The Effect of Surge Tank Throttling on Governor Stability, Power Control, and Hydraulic Transients in Hydropower Plants

Kaspar Vereide, Bjørnar Svingen, Torbjørn Kristian Nielsen, and Leif Lia

***Abstract*—This paper investigates the effect of surge tank throttling on governor stability, power control, and hydraulic transients in hydropower plants. The work is intended to be practical, but includes some new research. The practical contributions include a methodology for a combined evaluation of the effects of installing surge tank throttles in hydropower plants, and a demonstration of the throttle effects through a case study. The research contributions include the evaluation of the throttle effect on power control, and a comparison of the throttle effects on power control for governor systems with speed feedback exclusively versus combined speed and power feedback. Field measurements are used to calibrate a numerical model of the case-study hydropower plant. The results from the case study show that the throttle has an insignificant positive impact on governor stability. Power control is improved when a throttle is installed; the overshoot of produced power and the time until steady-state conditions occur are reduced. The throttle has a significant effect on the hydraulic transients, and increases the water hammer and reduces the mass oscillations in the system.**

***Index Terms*—Control systems, fluid dynamics, hydroelectric power generation, stability analysis.**

I. Introduction

THIS work investigates the effects of surge tank throttling on governor stability, power control and hydraulic transients in hydropower plants. Governor stability is relevant for hydropower plants applied for automatic control of the frequency in the power grid. Hydropower plants are effective for this task due to their ability to increase or reduce power output within a limited time.

The automatic control of grid frequency by hydropower plants is enabled through turbine governors with speed feedback [1]–[3]. The governor measures the rotational speed of the units and closes or opens the guide vanes or injectors in order to maintain a reference speed. The governor may be programed with several variants of speed feedback and power feedback, depending on system stability and grid requirements. When a single turbine unit is operated on an isolated grid, only speed feedback is enabled. When operated on a large interconnected grid, droop is included and power feedback may also be included to improve control of the output power.

All governor systems for hydropower plants are potentially unstable, owing to the water inertia and elasticity. When the power plant reduces the water flow in order to reduce the produced power, water inertia and elasticity counteracts the desired change as the pressure will increase.

Power feedback may increase this effect compared to when only speed feedback is activated. To ensure stability, the appropriate turbine governor has to be selected based on thorough studies. In hydropower plants with long water conduits, restrictions to surge tank water-level amplitudes may require governors to operate exclusively through speed feedback and droop functionality without power feedback.

Implementing a surge tank is often necessary to improve governing when the water inertia otherwise would become too high [4]–[7]. The surge tank introduces a water reservoir with a free water surface close to the turbine, and thereby reduces the water inertia and the elastic water hammer [8]. For hydropower plants with long water conduits, the surge tank is often the only feasible measure from which governor stability may be achieved with adequate time constants that fulfill the relevant grid codes. However, the surge tank introduces the problem of mass oscillations, which occur in the form of U-pipe oscillations of the water between the reservoir and the surge tank. The resulting surge tank design is determined based on the maximum amplitudes of the mass oscillations. The minimum size of the surge tank is selected to ensure that the mass oscillations are dampened over time [5]. If the surge tank is of insufficient size, the governor system can become unstable and amplify the mass oscillations to damaging magnitudes.

The mass oscillation amplitudes may be reduced by increasing surge tank size or by installing a throttle in the surge tank inlet [9]. Fig. 1 present a principle diagram of a hydropower plant with surge tanks where throttles may be installed. The throttle is constructed as an orifice that restricts the water flow in and out of the surge tank. The throttle may be symmetric, with equal loss coefficients in both flow directions, or it may be asymmetrical, with different loss coefficient depending on the flow direction.

This work investigates the throttle's effect on three key aspects of hydropower plants; governor stability, power control, and

Manuscript received October 10, 2015; revised April 28, 2016 and August 9, 2016; accepted September 27, 2016. Date of publication October 31, 2016; date of current version February 16, 2017. Paper no. TEC-00741-2015.

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Digital Object Identifier 10.1109/TEC.2016.2614829

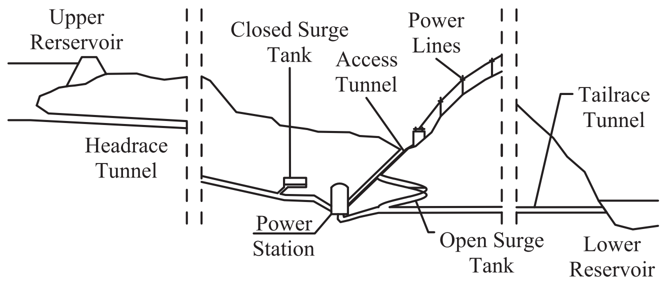

Fig. 1. Principle diagram of a hydropower plant.

hydraulic transients. These aspects are multidisciplinary and are normally evaluated with different methods. In the present work, a methodology for a combined evaluation is presented.

Previous publications on the effect on governor stability include Escande [10] who was among the first to investigate the impact of throttling on governor stability for small oscillations in hydropower plants. Li and Brekke [11] investigated the stability of large amplitude water-level oscillations in throttled surge tanks, and found that the throttle improves governor stability for large oscillations. Yang and Kung [12] investigated governor stability of closed surge tanks with throttles, and confirmed that the throttle has a positive effect on stability. The last two studies were conducted with the phase plane method as described in [13]. A limitation of these studies is that they assume an ideal governing of power, i.e. the product of pressure, discharge, and unit efficiency is constant.

The throttle's effect on power control in terms of overshoot and time until convergence after load changes has not been properly addressed previously. This topic has most likely been overshadowed by the governor stability and the hydraulic transients, as the throttle's effect on the power control is not design critical. However, in the modern power market, the power control is of increasing importance, both to improve the control of the grid frequency and to optimize the production of electrical power and energy. The methodology presented in this work allows for testing of the throttle's effect on power control, and compare the effects when different governor systems are implemented.

The throttle's effect on hydraulic transients is well known and has been investigated in previous works including [2], [14], [15]. The throttle's effect on hydraulic transients is included in this work to provide a more complete overview of the effects of installing a surge tank throttle. By conducting a combined evaluation of all the throttle effects, the basis for decision-making is improved, and by applying the same methodology for all the effects of the throttle, the connection between the different effects become more transparent.

The presented methodology employs one-dimensional (1D) numerical modelling, and includes the effects of non-linearities and non-ideal governing systems. The methodology is demonstrated on a case-study hydropower plant, namely the 150 MW Torpa power plant in southern Norway. The throttle effects are evaluated separately for a governor system with speed feedback exclusively and a governor system with combined speed and power feedback.

Section II of this work presents relevant theory for 1D numerical modelling of the hydraulics and governing systems in hydropower plants. Section III outlines the methodology for evaluation of the throttle effects, and presents the case-study hydropower plant on which the methodology is demonstrated. Section IV gives the results, while Sections V and VI present a discussion and the conclusions.

II. Theory

This section outlines the theory for the 1D numerical modelling of hydropower plants applied in this work. The following components are described; upper and lower reservoirs, pressurized pipe and tunnel flow, Francis turbines, turbine governors, generators, open surge tanks, closed surge tanks, throttles, and the power grid frequency and power consumption. The voltage governor of the generator is not included in the present work to isolate the effects of throttling on the frequency governing. For theory on additional types of hydropower plants and components, one may consult the work of Chaudhry [2] and Wylie and Streeter [15].

The boundaries of the system include the upper and lower reservoirs. These boundaries dictate the available pressure head in the system, and can be modelled as constant pressure boundaries.

The governing equations for pressurized pipe flow are the equation of continuity (1) and the equation of motion (2):

$$\frac{\partial p}{\partial t} + \frac{Q}{A}\frac{\partial p}{\partial x} + \frac{\rho a^2}{A}\frac{\partial Q}{\partial x} = 0 \tag{1}$$

$$\frac{1}{A}\frac{\partial Q}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{Q}{A^2}\frac{\partial Q}{\partial x} + g\sin\alpha + \frac{fQ|Q|}{2DA^2} = 0 \tag{2}$$

where p is the water pressure, ρ is the water density, g is the acceleration of gravity, Q is the water discharge, A is the cross sectional area perpendicular to the flow, x is the length coordinate, t is the time, α is the angle to the horizontal plane, a is the speed of sound in water, f is the Darcy-Weisbach friction factor, and D is the pipe diameter. Several methods for solving these equations exist [2], whereof the method of characteristics (MOC) is the most frequently applied. The MOC [15] combines the two equations above and transforms them into two ordinary differential equations, which are integrated to obtain two algebraic relationships:

$$\begin{aligned} p_{x,t+1} = p_{x-1,t} &- \frac{a\rho}{A}(Q_{x,t+1} - Q_{x-1,t}) \\ &- \frac{f\Delta x\rho}{2DA^2}Q_{x,t+1}|Q_{x-1,t}| \end{aligned} \tag{3}$$

$$\begin{aligned} p_{x,t+1} = p_{x+1,t} &+ \frac{a\rho}{A}(Q_{x,t+1} - Q_{x+1,t}) \\ &+ \frac{f\Delta x\rho}{2DA^2}Q_{x,t+1}|Q_{x+1,t}| \end{aligned} \tag{4}$$

where subscripts $x-1$, x, and $x+1$ are the previous, current and next length steps respectively, whereas t and $t+1$ are the current and next time-step. These two equations are only valid for calculation with time-step $\Delta t = \Delta x/a$ [15]. When the initial conditions are known, there are two equations and two unknowns, and the system can be solved. More detailed information of the MOC is found in Wylie and Streeter [15].

Surge tanks for hydropower plants can be separated in two main groups; open surge tanks and closed surge tanks. Open surge tanks are most common, and the equations are described in Wylie and Streeter [15]. Closed surge tanks for hydropower plants are less familiar, and are usually assumed to have adiabatic thermodynamic behavior, with an adiabatic gas constant of 1.4 for air [16], [17]. The adiabatic behavior in closed surge tanks can be modelled with the following relationship:

$$p_{\text{air}} {V_{\text{air}}}^{1.4} = k \tag{5}$$

where p_{air} is the absolute air pressure, V_{air} is the air volume, and *k* is a constant. Previous studies have shown that different closed surge tanks may have different thermodynamic behavior based on size and construction material [18], and that closed surge tanks may be influenced by heat transfer [19], [20]. Field measurements may be necessary to determine the appropriate thermodynamic model.

Surge tank throttles induce a singular loss in the connection tunnel between the surge tank and the main water conduit. The throttle loss is described by

$$p_{\text{loss}} = \xi \frac{\rho Q^2}{2A^2} \tag{6}$$

where ζ is the loss coefficient, *Q* is the water discharge through the throttle, and *A* is the cross-sectional area of the tunnel before and after the throttle. In this work, a symmetric throttle with equal head loss coefficient in both flow directions is tested.

Francis turbines may be modelled with three alternative approaches; simplified relationships, Hill charts, or analytical models [2]. Hill charts with the measured characteristics of the turbine are often regarded as the most accurate modelling method. However, Hill charts have two weaknesses; measurements are seldom available a priori, and available measurements are based on steady-state conditions and do not fully represent transient situations. In this work, an analytical model for Francis turbines is applied, for which a detailed description is provided in [21]. This turbine model analytically calculates the characteristics of the turbine.

Changes in the angular speed (ω) of the turbine and generator are calculated from

$$J\omega \frac{d\omega}{dt} = P_h - P_g - P_l \tag{7}$$

where *J* is the generator's inertia, P_h is the utilized hydraulic power, P_g is the power consumed by the grid, and P_l is the sum of smaller hydraulic and electromechanical losses. These losses include friction loss in the turbine and distribution ring, which can be estimated from empirical relationships suggested by Nielsen [21]. The losses from generators and transformers are modelled with standard characteristics obtained from the manufacturers.

For synchronous alternating current (AC) generators, the rotational speed of the unit is dependent on the grid frequency:

$$\frac{n_r}{60} = \frac{f_r}{\#} = \frac{\omega}{2\pi} \tag{8}$$

where f_r is the reference grid frequency (Hz), # is the number of pole pairs on the generator, and n_r is the generator's reference rotational speed (r/min).

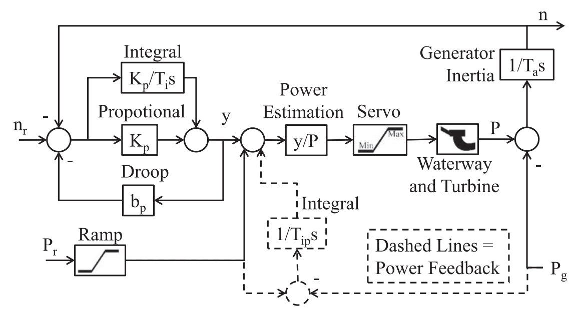

Fig. 2. Block diagram for a PI governor.

The turbine governor controls the opening degree of the turbine in order to maintain a constant rotational speed. A typical Proportional-Integral (PI) type governor is presented in the block diagram in Fig. 2, where *n* is the turbine's rotational speed, n_r is the turbine's reference rotational speed, K_p is the proportional constant, T_i is the integral constant, *s* is the Laplace complex number frequency, b_p is the speed droop constant when running on a large interconnected grid, *y* is the desired opening of the turbine, *P* is the produced power, P_r is the power set point, T_{ip} is the integral constant for the power feedback, and T_a is the time constant of the generator inertia at nominal power. The y/P function adjusts the opening of the turbine according to curves of estimated resulting produced power. The ramp controls how fast the produced power is increased or decreased when the power set point is changed.

III. Methodology

This section outlines the methodology for testing of the governor stability, power control, and hydraulic transients in hydropower plants.

Governor stability can be tested with frequency-response tests evaluated in Bode plots [22]. Traditionally, the governing equations are linearized to analytically calculate the response function through Laplace transforms. This approach is acceptable for small oscillations around a stationary point but is less accurate for large amplitude oscillations, as the system is influenced by non-linearities [11]. Non-linear methods, such as the phase-plane method [13], or numerical solution of the governing equations are necessary for evaluating the stability both for small and large oscillations. In the present work, the one-dimensional governing equations are solved with the MOC in the time domain, and transformed into the frequency domain with the Fast Fourier Transform (FFT). A similar methodology for evaluation of hydropower plant governing is reported by Nicolet *et al.* [23], who uses a finite difference numerical scheme. In this work, the MOC is selected as it is known to yield accurate results for calculating hydraulic systems with throttles [24].

Frequency-response tests are conducted with simulated isolated grid operation. The feedback is disabled (open loop configuration), and the tests are performed with the following procedure:

1) State an oscillating disturbance with a fixed amplitude and frequency in the reference rotational speed (n_r) of the unit to be tested.
2) Run the simulation until steady-state oscillations occur.
3) Measure the response (n) from the system and calculate the phase and gain.
4) Repeat with new frequencies of the oscillating disturbance until the desired spectrum is covered. The amplitude of the disturbance is fixed for all frequencies.

An amplitude of 0.001 p.u. of the reference rotational speed (n_r) is suggested, which equals a disturbance amplitude of 0.05 Hz in the grid frequency. The oscillation frequency of the disturbance should range from 0.001 to 10 Hz. The results are evaluated in a Bode plot, and the system is regarded as stable if it satisfies the Nyquist stability criterion [25]. The criterion states that the system is stable if the phase angle (φ) between the disturbance and the response is higher than $-180°$ at the frequency where the gain (h) first becomes equal to unity (the crossover frequency). The gain (h) is the ratio of the amplitude of the response (n) to the amplitude of the disturbance (n_{ref}).

Governor stability may be quantified with the phase margin ($\Delta\varphi$) and the gain margin (Δh) [22]. The phase margin is the margin between the actual phase and $-180°$ at the crossover frequency. The gain margin is the margin between the gain and unity ($h - 1$) at the frequency where the phase crosses the $-180°$ line. Large margins equal more stable systems, but too large margins also yield slow-acting systems.

Power control is evaluated in a load acceptance test. The power control is quantified with the amplitude of the overshoot or undershoot of the produced power, and the time from the load change is initiated until the power reaches steady state. In this work, a load acceptance from 0% to 100% load with a ramping time of 60 s is applied.

The hydraulic transients are evaluated with an emergency shutdown test from 100% load. During emergency shutdowns, the circuit breaker in the switchgear is activated and disconnects the power plant from the power grid. This initiates a fast closing procedure of the turbine guide vanes or injectors, to avoid dangerous runaway speed of the units. In the numerical model, this event is simulated with a linear closing of the turbines in 10 s from full load. The hydraulic transients are quantified with the amplitude of the water hammer and the amplitude of the mass oscillations.

A. A Case-Study

The methodology is applied on a case-study hydropower plant, namely the 150 MW Torpa hydropower plant located in Nordre Land County in southern Norway. This power plant was commissioned in 1989 and is owned and operated by Eidsiva Vannkraft AS. Fig. 1 shows the principle diagram of the power plant. Two 75 MW Francis turbines with a nominal head of 430 m and a nominal discharge of 40 m^3/s are installed. The headrace is 9.6 km long with an unlined cross-sectional area of 35 m^2 and is inclined directly between the reservoir and the power station, without a pressure shaft. A closed surge tank is located 300 m upstream from the turbines and is constructed as an unlined rock cavern filled with 13 000 m^3 of air at pressure 4.1 MPa. The tailrace is 10 km long with an unlined cross-sectional area of 35 m^2 and an open surge tank with 1650 m^2 of water surface area, which is located 100 m downstream of the turbines. The turbine governors are PI-type speed governors, with proportional constant (K_P) equal to 4.0 and integral constant (K_I) equal to 6.0. The governor constants are tuned according to the national grid code, and are effective for operation on both isolated grid and a large interconnected grid. The turbines closing times from full opening is 10 s. When running on an interconnected grid, the governor has a permanent speed droop (b_p) of 6%. The two generators each have an inertia (J) equal to 114 250 kg m^2 resulting in time constant for the generator inertia (T_a) of approximately 6 s. The nominal rotational speed (n_0) is 600 r/min. The existing governors do not have activated power feedback, but for simulations with power feedback in this work, the time constant T_{ip} is set to 100 s. Anti-windup is implemented. The ramp for the power set point were 180 s for unit nr. 1 and 60 s for unit nr. 2 in the period when the field measurements where conducted.

None of the surge tanks are equipped with throttles in the existing power plant. A numerical model has been established and used to simulate the current situation, and compare with situations where throttles with different head loss are installed in the headrace surge tank.

B. Numerical Model and Field Measurements

The numerical model of the power plant is established with the 1D numerical simulation program LVTrans [26]. This program solves the MOC and the governing equations for hydropower plants as described in the theory section, and applies the analytical Francis turbine model. The simulations were carried out with a time step (dt) of 0.001 s and a length step (dx) of 1.2 m. The singular loss coefficient (ζ) of the T-connection between the headrace tunnel and the surge tank was 60 (–) for the existing power plant, when A in Eq. (4) is 35 m^2. In addition to the existing non-throttled configuration, two different throttles were tested, with $\zeta = 300$ (–) and $\zeta = 1200$ (–). These three different cases are hereby respectively referred to as no throttle, medium throttle, and strong throttle configurations and will be plotted in green, blue and red respectively.

To calibrate the numerical model, field measurements were collected from Torpa power plant. The measurements include reservoir water levels, produced power, air pressure, water level in the closed surge tank, and water pressure upstream from the turbine. The measurements were collected during a load acceptance from 72 to 130 MW, and the initial and end conditions and input data for the numerical model calibration are given in Table I.

A PARO scientific DIQ 73K pressure sensor with error less than 0.04% of full-scale 20 MPa was used to measure the pressure upstream from the turbine. A permanently installed PARO scientific 8DP000-S with error less than 0.01% of full-scale 6 MPa was applied to measure the air pressure in the closed surge tank. Measurements of the produced power, the closed surge tank water level, and the reservoir water levels were taken and provided by the power plant owner. The sampling rate from

TABLE I
INPUT DATA FOR THE NUMERICAL MODEL

Parameters	Initial Conditions	End Conditions
Upper reservoir level (m)	707	707
Lower reservoir level (m)	263	263
Closed surge tank water level (m)	292.3	292.2
Closed surge tank air pressure (kPa)	4140	4120
Power output (MW)	36 + 36	62 + 68
Turbine flow (m^3/s)	9.5 + 9.5	15.7 + 17.2
Tunnel friction factor (−)	0.07	0.07
Speed of sound in water (m/s)	1200	1200

the pressure upstream the turbine is 1 Hz, while the sampling rate from the closed surge tank and produced power is slower and unsteady, because samples are only stored after a minimum change in the measured parameter.

These measurements allow for calibration of the numerical model of the power plant. However, the field measurements were only collected upstream from the turbine and in the closed surge tank, and it is therefore not possible to fully verify the hydraulic behavior of the tailrace tunnel and open surge tank. To account for this, the present study only evaluates the effect of installing a throttle in the closed surge tank in the headrace tunnel. The accuracy of the tailrace modelling is thereby of limited importance for evaluation of the throttling effect.

C. Calibration of the Numerical Model

The numerical model is calibrated with the field measurements. A comparison of the output power production from the two units, the pressure upstream from turbine 1, the air pressure in the closed surge tank, and the water level in the closed surge tank is presented in Fig. 3.

IV. RESULTS

The proposed methodology is applied on the case-study hydropower plant, and this section present the results of the analysis. The governor stability, power control, and hydraulic transients are evaluated for the no throttle, medium throttle, and strong throttle configurations.

A. Governor Stability

Frequency-response tests are conducted on unit nr. 2, while unit nr. 1 is closed. The power plant is operated on an isolated grid with zero damping. The disturbance is an oscillation in the reference rotational speed of the unit with an amplitude of 0.001 p.u. The gain and phase angle between the response and the disturbance is found for frequencies of the disturbance ranging from 0.001 to 2 Hz. Fig. 4 presents the resulting Bode plots.

The phase margin ($\Delta\varphi$) is 76.9°, while the gain margin (Δh) is 4.3 (−) for all configurations. The crossover frequency is 0.11 Hz for all the configurations. There is a local peak in the phase and gain at frequency 0.006 Hz owing to the mass oscillation phase angle peak is −94.4°, −94.3°, and −94.1° the no throttle, medium, and strong configurations respectively. The equivalent mass oscillation gain peak is 8.3 (−), 8.4 (−), and 8.7 (−).

B. Power Control

Power control is evaluated with operation on a large interconnected grid with load acceptance tests from 0% to 100% load for both units. The ramping times are set to 60 s for both units in this test, and owing to similar design and behavior of the two units only the results from unit 2 is plotted and discussed in the following. The power control is evaluated for two different governor systems; with speed feedback exclusively and with combined speed and power feedback. Fig. 5 illustrates the results from simulations of produced power and pressure in front of the turbine. As Fig. 5 depicts, throttles improve the control of the produced power and reduces both the overshoot and the time before the desired steady-state power is reached.

The power feedback improves control of the produced power, and it can be seen that the power estimation in the governor is inaccurate. The set point is set to 75 MW, but the resulting steady-state power is only 73.6 MW. In this power plant the gain/power estimation does not consider whether one or two units are running, resulting in different power output for the same set-point in these two situations. A negative impact of the power feedback is increased time before steady-state conditions occur and increased pressure amplitudes in the system.

The time until steady-state for power is defined as the time from the load change is initiated until the produced power converges to the second decimal, in this case that the remaining oscillation amplitudes are within 74.99 and 75.01 MW. Overshoot and undershoot are defined as the maximum deviation from the final steady-state value after the load acceptance. Table II lists the time before steady-state conditions occur, the overshoot of produced power, and undershoot of pressure upstream from the turbine. The time before steady-state conditions occur in the system with speed feedback exclusively is reduced by 15% or 29% by installing a medium or strong throttle respectively. The time before steady-state conditions occur in the system with speed and power feedback combined is reduced by 6% or 19% by installing a medium or strong throttle respectively.

The overshoot of the produced power for the governor system with speed feedback is reduced by 25% or 66% by installing the medium or strong throttle respectively. By comparison, the overshoot for the governor system with speed and power feedback combined is reduced by 18% or 45% by installing the medium or strong throttle respectively. Finally, the undershoot of the pressure upstream from the turbine for the governor system with speed feedback is reduced by 8% or 15% by installing the medium or strong throttle respectively. The undershoot for the governor system with speed and power feedback combined is reduced by 8% or 16% by installing the medium or strong throttle respectively.

C. Hydraulic Transients

The throttle's effect on the hydraulic transients is evaluated with an emergency shutdown test from full load. The emergency closing in Torpa power plant uses approximately 10 s from full opening.

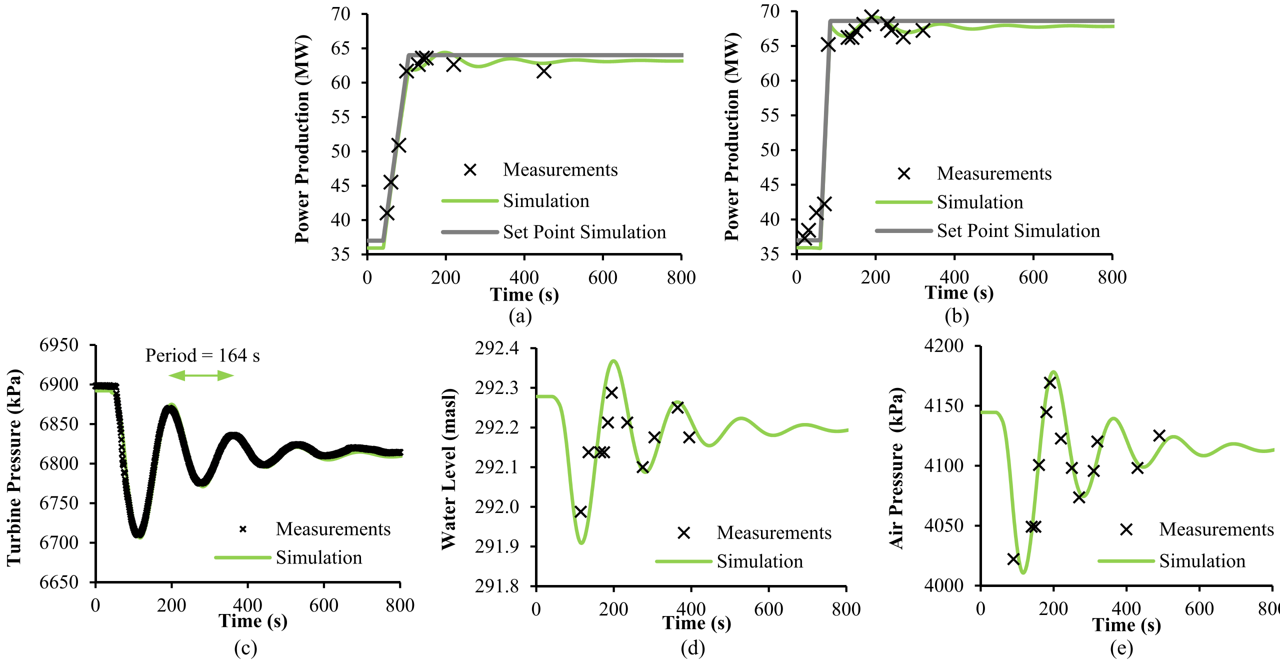

Fig. 3. Comparison of field measurements and simulations of (a) the power output from unit 1, (b) the power output from unit 2, (c) the pressure upstream from the turbine, (d) the closed surge tank air pressure, and (e) the closed surge tank water level.

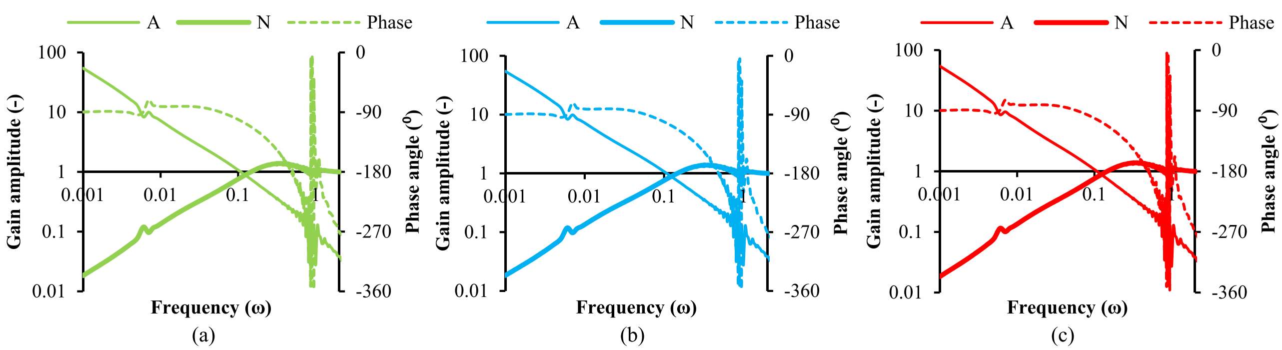

Fig. 4. Frequency-response plot for the system running on isolated grid with (a) no throttle, (b) medium throttle, and (c) strong throttle. The signal input of the frequency-response test is n_{ref} with the disturbance, and the signal output is the resulting n. A is the gain amplitude of the system without speed feedback (open loop), and $N = 1/(1 + A)$ is the offset ratio for the governor system when speed feedback is enabled (closed loop).

Fig. 6 presents the resulting pressure upstream of the turbine during an emergency shutdown, for the three different throttle configurations. The simulations show that the throttle reduces the mass oscillations, but increases the water hammer in the system. The time before steady-state pressure is reduced with stronger throttling. Table III presents the maximum water hammer and mass oscillations amplitudes for the three different throttle configurations.

V. Discussion

The methodology is demonstrated on a case-study. The results show that throttling has a positive impact on governor stability and power control. The Bode plots reveal that the improvement in governor stability is negligible. This limited effect was expected as the maximum frequency disturbances in the grid are relatively small and only result in a limited water flow through the throttle. The load acceptance tests show that the throttle enables more accurate control of the power output and reduces the time before steady-state conditions are restored in the system. This conclusion is valid for systems both with and without power feedback. The power feedback increases the control of output power but increased the time before steady-state and the pressure amplitudes.

The shutdown tests reveal that the throttle reduces the mass oscillation amplitudes and increases the water hammer amplitudes. The tests show that excessive throttling increases the maximum pressure amplitude in the system. In the presented case-study the medium throttle resulted in the lowest maximum pressure.

Owing to the design criteria for throttles, it is assumed that the positive effect of the throttle on governor stability and power

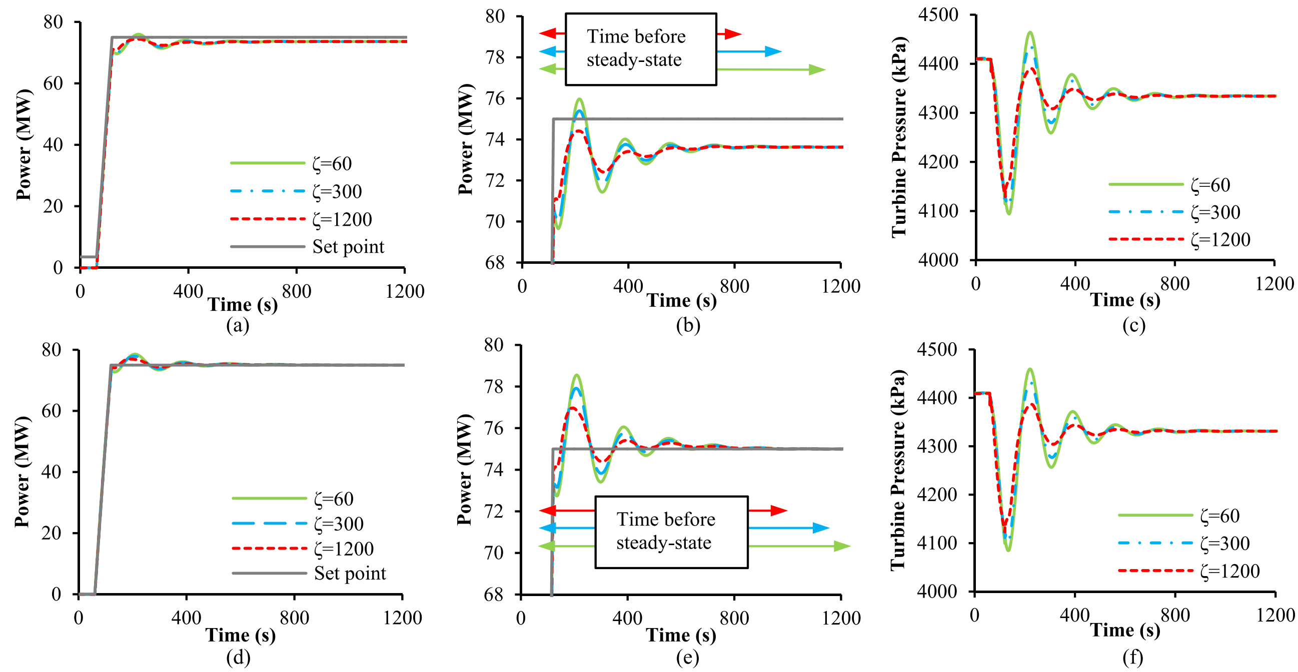

Fig. 5. Load acceptance tests from 0% to 100% load on interconnected grid. Plot (a) shows total range of the power change during the load acceptance, (b) shows close-up of the overshoot and convergence of the produced power, and (c) shows resulting pressure in front of the turbine for the system with speed feedback exclusively. Plots (d)–(f) show the equivalent results for the system with speed and power feedback combined.

TABLE II
RESULTS FROM THE LOAD-ACCEPTANCE TESTS

Throttle loss factor (ζ)	60	300	1200	Unit
Steady-state power (speed)	73.6	73.6	73.6	MW
Steady-state power (speed/power)	75.0	75.0	75.0	MW
Power overshoot (speed)	2.3	1.8	0.8	MW
Power overshoot (speed/power)	3.6	2.9	2.0	MW
Pressure undershoot (speed)	240	221	205	kPa
Pressure undershoot (speed/power)	247	227	208	kPa
Time before steady state (speed)	1140	974	812	s
Time before steady state (speed/power)	1238	1159	1002	s

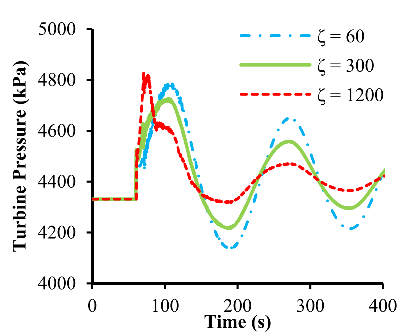

Fig. 6. Emergency shutdown test from full load.

control is similar for a wide range of hydropower plants. Design of surge tank throttling is performed with the same design criteria for different hydropower plants; the maximum water pressure in the conduit cannot exceed a maximum value (normally 15–20% increase from the static pressure), and

TABLE III
RESULTS FROM THE SHUT-DOWN TESTS

Throttle loss factor (ζ)	60	300	1200	Unit
Initial pressure	4331	4331	4331	kPa
Water hammer amplitude	205	296	498	kPa
Mass oscillation amplitude	453	395	283	kPa

local negative pressures cannot be allowed to occur during flow through the throttle. The results from the case-study may therefore indicate the positive effects of the throttle can be expected also in other hydropower plants.

Regarding the effect of throttling on hydraulic transients, the throttle can reduce the design pressure in the system, if the mass oscillations initially cause the maximum pressure amplitude in the un-throttled system. The optimal throttle can then be designed to reduce the mass oscillations amplitude to the equal magnitude as that of the water hammer. However, if the water hammer initially cause the maximum pressure amplitude in the un-throttled system, the throttle will inevitably increase the design pressure.

Additional effects of the throttle not previously mentioned in this work include the additional construction cost of the throttle and the related extra inspection and maintenance in the power plant. For closed surge tanks, the throttle may cause problems of accessing the surge tanks. Further, the throttle may have a small positive impact on the overall efficiency of the hydropower plant, owing to the reduced mass oscillations. Mass oscillations have a negative effect on the efficiency owing to increasing frictional and singular losses, and increased deviations from the turbine best-point operation.

VI. Conclusions

A methodology for a combined evaluation of the throttling effect on governing stability, power control and hydraulic transients is presented and demonstrated. The methodology includes non-linearities and allows for implementation of non-ideal discrete-time implemented governor systems. It may be applied to any hydropower project where throttles are considered in the surge tank design.

The methodology is demonstrated on a case-study hydropower plant, from which field measurements are applied to calibrate a numerical model. Based on frequency-response tests, it is concluded that the throttle has an insignificant positive effect on the governor stability for normal disturbances in the grid frequency. The disturbance is not large enough to activate any significant head loss through the throttle, except during the resonance frequency of the mass oscillations. It should be noted that this may not be the case for all hydropower plants.

The load acceptance tests reveal that the power control is improved by the throttle; the overshoot of output power and the time before the system reaches steady state is reduced. This conclusion is valid for both the governor system with speed feedback and with combined power and speed feedback. The improvement increases with stronger throttling. The load acceptance tests also demonstrate how the power feedback improves the power control with regards to the set-point for power. However, in this case-study, the power feedback also lengthens the time before steady-state conditions occur, and increases the overshoot for produced power and the undershoot for pressure upstream the turbine.

The shutdown tests show that the throttle reduces the mass oscillation, but increases the water hammer. Excessive throttling will result in too high water hammer pressure and must be avoided. However, in the presented case-study, the medium throttle configuration resulted in the lowest maximum pressure. The medium throttle reduces the mass oscillations amplitude, while the water hammer amplitude is still within acceptable levels.

Acknowledgments

This work would not have been possible without the scientific cooperation in the Norwegian Research Center for Environmental Design of Renewable Energy (CEDREN). Wolfgang Richter is acknowledged for assistance during the manuscript preparation.

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Authors’ photographs and biographies not available at the time of publication.