

Modeling Hydro Power Plants and Tuning Hydro Governors as an Educational Guideline

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Abstract – Appropriate modeling of components and related controllers are very significant in studying dynamic performance of power systems. In this paper, an educational procedure for modeling, simulation, and governor tuning of hydro power plants is presented. Different existing dynamic models of hydro plant components are reviewed. The procedure for calculating the required parameters from real plant data is also presented. Application and performance of reviewed models is discussed as well. In addition, appropriate methods for tuning different types of hydro governors are studied and a classical method is used for PID governors. Inclusion of the nonlinearity and elasticity of the detailed turbine-penstock model, and studying the effect of the servo transfer functions are the main aspects of the proposed tuning method. The paper is written in a way to be useful for an electrical engineering student or a novice engineer to model, simulate, and analyze a hydro power plant dynamic behavior, and finally tune its governor. This work can be used as a primitive guideline for educational and practical purposes. **Copyright © 2012 Praise Worthy Prize S.r.l. - All rights reserved.**

Keywords: Power Plant Dynamics, Hydraulic Turbines, Governor Tuning, Power System Education, PID Controller

Nomenclature

A_t	Turbine gain
D	Penstock pipe diameter (m)
D_a	Turbine damping
G	Ideal gate opening
H	Net head (m)
H_q	Inertia time constant of generator (MW.s/MVA)
K_s	Gain of servo-system
L	Length of penstock pipe (m)
P_m	Turbine mechanical power (MW)
P_r	Rated turbine power (MW)
Q	Water discharge (m ³ /s)
R_T	Transient droop
R_p	Permanent droop
T_{ep}	Elastic time of penstock pipe (s)
T_M	Mechanical starting time (s)
T_p	Pilot valve and servo time constant (s)
T_G	Main servo time constant (s)
T_R	Resetting time (s)
T_w	Water starting time (s)
U	Water velocity (m/s)
Z_p	Normalized hydraulic surge impedance of the penstock
a	Pressure wave speed in penstock pipe (m/s)
g	Real gate opening
a_g	Acceleration due to gravity (m/s ²)
a_{ij}	Turbine characteristic coefficients
ϕ_p	Friction coefficient of penstock
ω	Angular velocity of the generator (rad/sec)

–	Superscript for normalized values
r	Subscript for rated values
0	Subscript for initial steady state values
NL	Subscript for no-load values

I. Introduction

Accurate mathematical representation of power system components is significant for dynamic and transient stability studies. Therefore, some standard dynamic models for prime movers were introduced previously for simulation programs and other purposes in the literature [1]-[5].

The parameters of these models need to be determined by operators and engineers as accurate as possible to take into account the behavior of such elements in power system dynamic simulations.

Furthermore, one important application of such modeling is tuning of the parameters of the unit controller such as governor. In order to guarantee good performance of electric power generation process in different conditions, the parameters of the controllers of the components have to be properly tuned.

Hydroelectricity is an important source of renewable energy and provides significant flexibility from operating point of view. The dynamic behavior of hydro power plants is determined by the transients of water flow in the water column. The energy conversion process in turbine involves only nondynamic characteristics [4].

Hydraulic turbine is controlled by a governor system which consists of control and actuating equipment for regulating the flow of water, starting and stopping the unit, and regulating the speed and power output of the generator [6]. The proper tuning of these governor systems plays an important role in stable and acceptable performance of the connected power system.

There are some works for the modeling and simulation of hydraulic power generation systems and their governor tuning. Most of them are reviewed in [7]. Some textbooks present the basic knowledge about the subject [1]-[3].

In addition, there are some contributions for tuning of hydro governors considering different models and controller design methods. As mentioned in [7], the governor tuning of nonlinear models specially in the case of considering elasticity effect, needs more work. This subject is discussed in this work with a real case study and a simple approach is used for this purpose.

The first aim of this paper is to illustrate the procedure of the extracting hydro turbine different standard and detailed model parameters using real plant data. The response of all models is presented for the sake of comparison.

The second aim is to propose the application of a proper and simple controller design method for tuning of the plant governor for the prepared models. Linearization of nonlinear model with elasticity effect is also presented in this work for governor tuning purpose. Performance of the tuned governor is verified by comparing with previous method and for different models. This work is in succession with previous works by the authors, in which the derivation of thermal turbines and steam boiler dynamic model parameters are presented [8]-[10].

This paper is organized as follows: In Section II, a brief description of hydroelectric power plants is presented. Hydraulic turbine-penstock models for dynamic simulations are briefly described in Section III. Section IV is dedicated to present the required equations for parameter determination. Hydro governor models and their tuning are discussed in Sections V and VI respectively. Section VII presents the simulation results of the reviewed models of a real hydro power plant and performance of the tuned governor. Finally, Section VIII concludes the paper.

II. Hydroelectric Power Plants

In a hydraulic power generation plant, the stored energy in water as a hydraulic fluid is converted into mechanical energy by means of hydraulic turbine.

Hydraulic turbines are of two basic types: impulse turbines and reaction turbines. Selection of the type of the turbine depends upon the head and water flow rate of the dam. The shaft of the generating unit may be in a vertical, horizontal, or inclined direction depending on conditions of the plant and the type of turbine applied.

The majority of new installations are vertical.

More details about hydroelectric power plant structures are beyond the scope of this work and can be found in related textbooks such as subject [1]-[3].

Fig. 1 illustrates the main components of a typical hydroelectric generating unit. From the reservoir, water is drawn from an area called the forebay to the turbine through the water column. The water column comprises all of the structures used to convey water from the forebay to the turbine. It may include an intake structure, a penstock, one or more surge tanks, and a spiral case.

The composite water column inertias and elasticity of these structures contribute to the water hammer effect that impacts the performance of the turbine governing system.

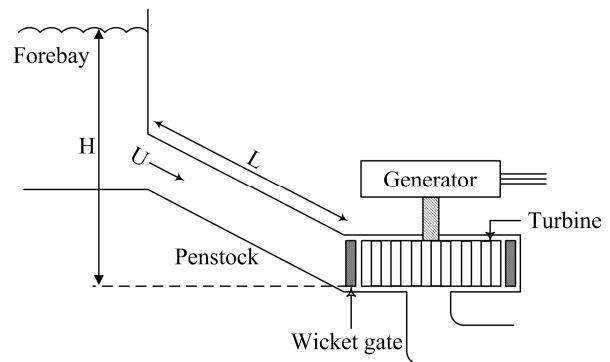


Fig. 1. Simplified schematic of a hydroelectric power plant

Wicket gates are adjustable and pivot open around the periphery of the turbine to control the amount of water admitted to the turbine.

These gates are adjusted by the servo actuators which are controlled by the governor. Fig. 2 demonstrates the simplified relationship between the basic elements of power generation process in a hydraulic power plant. Modeling of these elements is described in the following sections.

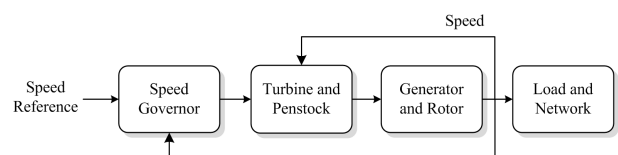


Fig. 2. Simplified functional block diagram of hydraulic power plants

III. Hydraulic Turbine Models

Hydroelectric power generating system exhibits a high-order and nonlinear behavior. Appropriate mathematical models are essential tools for simulation of such systems. The hydraulic system and turbine-penstock models have been analyzed in literature. The reader can find out that there are several models with different levels of details for hydraulic power plants. This section presents a brief overview and classification of the proposed models for representing turbine-penstock systems for dynamic simulations.

First of all, the models can be classified into *linear* and *nonlinear* categories based on the complexity of the equations. The proposed models can be classified further by considering *elasticity effect* of the water column (penstock). The mathematical equations of the models can be represented in several different ways.

Here, the transfer function approach is used for modeling due to the purpose of this work.

III.1. Linear Models

Since linear models are obtained around an operating point, they can also be called *small-signal* models. These models are extracted from basic equations of turbine and penstock characteristics with some simplifying assumptions for approximate modeling. The transfer function of linear models relates the turbine mechanical output power to the gate signal. The linear models are categorized and described as follows:

III.2. Simplified Linear Model

The simplest model of the hydraulic turbine-penstock component is the classical transfer function for an ideal lossless turbine-penstock system which is given by:

$$\frac{\Delta \bar{P}_m}{\Delta \bar{G}} = \frac{1 - T_w s}{1 + \frac{1}{2} T_w s} \quad (1)$$

The superbar “ $\bar{}$ ” indicates normalized values based on steady state operating point values. The normalized values of parameters will be used often hereafter. It should be mentioned that $\bar{G}$ is the ideal gate opening which is defined based on the change of real gate opening from no load to full load being equal to 1 per unit, i.e.:

$$\bar{G} = A_t \times \bar{g} = \frac{1}{\bar{g}_{FL} - \bar{g}_{NL}} \times \bar{g} \quad (2)$$

The above transfer function indicates that how the turbine power output changes in response to a change in gate position. This model is valid for small disturbances around the operating point.

III.3. Non-ideal Linear Model

Non-ideal turbine-penstock model can be expressed by the following transfer function [3]:

$$\frac{\Delta \bar{P}_m}{\Delta \bar{G}} = \frac{a_{23} + (a_{11}a_{23} - a_{13}a_{21})T_w s}{1 + a_{11}T_w s} \quad (3)$$

Coefficients a_{11} and a_{13} are partial derivatives of water flow with respect to head and gate opening and the

coefficients a_{21} and a_{23} are partial derivatives of turbine power with respect to head and gate opening respectively.

III.4. Non-Ideal Elastic Linear Models

Two previous conventional hydraulic system models neglect the effects of water compressibility and pipe elasticity.

In many applications, a more accurate hydraulic system model is necessary to take into account the compressibility of water and elasticity of penstock pipe.

These effects exhibit a dynamic interaction between the hydraulic and electrical systems. Pressure wave in the penstock as a hydraulic transmission line is terminated by an open circuit at the turbine end and a short circuit at the reservoir end.

In order to take into account the elasticity effect in the transfer function of the turbine-penstock system, (3) can be rewritten as [11]:

$$\frac{\Delta \bar{P}_m(s)}{\Delta \bar{G}(s)} = \frac{a_{23} + (a_{11}a_{23} - a_{13}a_{21})Z_p \tanh(sT_{ep})}{1 + a_{11}Z_p \tanh(sT_{ep})} \quad (4)$$

The above equation is a distributed parameter model and it is difficult to use it in system stability and governor tuning studies for linear control methods. Therefore, $\tanh$ function is usually replaced by using the Maclaurin series approximation as following [2]:

$$\tanh(T_{ep}s) = \frac{sT_{ep} \prod_{n=1}^{n=\infty} \left[1 + \left(\frac{sT_{ep}}{n\pi} \right)^2 \right]}{\prod_{n=1}^{n=\infty} \left[1 + \left(\frac{2sT_{ep}}{(2n-1)\pi} \right)^2 \right]} \quad (5)$$

Thus, the transfer function of (4) can be expressed in a rational polynomial form.

Linear models are generally useful for control system studies using linear analysis techniques (root locus, frequency response, etc.) and provide a good insight into the basic characteristics of the hydraulic system dynamics.

These models are inadequate for large variations in power output or system frequency [5].

III.5. Nonlinear Models

Appropriate nonlinear model is required for large-signal time-domain simulations such as islanding, load rejection, system restoration, etc. Hydrodynamics and mechanic-electric dynamics are included in nonlinear models.

This type of modeling is especially important for hydro power plants with long penstock.

The non-linearity of the model comes from the valve characteristic of the turbine. Nonlinear models can be generally represented by the block diagram shown in Fig. 3 [3]. It should be mentioned that $\bar{P}_r$ is used to take into account the selection of the power base such as turbine MW, generator MVA, or a common MVA base in the whole studied power system.

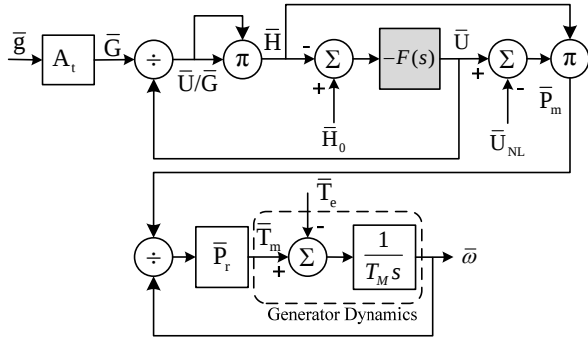


Fig. 3. General form of nonlinear hydraulic turbine-penstock model with isolated generator

The transfer function $F(s)$ relates turbine velocity to the turbine head:

$$F(s) = \frac{\Delta \bar{U}}{\Delta \bar{H}} \quad (6)$$

Depending to the selection of $F(s)$, nonlinear models can have one of the following categories:

III.6. Simplified Nonlinear Models

In this model, the traveling pressure wave and water compressibility is neglected. Then, $F(s)$ is described as:

$$F(s) = -\frac{1}{T_w s} \quad (7)$$

III.7. Nonlinear Models Assuming Elastic Water Column

The aforementioned model, assuming inelastic water column is adequate only in short or medium length penstocks. The pressure wave phenomenon or water hammer effect can be represented by considering $F(s)$ as [3]:

$$F(s) = \frac{-1}{\phi_p + Z_p \tanh(T_{ep} s)} \approx \frac{-1}{Z_p \tanh(T_{ep} s)} \quad (8)$$

It should be noted that ϕ_p can be neglected, especially for governor tuning studies because of its positive effect on damping of system oscillations and overshoot [12].

The parameters of the presented models need to be estimated before simulation.

The required equations for calculating these parameters are presented in the next section. The output power/torque of the turbine is transferred to a synchronous generator.

The speed of the generator is kept constant by the governing system.

IV. Parameter Determination

This section describes and provides the required equations for estimating parameters of the aforementioned models as following:

IV.1. Water Starting Time (T_w)

It stands for the time required for a head H_0 to accelerate the water flow in the penstock from standstill to U_0 . This time can be calculated by the following equation:

$$T_w = \frac{LU_0}{a_g H_0} \quad (9)$$

Therefore, T_w varies with operating point. Its value mainly depends on U_0 , because other parameters remain almost unchanged. Typical values of T_w vary between 0.5 s and 4.0s [3].

IV.2. Turbine Coefficients (a_{ij} Coefficients)

These coefficients depend on machine loading and may be determined from the turbine characteristics called Hill charts at the operating point. Since these charts are usually unavailable, some proposed equations which are extracted from turbine characteristic equations can generally be used (see Table I).

However, more accurate values of these parameters can be obtained from Hill charts of the turbine [1].

TABLE I
PROPOSED EQUATIONS FOR TURBINE COEFFICIENTS

Coefficient	Definition	Ref.[13]	Ref. [14]
a_{11}	$\frac{\partial \bar{U}}{\partial \bar{H}}$	$0.5\bar{G}$	$\frac{\bar{G}}{2\sqrt{\bar{H}}}$
a_{12}	$\frac{\partial \bar{U}}{\partial \bar{\omega}}$	0	0
a_{13}	$\frac{\partial \bar{U}}{\partial \bar{G}}$	1.0	$\sqrt{\bar{H}}$
a_{21}	$\frac{\partial \bar{P}_m}{\partial \bar{H}}$	$1.5\eta\bar{G}$	$A_t \bar{P}_r \left(\frac{3}{2} \bar{G} \sqrt{\bar{H}} - \bar{U}_{NL} \right)$
a_{22}	$\frac{\partial \bar{P}_m}{\partial \bar{\omega}}$	0	$D_a \bar{G}$
a_{23}	$\frac{\partial \bar{P}_m}{\partial \bar{G}}$	η	$A_t \bar{P}_r \bar{H}^{1/2} - D_a (\bar{\omega} - 1)$

IV.3. Elastic Time (T_e)

T_e is the traveling time of the pressure wave in penstock pipe which can be calculated by the following equation:

$$T_{ep} = \frac{\text{penstock length}}{\text{pressure wave velocity}} = \frac{L}{a} \quad (10)$$

Typical values of pressure wave velocity (a) are 1220 m/s for steel pipes and 1420 m/s for cement tunnels [3].

IV.4. No-Load Velocity (U_{NL})

The normalized value of U_{NL} which is required for the nonlinear model is calculated by [3]:

$$\bar{U}_{NL} = A_t \bar{g}_{NL} \sqrt{\bar{H}_0} \quad (11)$$

IV.5. Hydraulic surge impedance of the conduit (Z_p)

Z_p is expressed by:

$$Z_p = \frac{T_w}{T_{ep}} \quad (12)$$

These equations are used for calculating parameters of a real hydro power plant in Section VII.

V. Hydraulic Turbine Governor Systems

A governor regulates the speed and power output of a prime mover as a control system. The governor includes mainly a controller function, and one or more control actuators [6].

It should be mentioned that hydro turbines have initial inverse response characteristics of power to gate changes due to water inertia.

Therefore, a hydro governor needs to provide a transient droop in speed controls to limit the overshoot of turbine gate servomotor during a transient condition. This means that for fast deviations in frequency, the governor should exhibit high regulation (low gain) while in slow changes and steady state it should exhibit the normal low regulation (high gain).

Therefore, a large transient droop with a long resetting time is required [5]. This feedback limits the movement of the gate blades until the water flow and mechanical power output have time to overtake.

V.1. Hydraulic-Mechanical Governor

The governing function of such governors is realized with the use of mechanical and hydraulic components [15]. A large temporary droop compensation for stable operation of the governor is provided by a dashpot. This

can also be bypassed if desired. The transfer function of transient droop compensation is given by:

$$G_{TD}(s) = R_T \frac{sT_R}{1 + sT_R} \quad (13)$$

The model of the typical hydraulic turbine governor is shown in Fig. 4.

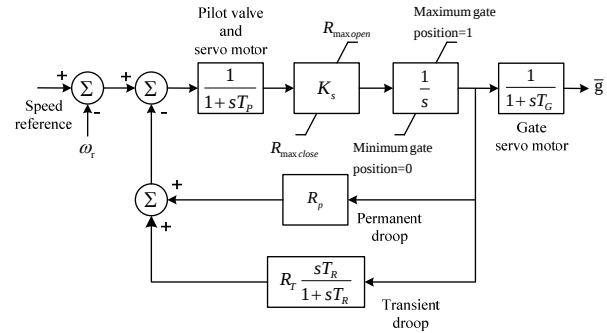


Fig. 4. Hydraulic turbine governor model [3]

V.2. Electrohydraulic (PID) Governors

Nowadays, speed governors for hydraulic turbines use electrohydraulic systems.

These governing systems are realized by electric components which provide greater flexibility and better performance. These types of governors are mostly designed in PID controller form which is widely applied in industry.

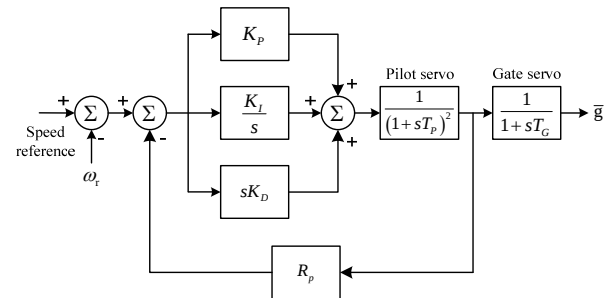


Fig. 5. Block diagram of PID governor for hydraulic turbines.

As shown in Fig. 5, the three terms of the controller treat the present control error (Proportional), past control error (Integral), and predicted control error (Derivative). PID controllers ensure faster speed response by providing both transient gain reduction and increase. The derivative term in the control action is important in the case of isolated operation [7].

The use of high derivative gain will lead to oscillations or instability in the case of connection to a strong interconnected system.

It should be mentioned that the transfer function of PID without derivative part (PI governor) is actually equivalent to that of the hydraulic-mechanical governor.

VI. Tuning of Hydro Governors

The most critical condition for governor tuning would be with the unit supplying an isolated load at maximum output [6].

Therefore, this point is considered for governor tuning in present work.

To the knowledge of the authors, there is no work that has considered elasticity effect of water column and nonlinear characteristic of the turbine simultaneously for tuning hydro governors. The elasticity effect represents a delay e^{-2sT_e} ; which is irrational term in the hydraulic structure.

A transfer function with irrational term is difficult to solve and sometimes cannot be used directly in stability studies. The study made for relatively long penstock on the basis of non-elastic water column effect will have significant error. Such approach in the governor design downgrades its effectiveness. The approximation of a high order system by a low order seeks importance particularly in the controller design and control system analysis. A simple model involves less computation time of the transient response.

Thus for designing efficient control system, it becomes necessary to use reduced order turbine-penstock model with elastic water column effect of a long penstock in hydropower plant.

VI.1. Tuning of Hydraulic-Mechanical Governors

The steady state performance of governors can be determined by the *permanent* droop or *speed regulation* (R_p). This droop is typically set at the range of 0.03 to 0.06. In addition, in the case of two or more generator units in parallel, the purpose of the droop is to ensure equal power sharing between the units [3].

The hydraulic-mechanical governor settings that can be tuned to control the *dynamic* performance of the generating unit are T_R , R_T , and K_s . For stable operation under islanding conditions (worst case), the proposed proper choice for the temporary droop R_T and reset time T_R is given in Table II ($T_M=2H_g$).

TABLE II
PROPOSED SETTINGS FOR HYDRAULIC-MECHANICAL GOVERNORS

Ref	Temporary Droop (R_T)	T_R
[16]	$2.5T_w/T_M$	$5.9T_w$
[17]	$2T_w/T_M$	$4T_w$
[18]	$2T_w/T_M$	$5T_w$
[19]	$[2.3 - (T_w - 1.0)0.15]T_w/T_M$	$[5.0 - (T_w - 1.0)0.5]T_w$

Moreover, K_s should be set as high as possible. The above settings provide good performance during the most severe isolated conditions and slow response during loading in normal interconnected operation.

Therefore it is recommended that the reset time T_R be preferably set to 0.5 s [3].

VI.2. Tuning of PID Governors

Conventional PID governors are usually tuned using linear control theory tools. In order to use these methods, the turbine-penstock model has to be linear. Therefore, nonlinear models have to be linearized around the operating point. This procedure is described later in this section.

However, most of the works in the subject have considered linear models for this purpose. A general method is presented by Hagihara [20] which is presented in Table III. It is worth nothing that if the gains K_p , K_i , and K_d of the PID governor set to the values of first row in Table III, the transfer function of the PID governor would be the same as that of the temporary droop governor represented by Hovey [17]. The setting of permanent droop (R_p) is the same as for hydraulic-mechanical governors described earlier.

TABLE III
PROPOSED SETTINGS FOR PID GOVERNORS [20]

K_p	K_i	K_d
$T_M/(2T_w)$	$T_M/(8T_w^2)$	0
$0.8T_M/T_w$	$0.24T_M/T_w^2$	$0.27T_M$

VI.3. Linearization of the Nonlinear Model

The linearization procedure is described here for the nonlinear model shown in Fig. 3. The equations given for the penstock-turbine system in per unit are given as following [1]:

$$\bar{U} = \bar{G}\sqrt{\bar{H}} \quad (14)$$

The $\tanh(T_{ep}s)$ term in (8) can be rewritten in polynomial form using (5). So $F(s)$ for $n=0$ to 2 can be calculated by the following equations:

$$F(s) = \frac{\Delta\bar{U}}{\Delta\bar{H}} = \frac{-B}{Z_p T_{ep} A s} \quad (15)$$

$$n=0 \rightarrow A=1, \quad B=1 \quad (16)$$

$$n=1 \rightarrow A=1+s^2\left(\frac{T_{ep}}{\pi}\right)^2, \quad B=1+4s^2\left(\frac{T_{ep}}{\pi}\right)^2 \quad (17)$$

$$n=2 \rightarrow \begin{cases} A=1+\frac{5}{4}s^2\left(\frac{T_{ep}}{\pi}\right)^2+\frac{1}{4}s^4\left(\frac{T_{ep}}{\pi}\right)^4 \\ B=1+\frac{40}{9}s^2\left(\frac{T_{ep}}{\pi}\right)^2+\frac{16}{9}s^4\left(\frac{T_{ep}}{\pi}\right)^4 \end{cases} \quad (18)$$

The above equations can become linearized around the operating point x_0 by substituting $x=x_0+\Delta x$ for different variables and dropping the high orders of Δx .

As the first step:

$$\bar{U}_0 + \Delta\bar{U} = (\bar{G}_0 + \Delta\bar{G})\sqrt{\bar{H}_0 + \Delta\bar{H}} \quad (19)$$

Applying a Taylor expression of polynomial raised to the $\frac{1}{2}$ power, and dropping out the initial terms, the radical term can be linearized and the following equation is derived [21]:

$$\Delta\bar{U} = \Delta\bar{G}\sqrt{\bar{H}_0} + \bar{G}_0\sqrt{\bar{H}_0} \frac{\Delta\bar{H}}{2\bar{H}_0} \quad (20)$$

The relation of mechanical power, head, and flow in turbine is:

$$\bar{P}_m = \bar{U}\bar{H} \quad (21)$$

By applying the same procedure for the above equation, following expression can be obtained:

$$\Delta\bar{P}_m = \bar{U}_0\Delta\bar{H} + \Delta\bar{U}\bar{H}_0 \quad (22)$$

By substituting $\Delta\bar{H}$ from (15) into (20) and (22) $\Delta\bar{U}/\Delta\bar{G}$ and $\Delta\bar{P}_m/\Delta\bar{U}$ can be calculated.

Consequently, the final transfer function is determined as follows:

$$\frac{\Delta\bar{P}_m}{\Delta\bar{G}} = \frac{\Delta\bar{P}_m}{\Delta\bar{U}} \times \frac{\Delta\bar{U}}{\Delta\bar{G}} = 2\bar{H}_0^{\frac{3}{2}} \frac{\bar{H}_0 B - \bar{U}_0 T_w A s}{2\bar{H}_0 B + \bar{U}_0 T_w A s} \quad (23)$$

where A and B are defined in (16)-(18).

VII. Simulation and Discussion

Parameter determination and simulation of a real hydro power plant (Abbaspour power plant) is presented in this section. The required data for parameter determination is given. The prepared models are used for comparison of their performance and PID governor tuning.

VII.1. Case Study

A real hydro power plant of Iranian grid with Francis turbine is selected for this work. The required data of this plant is presented in Table IV.

VII.2. Parameter Determination

Calculation of required parameters for modeling Abbaspour power plant at nominal operating point is presented as an example:

$$T_w = \frac{L U_r}{a_g H_r} = \frac{L \frac{Q_r}{\pi \times (D/2)^2}}{a_g H_r} = \frac{514 \times \frac{72}{\pi \times (4/2)^2}}{9.8 \times 250} = 1.202 \text{ s}$$

$$T_{ep} = \frac{\text{penstock length}}{\text{pressure wave velocity in steel pipes}} = \frac{L}{a} = \frac{514}{1440} = 0.357 \text{ s}$$

$$Z_p = \frac{T_w}{T_{ep}} = \frac{1.202}{0.357} = 3.367$$

Turbine rated MW is chosen as base power for the sake of simplicity, so:

$$\bar{P}_r = \text{Turbine MW} / \text{Base Power} = 1$$

Turbine coefficients (a_{ij}) can be calculated by the values and equations of Table I. In order to use the proposed equations in [13] and [14] it is assumed that neat head (H) remains unchanged for all operating points, then:

$$\bar{H}_0 = 1.0 \text{ pu}, \bar{\omega}_0 = 1.0 \text{ pu}, \bar{U}_0 = 1.0 \text{ pu}$$

$$A_t = \frac{1}{\bar{g}_{FL} - \bar{g}_{NL}} = \frac{1}{0.96 - 0.06} = 1.111$$

$$\bar{G}_{FL} = A_t \times \bar{g}_{FL} = 1.111 \times 0.96 = 1.067$$

$$\bar{U}_{NL} = A_t \bar{g}_{NL} \sqrt{\bar{H}_0} = 1.111 \times 0.06 \times 1.0 = 0.067$$

Then, turbine coefficients are calculated by the above parameters and the results are presented in Table V. Ideal and proposed typical values are also presented in this table.

The values of the last column are used here for simulation of the studied power plant.

TABLE IV
ABBASPOUR HYDRO POWER PLANT DATA

	Parameter	Symbol	Value
Penstock	Length	L	514 m
	Diameter	D	4 m
	Pressure wave velocity	a	1440 m/s
Turbine	Rated mechanical power	P_{mr}	256 MW
	Rated discharge	Q_r	72 m ³ /s
	Rated head	H_r	250 m
	Rated efficiency	η	0.94
	Gate position at rated condition	$\bar{g}_{FL}$	0.96
Generator	Gate position at No-load	$\bar{g}_{NL}$	0.06
	Turbine damping	D_a	0.5
	Rated power	P_{gr}	250 MW
	Frequency	f	50 Hz
	Inertia time constant	H_q	4.5 s

It should be mentioned that the transfer function of pilot valve and main servo motor is not considered in most of the previous works. However, realistic values for T_p and T_G is reported by Sanathanan as 0.1 s and 0.15 s respectively [22]. These transfer functions are neglected

in the majority of the previous works. Their effect is studied here for governor tuning studies.

TABLE V
TYPICAL AND CALCULATED VALUES OF TURBINE COEFFICIENTS

Coefficient	Ideal lossless [3]	Typical at full load [23]	Typical at no-load [23]	Ref. [13]	Ref. [14]
a_{11}	0.5	0.58	0.57	0.534	0.5
a_{12}	0	0	0	0	0
a_{13}	1.0	1.1	1.1	1.0	1.0
a_{21}	1.5	1.4	1.18	1.505	1.630
a_{22}	0	0	0	0	0.5
a_{23}	1.0	1.5	1.5	0.940	1.111

VII.3. Comparison of Turbine-Penstock Models

In order to compare five different reviewed models of hydro turbine and penstock system, the step response of the models are presented. Fig. 6 and Fig. 7 show the step response of linear and nonlinear models respectively to a 10% reduction in gate opening.

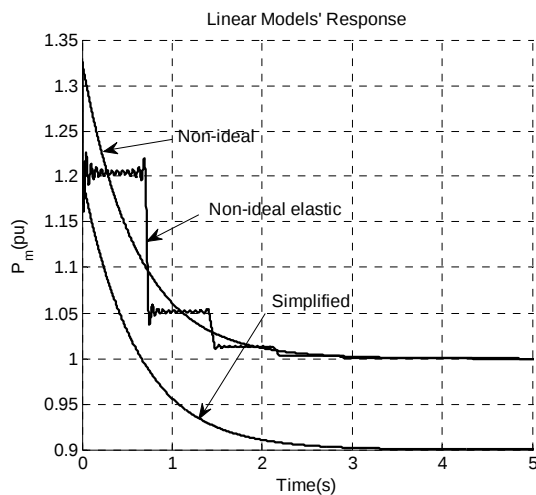


Fig. 6. Response of linear turbine-penstock models to a 10% reduction in gate opening

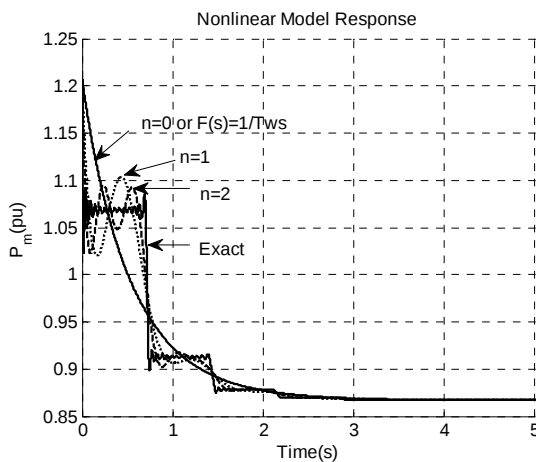


Fig. 7. Response of nonlinear turbine-penstock models to a 10% reduction in gate opening

Comparison of different models shows that linear or simplified models approximate the detailed models. The difference is more significant in transient state or higher frequencies. Pressure wave traveling effect can be observed in the responses of elastic linear and exact nonlinear models with a time period of about $2 \times T_{ep}$ (i.e. 0.72 s).

VII.4. PID Governor Tuning

The studied power plant is facilitated with a PID governor. The extracted models can be used here for tuning this PID governor by adequate techniques. The performance of the governor must be evaluated in worst condition to ensure the stable operation of the unit in different situations. According to the IEEE standard, hydro governors can usually control 10% of the rated load shedding in isolated conditions due to some practical considerations and limits [6]. Therefore, these conditions should be considered as the worst case scenario to verify the performance of governor controller. General rules for tuning a PID controller are:

1. Use K_P to reduce the rise time.
2. Use K_D to reduce the overshoot and settling time.
3. Use K_I to eliminate the steady-state error.

These rules work in many cases, but it is necessary to have appropriate starting points of above parameters. Some methods are proposed to find appropriate set of initial parameters such as Ziegler-Nichols (ZN) method.

This method was first introduced in 1942 [24]. This method can also be applied on the plants that are not mathematically known but with available experimental step responses on-site. Basically there are two ZN tuning rules including first method and second method [25].

The first method is only applicable when the step response of the plant is S-shaped. If the plant involves integrator or dominant complex-conjugate poles, then the second method is applied. The second method is used here for PID governor tuning.

The second ZN approach is based on determining the ultimate gain and period that result in marginal stability when only proportional control function is used. For linear systems, the continuous oscillation mode corresponds to the critical stable condition. Such condition can be easily determined through critical gain k_{cr} and critical oscillation period $P_{cr} = 2\pi/\omega_{cr}$ that ω_{cr} is the crossover frequency. The primary PID controller gains then could be calculated by expressions presented in Table VI. It is helpful to note that the critical gain and crossover frequency can be obtained by "margin" command in Matlab software environment [26]. These values can also be obtained for nonlinear models by trial and error.

TABLE VI
ZIEGLER-NICHOLS PID TUNING RULE

K_P	K_I	K_D
$0.6 \times k_{cr}$	$1.2 \times k_{cr} / P_{cr}$	$0.075 \times k_{cr} \times P_{cr}$

ZN tuning rule is applied to all of the studied models. In addition, the linearized model of the nonlinear model is used for this purpose. The tuned parameters of PID governor by ZN rule using different models are presented in Table VII and compared with the PID parameters which are calculated by proposed equations by Hagihara [20] (See Table III).

It should be mentioned that the proposed equation by Hagihara is only applicable to the simplified linear model without the effect of pilot and gate servo transfer functions [22]. However, ZN tuning rule is more general and easy to use for more complicated models. In order to compare the performance of tuned PID governor, the response of the system to a 10% rated load shedding is illustrated in Fig. 8 to Fig. 10.

TABLE VII
PID GOVERNOR SETTING RESULTS

	Model/Method	K_P	K_I	K_D
Without servo Transfer Function	Hagihara (PI)	3.747	0.780	0
	Hagihara (PID)	6.00	1.50	2.43
	Linear Simplified	6.75	1.69	3.07
With servo Transfer Function	Linear Simplified	7.06	0.90	3.55
	Linear Non-ideal	6.52	0.84	3.24
	Linearized with $n=1$	5.19	0.83	3.54
	Linearized with $n=2$	5.16	0.87	3.57
	Nonlinear	3.57	0.24	3.35

The settings of Table VII are applied to the studied models with isolated load condition to verify the performance of the governor. It should be mentioned that the initial K_P obtained by ZN method needs to be increased about 50% to improve the response speed by reducing the rise time and settling time of the response.

The results show that the performance of the PID governor tuned by ZN method is appropriate for stable operation of the studied hydro power plant.

As shown in Fig. 8, the response of the system with the governor settings determined by ZN method is slightly better than the response with settings of Table III. Fig. 9 shows that selection of linear model type has almost no effect on the performance of ZN method. Comparison of the response of simplified model with and without governor servo transfer functions indicates that they have no considerable effect on tuning results and can be neglected in tuning by ZN method. It can also be concluded that the order of linearization of the nonlinear model has also insignificant effect (See Fig. 10). The response of the nonlinear model has less overshoot but is slower than other models.

The experiment in this work showed that the response of the nonlinear elastic model as the most comprehensive one among the reviewed models is more sensitive to adjusting PID parameters compared to other simpler models. Finally, it is important to note that the proposed settings in [20] and initial governor settings obtained by applying ZN method on other models are unusable for nonlinear model, because they lead to instability.

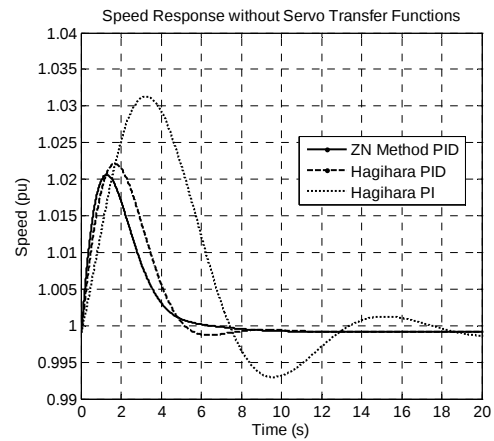


Fig. 8. Response of simplified linear turbine-penstock model to a 10% load shedding with PI and PID governor without servo transfer functions

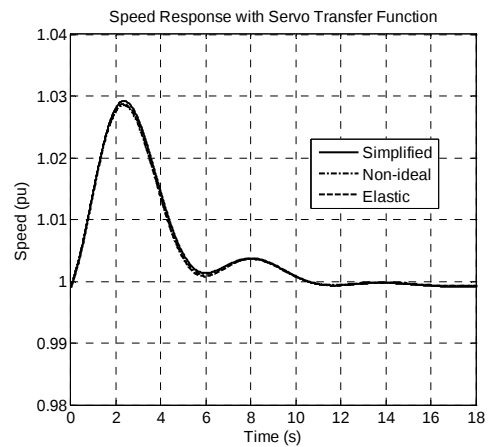


Fig. 9. Response of linear turbine-penstock models to a 10% load shedding with PID governor and servo transfer functions

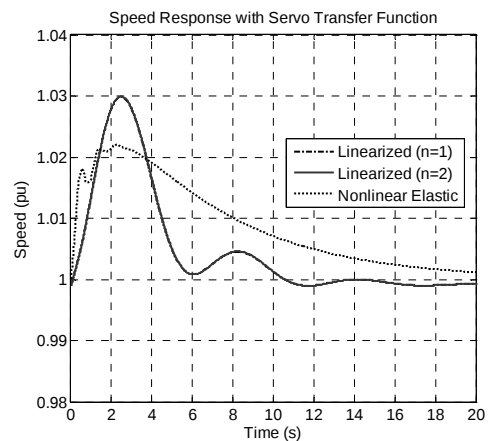


Fig. 10. Response of linearized and nonlinear turbine-penstock models to a 10% load shedding with PID governor with servo transfer functions

VIII. Conclusion

This paper reviewed the proposed models for dynamic simulation of hydro power plant components and

provided a proper view of their behavior. The required equations for the determination of the model parameters have been presented. Different models have been compared for simulation of a real hydro power plant. The extracted models are then used for hydro governor tuning. The previous methods for tuning hydro governors have been also reviewed and a classical heuristic method is utilized for this purpose.

This work can be a useful guide for power engineering students who want to use an appropriate hydro power plant model including the penstock, turbine, governor, and generator for stability studies. This paper provides required background knowledge for studying the effect of actual data on the model parameters and behavior. The review and comparison of different models help the electrical engineer to choose the best model for specific studies. Furthermore, the appropriate tuning of the conventional PID governor using a simple approach has been presented which can be used for educational purposes and real applications.

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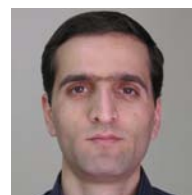
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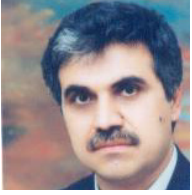
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