Control System for Kaplan Hydro-Turbine

MATEI VINATORU, EUGEN IANCU, CAMELIA MAICAN, GABRIELA CANURECI
Department of Automation
University of Craiova
13 A. I. Cuza str., Craiova, Jud. Dolj, 200678
ROMANIA

Abstract: - In hydro power plants from Romania, there is a major interest for the implementation of digital systems for monitoring and control to replace the conventional control systems for power, frequency and voltage. Therefore is necessary to develop mathematical models capable to accurately describe both dynamic and stationary behaviour of the hydro units, in order to be able to implement digital control algorithms. Moreover, it is necessary to implement systems for monitoring and control of hydro power plants in a cascade system along a river, in order to optimise the use of the river resources. This paper presents the possibilities of modelling and simulation of the hydro power plants and performs an analysis of different control structures and algorithms.

Key-Words: - Kaplan turbine, Control system, Turbine model

1 Introduction

The work was motivated by problems presented during studies of the Kaplan turbine control system at "Iron Gates" HPP Romania. The main problem was to find a new structure for the optimal control system capable to resolve the needs for electrical power and river flow conditions. The real-time optimal control imposes more stringent quality assurance requirements on the modelling of hydro-power equipment. There are many papers that present the non-linear and linear models of Kaplan hydro turbine, and/or control system with the relationships between the turbine head, the turbine flow and the gate opening, and the mechanical power. In this context we mention:

- The IEEE Working Group [1] presents the non-linear IEE turbine model

- Kundur [2] formulates different models of the hydroelectric system, where the dynamics related to elastic water columns are considered as non-linear.

- The models for particular applications were presented in many papers: Hannett et al [5] (present a methodology for validating different model structures of a turbine-generator group). De Jaeger et al. [4] propose a non-linear model for study of hydraulic dynamics and the estimated parameters.

-There are many papers that present methods for design of controllers for the plants with Kaplan turbine: Arnautovic and Skataric [3] (present methodology for choosing the controller structure and tuning of its parameters). Ng,Walker and Sargison (a non-linear mathematical model for hydro turbine), Kosterev, Pierce and Spence [7] present non-linear model for Kaplan turbines.

This paper deals with a control system suitable for optimal control of the Kaplan turbine from hydro power plants. In this context, we will treat the problem of the non-linear hydraulic model, and the control system structure capable to resolve the flow transfer from the reservoir and the optimal energy production corresponding to electrical grid demands.

In the first section, we present the command options for the Kaplan turbine at "Iron Gate" hydropower plant, Romania and the requirements for these hydro groups. The second section is devoted to the modelling of some components of a hydro turbine and its water supply conduit. In the third section, we present the laboratory studies of the control system structure. The model presented is able to simulate different conditions that appear to Kaplan turbines and hydro generators such as discharge, load rejections and the hydraulic water frictions.

2 Turbines' control systems Structure

The functions of the turbine's control system are:

-Positioning of guide vanes and the pitch of runner's blades (given by the control algorithm);

-setting of vanes' position limiters;

-control of turbine rotational speed using sequential switchers and net isolated functions;

-control of generator power and the water flow;

- control of set-points for the position controller.

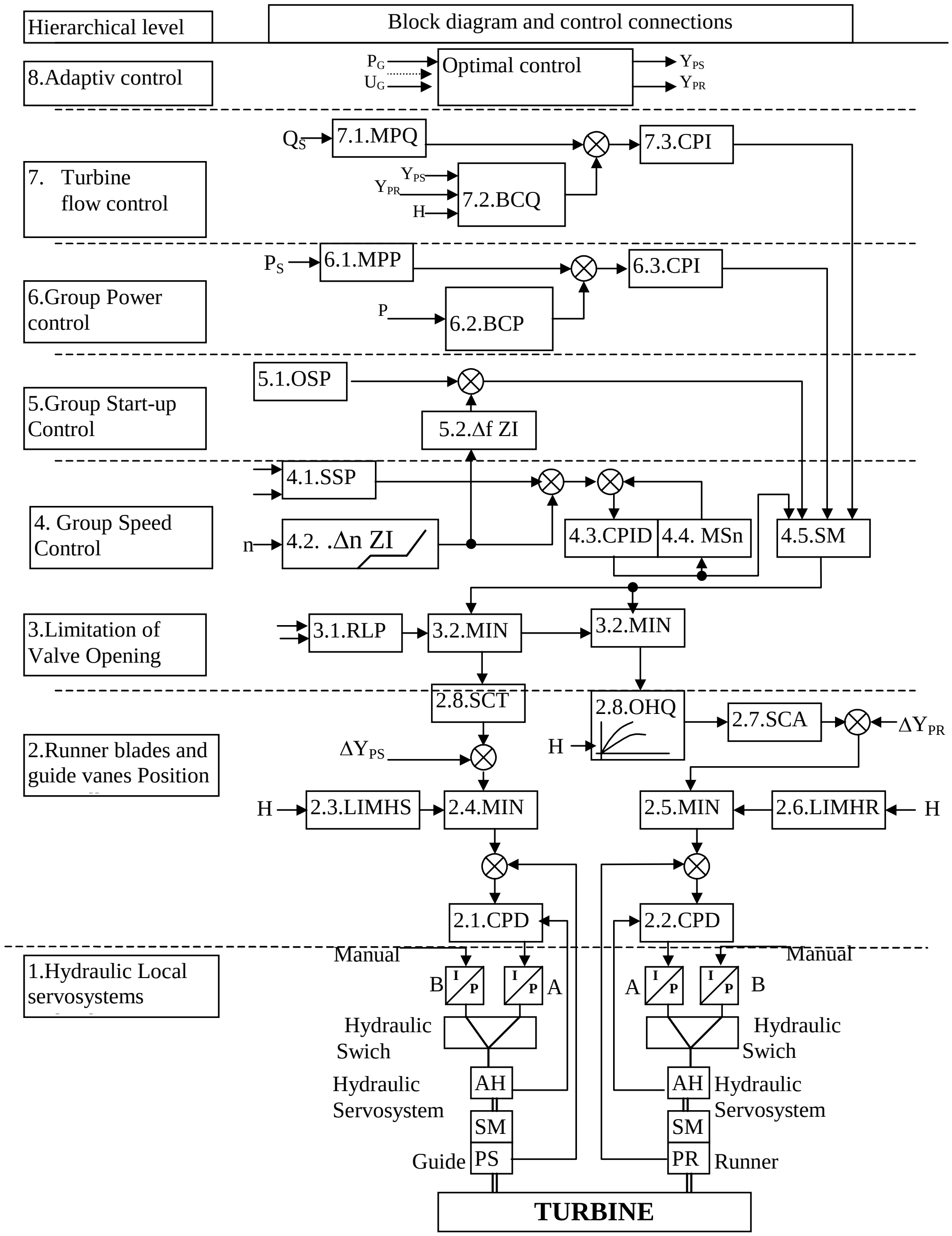

Fig.1. Block diagarm of hydraulic turbine control system

In order to handle emergency situations, it is necessary to impose a minimum and a maximum value of the net head for each turbine and an adaptive algorithm shall be implemented, linked with the flow monitoring system.

The conceptual scheme for the turbine control system is presented in figure 1. This structure is organized on eight hierarchical levels, including the field level containing the electro-pneumatic converters, the hydraulic amplifiers, and the hydraulic servomotors for the guide vanes and runner's blades positioning. The functions performed at each level and the subordination relationships can be easily determined from figure 1.

The control scheme is implemented both in hardware and software in a flexible control scheme for the turbine, to be able to reach all operational regimes requested by the hydraulic group operation, and to comply with all restrictions imposed by the energetic system and the limitations of the hydraulic group. The basic structure of the turbine controller consists of the control loops for the runner's blades pitch and guide vanes opening at control levels 1 and 2. In these loops, the controllers PD 2.1 and PD

2.2 receive a feedback from both the position sensors of the blades (main feedback) and the position sensors of the hydraulic servomotors (internal feedback). The set-points of these loops are calculated in the software modules at the higher levels. In order to avoid the over-opening of the blades for different operating regimes, the maximum opening value is limited to the number imposed by blocks 2.3 and 2.6; with blocks 2.4 and 2.5 transmitting to the controllers the lowest value received either from the set-point elements.

The controller for the guide vane opening shall always have an initial opening ΔYPS and the signal received from the upper level is added to that value (in block 2.9). Block 2.9 can stop the turbine control from the upper levels and allows the operator to manually set the controller's set-point from the local control panel through ΔYPS. In the controller for the runner's blades pitch, the prescribed value for the turbine flow, received from the upper levels, is processed by the optimisation module 2.8, which determines the optimum value of the pitch as a function of the required flow Q and the water level in the reservoir H (as per the H-Q diagrams specified above). Furthermore, for this control loop it is possible too, using the control module 2.7, to stop the pitch control from the upper levels and to allow the operator to manually control the pitch, by setting the set-point ΔYPR at block 2.5 MIN. Level 3 of control system performs the function of command rate of rise restriction, which is set by module 3.1 OPL and is controlled by module 3.2 MIN. The limitation of the command rate of rise is necessary to avoid pressure surges as well as cavitations, which may appear when the vanes or blades are opened suddenly. Level 4 allows selecting the operating regimes of the turbine controller, using the module 4.5 SM as follows:

- rotational speed control; in this regime, the rotational speed set-point and its rate of rise are imposed through block 4.1 MPn. This set-point is compared with the actual rotational speed received from the turbine speed sensor, which is processed in the dead-zone block 4.2.Δn ZI. The value of the dead-zone is chosen as a function of the allowed generator frequency shift and is necessary to avoid loop oscillations for small frequency changes.

-turbine control in start-up regime; this function is performed by level 5 of the control structure, in order to guarantee a certain rate of rise for the turbine rotational speed, and to facilitate the synchronization with the power grid when the generator is connected to the grid.

-the active and reactive power control is performed at level 6 of the control structure using blocks 6.1 MPP, which is providing the active power set-point for power controller 6.3 CPI, and feedback block 6.2 BCP, which is computing the power generated by the turbine using measurements taken from the electrical generator.

-the flow control is performed at level 7; the flow controller 7.3 CPI is providing the optimal values of the set-points for the guide vanes and runner's blades controllers. These optimal values are function of the imposed flow, through the set-point block 7.1 MPQ, and the hydraulic characteristics of the hydro plant such as net head H, and actual position of the gate YPS and blades pitch YPR.

Level 8 of the control structure is processing the parameters received from the central monitoring and optimisation system, such as required output power and required generator voltage, and calculates the necessary values for the position of the guide vanes YPS and blades pitch YPR to achieve those control goals. Calculations are based on the optimisation diagrams for the operation of the hydraulic group.

This control structure allows the reach of all operational regimes required by the common operational practice of hydraulic Kaplan turbines as well as those required by the position of the group in the national grid. The control structure allows both local control from the local control panels inside the machine room, as well as central control from the plant control room.

The hardware implementation shall consist of a hierarchical distributed structure, using specialized controllers or PLCs, monitoring and optimisation workstations, and data acquisition, processing and storage servers. The control structure shall offer maximum failure protection, using redundant equipment and functions.

3 Model of Kaplan Hydro-Turbine

The model will be used in transient stability studies and the power control system of hydro–groups. For this reason, it is necessary to study the turbine–governor response to system frequency deviations and oscillations. The turbine mechanical power can be expressed as [7]:

$$P = \eta H Q \qquad (1)$$

where: H is effective turbine head, Q is water discharge flow and η is turbine efficiency.

The turbine head H and turbine efficiency η are non-linear functions of gate position p and turbine blade angular position β:

$$H(t) = f_h (p, \beta, Q) \qquad (2)$$

$$\eta(t) = f_\eta (p, \beta, H) \qquad (3)$$

Also, the turbine discharge flow cannot be measured, but it can be determined as a dynamic function of effective head (H_o - steady–state head):

$$Q(s) = \frac{1}{sT_w}\left(H_o - H(t)\right) \quad (4)$$

A conceptual model of Kaplan turbine is presented in figure 2 [6].

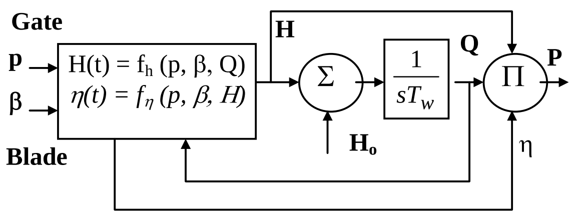

Fig. 2 Conceptual model of a Kaplan turbine

The IEEE hydro turbine model [1] was developed in many forms. Usually it is necessary to study the turbine model interconnected with the generator model and closed loop power and speed controller. In this case, the plant model (turbine + generator) has two poles placed in the axes origin and the stability studies are very important. For controller design and stability studies of closed loop system, we developed the linear turbine and generator models.

A. Modelling of the hydraulic system for run-on-the-river hydropower plants. These hydropower plants have a high water storage capacity in the reservoir; therefore the plant operation requires a permanent balance between the water flow through turbines and the river flow in order to maximize the discharge flow for a maximum efficiency of water use. We will determine the mathematical model for each component of the hydropower system.

* *Hydraulic turbine.* As parameters of the turbine, we will consider the discharge flow Q and the moment M generated by the turbine, transmitted to the electrical generator. These are non-linear functions of the turbine rotational speed N, the turbine gate position Z, and the net head H of the hydro system.

$$Q = Q(H, N, Z)\ ; \qquad M = M(H, N, Z) \quad (5)$$

In non-dimensional form (5) become:

$$q = a_{11}.h + a_{12}.n + a_{13}.z \quad (6)$$

$$m = a_{21}.h + a_{22}.n + a_{23}.z \quad (7)$$

where $q=\Delta Q/Q_o$, $n=\Delta N/N_o$, $q=\Delta Q/Q$, $m=\Delta M/M_o$, $h=\Delta H/H_o$, $z=\Delta Z/Z_o$, the non-dimensional variations of the parameters around the steady state values.

* *The hydraulic feed system.* The hydraulic feed system has a complex geometrical configuration, consisting of pipes or canals with different shapes and cross-sections. Therefore, the feed system will be considered as a pipe with a constant cross-section and the length equal with real length of the studied system to contain the same water mass: $Q=v.A=Q_i=v_i.A_i$ for i=1, 2,...,n, where v is the water speed in the equivalent pipe, and v_i is the speed in each segment of the real pipe.

The dynamic pressure loss can be computed as:

$$F_i = -m.a = -L.A.\rho.a = -A\frac{L.\gamma}{g}.\frac{dv}{dt} \quad (8)$$

where L is the length of the penstock or the feed canal, A is the cross-section of the penstock, γ is the specific gravity of water ($1000 Kgf/m^3$), a is the water acceleration in the equivalent pipe, and $g=9{,}81\ m/s^2$ is the gravitational acceleration.

In non-dimensional variables (8) become:

$$h_d = -T_W \frac{dq}{dt} \quad (9)$$

where h_d (net head), q (flow) and T_W (the integration constant) have the following meaning:

$$h_d = \frac{\Delta H_d}{H_{d0}}, \quad q = \frac{\Delta Q}{Q_0}, \quad T_w = \frac{Q_0}{H_{d0}} \cdot \frac{\rho L \cdot \sum l_i}{\sum l_i A_i} \quad (10)$$

Using the Laplace transform in (10), it results:

$$h_d(s) = -\,sT_w.q(s), \text{ and, } q(s) = -\frac{1}{sT_w} h_d(s) \quad (11)$$

Replacing (11) in (6) and (7), the mathematical model of Kaplan turbine is given by (12 - 14):

$$q(s) = \frac{a_{12}}{1 + a_{11}T_w s} n(s) + \frac{a_{13}}{1 + a_{11}T_w s} z(s) \quad (12)$$

$$h_d(s) = -\frac{a_{12}T_w s}{1 + a_{11}T_w s} n(s) - \frac{a_{13}T_w s}{1 + a_{11}T_w s} z(s) \quad (13)$$

$$m(s) = \left(a_{21} - \frac{a_{12}T_w s}{1 + a_{11}T_w s}\right) n(s) + \left(a_{23} - \frac{a_{13}T_w s}{1 + a_{11}T_w s}\right) z(s) \quad (14)$$

The internal feedback of the turbine between power q, head h and rotational speed n can be obtained from power relations $P=\eta.\gamma.Q.H$ [8], and $P=M.\omega=2\pi.M.N$. Using the non-dimensional variables this feedback can be presented as:

$$p = \eta.g.Q_o.h + \eta.g.H_o.q \quad (15)$$

$$n = \frac{2\pi N_o}{P_o} p - \frac{(2\pi N_o)^2}{P_o} m \quad (16)$$

where η is the turbine efficiency, $P_0=M_0.\omega_0$ is the steady state power generated for a given steady state flow Q_0 and head H_0, and rotational speed N_0.

Using these relations, the block diagram of the hydraulic turbine for small variations operation

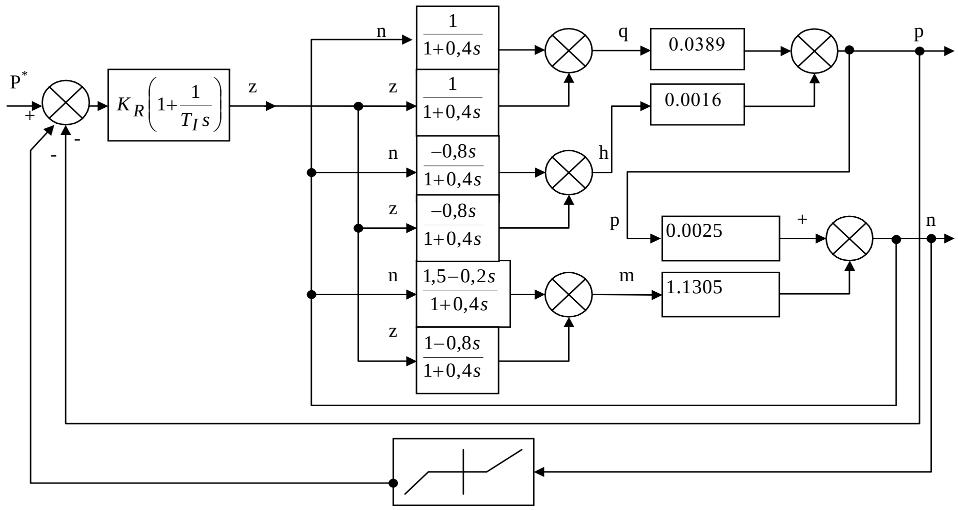

Figure 3. Block Diagram of the control system for hydraulic turbines

around the steady state point can be determined and we developed this diagram in [8].

The power control system is presented in figure 3. The transfer functions for different modules are given by the following relations, calculated for a Kaplan turbine:

$$H_{qn} = H_{qz} = \frac{1}{1+0{,}5.T_w s},\ H_{hn} = -\frac{T_w s}{1+0{,}5.T_w s} \quad (19)$$

$$H_{hz} = -\frac{T_w s}{1+0{,}5.T_w s},\ H_{mn} = \left(1{,}5 - \frac{T_w s}{1+0{,}5.T_w s}\right) \quad (20)$$

$$H_{mz}(s) = \left(1 - \frac{T_w s}{1+0{,}5.T_w s}\right) = \frac{1-0.5.T_w s}{1+0{,}5.T_w s} \quad (21)$$

B. Simulation results

Example. Let consider a hydroelectric power system with the following parameters:

-Water flow (turbines): $Q \in (500 \div 1000)$ m^3/s, $Q_N=725$ m^3/s; Water level in the reservoir: $H \in (17 \div 38)$ m, $H_N=30$ m; The equivalent cross-section of the penstock $A=60m^2$; Nominal power of the turbine P_N=178MW=178.000kW; Turbine efficiency $\eta=0{,}94$; Nominal rotational speed of the turbine N=71,43 rot/min; The length of the penstock $l=\Sigma l_i=20m$.

It shall be determined the variation of the time constant T_W for the hydropower system.

For the nominal regime, using relation (9), where $\Sigma l_i=20m$, the time constant of the system is:

$$T_W = \frac{725}{30} \cdot \frac{20}{9{,}81 \cdot 60} = 0{,}82s \quad (22)$$

This time constant has dependence with the variations of the water flow through the turbine Q and the water level in the reservoir H. In the normal conditions H=30m, and Q=725 m^3/s.

In figure 3 is presented the block diagram of the turbine's power control system, using a secondary feedback from the rotational speed of the turbine. It can be seen from this figure that a dead-zone element was inserted in series with the rotational speed sensor in order to eliminate the feedback for ±0,5% variation of the rotational speed around the synchronous value.

The constants of the transfer functions had been computed for a nominal regime T_w=0,8s. The optimal parameters for a PI controller are: K_R=10, TI=0,02s.

The results of the turbine simulation for different operational regimes are presented in figure 4, for a control system using feedbacks from the turbine power and rotational speed, with a dead-zone on the rotational speed channel for ±0,5% variation of the rotational speed around the synchronous value (a) Power variation with 10% around nominal value, b) Rotational speed variation for power control).

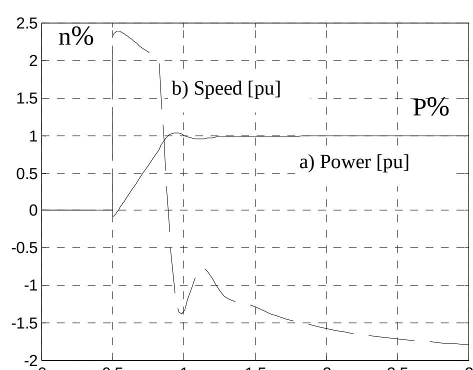

Fig. 4 Turbine Control simulations
a) Power, b) Rotational speed

In figure 5 are presented the variations of the turbine power (a) and rotational speed (b) for the control system using a feedback from the turbine power but no feedback from the rotational speed. The speed has big oscillations in the first period of transitory response (moore than 10%).

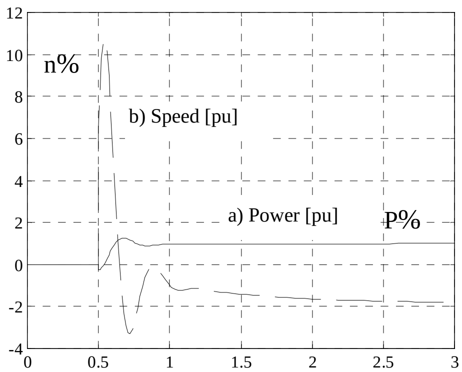

Fig. 5 Control structure with only power feedback
a) Power b) Rotational speed

This oscillation has no significant influence on the performance of the system but would have lead to permanent perturbation of the command sent to the turbine gate and it can produce dangereus situations. The hydro power groups are, usually, used for frequency control in power grid. In this case, the power group is set-up and set-down at the frequency variations outside of the normal limits. It is necessary to study the stability of the power control system after connexion.

The next set of experiments were realised with the hydro power group connected to the power grid. In this case the control system of the turbine power has oscillations in the first period of transitory response. In figure 6 are presented the rotational speed variation Sp (in percent units) and the power variation Power (in percent units).

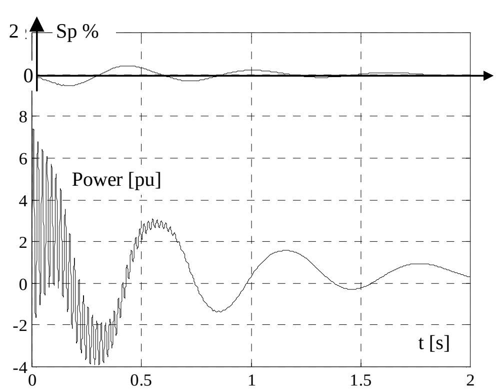

Figure 6. Power control response

4 Conclusion

The possibility of implementation of digital systems for monitoring and control for power, frequency and voltage in the cascade hydro power plant was discussed. The simplified mathematical models, capable to accurately describe dynamic and stationary behaviour of the hydro units have been developed and simulated. These aspects are compared with experimental results. Finally, a practical example was used to illustrate the design of controller and to study the system stability.

References:

[1] IEEE Committee Report, Hydraulic Turbine and Turbine Control Models for System Dynamic Studies, *IEEE Transaction on Power Systems*, vol. 7, no. 1, pp. 167-179, Febr. 1992.

[2] Kundur P., *Power System Stability and Control*, McGraw Hill, 1994.

[3] Arnaurovic D. B., Skataric D. M., Suboptimal design of Hydroturbine governers, *IEEE Transactions on energy Conversion*, Vol. 6/3, 1991.

[4] De Jaeger E., Janssens N., Malfliet B., Van De Meulebrooke F., Turbine Model for System Dynamic Studies, *IEEE Transaction on Power Systems*, vol. 9, 1994, pp. 1709-1715

[5] Hannet L. N., Fardanesh B., Field test to validate hydro-turbine-governor model structure and parameters, *IEEE Transaction on Power Systems*, vol. 9, 1994, pp. 1744-1751

[6] Kosterev D., Pierce M., Spence M., Hydro Turbine-governor model validation in Pacific Northwest, *IEEE Transaction on Power Systems*, vol. 19, May 2004, pp. 1144-1149

[7] Uddrill J., Woodward M., Nonlinear hydro governing model and improved calculation for determining temporary drop, *IEEE Trans. on Power Apparatus and Systems*, vol. PAS-86, no. 4, pp. 443-453, April, 1967.

[8] [4] M. Vinatoru, Monitoring and control of hydro power plant, *ICPS07-IFAC Workshop*, Mediamira Sc.Pub. 2007 pp.44-55 ISSN 978-973-713-180-5.

[9] Matko, ., Blazic, S., Geiger, G. Simulation of Multi-batch Driven Pipelines, 5th WSEAS International Conference on FLUID MECHANICS (FLUIDS'08) Acapulco, Mexico, January 25-27, 2008, ISSN: 1790-5117, ISBN: 978-960-6766-30-5, pp 53-59

[10] Moghaddam, M. A., Numerical Investigation on Transient Incompressible Fluid Flow, 5th WSEAS International Conference on FLUID MECHANICS (FLUIDS'08) Acapulco, Mexico, January 25-27, 2008, ISSN: 1790-5117, ISBN: 978-960-6766-30-5, pp 285-291

[11] Djanali,V.S.,K.C. Wong and S.W. Armfield, RANS Simulations of a Small Turbine Cascade, Proc. of the 4th WSEAS Int. Conf. on Fluid Mecha-nics, Queensland, Australia, Jan. 2007, pp. 113 117, ISSN: 1790-5095, ISBN: 978-960-8457-58-4 .