraulic torque. In the “S” zone three (dimensionless) discharge values of $Q'_1/Q'_10$, two negative and one positive, correspond to one value of $n'_1/n'_10$; also three (dimensionless) torques of $m'_1/m'_10$, two positive and

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one negative, correspond to one value of $n'_1/n'_10$. Such multi-valued nature would result in a difficulty in application of these curves to transient simulation. To overcome the challenge, various representations have been used in numerical modeling and each presents certain merits and limitations (Chaudhry, 1987 and Pejovic, et al., 1983). In Suter parameters, the head $W_H$ and torque $W_B$ characteristics are defined as single-valued in the entire area covered by the polar angle $\alpha = \tan^{-1}(\alpha/v)$ (Suter, 1966; Thorley, et al., 1966):

$$W_H(\alpha) = \text{SIGN}(h)\sqrt{\frac{|h|}{\alpha^2 + v^2}}; \quad W_B(\alpha) = \text{SIGN}(\beta)\sqrt{\frac{|\beta|}{\alpha^2 + v^2}}; \quad \text{here, } h = \frac{H}{H_R}, v = \frac{Q}{Q_R}, \beta = \frac{T}{T_R}, \alpha = \frac{n}{n_R}$$

When using the Suter curves, difficulties may arise from an uneven distribution of curves at the wicket gate opening. At small openings, the curves space themselves out further, and for a definite angle $x$ when the wicket gates are being closed, the values of $W_H$ and $W_B$ become indefinitely great. Consequently, the selection of interpolation and extrapolation methods of these characteristics has at least some influence on the accuracy of transient calculation results.

At the time of load rejection the generator is disconnected of the grid, neglecting the mechanical friction in the bearings and generator, the “S” type instability is described by $J \frac{\pi}{30} \frac{dn'_1}{dt} = -m'_1D'_1 \sqrt{H}$, here $J$ is the moment of inertia of rotating parts.

If the transient phenomenon begins at point A, in Fig.1, a normal turbine operation (Error! Reference source not found.) and the gate opening is not changing, $n'_1$ goes down ($dn'_1/dt < 0$) to point B since $m'_1 > 0$ (Error! Reference source not found.). At point B $m'_1 > 0$, however, when $n'_1 < n'_10$ the working point jumps from B to point C. At this moment $m'_1 < 0$ and $m'_1$ goes up to point D. At point D $m'_1D < 0$ and when we have $n'_1 > n'_1D$ the working point jumps from point D to point E, where $m'_1 > 0$. The process will be repeated passing through the points EBCDE and will never stop. Due to the waterhammer in the penstock and tailrace tunnel the head is changing quickly, which results in the operation passing through point B and D in the zone between B and D, as well as in the zone of $n'_1 < n'_1C$ and a new passage to the zone of $m'_1 > m'_1E$. Thus, all type of operations shown by the four-quadrant characteristics is possible. This nature of pump-turbine curves bring about the sudden changes in pressure, discharge, speed of rotation and torque with unpredictable superposition of pressure fluctuation, followed with resonance. Jumping from point B to point C, flow direction changes suddenly from turbine mode ($Q < 0$) to reverse pump mode ($Q > 0$), quadrant IV zone, and then from point D to point E back to the turbine flow direction.

![Bajina Basta pumped storage plant](Fig.2) (1982). Head 600 m, two units each 300 MW.

![Bajina Basta pump-turbine dimensionless “S” form unstable characteristics](Fig.1)
2.2 Computer Program and Mathematical Model

The discussed computer program was first developed in 1968 on an Elliote 803b papertape computer. The programs were subsequently transformed into FORTRAN, being gradually improved to be able to handle any system configuration, simulate waterhammer and hydraulic oscillations [33, 22]. A realistic application case of this computer program is for "Bajina Basta" pumped-storage hydroelectric plant (Fig.2). This plant is equipped with two 300 MW units, sharing a 8000 m long tunnel, a common 1200 m penstock, and a 300 m tailrace tunnel, as shown schematically in Error! Reference source not found.. The units are submerged 54 m below the tail water level to protect the system from cavitation and water column separation. Two identical pump-turbines, each developing 315 MW in turbine operation, with 600 m of net head, are installed in the power house. The maximum pump input is 310 MW for 50.8 m$^3$/s flow. The maximum pumping head is over 620 m.

The mathematical model represents the plant by a system of nodes and links. The nodes are as follows: storage basin (node 1), surge tank (2), penstock valves (3), upper (6) and lower (11) branching, pump-turbines (9 and 10), ball valves (7 and 8), lower reservoirs (13). Nodes (4), (5) and (12) are connection of pipes of different diameters.

![Diagram of the plant](image.png)

**Fig.3** Pumped storage plant “Bajina Basta”

2.3 Two Units Load Rejection

Simultaneous full load rejection of two turbine units with wicket-gates blocked in an open position (Fig.4) would cause dangerous pressure surges. It was clear that the machines tended to enter the reverse pump zone (so called "fourth quadrant"), alternatively, due to the slightly asymmetric branch pipes. The associated fluctuations in discharge were remarkable, with changes from the pump to turbine mode occurring in a matter of seconds as result of the instability of the “S” shaped pump-turbine characteristics in the runaway zone. These phenomena were described by Pejovic at al. [32] and Gajic [6] obtained similar results using another computer program. These results were initially viewed as so dramatic because only a few believed that a machine could enter the fourth quadrant; so some kind of verification was needed (1975). When the plant was finally completed (1982) the field tests fully confirmed the earlier computation results.

If the two units experience runaway simultaneously, the flow in the loop would go through phase shift; one machine would experience turbine flow and the other in the pump zone (see solid and dashed curves in Fig.4, respectively). The hydraulic torque acts to accelerate the shaft rotation of one unit and decelerate the other one. The shorter the pipes in the loop (the less inertia of water in them), the more violent the pressure surges; dangerous reverse waterhammer could damage the hydraulic machines and waterways and then flooding would be inevitable.

The peaks of pressure fluctuations exceeded 900 m, the design penstock pressure head. Therefore, the governing and protective system had been changed to prevent parallel runaway of both units to minimize the risk of a catastrophic accident. The problem was solved by ensuring that any of the four protective devices respond to all critical transients [20]. Though it is highly improbable that all four closing devices jointly fail, careful maintenance must ensure their continuous operation. The design team of this power plant carefully analyzed hydraulic transients and discovered a dangerous “S” form instability, which was published for the first time by Pejovic et al. 1976 [32]. However, the water column separation during runaway remained unnoticed as extremely dangerous by the design team (Pejovic was one member of the team), neither had it been indicated by turbine manufacturers nor other experts involved in the design and construction of the plant. Fortunately, the problem was resolved by preventing the turbine running through the unstable “S” zone of transient operation.
2.4 Load Rejection of One Unit

The response of load rejection from one unit and its resulting runaway are shown in Fig.5 [32]. The amplitude of pressure fluctuation in the draft tube, $H''$, is as high as $\pm 3$ bar, at the inlet, $H'$, up to $\pm 18$ bar, and the discharge is jumping from turbine to pump direction [32].

2.5 Comparing Simulation with Experiment

Field tests have been carried out several times in "Bajina Basta" plant both by the manufacturer and the asset owner. Several situations were tested, and various physical variables were measured and recorded. One such an event was taking place in February 1983, the testing procedure was:

- both units 1 & 2 operated as turbines, developing 298 MW each;
- Unit 1 dropped out and its wicket gates closed down rapidly;
- Unit 2 remained operational in spite of violent pressure surges.

A couple of multi-channel acquisition systems were used to record pressures in the spiral casings above and under the runners and in the draft tubes, as well as to record wicket-gates servo strokes, generators’ currents and rotating speeds of the shafts. The
discharge could not be measured. Fig. 6(a) displays the changes in Unit 1 which has dropped out. The wicket-gates remained open for some 0.2-0.3 s and then started to close down as prescribed, first rapidly and later slowly. The other lines represent other variables: pressures in the spiral casing $H_U$ and in the draft tube $H_D$, angular speed $\omega$, flow $Q$ and guide vane opening $a$. Fig. 6(b) shows pressure variations in Unit 2 which remained connected to the system with $\omega \approx$ constant and wicket-gates blocked in the open position. The other lines are, respectively, the pressure in the spiral casing $H_U$ and in the draft tube $H_D$, flow $Q$ and output $P$.

The agreement between measured and computed values is reasonable, although no calibration was carried out. However, some details are clearly visible:

- wave velocity used for computation is slightly greater than the real one (comparing peaks);
- full-sized prototype machine is different from the model, it is relatively stronger and more efficient, especially in the zone of medium and small openings.

Calibration would bring computational results closer to the measured one, but even this degree of accuracy was satisfied. The results showed that Unit 1 did enter the fourth quadrant. Discharge $Q$ was negative (pump direction) between 6.9 and 10.3 s after power failure (RP zone). These results are also plotted in Fig. 7 in the characteristic diagram of pump-turbine. Unit 1 follows the “S” shaped curve rather far in the fourth quadrant, goes back under the curve of zero efficiency, and so forth. Unit 2 remains in a relatively narrow range around the initial operating point. Thus, the machine enters this dangerous zone (reverse pump operation) characterized with strong vibrations, hydraulic unstable vortices, multiphase cavitation, all much more severe than in normal operation [1, 23, 24].

Long variable period of pressure fluctuation indicates that a void in the draft tube is acting as a closed surge tank (air vessel) all the time. The largest period of $T = 9 - 5 = 4$ s is followed with the time interval of runner filling with water $\Delta t = 12 - 7 = 5$ s, and rejoinder of separated water columns at peak pressure. Then a new void is formed, as shown in Fig. 6(a). The Unit 2, continuing operation, as shown in Fig. 6(b), also had a void all the time, with longest period of $T = 18 - 7 = 11$ s for the maximum pressure without a peak; the air in the draft tube cone is compressed but occurred without as sharp pressure peak: rejoinder of separated columns was mild and incomplete. The void visualized in the laboratory at steady operation of the model is shown in Error! Reference source not found.

In this example the designers identified the dangerous phenomenon and the manufacturer verified this instability. The control system was thus altered to prevent two units simultaneous runaway which was the most catastrophic case of concern. The probability that all four protecting devices – two spherical valves and two guide vanes – might fail to close is very small. But some risk is still present.

(a) Unit 1

(b) Unit 2

Fig. 6 Comparison between measurement and computation for “Bajina Basta” pumped storage plant
Fig. 7 Pumped storage plant “Bajina Basta”. Unit 1 load rejection; Unit 2 connected to the grid.

Fig. 8 Vortex core void in the model at steady operation

2.6 Additional Numerical Analysis at “S” Instability

The calculation shown in Figures 6-7 has been repeated and graph constructed with smaller time step to better analyze the unstable “S” zone. A sharp peak at maximum penstock pressure (Fig. 9) shows the instability at the most unstable point of zero instantaneous discharge. This corresponds to the unstable zone in diagrams of Error! Reference source not found.; the single value of $n_1/n_2$ corresponds to three discharge and three torque values. Interestingly, changing the moment of inertia from $1.500 \times 10^6$ kgm$^2$ to $1.502 \times 10^6$ kgm$^2$ (Fig. 10), less than 0.2%, the unstable peak shifts to entirely different shapes. To better understand this, the “S” phenomenon should be analyzed both theoretically and in laboratory. Computer simulation and mathematical analyses in such cases are invariably approximate and uncertain!
Fig. 9 "Bajina Basta" pumped storage plant load rejection. Both penstock pressure and pump-turbine head have unrealistic peaks (a). Magnified head and discharge curves (b) show that this peaks correspond to the zero discharge; discharge just change from generating into pumping direction.

Fig. 10 Changing moment of inertia for less than 0.2% leading to uncertain shapes and instability of pressure peaks

3. Water Column Separation

For a hydro project with long pressurized tunnel, it is particular important to understand the phenomenon of water column separation and subsequent column rejoinder arising from the reverse waterhammer in the draft tube [1, 2, 7, 15, 16, 17, 28, 29, 30, 31, 32, 33, 34, 35]. Water column separation could occur during either steady or transient conditions when low pressure establishes voids in the water column. Specifically, this phenomenon occurs when the local pressure drops below the vapour pressure of water or the partial pressure of dissolved gases (usually air). The dramatic pressure rise that typically accompanies the collapse of such voids (water column rejoinder) can be sufficient to crack the internal linings of conduits and damage both the turbine and other hydraulic components. Perhaps more insidiously, this damage can progress unnoticed, frequently observed only after repeated transient events, thus, creating a ‘time bomb’ of future hydraulic and structural problems. This phenomenon often occurs in the draft tube with long pressured tailrace. However, the phenomenon of draft tube surge has not been fully discussed in English although a few articles are listed in the References. A predominant condition associated with reverse waterhammer is that the runner is effectively dewatered when the flow reverses from the draft tube. Only if the axial component of flow velocity is lower than the allowable value (around 0.5 m/s) will the impact force on the runner blades from the reversing water be small and unlikely to cause damage [25]. In addition, runner lifting should also be checked during reverse waterhammer, that is, the axial

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hydraulic force during the entire transient must be smaller than the weight of the rotating parts.

In order to prevent severe rejoinder of partially separated columns of water, the large vortex void in draft tube should be avoided. As aforementioned, the pressure distribution in the draft tube cone is inherently uneven (due to the helical motion of the flow), with the maximum pressure occurring at the wall and the minimum near the centre (Fig.11). To be more precise, the minimum pressure occurs near the center of the vortex core, a complex rotating tube of water, vapor and air, which usual forms in the draft tube. The vortex core is unstable as a result of the swirl (tangential component of absolute velocity of water outflowing from the turbine runner) induced by the action of the turbine blades [14]. It is important to recall that transducers mounted on the draft tube wall invariably record the greatest cross-sectional pressures, instead of the lowest pressures in the vortex core at the centre of draft tube. In other words, even when the measured pressure is significantly high, the core pressure may well be at vapor pressure (or air release pressure), or perhaps slightly above the vapor pressure if air has been injected into the system [7, 31]. Clearly, then, it is inappropriate to rely on the pressures measured on the draft tube cone to judge whether the minimum pressure exceeds 0.05 MPa. Computational fluid mechanics (CFD) and numerical approaches clearly hold out some promise but they are unfortunately far from being sound solutions to the challenge of measuring and predicting the possible range of draft tube pressures and velocities due to the limited accuracy and numerical stability. The difficulty arises from many sources, including the multi-phase, unsteady, asymmetrical, three-dimensional and oscillatory nature of the flow, and the complex interactions between the flow and the machine unit (see photos in Fig.8 and Error! Reference source not found.11).

Overall, the inherent complexity and significance of these events demand an empirical approach to predict the occurrence of water column separation in draft tubes. It is essential that all pressure data be carefully evaluated. This is particularly true since, we emphasize, no similarity law is applicable to transient and/or two-phase flows [8, 18] while the equations based on similarity still apply to the boundary conditions of the pump-turbine. In practice, the lowest transient pressure in the turbine system is typically calculated based on one-dimensional waterhammer theory and thus represents the “cross-sectional average” value which is lower than the measured one but higher than that in the vortex core (Error! Reference source not found.).

A predominant condition associated with reverse waterhammer is that the runner is effectively dewatered when the reverse flow enters from the draft tube. Only if the axial component of flow velocity is lower than the allowable value (around 0.5 m/s) will the impact force on the runner blades from the reversing water be small and unlikely to cause damage [31]. Another condition should also be checked to prevent the so-called runner-lifting event; that is, the axial hydraulic force during the entire transient must be smaller than the weight of rotating parts. In horizontal machines thrust bearing controls this force.

Any mathematical solution in the zone of instability is highly inaccurate and uncertain; fluctuations in the multiphase unsteady vortex core further complicate the flow conditions, where model test data cannot be accurately transferred to the prototype.

![Vortex at turbine partial load](image1)

Fig.11 Pressure distribution at cross-section (A-A) of draft tube and several relevant terminologies/values. When turbine wicket gates close the inertia of water in tailrace tunnel pulls water out from the turbine and the void in the draft tube increases.

Photos: vortex core in the draft tube of the model at steady operation.

![Vortex at turbine overload](image2)
3.1 Case Study: Reverse Waterhammer in Kaplan Turbine

To prevent reverse waterhammer, it is necessary to avoid dewatering of the runner during guide-vane closure. As is known, it is impossible to prevent the emergence of absolute vacuum conditions under the turbine headcover (particularly in the case of high head Kaplan turbines, even if the closure time is very long). The condition in this case, as already discussed, is that the mean velocity of water outflow under the runner should be lower than a certain limiting small value [18, 19].

If measured data for the pressure distribution at the turbine head cover ($H_C$) is not available, the mean pressure drop through the runner can be calculated from the axial hydraulic thrust $F_a$. The condition which prevents the runner from dewatering is $H_I > H_{ip} + H_{rc} = 2$ to 3 mWC, here $H_{ip}$ is the vapor pressure and $H_{rc}$ takes into account pressure pulsations and calculation accuracy. If this condition is not satisfied, the mean maximum axial outflow velocity, $C_m$, should meet the following criterion, $C_m < C_r = 0.5 m/s$, which means that when the water flow reverses from the draft tube and strikes against the rotating runner blades, provided the water flow reverses at the same velocity with which it leaves the runner, the impact force will be small and not cause damage. The values of $H_{rc}$ and $C_r$ should be corrected on the basis of practical experience gained in turbine operation.

An additional condition, the time during which the runner remains waterless, is of equal importance because both the cavity size and the speed with which the water returning from the draft tube strikes at the runner depend on this duration. This condition is especially suitable for analyzing oscillograms recorded when examining transient processes in waterpower plants. If the duration of absolute vacuum is shorter than one second, it is considered that there is no danger of waterhammer accident. These short pressure drops are caused by pressure fluctuations and oscillations in the generating unit [18, 19].

Calculations for the above-described turbine which experienced an accident (Fig.12) are presented in Fig.13. The lines of guide-vane opening $A$, head $H$, discharge $Q$, and speed $n$, are obtained by calculating the unit quick closure. The line $F_n$ shows the hydraulic thrust variation as obtained by calculation from model test data. The mean pressure on the turbine headcover $H_C$ can be obtained from the pressure change in the draft tube and the hydraulic thrust variation. According to the adopted criterion, water column separation occurs in the eighth second (Fig.13 (a)). The water outflow speed is $C = 1.6 m/s > C_{rez} = 0.5 m/s$, and thus there is a risk of an accident. In the case of two-speed of turbine guide vanes closure (Fig.13 (b)) with the total closure time 14.5 s, the pressure $H_I$ drops below its limiting value, $H_I = -0.9 mWC < H_{ip} + H_{rc} = 2$ to 3 mWC. But $C_m = 0.3 m/s < C_r = 0.5 m/s$. Consequently, there is no danger of an accident.

During the past years, incidents due to quick closure have occurred in some Kaplan turbines and the consequent waterhammer resulted in great damages. The Faculty of Mechanical Engineering, University of Belgrade, and Association of Power Industry, Belgrade, then jointly undertook a study on the reverse waterhammer in hydraulic turbines (managed by S. Pejovic), the key results are summarized in a series of articles [7Error! Reference source not found., 17, 25, 31, 32]. The investigators also describe an accident that had occurred at a 22 MW turbine-generator unit (Fig.14) when it was operating at night with an output of 8 MW. The failure of a governor caused the guide-vanes to suddenly open leading to a rapid escalation in power output. Power oscillations between 20 and 25 MW were recorded by monitoring instruments and load rumbles were heard from the turbine room. In response, the unit was shut down by pressing the emergency shut-off button which disconnected the generator from the network and the unit overspeed control device went into operation. When the machine was to stop, a banging noise was heard from the turbine and water was observed leaking out through the turbine head cover. Subsequently, one of the runner blades was found to be broken at the root (Fig.1). The fracture line traversed the blade obliquely from the hub towards the inlet edge. A smaller part of the blade root and inlet section remained attached to the runner hub, while the larger part was broken off. The fracture revealed no cracks or casting defects and no signs of material fatigue. It was clear that the break had started on the lower part of the blade.

Another power plant equipped with identical turbine runners experienced a similar catastrophic accident one year later.
Fig. 12 Quick closure from power output of 16 MW. Turbine closure time is 4 s; \( a_0 \) (percent) = guide vane opening; \( \beta_0 \) = runner blade inclination, \( n_0 \) = rotational speed; \( h_{sp} \) (mWC) = pressure in spiral casing; and \( h_1 \) and \( h_2 \) = pressure on turbine head cover. It is important to notice that the pressure on the turbine head cover (\( h_1 \) and \( h_2 \)) becomes an absolute vacuum at the end of the servomotor stroke. This means that an air-vapor cavity is formed above the runner.

Fig. 13 Calculation of unit quick closure; \( A \) = guide vane opening; \( H \) = head; \( Q \) = discharge; \( n \) = rotational speed; \( H_k \) = mean pressure on turbine head cover; \( F_a \) = hydraulic thrust; \( C_m \) = meridional velocity of water at runner outlet.
3.2 Case Study: Turbine with Long Tailrace Tunnel

In this case, a power plant has eight 250 MW units with each pair of units sharing a common penstock and 500 m tailrace tunnel (Fig.15). The units are submerged 13 m below the tailwater to prevent cavitation. The simultaneous load rejection of two units would cause water column separation and subsequent rejoinder in the draft tube, an event that could result in serious damage to both machinery and the water conveyance system. Air admission was proposed to dampen the surge, but it is difficult to keep it operational at all times [4, 27, 28]. The proper amount of air injection is able to protect system from normal operational transients [1, 2, 2624] but not from full runaway conditions if too little air is pumped into the draft tube from the beginning to the end of the waterhammer transitions. Although the probability of runaway is low, the associated damages could be enormous with unpredictable catastrophic consequences. Therefore, the risks and potential costs of acceptable damage at full runaway condition should be carefully scrutinized. A proper submergence at no additional cost could have been done at the early design stage and during project reviews [23, 24]. The risk of draft tube water column separation was pointed out at an early stage [18, 19], but it was then too late to increase submergence as the plant was already under construction.

Upon commissioning of the plant, the hydraulic transient in the draft tube during load rejection had been excessive, as already described and calculated. It was apparent that the guide vane closing had resulted in water column separation during load rejection with only 75% partial load at one unit (Fig.16). The field tests confirmed that the partial water column separation could take place even if the measured minimum pressure at draft tube cone is above vapor pressure.

Significant effort was made, based on the water hammer calculations associated with site load rejections, to search for an improved break point and closing rates of wicket gate closure law. In fact, a severe surge during an emergency shutdown of one unit implied that the trial-and-error method of breakpoint adjustment was too dangerous, and a better solution was needed. Thus, a Panel of Experts was appointed to identify the origins of the pressure surges and to offer suggestions to solve the problem [4, 5].
3.3 Literature on the topic

As far as we know, the publications discussing water column separation in the turbine draft tubes is scarce, with little in English and only a small amount in other languages (Krivtchenko 1975; Pejovic 1977). Methods for evaluating draft tube water column separation are generally based on the one-dimensional waterhammer calculation of the minimum pressure below the turbine runner and on empirical data deriving from models and prototypes. By employing these approaches, the size of the void in the vortex core can be predicted. To prevent reverse waterhammer it is necessary to avoid dewatering of the runner during guide-vane closure because otherwise the resulting vacuum condition proves too severe. To this effect, the maximum pressure drops are not allowed to descend below a permissible positive value. The criterion introduced to ensure this can be achieved in a penstock when water flow is free of tangential component of velocity, but it is impossible in the draft tube. Therefore additional criterion is that the mean velocity of water outflow below the runner should be lower than 0.5 m/s as aforementioned.

This and many other cases of hydraulic transients and oscillations have shown that this is a serious problem which should never be overlooked during plant design, construction and operation. Therefore, the design must contain a complete analysis, covering the whole operating range of the plant. Natural frequencies of the system should be computed and compared with frequencies of the expected excitation forces. The instability and resonance must be avoided in any case. For existing plants and installations, this analysis should be also performed especially if some difficulties were noticed in a normal operation. The analysis should be performed by mathematical modeling and confirmed by field measurements and observations. As not all the problems have been resolved, close collaboration and frequent exchange of the ideas and results is still required.

4. Conclusions

Since the waterhammer calculation is usually one-dimensional and the results represent an “average” values of pressure heads; the variations in pressure distribution must be added to the calculated pressure in the draft tube. A required minimum submergence of turbine installation as a function of rotational speed of runner is established to control the water hammer pressure drop under a negligible level which does not lead to an excessive pressure rise to cause the damage of the unit and waterways; the runner overspeed must also be taken into consideration, since a very small pressure drop could further increase a void (a water column separation) under higher speed and the following rejoining creates forces that would damage turbine runners and non-rotating parts. Even more, if submergence is near the limit, a void formed by centrifugal force of rotating water is great enough and the pressure rise caused by very slow acceleration/acceleration of water in the tailrace tunnel could still be dangerous. Under this situation, additional transient analysis must be performed theoretically and confirmed experimentally. Finally when flow change direction from turbine direction to pump direction and back in turbine direction in short time period (here in 3 seconds) additional careful analysis is prudent; in addition, the full runaway condition must be simulated and analyzed as an Abnormal Condition [1, 23, 24].

Mathematical modeling has been an efficient “tool” in analyzing hydraulic transients, stability and resonance in hydroelectric plants and other hydraulic systems; in this case the “Bajina Basta” pumped storage hydroelectric plant. The pump-turbine characteristics, “S”-shaped, based upon model tests in “Toshiba” laboratory, were used for the analysis. The results of computation were confirmed by field-tests.

Points of interest for practice:
- similarity laws when applied to pump-turbines with high prototype/model ratio lead to under-estimation of actual pressure surges; therefore calibration procedure can be useful;

Fig.16 A severe surges during an emergency shutdown with one unit. This test showed that waterhammer was too dangerous to continue with commissioning tests. A satisfactory damping had to be found.
the similarity applicable to transients is untruth (inaccurate), therefore all data must be carefully evaluated as really there is no similarity between transient and/or two-phase flows.

- pump-turbines with “S”-shaped characteristics enter the fourth-quadrant (reverse pump operation) regularly after a trip-out from turbine operation, even if the wicket-gates are closed down rapidly as designed to prevent high speed rise and minimize time spent in unstable zone or operation decreasing resonance effects.

It will be interesting to monitor operational records of these machines with one key question in mind: will cavitation and hydraulic vibration cause more trouble than usual or not?

Those who design and construct complex systems have to face many challenges. Even routine issues such as the trade-off between capital and operating costs invariably involve an assessment of events that might happen in the future, a realm of great uncertainty. Although this paper discusses specific cases, the goal is not to cast blame, assign fault, or imply that the related decision-making is easy. In almost all such cases, only hindsight is sharp and clear. However, these issues have arisen before, and will continue to arise, until the associated challenges are brought more consciously into the open where they can be discussed and debated. The over-riding duty of engineers is to act in the best interest of both clients and the public [9].

Certainly the design and operation of any power system requires a delicate balance between certain competitive objectives. One crucial issue is related to the phenomenon of water column separation in tailrace tunnel, particularly following a transient turbine operation (e.g., the load rejection, emergency closure, and runaway).

Since water hammer calculations are usually based on one-dimensional models, the results represent average values of pressure heads and, thus, the variation of pressure over the draft tube cross-section must be carefully considered. Moreover, the speed increment of the runner, particularly in runaway condition, is crucial since the voids formed by the centrifugal force of the high speed rotating water can be large; the pressure rise caused by even slow accelerations/decelerations of the tailrace water could have a strong influence on void collapse. Therefore, transient analysis must be thoroughly performed and confirmed experimentally. A larger safety margin is prudent for the draft tube coefficient, particularly given the complex and difficult-to-model nature of these transient flows.

Air admission is way of controlling water column separation. Field tests show that the injection of compressed air reduces the severity of transients during load rejection, even for insufficiently submerged turbines. Thus, air injection is useful in underground power plants where the potential to increase the power output is constrained by the excessive negative surges in draft tubes. However, being a reactive and real time safety concept, it relies on the controller’s dexterity and skill, and it must be carefully designed tested and maintained. It also does not rectify shortcomings of inadequate submergence or defects in other design criteria, which are, in part, revealed through thoughtful case studies of actual events in operational facilities.

### Nomenclature

- $A$: amplitude of pressure pulsations [m]
- $A_{aw}$: average amplitude of pressure fluctuation in draft tube
- $a$: guide vane opening
- $a_0$: guide vane opening
- $a_0/a_{max}$: relative guide vanes openings
- $c_n$: tangential component of flow velocity
- $c_a$: tangential component of flow velocity
- $D$: draft tube tailrace diameter
- $e$: relative energy measured from model tests
- $g$: gravity acceleration
- $h$: turbine net head
- $h_{p2}$: pressure on turbine head cover
- $h_{ap}$: atmospheric pressure
- $h_{PR}$: pressure at runner outlet, pressure measured at draft tube cone
- $h_s$: suction head
- $h_{sp}$: pressure in spiral casing (mWC)
- $k$: empirical coefficient
- $k$: pressure distribution factor in the cross-section of draft tube cone
- $L_a$: length of tailrace tunnel [m]
- $n$: rotational speed of turbine unit
- $n_0$: unit speed
- $n_t$: rated speed of turbine unit
- $v_0$: flow velocity in the tailrace tunnel
- $v_o$: flow velocity at the runner outlet
- $T_c$: closing time of guide vanes [s]
- $\sigma_{w,av}$: average amplitude of pressure fluctuation [m]
- $\phi$: runner blade inclination
- $\beta$: plant cavitation coefficient
- $\sigma_r$: turbine dimensionless head rise
- $\xi$: dimensionless speed rise
- $\Delta h$: waterhammer pressure rise in the draft tube (pressure drop is negative)

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