

Modelling of Francis Turbine for Solving Parametric Discrepancies

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Abstract—This research work aimed at optimizing tuning parameters for the regulation of speed and power of Francis turbine. Here prototype calculations are used to model the Francis turbine. Here PID controllers are used to optimize the parameters of torque, power and time. Block diagram with control functions of turbine, actuator, generator and PID controller are employed here. Response of tuning of parameters of power and time are noted here for optimal models. Finally slow runner Francis turbine has selected for running with optimal speed and power.

Index Terms— Turbine, Optimal, Model, Torque.

I. INTRODUCTION

In the world scenario of hydroelectric power plants, Francis turbines are used in profuse. They can intake and discharge very large quantity of water than other type of turbines. It occupies speed governors which control the flow and speed of water entering the turbine. Along with this, hydroelectric controllers are also employed. Recently new techniques are under innovation to improve the regulation of speed and power of the hydroelectric turbines.

Evolutionary programming method to determine the optimized parameters of a PID controller of a speed governor for power hydroelectric turbines was proposed by Jiang, Ma and Wang (2006).

Qian, Yi and Liu (2011) applied sliding mode control (SMC). Also Ding and Sinha (2011) combined SMC and control techniques and this proposed control technique improves system performance against lead disturbances and parametric uncertainties with significant advantage. Liu, Li and Huang propose a robust nonlinear controller based on a high gain observer. Here only one control parameter needs to be tuned. The simulation showed that this method can provide a good and robust dynamic response. In the work of Hamarasheed, Haris and Nopiah (2012), they present adaptive multiple control technique which uses LQG/LTR and PI controllers. It has showed here that for smooth changes of response it exhibits oscillating response in situation having sudden changes.

In the work of Singh, Naresh and Gupta (2013), genetic algorithm was used to determine the tuning parameters of a controller for flow and velocity regulation of hydraulic turbines. In their research paper; Anbo, Xiangang and Hao (2013) presented distribution of multi agent genetic algorithm for optimizing parameters of a self-adaptive PID in World's largest hydroelectric power plant located in China.

$$P_n/g = (1-sT_w)/(1+sT_w/2) \quad (2)$$

The block diagram in figure 4 also represent the turbine transfer function of equation (2). The auxiliary variable x was chosen at output of the integrator so that the turbine can be fully represented in state space by means of equations (9) and (12).

As described by Kundur (1993), mechanical torque is being received on the generator from the turbine by the rotor shaft and produces an electromagnetic torque. In a simplified manner, the dynamics of the generator is of first order

$$\omega/(M_m - M_e) = i/(K_d + sT_m) \quad (3)$$

where M_m is the mechanical torque and M_e is the equivalent torque related to the electrical load. For the frequency and load study, it expresses an equation (3) in terms of mechanical and electrical power, rather than torque. Based on the work of Kundur (1993) and Machowski, Bialek and Bumby(2008), the torque deviation $(M_m - M_e)$ is equivalent to the power deviation $(P_m - P_e)$. Thus, the model of the generator can be represented by the block diagram in figure 5 so this model results in equation (10).

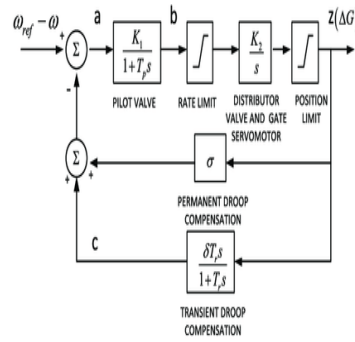


Figure 4: Detailed block diagram of the turbine

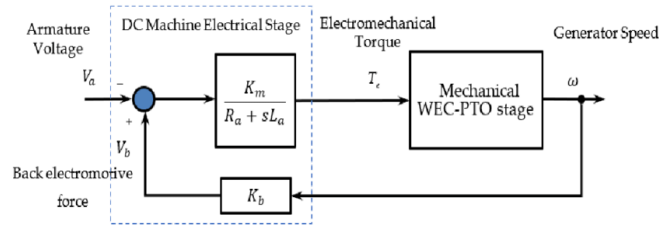


Figure 5: Detailed block diagram of the generator

In real time applications, the controller is typically a PID compensator whose transfer function is

$$u/(\omega_r - \omega) = K_p + K_i/s + K_d s/(T_d s + 1) \quad (4)$$

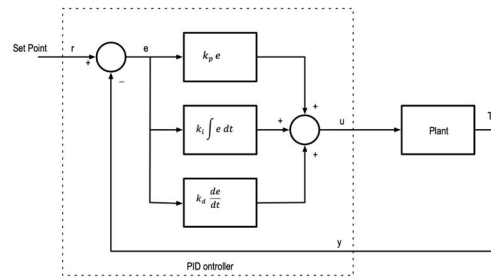


Figure 6: Detailed block diagram of the PID controller

The transfer function of PID controller is given by

$$C_{pid}(s)/E(s) = K_p + K_i/s + K_d s$$

$$\text{It can be simplified as } C_{pid}(s)/E(s) = K_p(1 + 1/T_i s + T_d s)$$

Where $T_i = K_p/K_i$ and $T_d = K_d/K_p$

From the research work of Roberto Canonico, Renato A Aguiar and Fabrizio Leonardi (2015), Equations (5) to (12) summarize the system model

$$Z = K_i(\omega_r - \omega) \quad (5)$$

$$V = 1/T_d((\omega_r - \omega) - v) \quad (6)$$

$$g = i/T_g(u - g) \quad (7)$$

$$2P_m + T_w P_m = 2(g - T_w g) \quad (8)$$

$$X = I/T_w(g - P_m) \quad (9)$$

$$\omega = 1/T_m((P_m - P_c) - K_d \omega) \quad (10)$$

$$u = K_p(\omega_r - \omega) + (z) + K_d(v) \quad (11)$$

$$P_m = 2(x - g) \quad (12)$$

III. OPTIMAL CONTROL

The numerical solution of an optimal control problem can be classified into two methods, the direct and indirect one. The indirect method is inefficient as in the case described by Murty(2008) where an application of the optimal control was designed to control the frequency of a hydraulic turbine constituted by one generator unit. This can be solved as a linear regulator problem by means of the algebraic matrix Riccati

In the direct method it is necessary to transform the original problem into non linear programming problem by means of the discretization of the dstate and control.

Direct methods can be classified into two types.

1. Discretization of both the state variables and control and then the model is solved by an algorithm for nonlinear programming problem. This method is highlighted in the work of Subchan and Zbikowsky(2009) and Betts(2010). This method has a large domain of convergence in Stryk and Schlemmer (1994).
2. Discretization of control can be obtained by numerical integration of the state equations. It is called as shooting technique. It gives high accurate solution in Stryk and Schlemmer (1994).

In this work, PROPT (Rutquist and Edval(2010) automates the conversion of an optimal control problem into an equivalent problem of parametric optimization to be solved by Sparse Nonlinear Optimization (SNOPT).

Considering a quadratic objective function, optimal control problem can be written as

$$\min J = \int_0^t (\omega_r - \omega)^2 + \lambda (g)^2 dt \quad (13)$$

ω is the rotor speed and λ represents balance between control and speed.

The system is subjected to dynamic constraints of equations (5) to (10), the algebraic constraints of equations of equations (11) to (12) and operational constraint(14).

It was found out that global minimum depends on

$$g_{\min} \leq (g) \leq g_{\max} \quad (14)$$

The minimum and maximum aperture constraints are physical limits. The inverse response of turbine denotes non minimum phase zero in its transfer function. The consequence is with the position of wicket gates, the flow may not change because of inertia of moving fluid. The pressure only changes. When forces are balanced, pressure goes to steady state value.

Based on the work of Kundur(1993) T_w values are within 0.5 to 4.0 s. T_g are close to 0.5s. T_m values are within the range of 5.0 to 10 s as seen in Qian, Yi and Liu(2011), Fang and Shen (2005). The nominal values of the plant parameters used are selected based on the work of Kundur(1993) and noted in Table 1. $T_d = 0.1$ s was suggested by Fang and Shen(2005).

TABLE 1-PLANT PARAMETERS

T_g	T_w	T_m	K_d
0.5s	2.0s	10s	1

The design method proposed in this research work is not like tuning method by Hagihara et.al.(1979)

The suggested values are

$$K_v=0.8T_m/T_w \quad (15)$$

$$K_i=0.24T_m/T_w^2 \quad (16)$$

$$K_d=0.27T_m \quad (17)$$

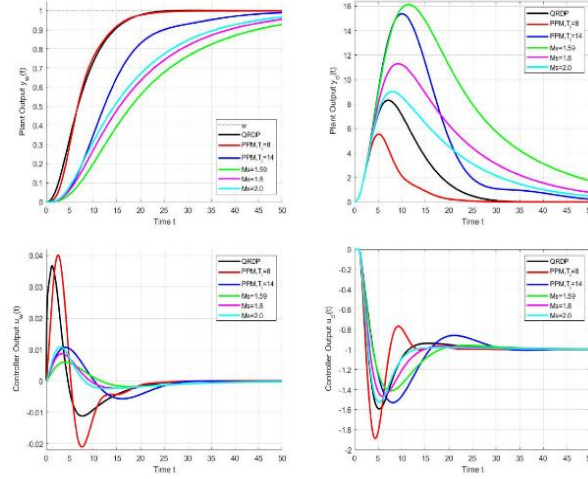


Figure 7: Time Response with Optimal PID

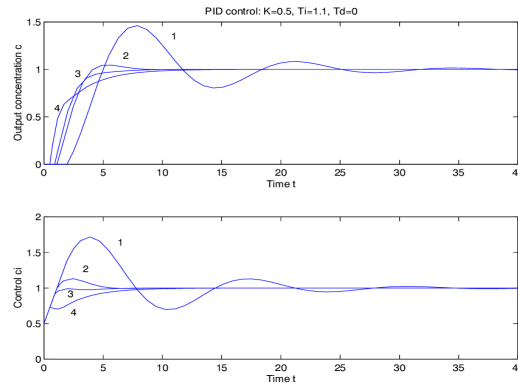


Figure 8: Response for mild and severe constraints

IV. DESIGN VIA SHOOTING

The optimization method of integration of differential equations are used for comparison. A new state variable, α was introduced in the dynamic model.

The method proposed here presents a high sensitivity, because a small change in the parameters at the beginning of the solution may result in large differences at the end of integration. Shooting method was quite sensitive to the initial estimate and only for values close to the optimal values there was convergence in the response.

V. MODELLING OF FRANCIS TURBINE

Modelling of Francis turbine was carried out in the research work of R K Tyagi (2012). For very rough and tidy operations, 20% to 35% of the rated discharge can be allowed. At lower output, strengthening mechanism have to be done in the draft tube below the runner discharge to minimize disturbance magnitude. Reduced efficiency noted at higher loads. When discharge % exceeds boundary limit, generator cannot attain maximum power. When % discharge is below lower limit, excess water is being bypassed and there is no power generation. When the head is above or below the boundary, there is no power produced. Francis turbine attains head between 40-200m. In optimal conditions, select turbine runner of high specific speed to generate more power for same head. At low heads, size of the alternator and the power house small and with economy. The vertical machines are

advantageous over horizontal turbines with variations of tail race water level. For horizontal machines, floor of turbine house should be at higher level because lower part of turbines house may be water tight.

Major dimensions for the Prototype are: Shaft Diameter = 10mm Shaft Length = 2 ft Turbine Blades = 15 Blade Length = 3 inches Nut & Bolt = 15mm Draft tube Length = 100mm Width at Runner Exit = 30mm Width at Tail race = 45mm Width B of the wheel = 35mm Diameter = 75mm Thickness of Blade = 3mm For 1 hour of operation, Francis turbine gives

Rated Discharge %	Rated Generator capacity
40	10
50	18
60	25
70	32
85	42

Type of turbine: Vertical Francis production capacity 5×90MW

Net Head	
Rated	121.92
Design	121.92
Maximum	156.06
Minimum	87.79

Turbine Output

Rated Head	112000
At Design head	112000
At maximum head	112000
At minimum head	53000
Rated discharge	102.3

Runner nominal diameter=4151mm

Rated speed=166.7rp

Output %	Gate Opening	Efficiency %
110	93.5	93.5
100	85.0	91.5
80	71.8	88
60	59.0	83

Power Developed = $\eta \rho g Q H$, $F_x = \rho a V_1 (V_{w1} + V_{w2})$

$V_1 = \sqrt{2gH}$

$W = F_x \times u$, Kinetic Energy = $\frac{1}{2} m V^2$, $m = \rho A V_1$ $\eta = (\rho A V_1 (V_{w1} + V_{w2}) \times u) / (\frac{1}{2} \rho A V_1^3)$

VI. PROTOTYPE CALCULATION

$\eta = B/D$ B-Width, D-Diameter = 35/75=0.45 H= 60mm $V_f = \Psi \times \sqrt{2gH} = 0.30 \times 32.5 = 9.75$ mm/s

$U = K_u \times \sqrt{2gH} = 0.8 \times 32.5 = 26$ mm/s $Q = \pi (D_1 - 2t) h V_{a1}$;

$V_{a1} = V_f \sin \alpha$; $\alpha = 37.5^\circ = \pi (75 - 2 \times 3) \times 18.75 \times V_f \sin 37.5^\circ h = 0.25 D_1 = \pi \times 69 \times 18.75 \times 12.25 \times 0.61 = 30400 \text{ mm}^3/\text{s}$

$P_o = \eta_o \times w Q H = 0.91 \times 1000 \times 9.81 \times 30400 \times 60 = 16.28 \text{ kW}$

$Pr = 10 \times h \times Q \times \eta \times t \times \eta_g = 10 \times 60 \times 0.92 \times 0.7 \times 30400 \times 1 = 11.62 \text{ kW}$

$P = P_o \times Pr \times 0.01 = 16.28 \times 11.62 \times 0.01 = 1.89 \text{ kW}$ P at 50% load = 0.95kW P at 25% load = 0.47kW

$$N_s = N\sqrt{P/H^{5/4}} = 75 \times \sqrt{1.89/1.04} = 97$$

This Specific speed is defined for a slow runner Francis Turbine.
Specific Speed

60-120	Slow Runner
120-180	Normal Runner
180-300	Fast Runner

Fast Runner Speed N can be 75-1000 rpm and head can be 40m- 600m for the model turbine.

VII. CONCLUSION

The work analysed the optimal control technique to optimize tuning parameters of a controller used in the regulation of speed and power of hydroelectric turbines. Prototype calculation and modelling carried out to assist the Francis turbine with wicket gates and also maximum opening speed. The findings were compatible. The numerical results suggest this technique is well suited to control system design for hydroelectric turbines. Here parametric discrepancies are corrected with prototype calculations.

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