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Analysis of transients of hydroelectric power plants with transverse turbines

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Abstract
Transients in hydropower plants can result in serious disturbances in a plant operation and damage of mechanical and civil components. The best way to prevent such adverse outcomes is to conduct a transient-state analysis using a mathe- matical model of the hydropower plant system. Based on the collected data on 270 hydropower plants with cross-flow turbines, regression equations were derived that relate a cross-flow turbine specific speed, rated speed, runner diameter and runner width to the rated turbine head and discharge. The obtained equations were used to estimate the turbine performance characteristics using available unit hill charts of three different cross-flow turbines. Finally, the estimated performance characteristics were used to form the boundary condition 'cross-flow turbine' within the unsteady 1D mathematical model. The model was verified by case studies by comparing the calculated and measured changes in turbine speed and inlet pressure due to sudden outage. The difference between the calculated and measured peak pressure is up to 5% in the most critical period, ie from the moment the charge is discharged until the fan line is closed. In the case of turbine speed, the difference between the peak values is at the same time less than 10%.

Keywords
Cross-flow turbine, unit characteristics, hill map, hydroelectric power plant, load suppression, transients, water shock

1. Introduction

The operation of the hydraulic system can be continuous or irregular (eg temporary). While steady state operation is characterized by constant flow over time, transient modes are characterized by dynamic changes in pressure and flow in the system. Unstable flow also results in dynamic changes in dependent hydraulic parameters, such as pipe friction factor (see Riasi bgl.

The intensity of pressure changes in the transient state depends on the flow change, but also on many system parameters such as pipe length and diameter, pipe material, compressibility. water, gas content in the water, characteristics of valves and used hydraulic machines and so on. . In some cases, the increase or decrease in pressure may be so severe as to cause it

serious damage to the inside of the system, which is an explosion part of the system.
The transient modes of the hydraulic system can be triggered by a number of events, which can be divided into two main groups: uncontrolled / unintended events (eg hydraulic machine, change of control valve opening, etc.). In both cases, system security is essential

A good example of the complexity of transients in a hydraulic system is the Garga and Kumara2 study of water hammer in pipes made of different materials. They calculated and measured the lower water pressure of a pipe made of Glass Reinforced Fiber Plastic (GRP) compared to a GRP and Mild Steel (MS) pipe and the lower water pressure of a pipe made of Glass Reinforced Fiber Plastic (GRP). completely made of MS.
A hydropower plant is often considered a complex hydraulic system. Transient modes in hydropower plants arise due to changes in turbine operation or due to changes in suction valve opening. The research presented in this paper is focused on the transients generated by changes in the turbine operation, particularly sudden load rejection.
Transients in hydropower plants and in hydraulic systems in general have been the subject of research by numerous researchers for a long time. first pub- lished results related to the subject of water hammer date back to the beginning of the 20th century (Joukowsky3; Allievi4).

Over time, extensive research efforts have resulted in the partial standardisation of certain parameters that characterise unsteady operation of hydropower plants and in the formation of IEC standards used worldwide today (e.g. IEC 603085).

However, despite attempts to standardise the construc- tion and operation of hydropower plants, each hydro-power plant is still a unique facility whose behaviour depends on a large number of parameters, especially in transient-state regimes. In addition to the parameters that typically affect the transient state of any hydraulic system (ie pipe material, diameter, length, friction, etc.), in the case of a hydroelectric power plant, certain specific system parameters. These include the performance characteristics of installed turbines and the flow characteristics of intake valves (see Kodura6), but also the latest law of turbine flow control assemblies such as Pelton turbine nozzles (see eg Bergant et al. 7) or Francis turbine. guide vanes (see, e.g., Zhang et al.8).
The hydraulic turbine is the heart of the hydroelectric power plant, so its type and performance characteristics (eg hill maps) deserve special attention in transient analyzes. However, despite the large amount of data published in the scientific and scientific literature on the Francis, Pelton or Kaplan turbine, the available data on steady state and especially on the transient operation of transverse turbines are still limited.
The cross-flow turbine (Ossberger, Banki-Michell) is a slow-running pulse-type turbine. Unlike most water turbines, which have axial or radial flow, the water in the cross-flow turbine passes through the turbine or through the turbine blades.9

In practice, cross-flow turbines are commonly used in the grid range of 2.5 to 200 m and in the flow range of 0.04 to 13 m3 s21 (Adhikari10; AHEC-IITR11; Ossberger Hydro12). Cross-flow turbines are one of the cheapest and most robust turbines used in small hydropower plants due to their simple design, good technical performance and high adaptability to flow changes, especially if the turbine is built as a two-cell. units (usually in a 2: 1 ratio).
The turbine is usually part of the hydropower system, which is the most difficult mathematical model. Turbine flow is extremely complex, spatial, non-uniform, uneven and sometimes multi-phase (mixture of water, steam and air). Even at steady state, turbine flow is prone to instability, which may be due to various turbine-specific phenomena, such as leakage vortices caused by tip clearance between impeller and casing (Ma et al.13) or intake tube vortices. (Zhou et al., 14). In continuous modes, the situation is complicated because the turbine speed is highly variable due to the impeller torque imbalance.
Modern CFD techniques are an absolutely powerful and widely used tool for analyzing turbine flow, evaluating their performance characteristics and predicting their behavior under different operating conditions. Cross-flow turbines are no exception, as evidenced by numerous published scientific papers based on the application of CFD for the analysis, development and design of this type of turbine. One of the latest advances in this area was reported by Mehr et al.15 They used commercial CFD software to develop a three-step numerical method for cross-flow turbine nozzle design (first step), impeller optimization (second step)) and turbine performance improvement (third step). . Such an approach allows them to optimize turbine design to achieve increased efficiency, evaluate turbine performance characteristics, and analyze turbine performance under various load conditions. . It is possible that in combination with a conventional 1D model of inconsistent pipeline flow, such a CFD model of a cross-flow turbine may provide a comprehensive tool for transient analysis. operation of hydroelectric power plants. In the case of Francis turbines, a similar approach has already been proposed by Zhang et al.8 However, in the case of cross-flow turbines, such a test has not yet been published.
On the other hand, all CFD methods require that the internal geometry of the turbine be known or evaluated in advance. This paper proposes another, simpler method for predicting cross-flow turbine behavior during transient operation. This method only reqires that the designation as projected network and turbine discharge be detected. No pre-turbine design information is required, which requires this method to be used at the design stage

2. Method statement

hydropower plants. Such an approach to the knowledge of the authors has not yet been published in the available literature.
The presented method uses a new empirical equation using the specific cross-flow turbine speed, nominal speed, impeller diameter and impeller width to the nominal head and turbine outlet. These comparisons were obtained by regression analysis of data collected at 270 transverse turbine hydropower plants. The resulting regression equation is then used to estimate the performance characteristics of the turbine base.

test equipment manufacturers. The results of such model tests are usually given in the form of hill diagrams n11 - Q11 representing the relationships between unit flow Q11, unit speed n11 and efficiency h, as a distribution vane

3. Declaration of procedure

Transverse turbine characteristics
In transient mode, it is important to take into account changes in flow and pressure in the system over time. In the case of continuous mode activation by changes in turbine operation, the hydraulic response of the system depends not only on the opening / closing time of the turbine distribution fans (or gates), but on the (variable) impeller speed. .
To understand the behavior of a turbine in transient modes, its performance characteristics must be known in advance. These characteristics include the relationships between velocity n, discharge Q, head H, torque T, force P, hole opening - and
case of Kaplan turbines - impeller angle f. Most of the above characteristics are usually shown in the form of unit hill diagrams, where the so-called unit quantity refers to a turbine with an impeller diameter of 1 m and works on a 1 m net slope.
(Jordan, 16 Benishek, 17 Kovalev18):
there is no quantitative dimension, although according to equation (4) it has a certain (unimportant) dimension. Specific speeds are used to classify, compare and scale turbines, but they are also the starting point for measuring a turbine and estimating its performance char
The obtained results show very good agreement andhigh reliability, especially in the case of equation, which it shouldbe noted that all the diameters of the actual turbines arerounded to (manufacturer's) standard values (200, 300,400 mm, and so on). A similar procedure may be appliedwhen using equation (10), that is, the obtained result forthe particular head and discharge could be rounded tothe closestmultiple of50 or 100mm.Figure 1. Rated speed none for cross-flow turbines: (a) actual rated speed none against calculated speed none (H, Q) and (b) actual rated speed none against calculated speed none (H, D).
speed booster between turbine and generator. Therefore, if the turbine is directly connected to an electric generator (usually without ø 500 min - 1), the result of equation (8) or equation (9)
must be completed at the nearest synchronous speed
rotation.
Using the same sample data and the same method, two more regression equations with the diameter and width of the runner on the observed head and outlet were obtained:

The obtained results show very good agreement and high reliability, especially in the case of equation (11), which is intended for estimating the impeller width of a cross-flow turbine. In the case of impeller diameter, it should be noted that all actual turbine diameters are surrounded by initial (production) values (200, 300, 400 mm, etc.). A similar method can be used using equation (10), ie the result obtained for a particular slope and discharge can be rounded to the nearest multiple of 50 or 100

are able to assess the specific speed ns, the rated speed no, the impeller diameter D and the impeller width B for the known rated or design head Ho and the displacement Qo in the random cross-flow turbine. The specific speed is the initial parameter for evaluating the power characteristics of the turbine unit, while the marked speed and impeller diameter are needed to calculate the turbine power parameters (head, discharge, etc. Power, torque and speed) of the estimated unit characteristics.
The performance characteristics of a cross-flow turbine unit, where only the design or nominal head and flow are known, can be estimated if the unit hill diagram is for at least two cross-flow turbines that are otherwise specific. speeds are known. For this purpose, a map of the unit hills of three excursion turbines was found in the literature and on the Internet. The specific speeds of these turbines are ns = 45.7, ns = 68.8 and ns = 93.4. A map of the unit hills of selected cross-flow turbines is shown in Figure 4.
The procedure for assessment of the unknown per- formance characteristics of an arbitrary cross-flow tur- bine is briefly described in five steps as follows.
Step 1. All three hill charts given in Figure 4 must be digitalised, that is, transformed into the dimensionless matrix form by introducing the following dimensionless unit quantities:

where the unit quantities n11o, Q11o, and P11o refer to the turbine best efficiency point.
At this point some assumptions must be made regarding the off-design regimes of a turbine. Namely, the unit hill charts given in Figure 4 cover only the regimes expected in normal that is, designed steady- state operation of the turbine.

However, as stated by Iovaˇnel et al., 23 Due to the growing instability of the energy market, hydropower plants often operate on unstructured parameters, even in modes that are consistent. In the case of transient operation, for example after an abrupt load shutdown, the turbine will always go (even for a short time) in modes that fall outside the designed operating range. Therefore, out-of-design operating system modes are particularly important in terms of mathematical modeling and analysis of turbine behavior during transient operation.
An analysis of the performance characteristics of various cross-flow turbines found in the literature (Durgin and Fay24; Adhikari10; Mockmore and Merryfield25) concludes that the performance characteristics of the turbine unit at their left and right edges can be as rough as:

4. Transient-state model of hydro power plants with cross-flow turbines

known boundary and initial conditions, these equations are usually solved by numbers using the property method. As a result, the changes in flow Q (x, t) and head H (x, t) are obtained at discrete distance times in all pipes in the hydraulic system.
A general sketch of a transverse turbine hydroelectric power plant is shown in Figure 6. The calculation area consists of a feeder (2) and a short inlet pipe (1). All required boundary conditions (BC) are also indicated in Figure 6. The presented mathematical model of transient operation of a hydroelectric power plant with a transverse turbine was solved by software developed by the authors. The calculation procedure used to solve the unstable current equations (22) and (23) by the property method is described in detail in Chaudry26 and Watters.28 Similarly, the boundary conditions that represent the pipe node and the tank are defined. The friction term in equation (23) is mixed with the first order approximation. The Darcy-Weisbach coefficient of friction f is assumed to be constant at a non-constant current. The involvement of higher order estimates in terms of friction or non-constant friction is considered superfluous in this case because the great uncertainty in the mathematical model comes from the robust performance characteristics of the turbine. The marginal situation is already a "cross-flow turbine"
evaluated using an iterative procedure similar to that proposed by Chaudry26 for the Francis turbine. The method for estimating the generator inertia hour is given in the same literature. The total moment of inertia of the impeller, clutches and speed booster is estimated at 10% of the moment of inertia of the generator. The evaluation of the cross-flow turbine performance characteristics is described in detail in the previous section.

Because the computational area shown in Figure 6 consists of two different tubes, the temporal and spatial measures of the numerical (i.e., characteristic) grid must satisfy the equation.
Dt = Dxi = Li, i = 1 to N24
aini
where ni is an integer equal to the number of sections where the pipeline is divided, and N is the number of pipelines in the system (N = 2). Since the velocity of the wave ai cannot be accurately calculated, small adjustments to its value are acceptable to obtain an integer number of sections (Chaudry26).
In the three case studies presented in the following section, the characteristic grid was defined by setting the spatial step Dx1 of the shortest system pipe (supply pipe) equal to the length of this pipe (see Table 1).). This results in a time step Dt of about 10 to 20 ms. The spatial step Dx2 of the spindle is then calculated using equation (24) and ranges from about 6 to 11 m. Small measures did not lead to any improvement in the results, indicating that the numerical scheme was convergent and stable. Case studies
● The presented mathematical model of the transient state of operation of a transverse turbine hydropower plant is investigated and validated by comparing the calculated changes in flow and slope (ie pressure) with the changes measured by three hydropower plants: HP De Belci op the River Josˇanica in Serbia.
● HP Zabukh 2 on the Aghavno River in Armenia.
● HP Velezˇ on the Samakovska River in Serbia.
Table 1. Basic data on hydropower plants.

5. Casestudies

Table1

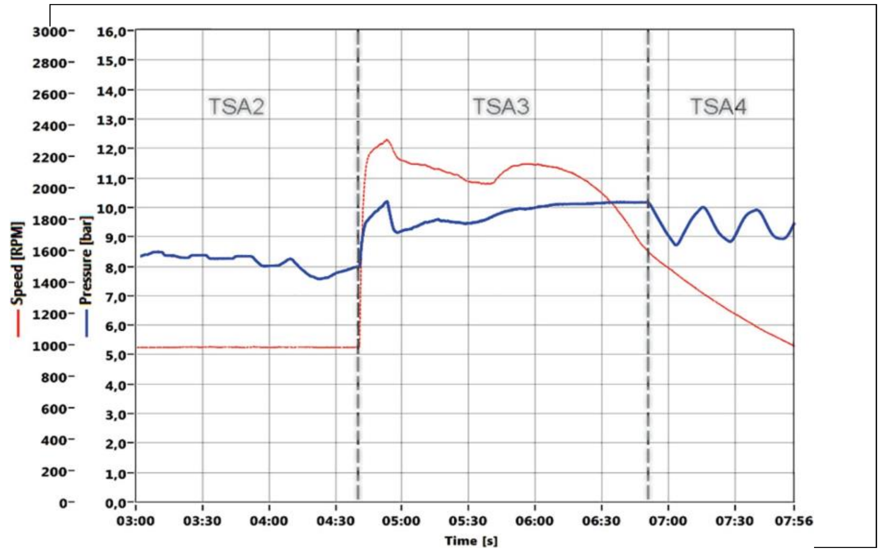
HP	Parameter	Belci	Zabukh2	Velezˇ
Penstock	Length	1964m	4510m	1690m
	Diameter	DN1700	DN1500	OD630
	Material	Steel	GRP	PEHD
Inletpipe	Length	10m	15.4m	20m
	Diameter	DN1700	DN1500	DN600
	Material	Steel	Steel	Steel
Designdata	Grosshead	30.6m	92.6m	103.5m
	Flowrate	5.65m^{3}s^{21}	4.5m^{3}s^{21}	0.96m^{3}s^{21}
Noofturbines		1	1	1

Table2.Basicdataontheinstalledcross-flowturbines.

HP	Belci	Zabukh2	Velezˇ	Note
RatedheadH_o(m)	31.6	85	88.4	
RatedflowQ_o(m^{3}s^{21})	5.65	4.5	0.96	
RatedpowerP_o(kW)	1471	3227	707kW	
Efficiencyh$_{bep}$	0.84	0.86	0.85	
Generatorspeed(min^{-1})	1000	1000	600	
Turbinespeedn_o(min^{-1})	169	298	600	Actual
	197(+16.6%)	308(+3.6%)	587(–2.2%)	Equation(8)
	192(+13.5%)	310(+4.3%)	587(–2.1%)	Equation(9)usingassessed$D$
	194(+15%)	309(+3.9%)	600(SYN)	Mean valueofequations(8)and(9)
Runnerdiameter$D$(m)	1.25	1.25	0.6	Actual
	1.15(–8%)	1.2(–4%)	0.65 (+8.3%)	Equation(10),rounded
Specificspeedn_s	86.5	65.6	58.9	Actual,equation(4)
	86.7(+0.2%)	59.4(–9.4%)	58.5(–0.5%)	Equation(6)
	100.7(+13.5%)	67.8(+3.3%)	57.7(–2%)	Equation(7)
	93.7(+8.3%)	63.6(–3%)	58.1(–1.3%)	Mean valueofequations(6)and(7)
Unitspeedn_{11o}(min^{-1})	37.6	40.4	38.3	Actual,equation(1)
	39.8 (+5.8%)	40.3(–0.3%)	41.5 (+8.3%)	Equation(1)usingassessed$n$,D
UnitflowQ_{11o}(m^{3}s^{21})	0.64	0.31	0.28	Actual,equation(2)
	0.76(+18.1%)	0.34(+8.5%)	0.24(–14.8%)	Equation(2)usingassessed$D$
UnitpowerP_{11o}(kW)	5.3	2.64	2.36	Actual–equation(3)
	6.26(+18.1%)	2.86(+8.5%)	2.01(–14.8%)	Equation(3)usingassessed$D$

Basic data on these hydro power plants are given in table 1 while the data on the installed cross-flow tur-binesaregiveninTable2.,and magnetometer, calculated strengthen proposed regression equations (6) through (10) arealsogiveninTable2.he calculated and actual values are given in parentheses.

Table3



The speeds of the electric generator are the same, which is a good indication that the model hydraulic transition state of the system is well defined.

(1) From the moment the load is disengaged until the hour t B 380 s, the calculated inlet pressure curves approach the measurement, especially the pressure curve "A". The difference in the calculated pressures "A" and "R" is probably due to a significant difference between the actual and gross unit flow Q11o, the unit power P11o and especially the nominal speed no (see table 2). However, it should be noted that the measurement and both calculated peak pressures are almost the same at the same time, even if it is shifted in time. Such a result should be considered acceptable for an inconsistent 1D model based on robust turbine performance characteristics. (2) At the end of the TSA3 time interval, the two calculated inlet pressure curves start to be based on the measurement and reach a maximum difference of about + 16% at the time of closing the turbine line. A possible explanation for this difference may be that the hydraulic cylinders that control the opening of the veneer guides are read near the lower end of their stroke (for example to prevent hammer irrigation). In other words, the actual closing law of turbine conductors is not linear, as the model assumes.

(3) In the period after the closing of the distribution vanes (TSA4), the calculated and both measured pressure changes showed an oscillating pattern typical of the pressure oscillation of the water shock when closing the valve. However, the calculated amplitudes of the inlet pressure oscillations were larger than measured (approximately 15%). Such a result supports the hypothesis that the hydraulic cylinders that guide the guide vanes are read to reduce the closing speed of the vanes at the end of their path. The lower closing speed resulted in a slower delay in fluid flow, and thus a slower pressure oscillation after the guide fans were closed.

generator) can be achieved by suppressing the load. In the analyzed case, the calculated peak speed of these measurements differs only by -9% (speed 'R') to -9.5% (speed 'A'). This result suggests that the proposed unstable model with sufficient accuracy can predict the maximum speed exceeded when shutting down the emergency turbine caused by load shedding. The maximum overspeed should be below the continuous turbine speed, which is in the range (1.9–2.3) 3 of the rated speed for cross-flow turbines (IEC 6200629).

ZABUKH Hydroelectric Power Plant 2

Figure 9 (a) shows a comparison of measured and calculated cross-flow HP Zabukh 2 turbine inlet pressures. The calculated flow change is also shown in Figure 9 (a), even if the flow is not measured on the page. Figure 9 (b) shows a comparison of the measurement and the calculated speed of the electric generator. The marked time intervals and the names 'A' and 'R' have the same meaning as in the previous case study.

A brief analysis of the presented results leads to the following conclusions:

(1) The calculated inlet pressures and generator speeds "A" and "R" are almost the same during the observed period. This shows that the values of specific speed, observed speed and runner diameter were approximated by the proposed regression equations (6) to (10) close to reality (see Table 2). . (2) The calculated inlet pressure curves exactly follow the measurements in one of all three time intervals. The maximum difference between the calculated and measured pressure of 6% was approximately 30 s after the moment of load suppression. This small difference may be due to the non-linear stroke of the hydraulic cylinders that control the opening of the vanes in the actual cross-flow turbine. Most importantly, however, the calculated and measured peak pressures are almost identical. In addition, unlike the previous case study, the calculated and measured peak pressures occur simultaneously, suggesting that the gross performance characteristics of the turbine are usually true. (3) Interestingly, the measured amplitudes of the pressure oscillations during TSA4 are now much closer to the calculation and are larger than in the previous case study. The hydraulic cylinders in the guide sockets do not seem to be too wet.

(4) The calculated and measured electric generator speed change curves are almost the same until the end of the most important time interval TSA3. The calculated maximum speed after load suppression differs from the measurement by only -5% at t B 295 s. The slower delay calculated at the end of TSA3 and TSA4 can be attributed to higher moments of inertia of the rotating masses . Velezˇ hydroelectric power plant

Figure 10 (a) shows the measured and calculated changes in cross-flow turbine inlet pressure installed at HP Velezˇ. Because generator speed measurement is not available, Figure 10 (b) shows only the calculated speed. By analyzing the results shown in Figure 10, the following

The following conclusions can be drawn:

(1) The calculated pressures and speeds "A" and "R" are the same, mainly due to the similarity between gross and actual speed (simultaneously), but also due to the estimated value of the specific speed and average are almost real (see Table 2).

Figure 10. HP Velez: (a) turbine inlet pressure and flow, and (b) calculated generator speed. (2) The calculated and measured inlet pressures are almost the same, except for almost the end of the TSA3 period, where the largest difference is almost + 5%.

(3) During TSA4, the calculated pressure oscillation amplitudes are larger than measured. The reasons for these small deviations are probably the same as for HP Belci.

6. Conclusion

Turbine performance characteristics (eg in the form of unit hill diagrams obtained from model tests) are important data for the analysis of continuous and transient operation of a hydroelectric power plant. However, for cross-flow turbines, the lack of their performance characteristics is the rule because they are often not supplied by manufacturers. This document presents a direct and circular approach for estimating the performance characteristics of a random cross-flow turbine, even if the grid height and turbine power are known, for example at the design stage of a hydropower plant. The monitored power characteristics allow the representation of the transverse turbine as a specific boundary condition within the mathematical model of transient flow in hydropower plants.

A comparison of numerical results and measurements performed on real hydropower plants with transverse turbines shows a very good agreement between the measurement and the calculated turbine inlet pressure and also between the measurements and the calculated generator speed (ie turbine). Such a result suggests that the developed mathematical model can be successfully used to predict the behavior of a cross-flow turbine during transient modes.

Further advances in the presented mathematics

models can be achieved by combining more than three known unit hill diagrams, which should lead to a better assessment of the performance characteristics of the cross-flow turbine. Accurate estimation of the moments of inertia of the rotating mass (generator, runner, speed booster, clutches) will also lead to greater accuracy. Finally, the introduction of a more realistic manual for concluding the Fan Act will provide a better prediction of cross-flow turbine behavior during transient operation. This is clearly demonstrated in the explained case studies, where the largest difference between the calculated and measured inlet pressure and the turbine speed is due to the statutory closed linear line fan. On the other hand, the presented mathematical model should serve as a tool for the analysis of different methods (including different laws of closed guide fans) to prevent or suppress the adverse consequences of transients in hydropower plants. there are cross-flow turbines.

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