Steady and transient regimes in hydropower plants

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Abstract. Hydropower plant that has been in operation for about 30 years has to be reconstructed. They have already installed 12 Kaplan turbines, the largest in the world at that time. The existing CAM relationship was determined based on hydraulic model tests and checked by efficiency on-site tests. It was also tested based on turbine bearing vibrations. In order to discover vibrations and long cracks on stay vanes detailed on-site measurements were performed. Influence of the modification of the trailing edges on the dynamic stresses of the stay vanes is also shown. In order to improve power output transient regimes were analyzed, both experimentally and numerically.

Reversible hydropower plant, a pioneer in Europe since it was the first Pump storage power plant constructed with the highest head pump-turbines in the world. Analyses of transient regimes discover some problems with S-shaped characteristics coupled with non-symmetrical penstock.

1. Introduction
River-off hydropower plant that has been in operation for about 30 years has to be reconstructed. The existing CAM relationship was determined based on hydraulic model tests, with very high scale effects. Because of that, serious field tests were performed, with problems in the flow measurements in range 100 m$^3$/s to 840 m$^3$/s and in huge cross-section areas, the turbine power output measurements, the variation of the turbine net head due to electricity consumption limitations, etc. High measurement accuracy and repetition of measurement results were obtained. The CAM relation was also tested by measuring turbine bearing vibrations.

Measuring of stresses in the upper and in the lower zone along the stay ring are completed in the purpose of discovering causes of high vibrations and long cracks on turbine stay vanes. Investigations performed in order to find out the excitation of vibrations, included measurements of the pressure along the trailing edge and stresses on stay vanes. The main frequency was about 36 Hz, which is very close to the natural frequencies of some stay vanes. It was concluded that there were two excitations, one belonging to Karman vortexes and the other to attack flow of the guide vanes, both with very close frequencies, causing beating oscillations. Modification of the stay vanes was made by grinding of the convex part of the trailing edges until half-circular concave of the same radius of 25 mm was reached. Influence of the modification on the static and dynamic stresses of the stay vanes, as well as their frequencies of oscillations, is shown in the paper.

Numerical and experimental investigations were performed in order to prevent reverse water hammer. Confirmed numerical programme was used for the analyses of normal transient regimes, but also for emergency and catastrophic ones.
The lowest specific-speed pump-turbines $n_p=27$ installed in Reversible Hydropower Plant have an unusual characteristic in turbine operating mode. The results of computation which were confirmed by field tests explained influence of S-shaped characteristics on hydraulic transients.

2. Steady regimes of Kaplan turbines
Prior to turbine manufacture, CAM relation was determined by hydraulic model tests carried out in the laboratory of turbine supplier LMZ 1968/69 [1]. Main data of installed Kaplan turbines is shown in Fig. 1.

Due to a huge turbine size (runner diameter is 9.5 m) and significant discharge capacity, resulting from high specific speed, model dimensions had to be reduced, so that a runner blade model diameter was only 460 mm, head being 3 m, as a result length scale ratio was $\lambda_L=20.65$. According to thus obtained data a three-dimensional crankshaft had positioned for adjusting CAM relation, depending on head and required power output. Field tests, as stated in the heading, were carried out on Iron Gate I hydropower plant over several past years, [2, 3, 4]. Index methods were used for flow measurements on Unit 1 and Unit 3. The influence on nonsymmetrical inflow on net head was taken into account [4, 5].

To determine the curves for turbine efficiency ($\eta$) dependence of discharge ($Q$) and a corresponding combination of guide vanes opening ($a$) and runner blades angle ($\phi$) at the existing head, first a number of operating regimes with a retained CAM relation were registered [2,3]. The CAM relation was broken-off at the second stage of tests, and a series of propeller operating regimes with constant runner blade inclinations ($\phi = -10^\circ, -5^\circ, 0^\circ, 10^\circ, 15^\circ, 17^\circ$) were tested. On the basis of data analyzed, efficiency curves were drawn, depending on discharge for each $\phi$, i.e. curves $Q-\eta$ for $\phi =$const., see Fig. 2.

By drawing envelope curves around efficiency curves, optimal efficiency values were determined. Vertical lines, drawn through envelope curve contact points with propeller curves to the section with guide vanes inclination change curves for different runner blade angles $\phi$, determine optimal CAM characteristic. A curve was drawn through points thus obtained, defining optimal combination; see lower part in Fig. 2. [4, 5].
Figure 2. Efficiencies at broken-off combination with envelope curve defining optimal values (Unit 3) \( H_n = 27.8 \text{m} \)

Figure 3. CAM relation test by turbine bearing vibrations and their harmonics

In addition, the CAM relation on unit 3 was also tested based on turbine bearing vibrations in two radial directions: dam direction (x - axis) and in flow direction (y - axis). Only radial vibrations of
turbine bearing in flow direction are presented here. It is evident from Fig. 3 that vibration minima on propeller regimes do not correspond to combination regimes. Exception occurs at runner blade angle $\varphi = 5^\circ$ (optimal angle).

Expected vibration minimal match with corresponding optimal combination of guide vanes opening and runner blades angle is not completely fulfilled. Deflection for certain regimes can be seen in Fig. 3 ($\varphi = -5^\circ, 0^\circ, 15^\circ$). This appearance can be explained by taking into account vibrations source. Beside excitation that originated in runner itself, vibrations can be the result of other excitations in system (mechanical, electrical, etc.). Measured radial vibrations of turbine bearing are the result of superposition of few different excitations in system.

Amplitude of main harmonics related to discharge for certain propeller regimes are shown in the lower part of Fig. 3. It can be seen that harmonics minimal for different hydraulic excitations mainly correspond to the appropriate regime optima. Frequencies of the first, the second and the third harmonic, for $z=6$ runner blades and rotational speed $\omega=1.2\ \text{s}^{-1}$ are: $f_1=7.2\ \text{Hz}, f_2=14.4\ \text{Hz}, f_3=21.6\ \text{Hz}$.

Field tests on unit 1 were conducted in 1997 [2] (net head $H_n=27.4\ \text{m}$), and on unit 3 in 2003 [3] (net head $H_n=27.8\ \text{m}$).

In Ref. [5] is shown that unit 3 efficiency is cca 1% higher than efficiency of unit 1. It should be emphasized that unit 1 is placed on right river bank where the water inflow is inconvenient. Also discharge measurement accuracy (index method) is relatively low to make general conclusions. Tests of both units showed insufficient reliability of Winter-Kennedy inertia method for discharge measurement in wide range of its values, especially for low discharges, that is, for low runner blade inclination.

2. Modification of the trailing edges of the stay vanes

After the increase of the power and the flow through the turbines, the phenomenon of the cracks on the stay vanes was detected. The cracks were first noticed in the 1983. After the repairs which followed, the cracks were recorded on all 6 turbines, but not all vanes were equally attacked, which is the consequence of the difference in the dimensions, shapes and positions in the turbine. The height of the vanes is 3580 mm, see Fig. 5.

The first analyses were made at the end of 1986 on the vanes No. 3, No. 4, No. 5 and No. 9. It was concluded that the resonance in the field of the frequency from 36 to 38 Hz was the possible cause of the cracks. The two dominant frequencies of the dynamic pressures were - the first around 7 Hz, and the second was between 36 and 38 Hz. The frequency between 36 and 38 Hz was especially interesting, because the same was very close to the frequency of the conductive machine.
The new program of the analyses was made in 1989 [6,7], and it included the research of the natural frequencies of the vanes in the water and in the air. Excitation loads were performed by special pneumatic impact hammer. The static and dynamic condition of the stay vanes of the turbine and the pulsation of the pressures on the contour of the vanes were measured. Fig. 6 shows the pressure oscillations related to the discharge. The measured natural frequencies of the vanes in the air (NFₐ) and in the water (NFₕ) and Strouhal number of Karman vortexes on stay vane No. 5 is shown in Fig. 7.

Figure 5. Disposition of the spiral case and the stay vanes

Figure 6. Pressure oscillations related to discharge

Figure 7. Measured natural frequencies and Sh-number
Spectral analysis (PSD) of the measured data shows that the main harmonics are 36 Hz, 21 Hz, 7 Hz etc. and that they correspond to frequency of rotation, of runner blades, of the turbine stay vanes, of the turbine guide vanes, but also of Karman vortices at the trailing edge [9]. Distribution of the pressure amplitudes harmonics of 21 Hz and of 36 Hz are presented in Fig. 8.

The results clearly showed that there were conditions for the phenomenon of the resonance in the field of the larger power with the frequency between 36 and 38 Hz. Due to the economic situation, nothing was done for solving this problem. The cracks were repaired from the one to the other repair, occasionally, when they appeared.

With the new growth of the power output after the 1997, the action of finding the solutions for the stay vanes was restarted, now also in the light of the reconstruction. The new program of the analysis was set off on the Unit 1 in 1999, referred only to the static and dynamic stresses of the vanes, at the increased maximum power output of 200 MW.

The measurements were made on the vanes No. 2 and No.12. The results have completely confirmed the previous conclusions and secured the reference values for the evaluation of the solution which will be applied. Turbine manufacturer decided to reconstruct the shape of the stay vanes, by changing the trailing edges, according to their own experiences applied several times from 1963 until nowadays [10].

Figure 8. Distribution of the amplitudes of the harmonics 21 Hz and 36 Hz along the stay vane No. 5

Figure 9. Modification of the vane’s trailing edges from the convex into the concave shape
The adopted solution, shown in Fig. 9., reminds on the tail of the swallow, and was made by grinding the half-circle of the radius of 25 mm on the trailing edge and in the vane the half-circle with the radius of 25 mm was concaved.

2.1 Analyses of the modified stay vanes

After the modification of the trailing edges of the stay vanes the analyses were made with nearly the same steady regimes to 200 MW, as before, but also above this level, even to the power output of 208 MW. The flow was about 900 m$^3$/s, which compared to the designed flow represents the growth of around 30 %. According to the results of the measurements of the dynamic stresses, there are indications that the frequency of the Karman vortexes was changed on the values over 40 Hz, with the relatively small amplitudes, as shown in Fig.10. It is possible that the power of these vortexes wasn't larger, or much larger, before the modification, but now is changed beyond the boundaries of the natural frequencies of the stay vanes, and there is no resonance.

Regarding the static component of the stress, as it was shown in Ref [10], there were no changes, if we compare the data for 200 MW we will see that they are the same. This, obviously, wasn't expected, but it confirmed to the analysts and to the customer that the measurements were correct.

The modification made on the trailing edges of the stay vanes of the turbine brought the change in the conditions of the flow on the trailing edges and the decrease of the intensity of the dynamic component of the stresses. The previous peak at the power output between 160 and 190 MW didn't show up, which at the power to 205 MW and at the flow to 840 m$^3$/s secured the smooth working of the units and the larger resistance of the stay vanes on the material fatigue.

(a) Turbine 1, stay vane 12 $\phi=+10^\circ$
3. Transient regimes of Kaplan turbines

Two big accidents had occurred in Yugoslav hydropower plants in the period of 1975-77. The first one in HPP with Kaplan turbine rated data are: power output 22.4 MW, nominal head of 19.3 m, discharge 136 m³/s, runner diameter 4.65 m. The unit was closed by pressing the turbine emergency shut-off bottom and generator was disconnected from the network. When the machine came to stop, a banging noise was heard from the turbine and water was leaking out through the turbine head cover. One of the runner blades was broken at the root. More details are presented in ref. [11, 12, 13]. At that moment only a few data were published about reverse waterhammer [14].

The development of waterhammer theory has started by using the solid body type analysis called "Rigid pipe - incompressible fluid" theory. In this type of analysis it is assumed that pressure surges are not violent, therefore the deformations of the pipes are negligible, and the water may be regarded as non-compressible fluid. This is the case of Kaplan and bulb turbines, when the time for a pressure and flow change is shorter than the pipeline reflection time 2 l/a. The analysis in those cases is usually based upon the rigid waterhammer [14, 15, 16]. Our mathematical model which includes pipe and turbine characteristics and governing system is described in [17, 18, 19, 20]. Numerical programme has been confirmed by on-site measurements, and then used for various low head HPP.

Beside the fact that the numerical program was already tested and confirmed during the several decades of use, it is necessary to start every new analysis with a verification of input data: characteristic of the flow system, turbines, governors, gates and valves and other hydro mechanical equipment. The best method for that verification is a comparison with the field tests. The reasons for that are: use of turbine characteristics obtained by model tests in stationary regimes, unsatisfactory reliability in small openings zone, insufficient data about the plant, etc.
Results computed in a study described here, for HPP presented in Fig. 1, were compared with those obtained by on-site measurements. The analyses of the possibility of RWH occurrence were performed according to the mean pressure method, because the model tests included axial hydraulic thrust measurements. Very good agreement is received for all physical values analyzed.

Load rejection from power output of 162 MW to the idle run is presented in Fig. 11.a. Experimental results are given by the solid lines, and computed results by dotted lines. Notations are: \( Y(\text{mm}) \) - guide vanes servomotor stroke, \( \beta(\circ) \) - runner blade inclination, \( n(\%) \) - runner speed, \( p_{sp}(\text{m}) \) - spiral casing pressure, \( p_{dt}(\text{m}) \) - draft tube pressure, \( F_a(\text{kN}) \) - runner axial thrust. The turbine quick closure and stoppage are presented in Fig. 11.b. The agreement of the experimental and computed results is obvious.

The analysis of the various transient regimes can be done using the programs described in the paper. For instance, manual stoppage from the runaway and the optimization of the runner blade and guide vanes closure modes and other emergency cases which have not been tested on site, and which were, consequently, analyzed mathematically. Study of possible power output increase at the hydropower plant requires similar analysis for all normal and emergency transient regimes which can occur in future operation. Reverse waterhammer may be prevented by adjusting the time of turbine closure. The comprehensive and expensive investigations must be conducted when governing conditions require a minimal closure time. The pressure distribution on the turbine head cover and the hydraulic thrust should be measured on the model, under steady operating conditions. It must be noted that in the case of cavitation, air introduction, or any kind of two phase flow, there is no similarity between the model and prototype. The results should be critically analyzed.

4. Transient regimes of reversible hydropower plant

High-head reversible pump-turbines with low-specific-speed \( n_q = 27 \) have an unusual characteristic in turbine operating mode; beyond the runaway zone any decrease in speed reduces the discharge. In the
\( n_1 \cdot q_1 \) and \( n_1 \cdot m_1 \) diagrams partial load curves for a pump-turbine are markedly "S" shaped, as shown in Fig.12. 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 guide vanes. The net discharge might be either negative (turbine) or positive (pump). These operating modes are, naturally accompanied with violent vibrations and highly developed cavitations [21, 22]. The numerical simulations exhibited intensive instability and resonance, though the accuracy of analysis is reasonably low. 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. But when using the Suter curves, difficulties may arise from an uneven distribution of curves at the guide vanes opening. At small openings, the curves space themselves out further, and for a definite angle when the guide vanes are 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 [23].

![Diagram](image)

**Figure 12.** Reversible HPP “B. Basta” with pump-turbine model characteristics

### 4.1 Computer program and mathematical model

Assume that the transient phenomenon begins from point A in Fig.12, a normal turbine operation regime, and the gate opening is not changing, \( n_1 \) goes down to point B, than the working point jumps from B to point C. Then, at this moment \( m_1 < 0 \) it goes up to point D. At point D we have \( n_1 > n_1 C \) the working point jumps again from point D to point E. [24]. The process will be repeated passing through the points EBCDE and will never stop. Thus, all type of operations shown by the four-quadrant characteristics are 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.

The computer programs were based on elastic water hammer theory and Method of Characteristics (MOC), and later improved to be able to handle any system configuration, simulate water hammer and hydraulic oscillations [25,26]. The second one is based on linear mathematical model, solved by Transfer Matrix Method (TMM). A realistic application case of this computer program is for pumped-storage hydroelectric plant shown in Fig.12. This plant is equipped with two 300 MW units, sharing a 8000 m long tunnel, a common 1200 m penstock, and a 300m tailrace tunnel. It has to be noted that the upper brunch is asymmetrical, left pipe is 12 m shorter than the right one. 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 628 m.

Field tests have been carried out several times both by the manufacturer and by the asset owner. Several situations were tested, and various physical variables were measured and recorded. One transient regime was:
- both Units 1 and 2 operated as turbines, developing 298 MW each;
- then, Unit 1 dropped out and its guide vanes closed down rapidly, while 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 guide vanes servo strokes, generators’ currents and rotating speeds of the shafts. The discharge could not be measured.

Fig. 13.a displays the changes in Unit 1 which has dropped out. The guide vanes 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. 13.b shows pressure variations in Unit 2 which remained connected to the system with $\omega \approx$ constant and guide vanes blocked in the open position. The other lines are the same as in fig. 13.a. including power 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 satisfying. 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.14 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 the strong vibrations, hydraulic unstable vortices, multiphase cavitation, all much more severe than in the normal operation [24, 25].

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. 13(a).The Unit 2, continuing operation, as shown in Fig. 13(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 Ref. [24].

Different hydraulic transient regimes may be analyzed in time domain, usually by numerical programs based on Methods of Characteristics, or in frequency domain, based on Transfer Matrix Method. Resonance in hydraulic systems, fluid structure interaction, earthquake effects on closed conduits, auto-oscillations of the valves and gates, effects of air supply to the vibrations and oscillations are computed and measured [37, 38, 39, 40, 41, 42, 43, 44, and 45].
Figure 13 - Comparison between measurement and computation results

Figure 14 - Unit 1 load rejection; Unit 2 connected to the grid.

5. Conclusion
In order to improve the power output and turbine efficiency CAM characteristics may be tested by on-site measurements. Main task in reconstruction of hydropower plants is usually to increase the turbine discharge, which may introduce some additional problems, like Karman vortexes, draft tube vortex core, vibrations etc. On site measurements are very useful in finding sources and finding final solutions in revitalization or reconstruction of hydro mechanical parts in power plants.

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.

Since the water hammer results represent an “average” value 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. The runner overspeed must also be taken into account, 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/deceleration of water in the tailrace tunnel could still be dangerous. In this situation, additional transient analysis must be performed theoretically and confirmed experimentally.

Mathematical modelling has been an efficient "tool" in analyzing hydraulic transients, stability and resonance in hydroelectric plants and other hydraulic systems. The pump-turbine characteristics, (“S”-shaped based upon model tests), were used for the analysis. The results of computation were confirmed by the 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;
- the similarity applicable to transients is 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.

Finally when the flow change direction from turbine direction to pump direction and back in turbine direction in short time period additional careful analysis is prudent; in addition, the full runaway condition must be simulated and analyzed as an Abnormal Condition.

Certainly, design and operation of any power system require 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).
Nomenclature

\begin{itemize}
\item \( A \) : cross-section area
\item \( a_o \) : guide vane opening
\item \( c \) : flow velocity
\item \( D \) : diameter
\item \( Fa \) : axial hydraulic thrust
\item \( f \) : frequency
\item \( g \) : gravity acceleration
\item \( h \) : turbine net head
\item \( h_{at} \) : atmospheric pressure
\item \( h_{DT} \) : pressure measured at draft tube cone
\item \( h_s \) : suction head
\item \( h_{sp} \) : pressure in spiral casing
\item \( h_{vp} \) : vapour pressure
\item \( L \) : length
\item \( n \) : rotational speed of turbine unit
\item \( n_q \) : specific speed
\item \( n_1' \) : unit speed
\item \( q_1' \) : unit discharge
\item \( Q \) : discharge, flow
\item \( z \) : elevation
\item \( \eta \) : efficiency
\item \( \sigma \) : stress
\item \( \sigma_{plant} \) : plant cavitation coefficient
\item \( \phi, \beta \) : runner blades inclination
\item \( \Delta h \) : turbine head rise due to water hammer
\item \( W_H \) : head characteristic
\item \( W_B \) : torque characteristic
\end{itemize}

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References
[1] LMZ 1968/69 Turbine model acceptance tests for hpp iron gate i, in Russian, Leningrad, Ed. I-Energy tests, Ed. II – Cavitation tests.
[2] Gajic A, Krzmanovic L, Makivic Z, Kapor R and Predic Z 1999 Complex field tests of hpp iron gate i – Unit No. 1, Vol. 3. – Energy Characteristics, in Serbian, Institute "J. Cerni" (Rep. 01-910) (Belgrade: Faculty of Mechanical Engineering) (Rep. 06.10-01-99)
[3] Gajic A, Krzmanovic L, Komarov D, Stojkovic D, Ivljanin B, Djindo E and Predic Z 2004 Field tests of hpp iron gate i – Unit No. 3, Vol. 3. – Energy Characteristics(Belgrade: Faculty of Mechanical Engineering) (Rep. 06.10-01/04),
[4] Gajic A., Krzmanovic L, Ignjatovic B, Predic Z, Cusic M and Makivic Z. 2000 Control of CAM characteristics of the kaplan turbine by on-site measurements. Hydraulic Machinery and Systems 20th IAHR Symposium (Charlotte, USA, 6-9 August 2000)
[5] Gajic A, Komarov D, Stojkovic D, Ignjatovic B, Daskalovic L and Predic Z 2004 CAM characteristics of the kaplan turbine determined by efficiency and bearing vibrations 6th Int. Conference on Hydraulic Machinery and Hydrodynamics(Timișoara, Romania, 21-22 October 2004)
[6] Muškatirović J and Predić Z 1989 Analysis of hydrodynamic pressures acting on stay vanes of kaplan turbines, Proc. International Symposium on Large Hydraulic Machinery & Associated Equipments(Beijing, China, 28-31 May 1989)
[7] Gajić A, Predić Z, Muskatirović J, Pejović S and Manzalović P 1990 Investigations of cracks on the kaplan turbine stay vanes 15th IAHR Symposium(Belgrade, Yugoslavia, 11-14 September 1990)
[8] Pulpitel L and Robert P 1990 Discussion on [7] Proc. 15th IAHR Symposium(Belgrade, Yugoslavia, 11-14 September 1990)
[9] Gajić A, Oba R, Ikohagi T, Ignjatović B and Muškirović J 1994 Flow induced vibrations and cracks on stay vanes of a large hydraulic turbine 17th IAHR Symposium(Beijing, China, 15-19 September 1994)
[10] Gajic A, Manzalovic P and Predic Z 2010 Modification of the large stay vanes and their influence on dynamic stresses 25th IAHR Symposium(Timisoara, Romania, 20-24 September 2010)

[11] Pejovic S, Krmanovic L and Gajic A 1978 Reverse Waterhammer and accident in hydro power plant “zvornik” (in Serbian), Faculty of Mechanical Engineering, Belgrade, pp. 90.

[12] Pejovic S, Krmanovic L, Gajic A and Obradovic D 1980 Water Power and Dam Construction 1980 36-40.

[13] Pejovic S, Gajic A and Obradovic D 1980 Reverse water hammer in kaplan turbines, 10th IAHR Symposium on Hydraulic Machinery and Cavitation(Tokyo, Japan, 1980) pp 489-499

[14] Krivchenko G I, Arshenevsky N N, Kvyatkovskaya E V and Klabukov V M 1975 Hydraulic transients in hydroelectric power plants (in Russian), Energy, Moscow.

[15] Liu S Z 1987 Experimental research on the lifting-rotor characteristics of axial-flow turbine Proc. 2nd China-Japan Conf. on Fluid Machinery pp. 239-248.

[16] Duan C G 1985 Kexue Tongbao 30(7) 976-80.

[17] Gajic A, Pejovic S, Arnautovic D and Ignjatovic B 1992 Reverse waterhammer analysis in kaplan turbines, Proc. 16th IAHR Symposium(Sao Paulo, Brazil, 14-18 September, 1992) pp 161-172.

[18] Gajic A 1993 Kaplan turbine incidents due to reverse waterhammer and mathematical model confirmed by the field tests Proc. Int. Symposium on Aerospace and Fluid Science(Sendai, Japan, 14-16 November 1993) pp. 741-766.

[19] Gajic A, Krmanovic L, Ignjatovic B, Predic Z 2000 Transient regimes of the huge kaplan turbine measured in situ Proc. 2nd Int. Symposium on Fluid Machinery and Fluid Engineering(Beijing) 1 pp. 480-486.

[20] Gajic A, Pejovic S and Ivjlanin B 2003 Reverse waterhammer- case studies, (Inv. Paper) Proc. Int. Conference on Case Studies in Hydraulic Systems- CSHS-03, (Belgrade, Serbia,2003) ISBN 86-7083-469-3 pp 89-104.

[21] Pejovic S, Krmananovic L, Jemcov R, Crnkovic P 1976 Unstable operation of high-head reversible pump-turbines IAHR 8th Symposium(Leningrad, USSR, 6-9 September, 1976)

[22] Pejovic S, Obradovic D and Gajic A 1984 Hydraulic transients in a power plant - mathematical modeling confirmed by field tests, Hydrosoft 84, Portoroz, pp. 5.57-67.

[23] Pejovic S, Gajic A and Obradovic D 1983 ASME meeting Performance Characteristics of Hydraulic Turbines and Pump 6 15-21

[24] Pejovic S, Zhang F Q, Karney B and Gajic A 2011 Analysis of pump-turbine s instability and reverse waterhammer in hydropower systems, 5th IAHR-WG2011 Meeting(Belgrade, Serbia, 26-28 October, 2011) pp. 11-26.

[25] Obradovic D, Arnautovic D, Pejovic S and Gajić A 1988 Mathematical modelling of transient regimes in multy units hydro power plants 14th IAHR Symposium(Trondheim, Norway, 20-23 June, 1988) pp 163-176.

[26] Pejovic S and Gajic A 1992 Cases and incidents due to hydraulic transients – yugoslav experiences Int. Congress on Cases and Accidents in Fluid Systems(Sao Paulo, Brazil, 14-18 September, 1992) pp181-223

[27] Karney B, Gajic A and Pejovic S 2003 Case studies – our experience in hydraulic transients and vibrations (invited paper) Int. Conference on Case Studies in Hydraulic Systems CSHS-03 (Belgrade Serbia 2003) pp. 131-154.

[28] Gagić A, Pejović S and Stojanović Z 1996 Hydraulic oscillation analysis using the fluid-structure interaction model 18th IAHR Symposium(Vallencea, Spain, September 16-19, 1996) pp. 845-854.

[29] Swingen S, Stojanovic Z, Brekke H and Gajic A. 1998 Two numerical methods of hydraulic oscillation analysis in piping systems including fluid-structure interaction JSME CENTENNIAL GRAND CONGRESS, International Conference on Fluid Engineering ICFE 97(Tokyo, Japan, 13 – 16 July 1997), 3 pp. 1601-06.
[30] Obradovic D, Pejovic S and Gajic A, 1986 Analysis of earthquake effects upon hydraulic structures 5th International Conference on Pressure Surges (Hannover, F. R. Germany: 22 - 24 September, 1986) pp. 35-41.

[31] Pejović S, Krsmanović L, Gajić A and Obradović D 1982 Resonance in the piping system of a power plant during filling-in 11th IAHR Symposium (Amsterdam, Netherland, 13-17 September 1982) pp 43.1-10.

[32] Pejović S, Gajić A and Obradović D 1984 The effects of air supply to the draft tube upon hydraulic oscillations in a hydropower plant 12th IAHR Symposium (Stirling, UK, August 1984) pp 242-53.

[33] Pejović S, Obradović D and Gajić A 1985 Resonance in hydropower plants experiences in yugoslavia, IAHR Work Group on the Behaviour of Hydraulic Machinery under Steady Oscillatory Conditions (Mexico City 1985) pp 1.1-1.13

[34] Pejović S, Gajić A and Obradović D 1988 Auto oscillations of the intake wheel gate, 14th IAHR Symposium (Trondheim, Norway, 20-23 June 1988) pp 443-50

[35] Pejović S and Gajić A 1992 Sensitivity and stability analysis of pumping systems and air chamber influence on hydraulic vibrations Int. Conference Unsteady Flow and Fluid Transients (Durham, United Kingdom, 29 September-1 October 1992) pp 229-39.

[36] Gajic A 2008 Hydraulic oscillations caused by the earthquake The 4th International Symposium on Fluid Machinery and Fluid Engineering (Beijing, China, 24-27 November 2008) pp 97-106.

[37] Gajic A 2003 Case studies in hydraulic systems- CSHS’03 (editor), Faculty of Mechanical Engineering, IAHR (Belgrade, Serbia, 2003) p 233

[38] Gajic A, Benisek M and Nedeljkovic M 2011 Cavitation and dynamic problems in hydraulic machinery and systems 5th IAHR-WG2011 Meeting (Belgrade, Serbia, 26-28 October, 2011) p 327