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Physics Based Hydraulic Turbine Model for System Dynamic Studies

Dean R. Giosio, Alan D. Henderson, Jessica M. Walker and Paul A. Brandner

***Abstract*—A one-dimensional numerical model of a Francis turbine hydropower plant for dynamic response studies is presented with an alternate representation of the turbine unit component. The conventional, simplified representation of the hydraulic turbine is replaced by a consideration of the conservation of angular momentum using inlet and outlet velocity vectors calculated based on effective turbine geometry. Specific energy loss components associated with off-design conditions such as runner blade inlet incidence loss and draft tube residual swirl flow loss are determined. Estimates for mechanical frictional losses and churning losses are calculated to ensure accurate simulation across the entire turbine operating range. The resulting model therefore takes into consideration real sources of major loss, eliminating the use of ambiguous correction factors, while remaining equally simple to implement into current power system models. The new turbine formulation is validated against transient test data from a 119 MW Francis turbine unit, while simulations based on two existing conventional models are included for comparison.**

***Index Terms*--Hydroelectric power generation, power system dynamics, hydraulic turbine dynamic simulation model.**

I. Introduction

EVOLVING demands of the modern electricity market over recent decades has seen changes in the way hydro-electric power plants are utilised in electricity systems across the globe.

Increasingly, due to both the deregulation of markets and the growing contribution of non-baseload renewables, hydro-electric power plants are being relied upon to stabilise electricity networks. As such, the power plants are routinely subject to low-load operation, frequent changes in operating set-point, as well as load acceptance and load rejection procedures, leading to potentially dangerous transient phenomena.

Additionally, transient plant behaviour may arise through unplanned events. Electrical faults such as short-circuit, earth faults, loss of synchronisation, or numerous other scenarios which may lead to an emergency plant shutdown will also initiate transient behaviour, both mechanical and electrical in nature. Anticipated response and safety margins of operation may also be considered following large magnitude disturbances, while any proposed modification in regards to plant operation, system control or component upgrades will certainly require adequate transient analysis before implementation. Transient analysis studies are therefore of paramount importance. As such, the relatively recent restructuring and liberalisation of electricity markets around the world has seen an increased effort to produce more accurate dynamic models to improve control systems, estimate stability limits, predict restoration scenarios and increase overall performance of installed plants.

The seminal work by the 1992 IEEE Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies [1] presented early formulations of linear and nonlinear hydraulic turbine models, taking into account both inelastic and elastic water column, for use in a variety of plant configurations. A slightly different formulation was provided by Kundur [2]. Both models have since been used extensively in the literature, and in industry, often as a base model requiring refinements depending on the nature of the simulation.

Hannett *et al.* [3] provided early validation tests of the simple turbine model presented in the 1992 IEEE report [1] by comparing the output of the model simulation with field tests conducted on hydro units at three separate stations. The recommended transfer function for mechanical-hydraulic governors as given by [4] was found to be adequate in representing the governor controls, however the model structure of the turbine and hydraulic dynamics of the penstock as given by [1] was found to not fully represent the characteristics of the units as indicated by the field tests.

As such, Hannett *et al.* [3] offered two main model refinements. The first was to introduce additional damping for no load conditions as a result of discrepancies in observed and simulated speed excursions following a 25% load rejection. The second was to correct for the assumption of a linear turbine characteristic over the full range of guide vane positions present in the IEEE model.

Similarly De Jaeger *et al.* [5] also proposed a nonlinear turbine model supported by field testing of load rejection for both the single unit case and the case with multiple turbines sharing a common tunnel. Hannett *et al.* [6] later re-examined

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D. R. Giosio, J. M. Walker and P. A. Brandner are with the National Centre for Maritime Engineering & Hydrodynamics, Australian Maritime College, University of Tasmania, Launceston 7250 AUSTRALIA (e-mail: Dean.Giosio@utas.edu.au, Jessica.Walker@utas.edu.au and P.Brandner@utas.edu.au).

A. D. Henderson is with the School of Engineering & ICT, University of Tasmania, Hobart 7005 AUSTRALIA (e-mail: Alan.Henderson@utas.edu.au).

both the IEEE Working Group [1] and De Jaeger *et al.* [5] models for multiple units for a system in which dissimilar turbines with different characteristics share a common tunnel.

While these earlier researchers recognised the limitations of the conventional models – particularly concerning prediction away from the designed operating condition, and for large power and frequency deviation studies – the over-simplified representation of the turbine unit has remained unchanged.

Recent literature has largely focused on developments in plant control and governor models [7], [8] with the turbine energy transfer still modelled using a linearization about an operating point [9]–[13], or as described by the IEEE nonlinear model [14]–[16] presented in [1], while other studies require extensive knowledge of turbine efficiency performance data [17]–[19]. Such detailed data is often either unavailable or obtained from model tests performed by the manufacturer which invariably deviates from on-site performance, particularly in the small-opening range [19]. There has been little development in the treatment of the turbine component since the IEEE formulation, with the exception of work stemming from Nicolet *et al.* [17]. The model offered however; while providing significant advances and flexibility in the physical phenomena able to be modelled; still relies heavily on turbine static characteristic curves to calculate Suter parameters from which an equivalent electrical circuit representation may be generated.

This paper presents a one-dimensional hydraulic model of a single machine Francis turbine power station for transient simulation and analysis. The conventional representation of the hydraulic turbine is refined by considering the actual flow physics to account for machine behaviour away from design. By basing calculations on the actual mechanism of energy transfer and considering real sources of loss, the resulting model is valid over the entire operating range without requiring re-tuning, or the use of ambiguous gain and dampening factors. The numerical model is validated against transient tests performed on a 119 MW Francis turbine unit at the Reece Power Station on the West coast of Tasmania, with simulation results also compared to two conventional models.

II. Fundamental Equations

The study of unsteady flow, and the propagation of hydraulic transients brought about by flow disturbances in piping systems, is governed by the equation of motion (conservation of momentum) and the continuity equation.

In developing a one-dimensional model of the dynamic behaviour of a water filled pipe the following assumptions [20] are made:

i. The pipe is uniform, and the pipe length (L) is much greater than the internal diameter (D_p).
ii. Pressure and velocity do not vary across the conduit cross-section and flow is in the direction of the centreline axis,
iii. The pipe remains full of water and the pressure inside the pipe remains above the vapour pressure of water (i.e. no cavitation, or column separation).

Applying the equation of momentum to a control volume of length dx along the x-axis yields:

$$\frac{\delta Q}{\delta t} + gA\frac{\delta H}{\delta x} + f\frac{Q|Q|}{2D_pA} = 0 \tag{1}$$

where Q is the flow rate, H the hydraulic head, g is the acceleration due to gravity, f the friction factor, A the conduit cross-sectional area, and D_p the internal pipe diameter. Similarly, performing a mass balance on the control volume the continuity equation may be expressed as:

$$\frac{gA}{a^2}\frac{\delta H}{\delta t} + \frac{\delta Q}{\delta x} = 0 \tag{2}$$

where a is the wave speed of a given pipe. Equations (1) and (2) are the well-known water hammer equations - coupled partial differential equations with dependent variables of hydraulic head, $H(x,t)$, and discharge, $Q(x,t)$, while the distance along the pipe, x, and time, t, are the independent variables.

A. Elastic Waterway Model

For the case in which the penstock length is significant, or the guide vane opening is considered rapid, the effects of water compressibility and pipe elasticity should be considered. The governing equations of momentum (1) and continuity (2) may be solved using the linear impedance method, based on electrical transmission line theory where the head and flow are analogous to the transmission line voltage and current, respectively [2], [21]. The general solution of the partial differential equations in time and space, normalised by rated head and flow, is given by:

$$\overline{H}_2 = \overline{H}_1\,\mathrm{sech}(T_e s) - \overline{Z}\,\overline{Q}_2 \tanh(T_e s) - \overline{H}_f \tag{3}$$

$$\overline{Q}_1 = \overline{Q}_2 \cosh(T_e s) + \frac{1}{\overline{Z}}\sinh(T_e s) \tag{4}$$

where $\overline{H}_2$ = per-unit hydraulic head at turbine inlet
$\overline{H}_1$ = per-unit hydraulic head at reservoir
$\overline{H}_f$ = per-unit flow conduit head loss
$\overline{Q}_2$ = per-unit turbine flow rate
$\overline{Q}_1$ = per-unit upper penstock flow rate
$\overline{Z}$ = normalised hydraulic surge impedance = T_w/T_e
T_e = elastic water time constant = L/a
T_w = inelastic water time constant = LQ/gAH
s = the Laplace operator

For a hydro power plant with no surge tower and a constant upstream reservoir head, (3) may be simplified to:

$$\overline{H}_2 = \frac{-\overline{Z}\,\overline{Q}_2(1-e^{-2T_e s})}{(1+e^{-2T_e s})} - \overline{H}_f \tag{5}$$

The functional block implementation of the elastic waterway model as given by (5) is illustrated later in Fig. 5.

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B. Conventional Francis Turbine Representation

The representation of the Francis turbine itself is a key component of the nonlinear one-dimensional model. While the waterway model is well tested and has been thoroughly validated by experimental data [5], [21], [22] the turbine model is a complex and much less understood element. Accurate modelling of the turbine characteristics is essential for predicting the plant behaviour over the full range of operating conditions. For an ideal, lossless turbine the power output of a Francis turbine is equal to the available power, $P = \rho QgH$. In order to account for turbine inefficiencies conventional models typically employ variations of the following equation for calculating the turbine output, P_m, in per-unit form:

$$\overline{P}_m = A_t\overline{H}(\overline{Q} - \overline{Q}_{nl}) - D\overline{G}(\overline{N} - \overline{N}_R) \tag{6}$$

where $\overline{P}_m$ = per-unit mechanical output power
A_t = turbine gain factor
$\overline{Q}_{nl}$ = per-unit no load flow
D = turbine damping factor
$\overline{G}$ = per-unit guide vane function
$\overline{N}$ = per-unit turbine rotational speed
$\overline{N}_R$ = per-unit turbine rated rotational speed

The above equation assumes a number of somewhat arbitrary and unrealistic correction factors. The no-load flow, $\overline{Q}_{nl}$, is used to take account for a collection of losses including bearing friction, internal flow losses and windage losses in both the turbine and generator. A damping effect constant, D, is also included based on the speed deviation which for Francis turbines is given the arbitrary value of 0.5. The turbine gain factor constant, A_t, in some models may convert the calculated power output to turbine power in terms of the generator MVA base, or take into account the turbine gain associated with the gate position at no load.

While in the conventional models the guide vane function was assumed to vary linearly with servomotor position, this oversimplification has since been addressed by numerous authors in the literature [3], [5], [13].

The above formulation, while satisfactory in the past for many situations, does not perform well in the context of the current energy market where large set-point deviations and operation away from design conditions are becoming unavoidable, and even commonplace. Even the more complex models that rely on turbine performance characteristic curves display large errors in very low load operation, the cause of which has been attributed to a number of factors [19], which are then addressed by adjusting correction factors or altering efficiency curves to match the actual observed response.

The new turbine unit formulation described in the following section identifies the true sources of energy loss over the full turbine operating range eliminating the use of ambiguous correction factors or the need for the user to re-tune models to match observed output.

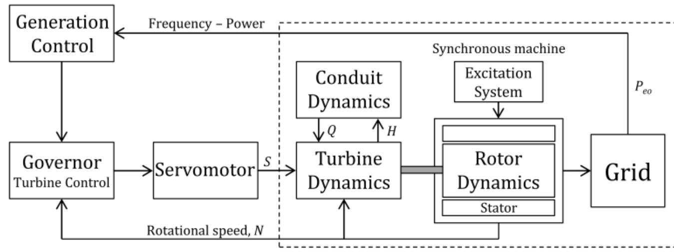

Fig. 1. Functional block diagram of the power system indicating the relationships between various system components. The components included in the present model are shown by the dotted boundary.

III. New Turbine Model Development

A more realistic turbine model, based on the actual principle of conservation of angular momentum, and the energy transfer for turbomachines, which includes loss factors of known origin that can be readily determined from field test data, is presented in the following.

The functional block diagram of a power system is given in Fig. 1 indicating the scope of the presented model. From a system point of view the fundamental control variable is the frequency set-point, while the servomotor position (controlling guide vane opening) is a dependent variable. However, in order to validate the hydraulic aspects of the new model the servomotor position is the input of primary interest as this is what physically initiates any hydraulic transient event. Servomotor position, $S(t)$, is therefore taken as a system input, recorded from actual field test measurements, while electrical power output, P_{eo}, is the output variable of primary interest.

The model was implemented in the Simulink environment, a graphical simulation package within the MATLAB commercial software platform. As with the conventional model, the turbine flow and penstock pressure are calculated and used in the computation of turbine mechanical power output. Additionally, the generator rotor dynamics are considered using the Simulink Synchronous Machine block from the SimPower library from which the rotor speed deviation is calculated and fed back into the turbine model.

The new model retains the conventional IEEE elastic waterway model, however, the turbine model is replaced by a formulation of the Euler equation with appropriate loss components as determined in the following sections.

A. Calculation of Torque from Conservation of Angular Momentum

The proposed new turbine model is based on the principle of conservation of angular momentum as expressed by the Euler turbine equation as given in (7):

$$E_{th} = \omega\left(C_{u_1}R_1 - C_{u_2}R_2\right) \tag{7}$$

where E_{th} = ideal specific energy (J kg^{-1})
ω = rotational speed (s^{-1})
C_u = tangential velocity component (m s^{-1})
R = radius (m)

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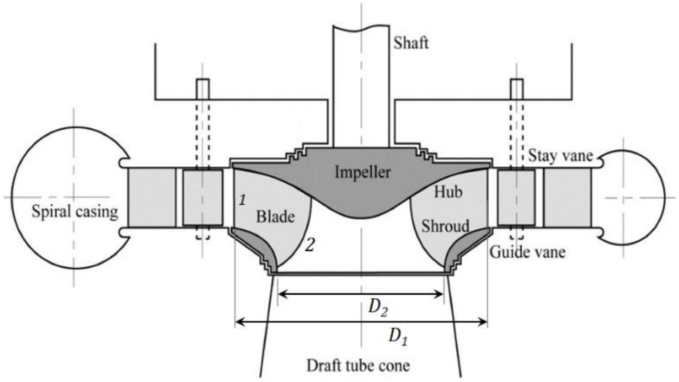

Fig. 2. Turbine section view and geometrical definitions used in the formulation of the simulation model (image adapted from [23]).

subscripts 1 and 2 representing conditions at turbine inlet and outlet respectively as illustrated in Fig. 2. In the development of the model the following assumptions were made:

i. The Euler turbine equation holds over the entire operating range,
ii. Original manufacturer scaled model test data accurately represents the installed turbine performance at the designed operating point,
iii. The turbine is considered axisymmetric, with geometrical parameters as defined in Fig. 2, and
iv. The working fluid is water in its liquid phase (i.e. the effects of cavitation on performance and pressure transient wave speed are neglected)

The model requires a set of steady-state data for calibration of model parameters of the unit to be studied that gives turbine performance, at rated head, over an adequate operational range in terms of servomotor position, flow rate, and output power. Additionally, rated speed must be known as well as some required unit geometry, namely the runner diameter at inlet mid-section, shroud outlet diameter, and diameters at the pressure measuring sections (generally spiral inlet and draft tube outlet).

B. Inlet and Outlet Turbine Blade Angles at Design

In order to calculate the energy transferred to the runner by the incoming water, the blade angle, at both inlet and outlet, must be known. In reality the shape of the turbine runner blade is highly complex and three-dimensional and will not be adequately described by a single value. Euler theory simplifies the analysis by considering a two-dimensional, axisymmetric geometry whereby the change in angular momentum between blade inlet and outlet is calculated.

The runner blade outlet angle is calculated based on the assumption of zero swirl component at best efficiency operating point. As such, the tangential component of absolute outlet flow, C_{U_2}, is reduced to zero such that $C_{m_2} = C_2$. The assumed runner blade outlet angle, β_{2_b}, can be determined from (8) and (9) considering Fig. 3.

$$C_{m_2} = Q/A_2 \tag{8}$$

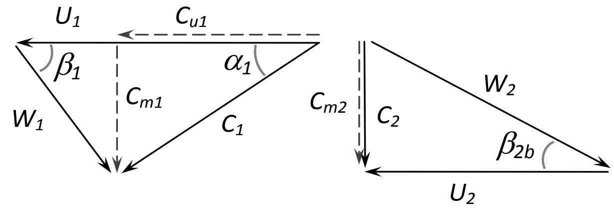

Fig. 3. Velocity triangles at turbine inlet and outlet at design where the zero outlet swirl and zero inlet incidence is assumed. Absolute (C), rotational (U) and relative (W) velocities are shown along with flow angles α and β.

$$U_2 = \left(\frac{C_{m_2}}{\tan\beta_{2_b}}\right) \tag{9}$$

Similarly, the runner blade inlet angle may be determined assuming that the flow approaches the runner blade with zero incidence at design operating point, i.e. $\beta_1 = \beta_{1_b}$. The meridional flow component, C_{m_1}, is calculated taking into account a blockage effect of the runner blade inlet thickness, t, for the z_b blades

$$C_{m_1} = Q/[B_R(\pi D_1 - z_b t)] \tag{10}$$

where B_R is the runner passage height. From the Euler equation (7) multiplied by the mass flow rate (ρQ), the peripheral component of the inlet absolute velocity, C_{U_1}, is determined by (11) based on the output power and flow from test data and the zero outlet swirl assumption. The assumed runner inlet angle, β_{1_b}, is then determined from the inlet velocity triangle as given in (12).

$$P_{th} = \rho Q\omega\left(C_{u_1}R_1 - 0\right) \tag{11}$$

$$C_{U_1} = U_1 - \left(\frac{C_{m_1}}{\tan\beta_{1_b}}\right) \tag{12}$$

Equations (9) and (12) provide a calculation of the effective blade angles at turbine outlet and inlet based on velocity triangles and Euler turbine theory, calculated at the best efficiency point. The actual blade angles may vary from those calculated and will likely also vary from hub to band.

C. Determination of Loss Components and Effective Inlet Flow Angle

The Euler turbine equation calculates the theoretical maximum specific energy transfer possible, E_{th}, based on the incoming flow velocity, flow angle and turbine blade geometry. In reality there exist a number of system losses, E_L, limiting the actual energy transfer, E_t, achievable at a given operating point.

$$E_t = g\left(H - \sum h_L\right) = E_{th} - \sum E_L \tag{13}$$

Specific energy loss components are determined according to formulae given by Ida [24] in examining the scale effects of a Francis turbine away from the optimum operating condition. Specific energy loss components are incurred within each flow field of the hydraulic turbine, from upstream pressure measurement section to draft tube exit, as considered along a representative streamline. Specific energy loss formulae are proposed for each loss type within the five identified flow domains such that the total specific energy loss may be expressed as:

$$\sum E_L = E_c + E_s + E_G + E_R + E_D \tag{14}$$

where c represents the turbine spiral case, s the stay vane ring, G the guide vane distributor, R the runner and D the turbine draft tube. Losses can be classified into being either frictional losses, or kinetic losses which include incidence loss, wake loss, residual swirl loss and draft tube diffusion and bend losses.

The incidence loss at runner inlet is based on the difference in the tangential components of the relative flow velocities (ΔW_R) entering the runner, and just beyond runner inlet, as expressed in (15).

$$E_{R_\alpha} = \xi_{R_\alpha}(\Delta W_R^2/2) \quad \text{for positive incidence} \tag{15a}$$

$$= \xi_{R_\alpha}\left[\left(\Delta W_R^2 \times \cos\alpha_{1_b}\right)^2/2\right] \quad \text{for negative incidence} \tag{15b}$$

where the loss coefficient $\xi_{R_\alpha} = 0.75$ [24].

The residual swirl flow loss is calculated using the tangential component of the absolute flow velocity at runner exit, C_{U_2}, as shown in Fig. 4 according to:

$$E_{D_u} = \xi_{D_u}\left(C_{U_2}^2/2\right) \tag{16}$$

where the loss coefficient $\xi_{D_u} = 1.0$ [24].

Minor kinetic energy losses such as wake and shock losses within the distributor, and diffusion and bend losses with the draft tube, were considered negligible.

Additionally, hydraulic friction losses are present in each of the flow zones. In general, the friction loss formulae take the form:

$$E_{i_f} \propto v_i^2/2 \tag{17}$$

where E_{i_f} is the specific energy frictional loss of the particular flow domain, i, and v_i is the characteristic velocity of the flow domain. Considering the system as a whole, the total frictional losses, $E_{tot.f}$, may be combined such that:

$$E_{tot.f} = k_f Q^2 \tag{18}$$

where k_f is the combined friction loss coefficient. Since at design operating point both the runner inlet incidence and residual outlet swirl velocity are close to zero, losses at best efficiency point can be assumed to be almost entirely due to hydraulic friction losses across the various fluid domains. As such (13) becomes:

$$\begin{aligned} E_t &= gH - E_{tot.f} \\ &= gH - k_f Q^2 \end{aligned} \tag{19}$$

where $k_f = 0.022$ for the system under consideration, calculated at best efficiency. For operation away from design, the assumed model for turbine specific energy transfer is

$$E_t = gH - \left(E_{R_\alpha} + E_{D_u} + k_f Q^2\right) \tag{20}$$

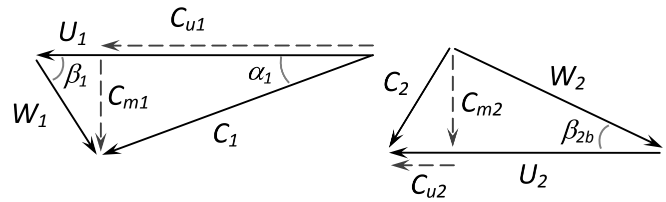

Fig. 4. Velocity triangles at turbine inlet and outlet for $Q < Q_r$. Absolute (C), rotational (U) and relative (W) velocities are shown along with flow angles α and β.

Equation (20) can therefore be solved over the entire operating range to determine the effective runner inlet flow angle as a function of servomotor stroke.

D. Estimation of Additional Operational Losses

External mechanical losses in bearings, shaft seals and wear rings are estimated by the power required to operate the unit in synchronous condenser mode – a mode of operation used to control reactive power and provide voltage support to the system. During operation in synchronous condenser mode the guide vanes are fully closed and the water is drained from the runner chamber and upper draft tube so that the machine may spin at rated speed in air. The power drawn from the grid to motor the turbine unit at rated speed is assumed to approximate the power loss due to mechanical friction, P_{fm}, and is assumed constant over the operating range.

Finally, in order to correctly represent behaviour at very low flow operation, churning losses are included into the model. Churning losses are a result of the resistance to rotation applied to the turbine runner by low momentum fluid within the runner proper. As identified by Daugherty [25] in an examination of centrifugal pumps, this loss is neither a mechanical loss, as it is clearly a function of flow through the turbine, nor is it classed as a typical hydraulic loss as this would tend toward zero as the flow rate is decreased. On the contrary, the churning loss is greatest at zero through-flow and reduces in magnitude as a more favourable flow is established within the runner as flow rate is increased. For this reason it must be considered separately. Churning losses, P_{Ch}, are determined by inspection of test data according to the difference between calculated losses and reported turbine performance at low flow [25]. This is implemented into the model as a lookup table based on flow rate.

The equation for mechanical output power, P_m, of a Francis turbine unit, assuming a constant volumetric efficiency, $\eta_v = 0.99$ [24], is therefore:

$$\begin{aligned} P_m &= P_t - P_{fm} - P_{Ch} \\ &= \rho\eta_v Q E_t - P_{fm} - P_{Ch} \\ &= \rho\eta_v Q[gH - (E_{R_\alpha} + E_{D_u} + k_f Q^2)] - P_{fm} - P_{Ch} \end{aligned} \tag{21}$$

where $P_t = P_{th} - P_L$ is the power transferred according to Equation (20). The functional block implementation of the new hydraulic turbine model is illustrated in Fig. 5.

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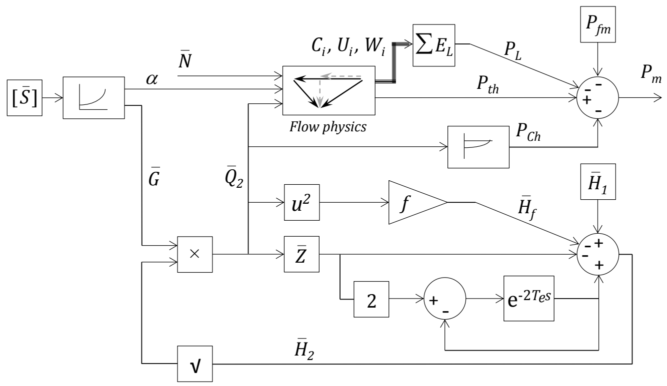

Fig. 5. Functional block implementation of the new hydraulic turbine model with standard elastic waterway model for determination of conduit dynamics.

TABLE I
HYDROPOWER PLANT SPECIFICATIONS FOR SIMULATIONS

Symbol	Parameter	Value
P_R	Rated power	119 MW
H_R	Rated head	92.0 m
Q_R	Rated discharge	142 m^3/s
N_R	Rated rotational speed	167 rpm
P_{Gen}	Synchronous generator rated output	136 MVA
V	Rated voltage	13.8 kV
ω_0	Power system frequency	50 Hz
T_w	Inelastic water time constant	1.65 s
T_e	Elastic water time constant	0.223 s
$\bar{Q}_{nl}$	Per-unit no-load flow	0.089 -
D	Turbine damping factor	0.5 -
A_t	Turbine gain factor (IEEE)	0.947 -
A_t	Turbine gain factor (Kundur)	1.32 -

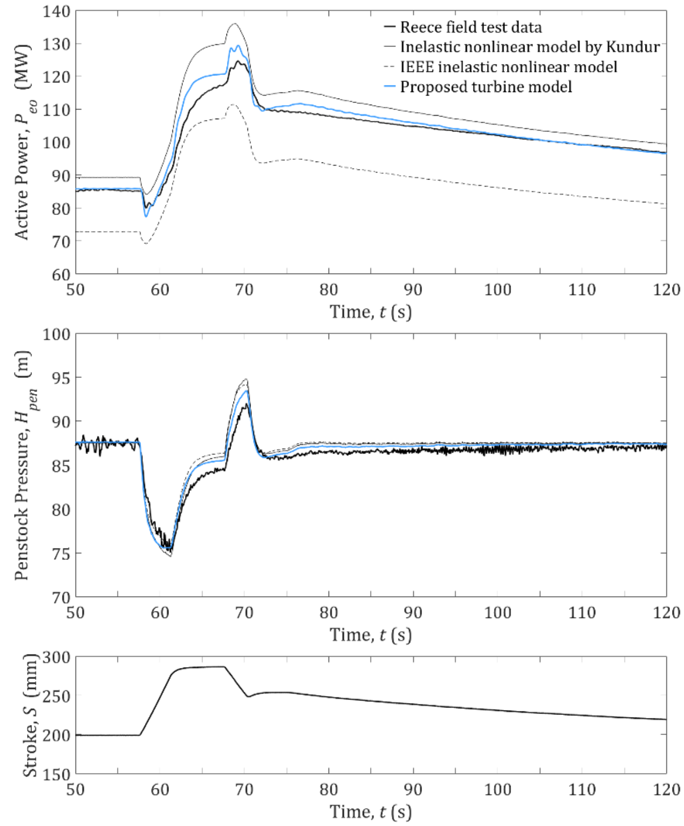

Fig. 6. Fast raise test from 86 MW (negative input frequency ramp). Simulation results are given and compared to full-scale data for active output power (top) and penstock pressure (middle) based on the measured full-scale servomotor stroke (bottom).

IV. MODEL VERIFICATION

The ability of the new turbine model to simulate the transient response of a Francis turbine hydro power station is assessed against full-scale test data for a number of fast raise and fast lower tests performed by Hydro Tasmania at the Reece Power Station. The power station was chosen, in part, for its simple hydraulic circuit – reducing the factors affecting the plant dynamic response and allowing a proper evaluation of the new turbine model formulation. The hydropower plant key specifications are given in Table 1. Additionally, simulations are performed using two well-known conventional models: the 1992 IEEE inelastic waterway model [1] and the model presented by Kundur [2]. While these are relatively simple models, they are still widely used, and provide a tested and well documented basis for comparison. All models were tuned for operation at rated conditions based on the measured active power output.

A synchronous machine model is utilised in all simulations presented in this study to more accurately simulate the electrical power response from a rapid transient event by taking into account the electro-mechanical rotor dynamics. The Synchronous Machine pu Standard block from the MATLAB SimPower library, which operates in both motor and generator modes, is used in conjunction with a standard excitation system model (Type DC1A) as described in the IEEE standard [26]. The network is represented by a 13.8 kV, 50 Hz infinite bus, approximating an infinitely stiff system [2].

The synchronous machine model enables the simulated active power output from each model to be directly compared to the active power recorded during field tests. In the case of the conventional IEEE model the speed deviation output is used in (6) while for the new model the speed deviation is fed into the Euler turbine equation (7).

Fig. 6 presents the active power output, P_{eo}, and penstock pressure, H_{pen}, during a fast raise test from 86 MW, or $0.72{\cdot}P_R$, initiated in the field tests by a negative input frequency ramp resulting in the measured servomotor position as given in the lower plot. Fig. 7 and Fig. 8 present similar results during fast lower and fast raise tests, respectively, from very low load turbine operation.

The simulation results show that the new model performs well over the entire operating range, without the requirement to re-tune the model at the operating point of interest. Conversely, the two conventional models, without re-tuning, display large errors in steady-state value prior to the transient events, which increase as the operating point moves away from design. While these would generally be adjusted, the adjustments performed would be achieved by arbitrarily increasing or decreasing various gain or dampening factors which could potentially lead to misleading simulation results, and do not realistically simulate the actual physical process occurring within the turbine. In general terms, the transient response in all cases is well predicted, although the model by Kundur does tend to systematically over predict the transient response, while the new model shows consistent agreement.

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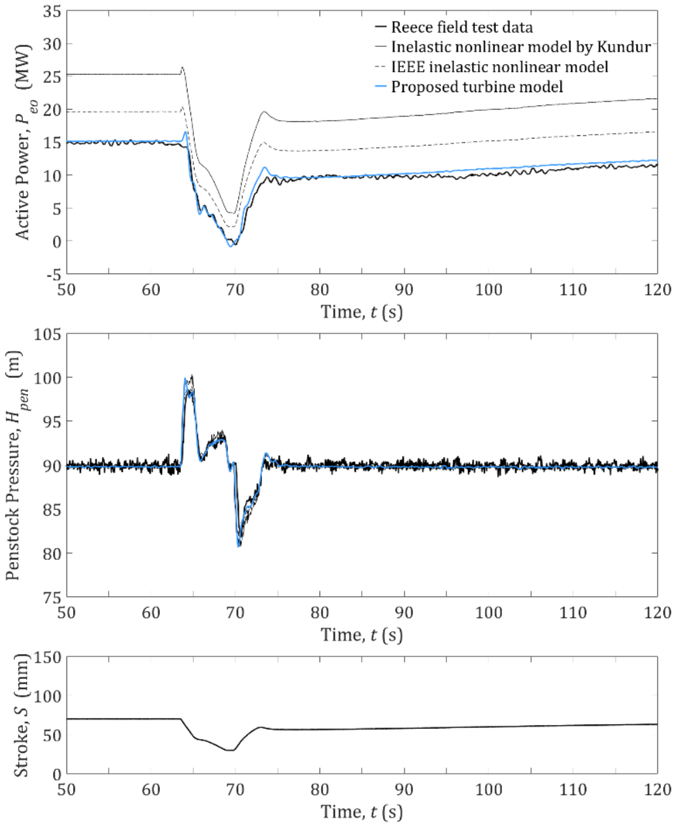

Fig. 7. Fast lower test from 15 MW (positive input frequency ramp). Simulation results are given and compared to full-scale data for active output power (top) and penstock pressure (middle) based on the measured full-scale servomotor stroke (bottom).

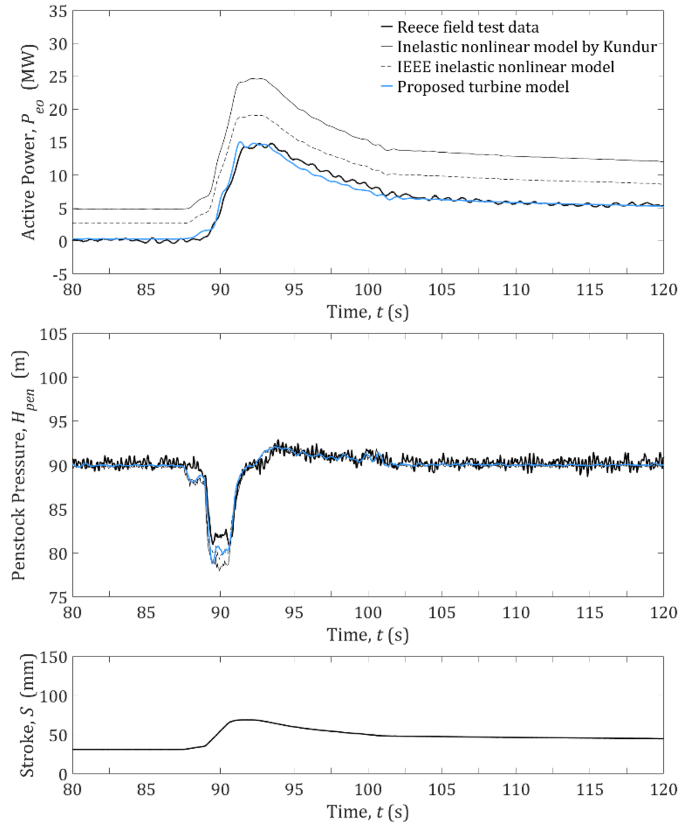

Fig. 8. Fast raise test to 15 MW from no-load (negative frequency ramp). Simulation results are given and compared to full-scale data for active output power (top) and penstock pressure (middle) based on the measured full-scale servomotor stroke (bottom).

It may be noted that the new model does tend to over predict the initial response following a rapid guide vane movement and can be seen to be related to an associated over-prediction of penstock pressure response, in fact this is a characteristic of each of the simulation models. This is due to the servomotor recording being used as a system input variable which is fed directly into the waterway model. In reality, this linear movement of the servomotor arm does not instantaneously translate into a guide vane movement as a degree of backlash is inevitably present in the guide vane linkage mechanism [13]. Additionally, the presence of cavitation and the associated increase in water column compliance, particularly at low load, may also contribute to the machine response.

The two conventional models are still widely used throughout the hydropower industry, however, they are often used as a base model and various modifications are made [5]. Regardless, the new formulation provides a relatively simple and much more physically realistic model that is thoroughly verified by full-scale test data.

The development of the model requires sufficient steady-state test data that gives turbine performance at rated head over an adequate operational range in terms of guide vane position, turbine discharge and output power.

From this, and some basic turbine geometry measurements; namely the mid-blade inlet diameter, outlet diameter and inlet blade height; effective inlet flow angles may be calculated over the entire range. In comparison to other hydraulic transient analysis techniques, such as a method of characteristics analysis, the model requires relatively little information to set up and may also easily be adapted for use in other hydropower plants, hydraulic configurations, and for other types of machines such as Kaplan and Pelton turbines.

V. Conclusion

A one-dimensional numerical model of a Francis turbine power plant for use in system dynamic studies is presented. The turbine unit - a key component of hydraulic models that has remained drastically simplified, is replaced by considering effective inlet and outlet velocity vectors and calculating machine output based on the conservation of angular momentum. Furthermore, individual loss components such as blade incidence loss, residual swirl flow loss and churning loss are calculated across the full operating range.

The new turbine model is validated against field data of fast raise and fast lower tests at conditions away from design. A major improvement in comparison with conventional models is seen in the simulation performance at off-design conditions owing to the realistic treatment of turbine flow physics.

By being able to accurately simulate the dynamic behavior of hydraulic turbines at off-design conditions, with quantifiable sources of loss and inefficiency, operators are able to gain a deeper understanding of the true dynamic capabilities of a given power plant – an issue that is becoming increasingly valuable in the current market.

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VII. Biographies

Dean Giosio is a Postdoctoral Research Fellow at the Australian Maritime College, a specialist institute of the University of Tasmania.

Dr Giosio graduated from the University of Tasmania in 2009 with a combined Bachelor of Science (Chemistry) and Engineering (Mechanical) with First Class Honours. He completed a Doctor of Philosophy from the University of Tasmania investigating the unsteady operation and rapid start-up ability of Francis turbines in 2015. His research interests include renewable energy and power generation, environmental and development engineering as well as fundamental fluid dynamics.

Alan Henderson is a Senior Lecturer in Mechanical Engineering at the School of Engineering and ICT, University of Tasmania.

Dr Henderson is Program Leader for Renewable Energy at the Centre of Renewable Energy and Power Systems. His research interests in fluid mechanics include applications in renewable energy, turbomachinery and biofouling. Dr Henderson graduated with a Bachelor of Engineering (Mechanical) with First Class Honours in 1997 and a Doctor of Philosophy from the University of Tasmania in 2006. He has worked as an engineer in the manufacturing, aerospace and energy sectors.

Jessica Walker is a Lecturer in Ocean Engineering at the National Centre for Maritime Engineering and Hydrodynamics at the Australian Maritime College, an institute of the University of Tasmania.

She was a 2012 Fullbright Postdoctoral Scholar and spent 12 months researching the impacts of roughness and biofouling on tidal turbine performance at the United States Naval Academy in Annapolis, Maryland. She graduated with a Bachelor of Engineering (Civil) with First Class Honours and a Doctor of Philosophy from the University of Tasmania in 2005 and 2010 respectively. She specialises in fluid dynamics and biofouling research with renewable energy and energy efficiency applications.

Paul Brandner is an Associate Professor of hydrodynamics at the Australian Maritime College, a specialist institute of the University of Tasmania.

He is Research Leader and Facility Manager of the Cavitaiton Research Laboaratory (CRL) at AMC. He obtained a Bachelor of Engineering with Honours (1991) and a Doctor of Philosophy (1996) from the University of Tasmania. He has worked in Naval Hydrodynamics since 1991 and from 1996 specialised in cavitation and bubbly flows with interests in basic cavitation physics including cavitation dynamics, nucleation and supercavitation; and more recently in fluid-structure interaction and hydroacoustics. He led the development of the CRL from 2001 funded under the Australian AusIndustry Major National Research Facilities Program.