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Modelling of Transient Behaviour in a Francis Turbine Power Plant

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Abstract

This paper presents a nonlinear mathematical model of the Francis turbine for a single-machine hydroelectric power plant. Several model refinements have been proposed to improve the capability of the existing industry models to simulate the transient operations of the power station. The new model is evaluated by full-scale field tests involving both steady and transient operations. Significant improvement in accuracy is demonstrated. However, there remain some frequency-dependent discrepancies for short penstock installation that appear to be associated with unsteady flow within the turbine.

Introduction

The increasing interconnection of individual power systems into major grids has imposed more stringent quality assurance requirements on the modelling of power plants. Power systems are nowadays operated closer to capacity limits than in the past. Hence, a review of the commonly used models for the hydraulic systems in the hydroelectric power plant is warranted to accurately identify and minimise transient stability problems. This is particularly relevant for islanding, load rejection and black start after power system restoration cases where large changes in the power output or system frequency are expected.

The commercial PSS/E package [5], which is commonly used to simulate the behaviour of the hydroelectric power plant, involves both hydraulic and electrical system components. It uses a conventional turbine model developed by authors from the Institute of Electrical & Electronics Engineers (IEEE) [10]. The current study is specifically concerned with the hydraulic modelling aspects of the Francis type reaction turbine incorporated in the PSS/E package. The IEEE model is improved to incorporate a nonlinear model, which is used to examine the transient phenomena associated with changing turbine load to meet fluctuating system demand.

The present paper will focus on the operation of a simple power plant with a single Francis turbine and a short penstock. This eliminates the need to consider travelling pressure wave phenomena in a long waterway conduit and the problem of more complex governor and hydraulic interactions that frequently occur in multiple-machine stations. Significant elements of the hydraulic model developed here are:

1. nonlinear modelling of Francis turbine characteristics;
2. allowance for water column inertia and unsteady flow effects in the turbine and draft tube;
3. nonlinear Guide Vane (GV) function for Inlet Guide Vane (IGV) operation;
4. correct allowance for effects of changing turbine speed and supply head.

Prediction of the original and improved IEEE models using Matlab Simulink software will be compared with the results of full-scale field tests on the Mackintosh power station conducted by Hydro Tasmania. Details of the Mackintosh power station are illustrated in Figure 1. The plant has a short penstock, unrestricted reservoir and tailrace, and no surge chamber. The turbine flow or power output is controlled by hydraulically operated guide vanes.

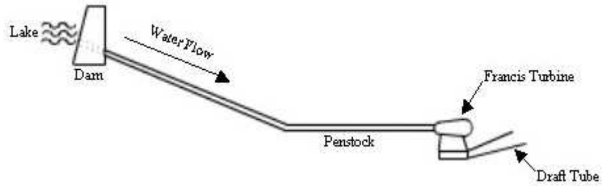

Figure 1: Scheme of the Mackintosh hydro power plant.

Description of the Power Plant Model

Conventional IEEE Model

The linearized equations originally designed for implementation on analogue computers are still widely used in the power industry. They are suitable only for investigation of small power system perturbations or for first swing stability studies. Nonlinear simulations have been increasingly utilized from the early 1990s [1,2,10] with the availability of greater computing power and the demands of more complex power system distribution grids. Although a nonlinear IEEE model [10] as shown in Figure 2 has been introduced in the time domain simulations, it has oversimplified some important features of the hydraulic system.

For a short-penstock, single-machine station where travelling pressure wave (water hammer) effects are relatively insignificant, the inelastic water column theory using the linear momentum equation for incompressible flow is usually applied in the waterway conduit:

$$\overline{Q} = \frac{1}{T_w} \int \left(\overline{H}_o - \overline{H} - \overline{H}_f \right) dt \qquad (1)$$

where $\overline{Q}$ = per-unit turbine flow
$\overline{H}_o$ = per-unit static head between reservoir and tailrace
$\overline{H}$ = per-unit static head at the turbine admission
$\overline{H}_f$ = per-unit conduit head losses
T_W = water time constant = $\sum Q_{rated} L_i / g A_i h_{rated}$
L_i = length of the conduit section i
A_i = area of the conduit section i
g = gravitational acceleration
Q_{rated} = rated flow rate
h_{rated} = rated head

The conduit head losses in equation (1) were usually ignored in the IEEE model for simplicity [10]. These losses could easily amount to around 5% of the total available head at rated flow and are not always a constant even for a simple hydro plant such as Mackintosh. Hence, the inclusion of the conduit losses is considered desirable [9].

No provision is made in the inelastic model to account for unsteady flow effects in the turbine and draft tube caused by changing GV position. Although these effects may be insignificant for a station with a relatively long penstock, they will be more important for station like Mackintosh where the water column inertia is small.

In this generic model, the Francis turbine is depicted as an orifice with constant discharge coefficient for a particular guide vane setting [6,9]. The flow rate through the turbine is modelled by a simple orifice flow relation:

$$\overline{Q} = \overline{G}\sqrt{\overline{H}} \qquad (2)$$

The guide vane (GV) function $\overline{G}$ in the existing model [10] is assumed to vary linearly with the guide vane opening only. In reality, the slope of this function will vary with flow coefficient and Reynolds number over the full range of turbine operations [4] and it should properly be modelled as a nonlinear function. A similar approach is implemented in the 1994 model of De Jaeger et al. [1].

The turbine power output for the IEEE model is evaluated from:

$$\overline{P_m} = A_t \overline{H}\left(\overline{Q} - \overline{Q_{nl}}\right) - D\overline{G}\left(\overline{N} - \overline{N}_{rated}\right) \qquad (3)$$

where $\overline{P_m}$ = per-unit turbine power output

A_t = turbine gain factor

$\overline{Q_{nl}}$ = per-unit no-load flow

D = speed-damping factor

$\overline{N}$ = per-unit turbine rotational speed

$\overline{N}_{rated}$ =per-unit rated turbine rotational speed

The no-load flow $\overline{Q_{nl}}$ is used to correct for the bearing friction and the windage losses in both turbine and generator [2]. The turbine gain factor A_t allows for other internal flow losses. However, the resulting linearized model is not very accurate [9]. The damping factor D in the IEEE model is introduced to allow for efficiency changes resulting from varied operating conditions; a constant value of D=0.5 has been used for Francis turbine modelling [10].

Problems with Existing IEEE Model

The current IEEE model does not use dimensionless turbine characteristics. Equation (3) is inappropriate and could lead to significant error when the change in turbine operating conditions is large. In particular, the speed-damping factor *D* used in the model is unrealistic for the Francis turbine operation. The power (and the efficiency) change with speed may be positive or negative depending on the GV position, and their rates of change also vary with GV position [4].

Damping effects due to head changes are also neglected in the existing model. In fact, changing the turbine net head (H) will also change the flow rate of the machine (Q). At a constant turbine speed (N), this also changes the flow coefficient $C_Q \propto Q/N$ and moves to a different turbine operating point and efficiency. The magnitude is similar to the speed damping effect and must be taken into account in the simulation [4].

Hence, dimensionless turbine performance curves should be employed to correctly represent the hydraulic turbine operation. Figure 3 shows a typical efficiency curve for the Francis turbine. For incompressible flow, the turbine operation is accurately described by the dimensionless relation:

$$C_H = f(C_Q, \mathrm{Re}) \qquad (4)$$

where C_Q= flow coefficient = Q / Nd^3

C_H= head coefficient= $gH / N^2 d^2$

Re= Reynolds number = $4Q / \pi \nu d$

d= characteristic turbine diameter

ν= dynamic viscosity of water

Changes with *Re* are relatively slow and for small variations in *Re* the turbine performance can be approximated by:

$$= f(C_Q) \qquad (5)$$

The net turbine head may vary due to transients or changes in the supply head. Similar operating conditions (C_Q, C_H constant) with varying speed require that $H \propto N^2$, $Q \propto N$ and therefore $Q \propto H^{0.5}$, as assumed in equation (2). This is incorrect for a power plant that has been governed to maintain a constant runner speed in order to keep the AC frequency constant within the grid, in which case C_Q must vary with H for GV fixed.

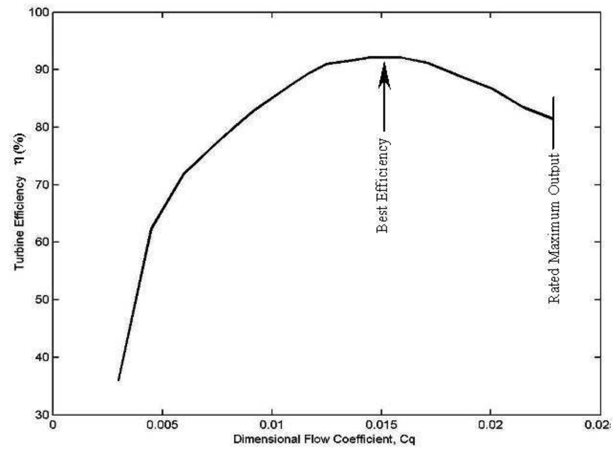

Figure 3: Typical efficiency curve ($\eta \sim C_Q$) for Francis Turbine.

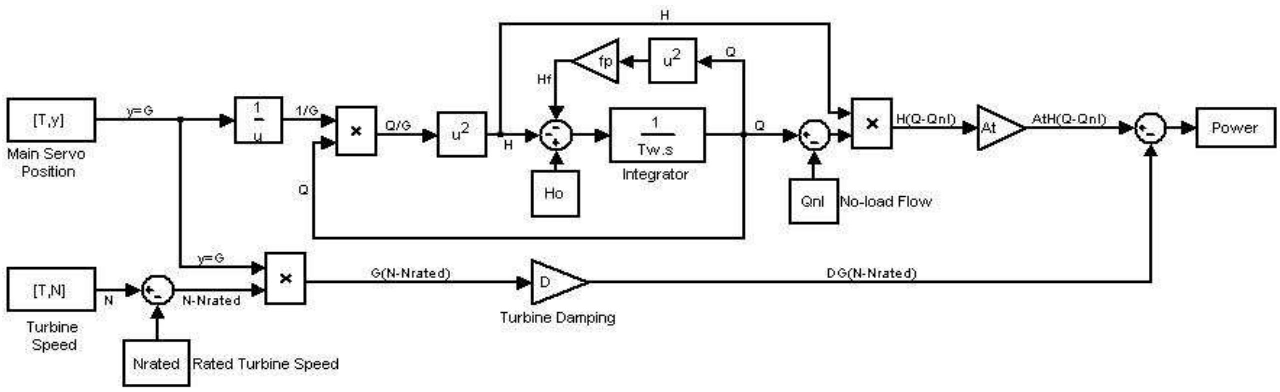

Figure 2: Block diagram for 1992 nonlinear IEEE turbine model [10].

New Features of the Proposed Model

The earlier IEEE model illustrated in Figure 3, with its simplified turbine and guide vane characteristics, could not adequately represent all the transient behaviour observed in the field tests. Such simplifications are no longer necessary with modern computing power. Thus, additional nonlinear features have been adopted here to improve accuracy of the turbine model.

1. A lookup table is included in the model to implement a nonlinear GV function. The table combines two nonlinear relationships: the GV angle varies nonlinearly with the main servo movement; and the GV function varies nonlinearly with the GV movement. A quadratic term is introduced to provide a simple non-linear relation between flow and gate opening. This term can be tuned to match the observed steady state power output.

2. A lookup table for the efficiency vs. flow coefficient is used to replace the turbine gain and damping factor. This procedure incorporates damping effects due to both speed and head changes as well as the losses in the turbine. The lookup table is constructed using a combination of data from full-scale steady-state tests, simulations and model test results. No further correction for variation from rated head is required with this arrangement [4].

3. A first order filter block (gate time delay) can be included to model the unsteady effects associated with gate movement. It has not been used in the present work, but will be implemented later when adequate data becomes available from computational studies, field tests or laboratory model tests.

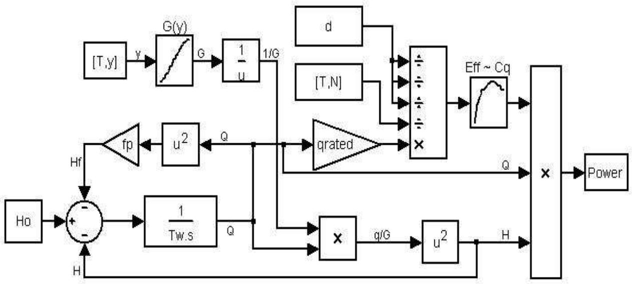

Figure 4: Block diagram for new proposed turbine & waterway model.

Field Test Procedure

A test program was developed in cooperation with Hydro Tasmania to evaluate the improved turbine and waterway model for the single-machine Mackintosh power station. The tests consisted of frequency deviation tests, Nyquist tests and the steady-state measurements [7]. The power output, main servo position, generator frequency (or turbine speed), and the static pressure at turbine admission were recorded during the tests [8].

The frequency deviation and Nyquist tests give a quantitative measure of the plant behaviour if the generator is supplying an isolated load [7]. A large injected signal to the governor is applied in the frequency deviation test to cause a large step change in the guide vane position. A smaller oscillatory signal is injected in the Nyquist test to move the guide vane sinusoidally about a given average position. This is repeated at various frequencies. Steady-state measurements were carried out to obtain the turbine characteristics with respect to the change in guide vane position [8]. The test results are used in combination with the model test data to determine the characteristic curves of the Francis turbine. Due to the influence of the remainder of the power system, it was not possible to vary the machine speed during field tests. Testing these aspects would require laboratory model tests or a full-scale machine isolated from the grid.

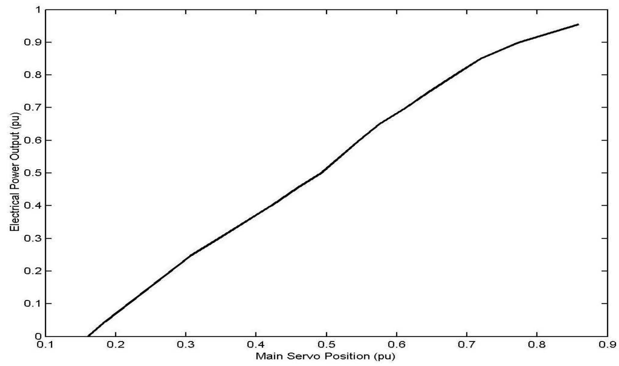

Figure 5: Steady state test measurement for Mackintosh power station.

Modelling and Simulation

A Matlab/Simulink program was used for testing of the new turbine model. The Simulink code can readily be translated into the Fortran-based PSS/E package used for predicting overall power system response to disturbances. Hydraulic parameters for the original and improved turbine models are listed in Table 1.

Description of the Model Parameter	Value
Rated flow rate, Q_{rated} (m^3s^{-1})	149.7
Rated power output, P_{rated} (MW)	79.9
Rated speed, N_{rated} (rpm)	166.7
Rated head, h_{rated} (m)	61
Water time constant, T_w (s)	3.16
Conduit head loss coefficient, f_p	0.0004
Damping factor, D	0.50
Turbine Gain, A_t	1.48
No-load Unit flow, Q_{nl}	0.16

Table 1: Hydraulic parameters for the original and improved turbine models of the Mackintosh power station.

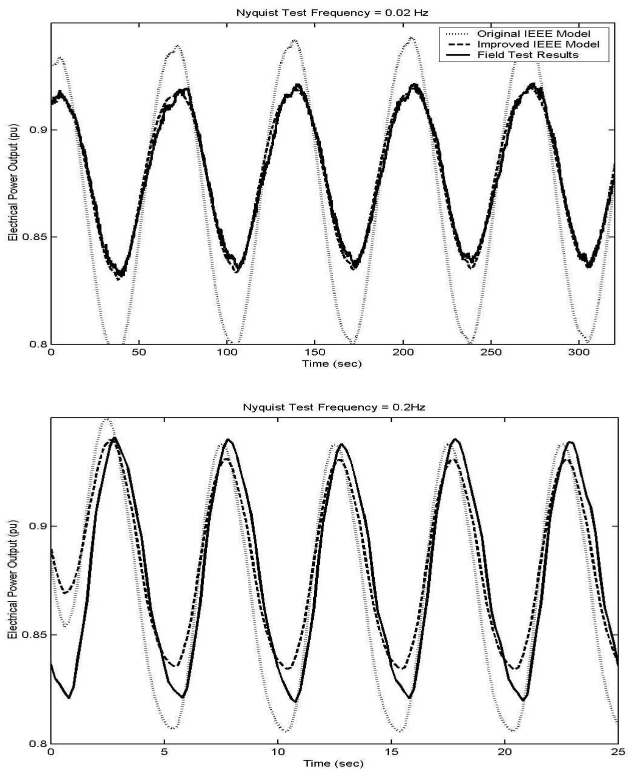

Figure 6: Simulated and measured responses of the Mackintosh power station. Nyquist tests are performed at test frequencies of 0.02 Hz (low speed) and 0.2 Hz (high speed) respectively. Available static head is 61m.

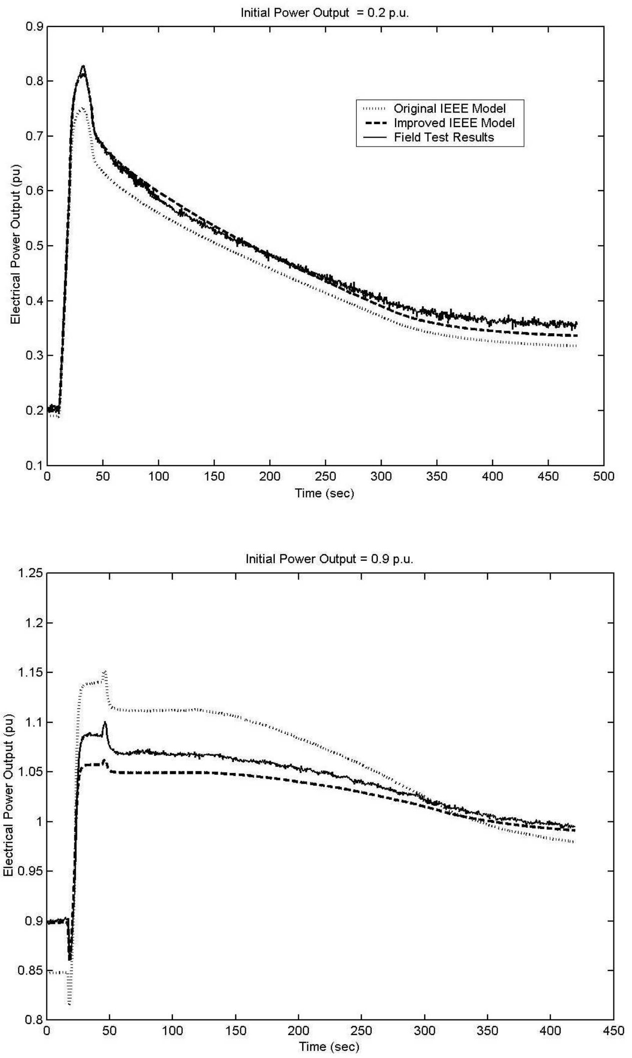

Figure 7: Simulated and measured responses of the Mackintosh power station following a step change in the load. The tests are conducted at low and high initial power outputs respectively. Available static head is 65m.

As shown in Figures 6 and 7, the new model has better simulated the magnitude of power fluctuations when the plant is subjected to a frequency disturbance. The improvements are more obvious when the turbine is operating at high load and the guide vane is moving at a faster rate. However, the new model still shows a retraceable phase lag between the measured and the simulated power outputs, which increases in magnitude with guide vane oscillation frequency.

The well-tested electro-mechanical model for the governor operation is unlikely to have been a significant cause of error. The remaining discrepancies are most likely due to unsteady flow effects in the Francis turbine. In general, the flow pattern in the Francis turbine does not change instantaneously with the gate movement and thus a time lag in flow establishment through the runner and draft tube may occur. The lag may change as the operating condition of the machine changes [4].

This unsteady effect, however, should not be such a significant problem for power stations with relatively long waterway conduits and high water inertia [4]. The inertia effect of the water column in such cases is expected to dominate any unsteady flow effects of the Francis turbine operation. Hence, unsteady flow studies should be focused on the stations with relatively short penstocks. This is the subject of the ongoing research.

Conclusions

An improved nonlinear turbine and waterway model suitable for Francis turbine operation has been proposed. Comparisons between simulation and full-scale test results have demonstrated significant improvements in accuracy. However, there remain some frequency-dependent discrepancies for short penstock installation that appear to be associated with unsteady flow within the turbine.

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