Adaptive multiple model algorithm for hydro generator speed and power control

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Abstract

Application of nonlinear models for local network speed and power control of hydro generator is considered. Such models are recommended under a wide range of load conditions. One way of achieving such control goals for systems with complex nonlinear dynamics is through application of multi-model adaptive control strategy. A bank of models is considered. Each of the models corresponds to a different load condition. An algorithm for probability estimation for each model is proposed. On their base "soft" adaptation is performed. The control signal is a weighted sum of particular controls designed for local load conditions. Such algorithm ensures smooth behaviour of the control signal. Simulations with proposed algorithm are carried out in order to explore the systems performance. The control system satisfies the technological requirements of such hydro engineering plants control.

Keywords : hydro turbine, modelling, multiple model algorithm, adaptive control

Introduction

The advanced turbine governors include the widespread application of electro hydraulic speed control both in the new construction and in modernization of older power plants. In modern programmable logical controllers more complicated and high effective control algorithms can be applied. These algorithms make most of the precise information about plant's model, load disturbances and measurement noises. Applications of computer control systems in hydropower engineering provide an opportunity to implement the recommended from *IEEE Working Group* more detailed models of the hydro generator power control under a wide range of load conditions [1]. One of the main challenges in the hydraulic turbine governor design is related to the nonlinear characteristics of the turbine. The turbine model parameters vary significantly with the unpredictable load variations. Such nonlinearities make the governor design a non-trivial task due to the fact that a controller designed for specific operating conditions may not perform well under different loads. This fact leads to the adaptive control algorithms [2]. However, in practice, the most governor design is based on the linearised turbine model at one load condition. The resulting controller is then de-tuned so that it can provide satisfactory performance for the worst operating conditions [3]. Jiang considers the analysis and design of a hydraulic turbine governor using optimal robust control theory [4]. The advantage of this approach is that the designed governor guaranties the stability and performance of the speed control for the entire load range.

One way of controlling hydro generators with complex nonlinear dynamics can be archived with multi-model adaptive control strategy [5, 9]. In this research a nonlinear model of the hydro turbine generator system is considered. The plant consists of three subsystems, the turbine wheel and rotor, the waterway with the nozzle and the servomotor to actuate the nozzle plunger. Models for initial speeding up of the turbine as well as four models for different load conditions are considered. These models are obtained using identification procedure. The multiple model algorithm consist of three stages: off-line system models determination; on-line mode probabilities estimation for each model; control signal computation via weighted sum of multiple local controls. The weighting coefficients are obtained by minimization of performance index. These parameters can be regarded as probabilities for the particular load condition, modelled in the model bank, to coincide with load conditions of the turbine. The adaptation provides smooth transition between local controllers as well as control of load conditions whish are convex combination of the modelled ones. The controller meets all requirements for water turbine process control. The results of the multi-model adaptive control strategy are compared with the control of the hydro generator by the same type of controller but designed with single model.

1. Turbine Dynamics Description

The research have been carried out by simulations with bank of *PI* control algorithms in MATLAB/Simulink environment with a nonlinear plant – hydro turbine generator, described in [6,7]. The dynamic behaviour of a hydroelectric power system is considered. It consists of a pipe that transports water from reservoir with water level h to a *Pelton* type impulse turbine. Between the outlet of the pipe and the turbine there is an actuator (*nozzle*) that adjusts the cross section A of the water flowing into the turbine. The input variable is a function of nozzle aperture variation while the turbine speed is the output.

The model of the system can be developed by deriving the model of the *water column*, the *control device model* and the *turbine shaft motion model*. The complete model can be obtained by connecting the three component models.

1.1 Turbine shaft motion mathematical model

The turbine shaft motion equation is given by the equation

$$J_t\dot{\omega}_t = M_t - M_l\,, \tag{1}$$

where J_t is the moment of inertia of the turbine shaft, ω is the turbine shaft speed and M_l is the load torque which is typically the driving torque for an electrical generator. All variables are presented in “per units” i.e. scaled by their rated values.

$$\frac{M_t}{M_r} = \left(\frac{\omega_{\max}}{\omega_r} - \frac{\omega}{\omega_r}\right)\frac{q}{q_r}\,, \tag{2}$$

$\frac{\omega_{\max}}{\omega_r} = 2$ for *Pelton* turbines.

Note that this will lead to a run away speed of twice nominal and a standstill torque twice the torque at nominal speed.

A small braking torque during the initial speeding up to the sub synchronous speed is due to the air and bearing friction. It is assumed to be proportional to speed:

$$\frac{M_l}{M_r} = a_f\left(\frac{\omega}{\omega_r}\right). \tag{3}$$

As typical value is used $a_f = 0.01$.

Thus

$$M_t - M_l = J\frac{d\omega}{dt}\,. \tag{4}$$

$$\frac{M_t}{M_r} - \frac{M_l}{M_r} = T_m\frac{d(\omega/\omega_r)}{dt}\,, \tag{5}$$

where

$$T_m = \frac{J\omega_r}{M_r}\,. \tag{6}$$

A typical value is $T_m = 3$ s.

Using equations (1), (2) and (3) it can be obtained

$$\frac{M_t}{M_r} - \frac{M_l}{M_r} = \left(\frac{\omega_{\max}}{\omega_r} - \frac{\omega}{\omega_r}\right)\left(\frac{q}{q_r}\right) - a_f\left(\frac{\omega}{\omega_r}\right). \tag{7}$$

1.2 Pipe dynamics mathematical model

The pipe is of length *L*, cross section A_p and has inlet at the elevation *h* where inlet pressure is zero. The outlet of the pipe has pressure p_p and volumetric flow *q* (water velocity $v_p = q/A_p$). The pipe can be regarded as a system with pressure and volumetric flow as output and input variables respectively. The inlet pressure is supposed to be the constant ambient pressure $p_a = 0$. Wherefore, the flow *q* at the outlet of the pipe will depend on the outlet pressure p_p. The linearized pipe dynamics is given is given by the transfer function

$$H_{pq}(p) = -\frac{\Delta p_p(p)}{\Delta q(p)}\,, \tag{8}$$

where $\Delta q = q - q_0$ and $\Delta p_p = p_p - p_{p0}$ are the deviation from a constant solution (q_0, p_{p0}). Note that the negative pressure change $-\Delta p_p$ is used in the definition of the transfer function to ensure that $H_{pq}(p)$ has positive gain.

Assuming that the water is incompressible with density ρ. The equation of the water motion in the pipe is

$$L\rho\dot{q} = mgh + A_p(p_0 - p_p)\,, \tag{9}$$

where mgh is the constant gravity force in the flow direction that acts on the water in pipe, p_0 is constant ambient pressure, and p_p is the pressure at the end of the pipe.

Then the water column transfer function is

$$H_{pq}(p) = -\frac{\Delta p_p}{\Delta q} = \frac{\rho L p}{A_p} \tag{10}$$

1.3 Water turbine control device mathematical model

The inlet of the control device has a constant cross section A_p and the inlet pressure is p_p. At the outlet of the control device the cross section is controlled to *A*, the pressure is *p*, and the water velocity is $v = q/A$. It is assumed that the outlet pressure is small and constant so that $p \approx 0$ can be used. The relation between the inlet and the outlet pressure and velocity can be described by *Bernoulli’s equation*

$$p_p = \frac{\rho}{2}\left(\frac{q^2}{A^2} - \frac{q^2}{A_p^2}\right). \tag{11}$$

Linearization of the control device equation (11) around the nominal area A_0 and a corresponding nominal flow q_0 results in

$$\Delta p_p = \frac{\rho q_0 \alpha}{A_0^2}\Delta q - \frac{\rho q_0^2}{A_0^3}\Delta A\,, \tag{12}$$

where $\alpha = 1 - \frac{A_0^2}{A_p^2}$.

Dividing equation (12) by Δq it can be obtained

$$\frac{\rho q_0^2}{A_0^3}\frac{\Delta A}{\Delta q} = \frac{\rho q_0 \alpha}{A_0^2} - \frac{\Delta p_p}{\Delta q}\,. \tag{13}$$

The turbine control device transfer function is

$$H_{qA}(p) = \frac{\Delta q(p)}{\Delta A(p)}\,,$$

where *A* is the input variable and *q* is the output variable. This transfer function is obtained after rearrangement equation (13) under the assumption $\Delta p_p(p) = -H_{pq}(p)\Delta q(p)$ in accordance with (10)

$$\left(\frac{\rho q_0 \alpha}{A_0^2} + H_{pq}\right)\Delta q = \frac{\rho q_0^2}{A_0^3}\Delta A\,.$$

Therefore the transfer function from the control input *A* to the flow *q* is

$$H_{qA}(p) = \frac{\Delta q}{\Delta A} = \frac{q_0}{\alpha A_0}\cdot\frac{1}{1 + \beta\frac{T_w}{2}p}\,, \tag{14}$$

where

$$T_w = \frac{2LA_0^2 q_{\max}}{\alpha q_0^2 A_p}\,,\quad \beta = \frac{q_0}{q_{\max}}\,. \tag{15}$$

Typical values are $T_w = 1$ s and $\beta = 0.01$ (during the initial speeding up to sub synchronous speed). This leads to a first order lag with unity gain and a very short time constant compared with the rotor run speed time constant T_m. Fig. 1 visualizes connections for the subsystems presented above.

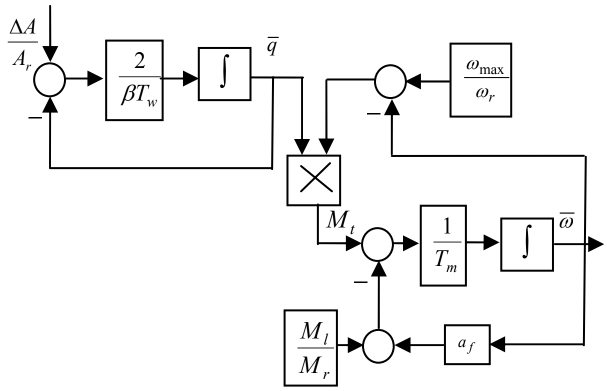

Fig.1 Model of the waterway and rotor dynamics

2. Speed and power control design of a power generating unit

Traditional approach in water turbine speed and power control is to use conventional controllers. In fig. 2 the simulation model for speed control of hydro turbine local network operation is shown. The main function of the turbine governor in this mode of operation is to compensate unpredictable load in order to keep the frequency on its nominal value. In order to obtain a zero steady state error in the case of load disturbance rejection an integral control low has to be used [8]. The cascade control system is proposed. It provides speed and turbine opening position feedbacks. The dead zone element in turbine opening position feedback is used in order to reduce the hydraulic servo-system fluctuations in accordance with the technological requirements.

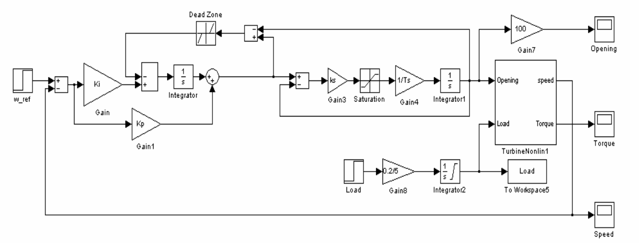

Fig.2 Simulation model of hydro turbine speed control

A continuous *PI* controller is designed using *LQR* technique. The following controller parameters are obtained:

$K_p = 2.54$, $K_i = 0.3$.

A linear plant model during initial speeding up is obtained using identification procedure. In fig. 3 a classical scheme of optimal model parameter tuning via identification procedure is presented. The optimal procedure is based on the minimization of performance index that presents the deviation between the measured output and the computed linear model output. The model input signal is the control signal. The linear model structure used by the identification procedure is shown in fig. 4

The obtained model parameters are *k*=98.33, $T_1 = 202.82$, $T_2 = 3.6$, $T_3 = 2.49$. Fig. 5 shows the original data and model response for the turbine speed and opening position during initial speeding up.

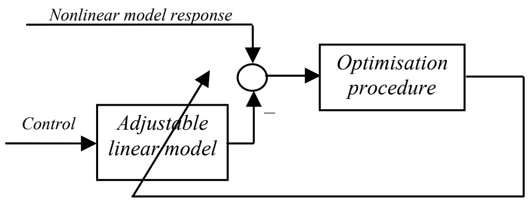

Fig.3 Model parameters tuning via identification

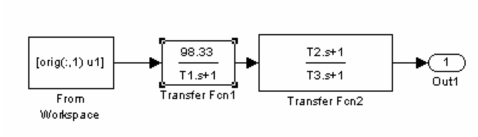

Fig.4 Identification model structure during initial speeding up

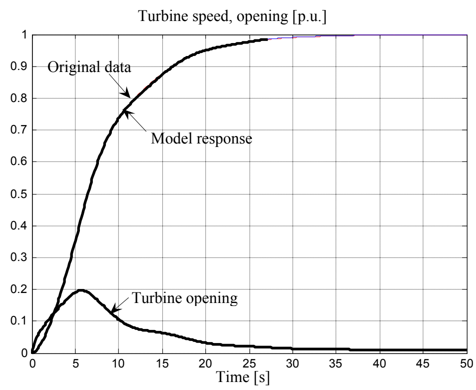

Fig.5 Turbine speed, opening position during initial speeding up (original data and model response)

The nonlinear characteristics of the hydraulic turbine vary significantly with the unpredictable load of the unit. Therefore the linearized plant models under different load conditions using identification procedure are obtained too. The model structure used by identification is shown in fig. 6. The model parameter values under different load disturbances are shown in table 1.

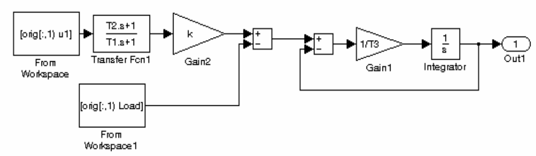

Fig.6 Identification model structure under load conditions

	k	T_1	T_2	T_3
Load 25%	4.84	11.83	14.13	24.74
Load 50%	3	14.55	18.54	20.94
Load 75%	2.29	16.09	23.06	20.3
Load 100%	1.95	18.87	28.6	21

Tab.1 Model parameters

3. Multiple model approach

In the considered multiple model approach there are three stages:

- The models of the system are obtained off-line;
- Probabilities for each of the model are computed on-line
- "Soft" control, as a weighted sum of local controllers, is computed.

3.1 Multiple models

Hydro turbine generator can be subjected to abrupt as well as gradual changes of the load. One way of describing this nonlinear system is by modelling it as a hybrid dynamic system, whose state may jump as well as vary continuously. These jumps between the different load conditions are used to model random abrupt changes in the generators demand, such as switching on or off other generators from the electrical grid. The dynamics between the jumps is used to model the system behaviour in the case of gradual load change. The hydro turbine generator is a system with two inputs and two outputs. The controlled output is the turbine speed. One of the turbine inputs is opening of the nozzle, which is control variable. The other one is the turbine load. The load is determined by the electricity demands and it is not possible to be controlled. Thus the second input has a characteristic of a disturbance. In order to be operational (connected to the electrical grid) the load of the turbine should be between 25% and 100%. In accordance to the multiple model (MM) approach it is proposed a set of four models to approximate the hybrid system. The chosen load models are 25%, 50%, 75% and 100%. They are presented in tab. 1.

$$M_l : y_l(k) = f_l(y(k-1), y(k-2), u(k), u(k-1)), \; l = 1, \cdots 4 \qquad (16)$$

Each of the equations corresponds to a particular system's load condition. Collectively these models are referred to as the model set *M*.

3.2 Adaptive control

Usually, control of the multiple model systems is done by choosing one model from the model set *M* (hypothesis testing) and then control action is applied based on the selected model as if it were the "true" one. The main drawback of such approach is that it only allows a hard decision i.e. only one model can be chosen at a given moment of time. This method does not give a good representation of partial turbine loads and hence the control performance can be poor in such case. The model set could be extended by adding partial loads modes to it. However this is not a solution, because when the models are too close to each other, problems with the statistical testing occur [10]. Furthermore, at the moment when the control system switches from one model to another (in case of a gradual change of the load) frequent jumps of the control signal may occur. Or even worse: in the case of a load condition which is between two models the switching between the models (jumps in the control signal) may occur very frequently. One way to solve this is by an extension of the method to linear differential inclusions. Considering the model set *M*, the linear differential inclusions are defined as a set of all systems that are a convex combination of the models in *M* [5, 9].

$$M : y(k) = \sum_{l=1}^{4} \mu_l(k) y_l(k) \qquad (17)$$

$$\sum_{l=1}^{4} \mu_l(k) = 1 , \qquad (18)$$

where μ is the vector containing the probabilities for each of the modes. Below this vector is called the mode probability vector. The probabilities in μ provides the certainty that this model is the true one. Probability 1 means the load conditions of the system are exactly the same as modelled for this particular model. Probability closed to 0 means that the load condition of the system is very different from the modelled ones. For the adaptation purpose, the mode probabilities must be calculated for each time instant.

The proposed controllers in this paper are from *PI* type. They have been designed on the base of *LQR* technique. A bank of such controllers has been created, in such way that there is a separate controller for each model (in *M*). In the paper it is proposed that the final control signal to be computed as a weighed sum from the control signals, obtained from the individual controllers. For the weights the mode probability vector is used. The input to the turbine is the opening of the nozzle.

$$o(k) = \sum_{l=1}^{k} \mu_l(k) w_l(k) , \qquad (19)$$

where the *o* is the input to the turbine, i.e. the opening of the nozzle and w_l are control signals computed from the individual controllers. Then the control of the system boils down to computing the probabilities for each model form *M*. The whole Simulink block-diagram is presented in fig. 7.

3.3 Probability estimation

The idea behind the probability vector is to make assessment of the load condition. In this paper as a performance index it is proposed to be used the difference between the real outputs of the system and the predicted ones [5]. Also in the working (on-line) regime the use of outputs is more convenient (less demanding from a computational point of view) than using estimates on their base. Of course, if the error in respect to certain model is small then the load of the system is close to the modelled one. In this paper it is proposed to be used the inverse of the square root of the error. The square root is chosen, because sign of the error is not important.

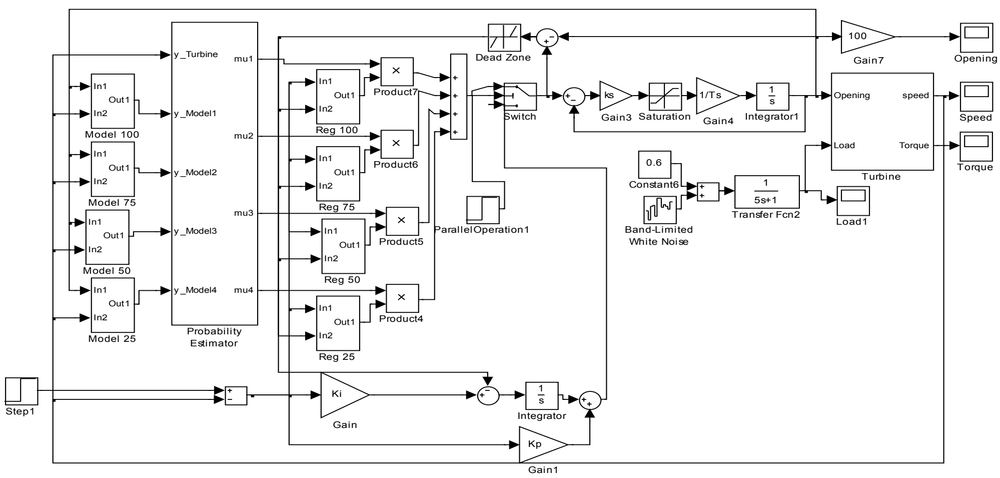

Fig.7 Simulink Block-diagram

$$\hat{\mu}(k) = \frac{1}{(y-y_l)^2} \Bigg/ \sum_{l=1}^{4} \frac{1}{(y-y_l)^2}, \; l = 1,\cdots 4 \,. \qquad (20)$$

In general, the real system is subjected to noise and disturbances. To make mode probabilities insensitive to the noise or to some disturbances, special methods have been developed. For linear systems, these include Kalman filtering, diagnostic observers, parity relations, parameter estimation and symptom analysis [10]. In the MM framework, techniques are available for linear systems with known mathematical models for all modes [10]. When probabilities are evaluated only on the base of the inputs and outputs of the system, only for the current time instant, it is possible that some momentary discrepancies may appear. In presence of strong noise the error for the correct model can become equal or even bigger than the other model(s). Another problem occurs in the transitions between two operating points (for example, after applying step signal as a reference). In order to overcome this problem it is proposed to be used the moving average of the probabilities estimation. This is done at the right side of fig.8. This slows down the probability transition process, but copes with these momentary discrepancies in the measured turbine speed.

$$\mu(k) = 0.98\mu(k-1) + 0.02\hat{\mu}(k) \quad l = 1,\cdots 4 \qquad (21)$$

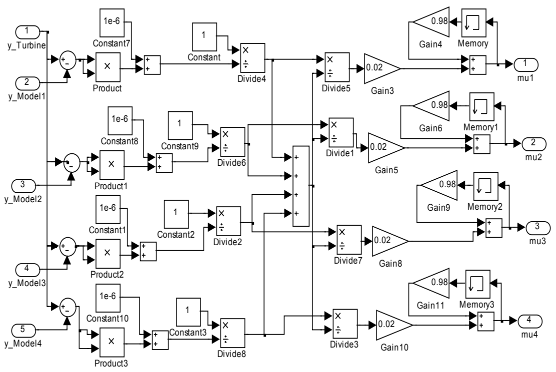

Fig.8 Model probabilities estimation

4. Simulation results

In this section the simulation results, obtained from the proposed algorithm are presented. Two experiments are carried out. The first experiments consist of varying the load conditions with a harmonic signal. The idea is to present the system performance in the whole operational range. Simulation is carried out for five hounded seconds. The results from the simulations are presented on fig. 9. In this figure the turbine speed, the opening and the load conditions are shown. In the first fifty seconds the hydro turbine generator reaches the nominal speed and in during the next fifty seconds the hydro turbine works in no load regime. Thus, the proposed algorithm starts from time $t = 100$ s. The goal of the proposed controller is to keep the speed close to its nominal value. It can be seen that only at the switching moment there is a speed deviation in about ten percent from its nominal value. After the initial switching of the proposed controller the speed deviation is less than two percent, while the load is changes in the whole operational range. This performance is in accordance with the UCTE primal frequency control requirements. This excellent quality of the system performance is due to the fact that the control signal is composed as a weighted sum from the outputs of multiple controllers, each of them designed for different load condition. Such results, for the whole load range can not be obtained with a single controller.

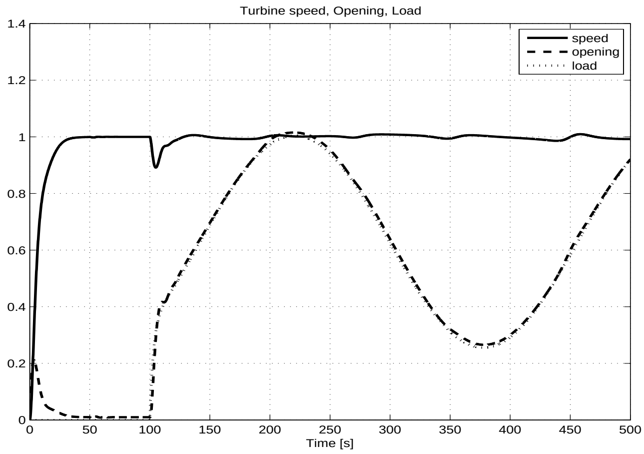

Fig.9 Turbine speed, opening in the case of sinusoidal load change

On fig. 10 the load conditions and the estimated probabilities are presented. It can be observed that the probability estimates track the load variation. This is true not only for modelled conditions, but also for any convex combination of the modelled ones.

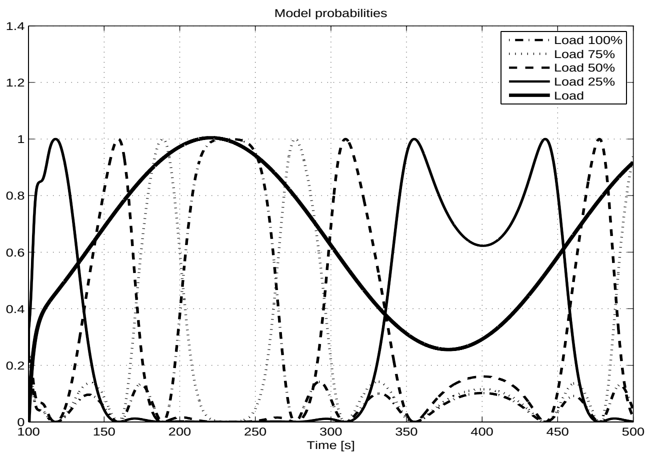

Fig.10 Model probabilities in the case of sinusoidal load change

In the next experiment random load conditions are applied. Every fifty seconds different load condition is simulated. Again, during the first hounded seconds the turbine speeds up to its nominal value and works in no load condition. The simulation time is again five hounded seconds. Although, hydro turbine generators typically work long time intervals under almost constant load conditions, such sudden changes in the load conditions can accrue. The turbine speed, opening and load conditions are presented in fig. 11.

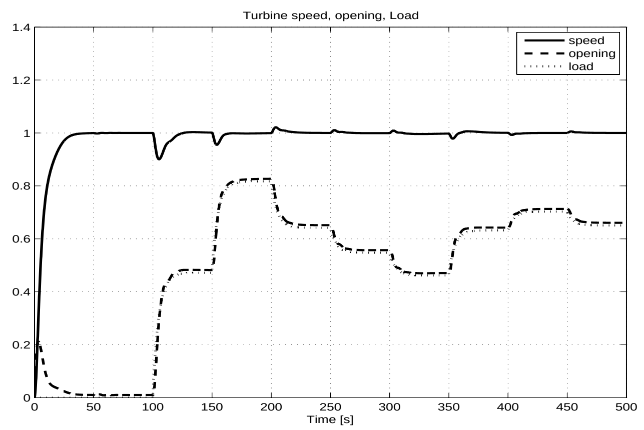

Fig.11 Turbine speed, opening in the case of random load change

It can be seen that the speed deviation is not greater that four percent and this corresponds to the load change in width range. The speed deviation is in the two percent margin when the load change is less than thirty percent. In fig.12 the mode probabilities are presented. The change of the load conditions can be compared with the mode probability estimates. Again it can be seen that the probability estimator correctly identifies not only modelled load conditions, but also each convex combination from the modelled ones.

The turbine speed, opening position and load conditions for the conventional turbine control based on a single controller are presented in fig. 13. It can be seen the speed deviation is about thirty percent for the fifty percent load change. Therefore the conventional controller designed at one load condition doesn't work well under different loads conditions.

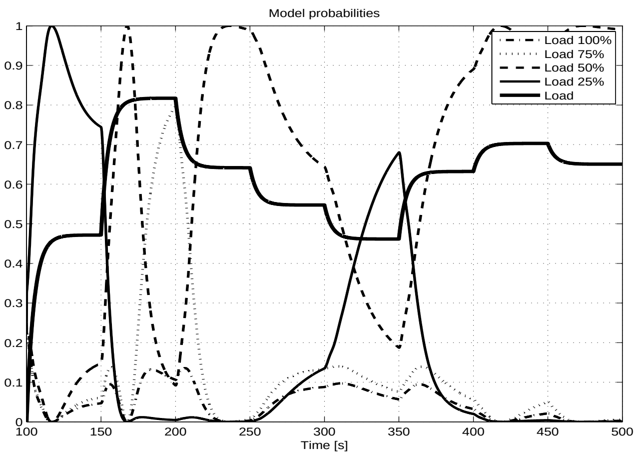

Fig.12 Model probabilities in the case of random load change

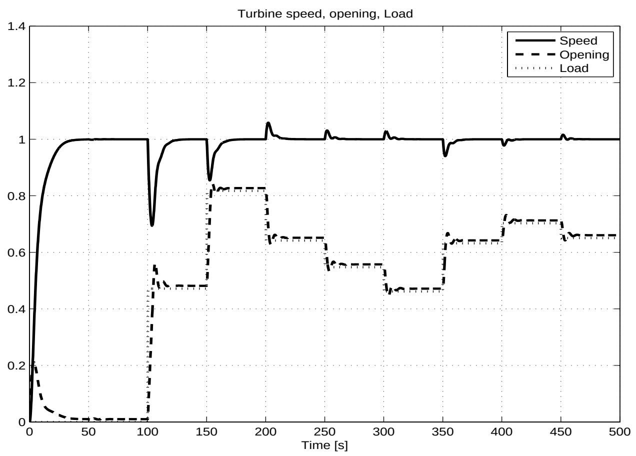

Fig.13 Conventional turbine control in the case of random load change

CONCLUSION

A nonlinear model of hydro turbine generator system is presented. A multi-model adaptive control algorithm is presented in this paper. This approach is applied to the hydro turbine generator speed and power control under wide range of load conditions. A bank of models is considered. Each of the models corresponds to a different load condition. An algorithm for probability estimation for each model is proposed. On their base "soft" adaptation is performed. The control signal is a weighted sum of particular controls designed for local load conditions. Different simulations are carried out. The results are discussed and compared with the conventional control based on a single controller.

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