Design of water turbine control based on localization method

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Abstract – This paper describes application principals of localization method to design control system for nonlinear dynamical model of water turbine. The purpose is to try to find universally applicable principal of water turbine control design. The model of system includes three different parts. The parts are servo governing turbine system model based on identification procedure, water turbine nonlinear model and the shaft balance model. All parts of the system are implemented in Matlab Simulink. In the paper are purposed two different structures of control system. The first structure includes all subsystems in one control loop. The second structure consists of two loops cascade where the internal loop consists of servo model and outer loop includes the turbine model and the shaft model. The control system's parameters are chosen on experimental basis due to the influence of non-linearity of the system which makes analytical approach ineffective. The robustness of the cascade control system is tested by altering parameters of the model. The parameters influencing system dynamics are altered in each subsystem of the model. The results are discussed and analyzed with focus on influence of the internal constants of the model and the disturbance to the output frequencies of the system.

Key words – Localization method, water turbine, control design, Matlab, nonlinear control system, non-linear model

Introduction

The stability of power grid is one of the most important requirements for the modern society. Fulfilling this request becomes more challenging since weather influenced renewable energy sources are implemented in the power grid in an increasing numbers. The way to solve this problem of weather dependent input of power to the grid is through storing the electric energy. The energy may be stored in electric form in battery storages which disadvantage is relatively small storage capacity [1]. Alternative way is presented in pumped storages power plant where the energy is stored in form of potential energy of water. The advantage is higher capacity compared to the battery storage. The disadvantage is the request to control of the water turbine [1–3].

Water turbine is a water drive of high efficiency and dynamics. The high dynamics makes possible to switch power output of the turbine in tens of seconds [6]. On the other hand the system includes several non-linear components which make design of an effective control system based on linear regulation principles complicated.

Localization method of control was purposed and studied at Novosibirsk State Technical University for more than 40 years [4]. The localization method is based on suppressing non-linear parts of the system in feedback through using high proportional gain of the internal loop with high dynamics.

Water turbines are now mostly governed by complicated structures based on linear control with improvements for compensating non-linear characteristics of the system (gainscheduling, predictors, ...) [5]. This approach allows to reach high dynamics and accuracy of the system output. On the other hand it is requested to design whole control system for each specific water turbine. Successful application of the localization method for water turbines control could simplify the turbine control design process.

1. Purposed Objective

The objective is to apply principals of localization method to a water turbine model and analysis of the feedback control loop. The main objective will be achieved in four steps:

- 1st step – implementation of the dynamical model of the system in Matlab Simulink.
- 2nd step – calculation of parameters for localization method control and its implementation in the model
- 3rd step – analysis transient process of the dynamical model and optimization of the feedback loop structure and its parameters
- 4th step – analysis of robustness of the feedback system and influence of disturbance

2. Implementation of the dynamical model

To design a control system based on localization method it is requested to know dynamic characteristics of the system [4]. These characteristics can be observed on a verified dynamical model of the system. Due the request to easy parameterization and the widest potential usage of the model it will be used simplified per unit model which can be applied for wide range of water turbine systems. For the purposes of analysis it is considered to simulate processes in Francis water turbine which widely used in all parts of the world [3, 6].

The model of the dynamical system can be separated into three different parts [6, 7].

The first part of the model is the actuator of system which is regulating the water flow through the water turbine. The type actuator of actuator depends on type of turbine. For purposes of modelling it is chosen hydraulic servo positioning the guide vanes. The basic scheme of the turbine is shown in fig. 1.

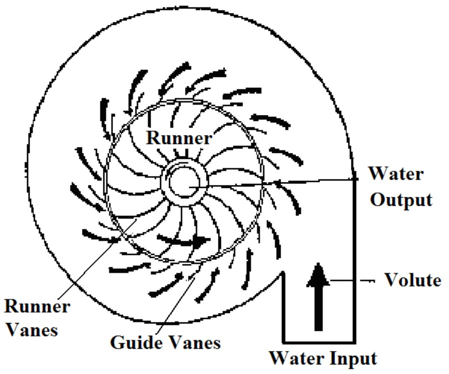

Fig. 1. Francis water turbine scheme

The model structure was chosen as linear time invariant transfer function of second order with one zero after analysis of data,

$$F_{servo}(s) = \frac{b_1 s + b_0}{a_2 s^2 + a_1 s + a_0}. \qquad (1)$$

The model parameters were estimated from identification data set and verified by verification data set reaching fit over 95 %. The model implementation in *Matlab Simulink* also includes constrain condition on output dynamics which was also estimated from data. The diagram of the servo model is shown in fig 2.

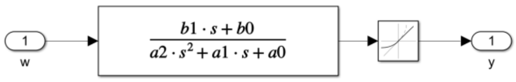

Fig. 2. Servo model diagram

The second part of the model dynamic relation between guide vanes position and power output of the water turbine. The presented model based on basic water turbine equation [7]

$$P = gHQ\eta, \qquad (2)$$

where P – output mechanical power of the turbine, g – gravitational acceleration, H – water pressure on the turbine, Q – water consumption of the turbine, η – efficiency of the turbine.

The basis for the model structure is based on the model recommended by IEEE commission. The model was modified to reach better fit during verification procedure. The proportional factor of the turbine was replaced by 1-*d* lookup table which characterizes the non-linear relation between water consumption of the turbine and output power. The model also contains a lot of saturation blocs since the signals are restricted to per unit range from 0 to 1. The modified model diagram of water turbine is shown in fig 3.

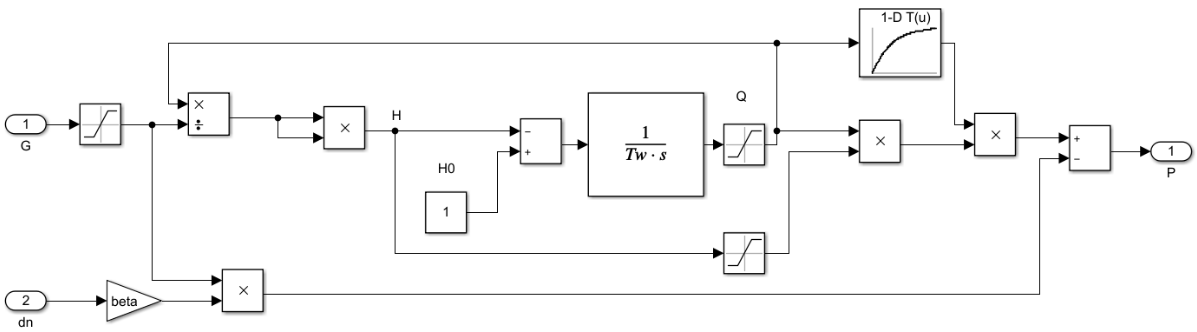

Fig. 3. Turbine model diagram

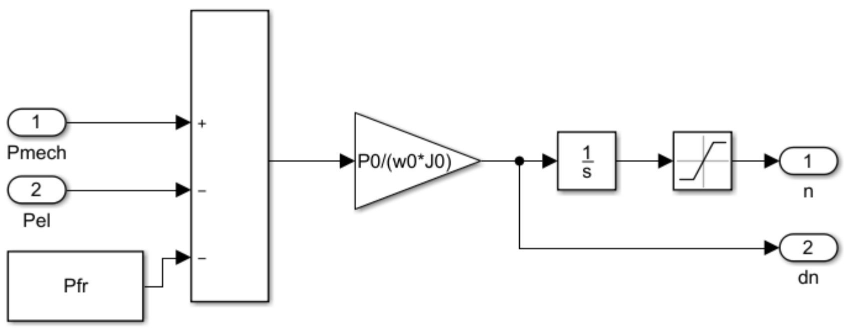

Fig. 4. Shaft model diagram

The third part of the model describes the shaft dynamics and relation between mechanical power output of the turbine, electric power output of the generator the friction losses and the shaft speed output. The friction losses were simplified to constant due to lack of data. The shaft is described by equation

$$\frac{d\omega}{dt} = \frac{P_0}{J_0\omega_0}(P_{mech} - P_{el} - P_{fr}), \qquad (3)$$

where P_0 – maximum power output, ω_0 – normalized shaft speed, J_0 - moment of inertia of the system, the constant $P_0/(J_0\omega_0)$ makes possible to calculate the output while maintaining the per unit value of the output. The equation is implemented in fig 4.

The saturation limits the output to values above zero since it is considered that the rotation orientation of turbine cannot switch to the other direction.

The whole system is created by connecting subsystems together as is shown in fig 5.

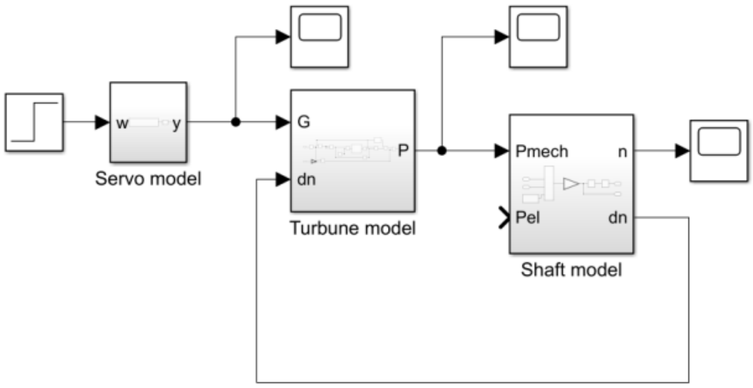

Fig. 5. Complete model diagram

3. Calculation of parameters for application of localization method

The common non-linear dynamic system can be described by equation

$$y^{(n)} = f(t, y, ..., y^{(n-1)}) + b(t, y, ..., y^{(n-1)}). \qquad (4)$$

where dependence of functions *f*(.) and *b*(.) on *t* parameter represents the disturbance influence and changes in system parameters over time. Even though *f*(.) and *b*(.) values are not necessarily known it is assumed that their range of possible values is known [4].

The goal of method is to find actuation which will satisfy the requirement

$$\lim_{t \to \infty} y = w, \qquad (5)$$

where y – output of the system, w – desired value.

This includes request to fulfil the steady state accuracy and dynamic behaviour

$$|\Delta(\infty)| = |w(\infty) - y(\infty)|, \qquad (6)$$

$$t_s \le t_{\max}, \sigma \le \sigma_{\max}$$

where Δ – error, t_s – setting time, σ – overshoot.

The feedback loop is focused to stabilize the shaft rotation speed at desired value and the requirements are defined as follows:

$$\Delta_{\max} = 0.01, t_{\max} = 40s, \sigma_{\max} = 0$$

To reach the desired dynamic behaviour of the system we must define desired equation in the form:

$$y^{(n)} = f(t, y, ..., y^{(n-1)}, w), \qquad (7)$$

The order of the equation must be calculated as

$$l = m - n, \qquad (8)$$

where l – desired equation order, m – number of integrators, n – number of zeros.

The order of the whole model is

$$l = 4 - 1 = 3, \qquad (9)$$

The desired poles are set to:

$$p_1 = -0.5, p_2 = -\frac{1}{3}, p_3 = -0.25,$$

The desired function has form

$$\dddot{y} = \frac{u}{24} - \frac{13}{12}\ddot{y} - \frac{9}{24}\dot{y} - \frac{y}{24}, \qquad (10)$$

To reach the steady state accuracy it is recommended to set the gain k is purposed

$$b_{\min}k \approx (20...100),$$

It will be considered: $b_{min} = 1, k = 100$.

To access the information about the output derivations we require to design the differentiating filter. The order of the filter is equal to the order of the desired function.

$$\mu^3 \dddot{\hat{y}} + a\mu^2 \ddot{\hat{y}} + b\mu \dot{\hat{y}} + \hat{y} = y, \qquad (11)$$

where μ - filter parameter describing filter lag.

For the purposes of the system we will choose parameters: $\mu = 10^{-3}$, $a = b = 7$. The differentiating filter is described

$$10^{-9}\dddot{\hat{y}} + 7 \cdot 10^{-6}\ddot{\hat{y}} + 7 \cdot 10^{-3}\dot{\hat{y}} + \hat{y} = y, \qquad (12)$$

The structure is implemented in block diagram (fig.6).

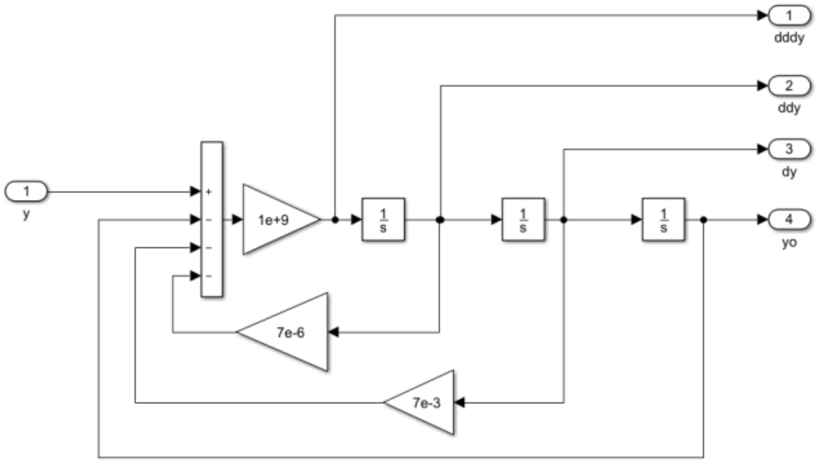

Fig. 6. Differentiating filter diagram

The regulator has got equation

$$u = 100 \cdot (c_1 w + c_2 y + c_3 \dot{\hat{y}} + c_4 \ddot{\hat{y}} - \dddot{\hat{y}}), \quad (13)$$

The values of the constants are:

$$c_1 = \frac{1}{24}, c_2 = \frac{-1}{24}, c_3 = \frac{-3}{8}, c_4 = \frac{-13}{12},$$

The structure is implemented in fig 7.

With all parts prepaired the feedback loop can be implemented in simulink (fig. 8).

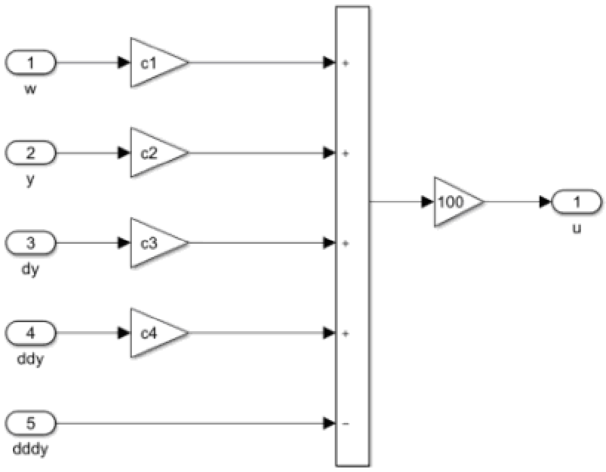

Fig. 7. Regulator diagram

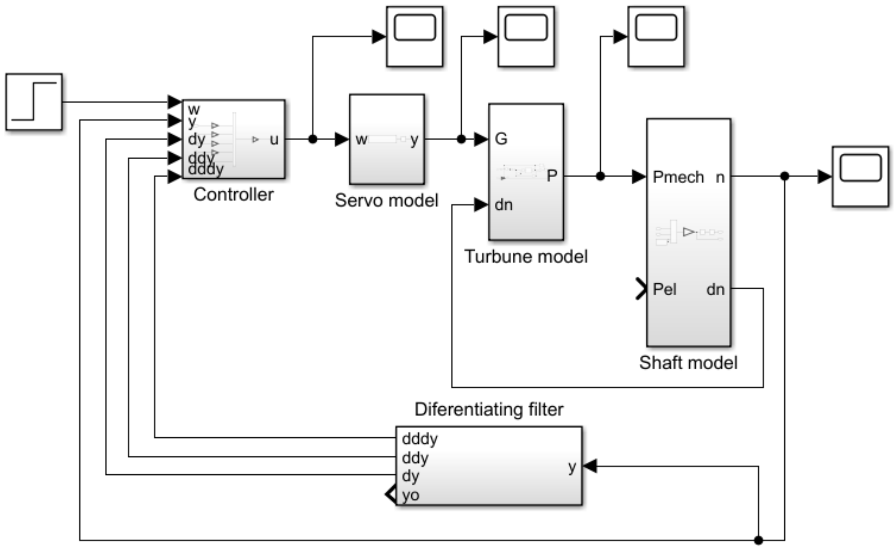

Fig. 8. Feedback control diagram

The designed feedback loop unfortunately doesn’t stabilize the output value and it leads to the unstable transients process which can be seen in fig 9.

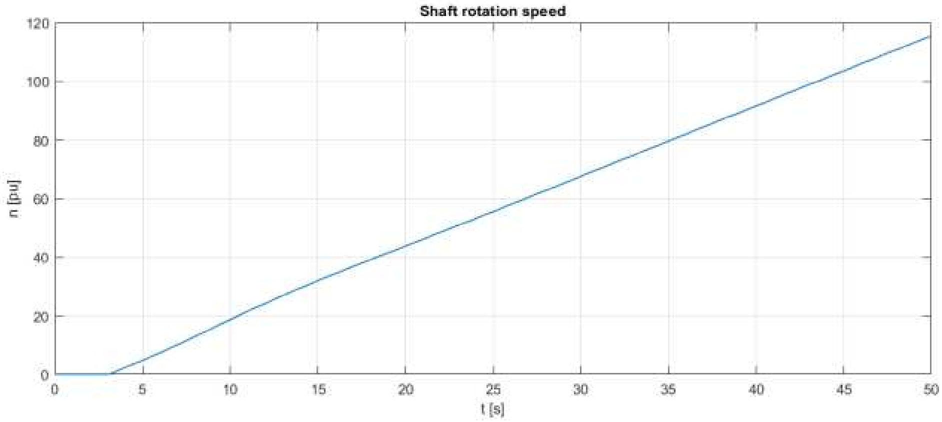

Fig. 9. Feedback control output

4. Transient process analysis and control optimization

The basic method of control design has failed and the non-linear features of the dynamical system do not allow usage of analytical methods to caltculate the control system paremeters. In this case the heuristic methods and manual setting take place. The paremeters are set on experimental basis.

Through the experimentation with the controller and the differntiating filter parameters was possible to reach the stabilization of the output but the value of the output did not correspond to the desired value. The system seems to display some sort of autonomy which can not be surpassed while using the current control structure of third order of regulator and the differentiating filter. The output can be seen in fig 10.

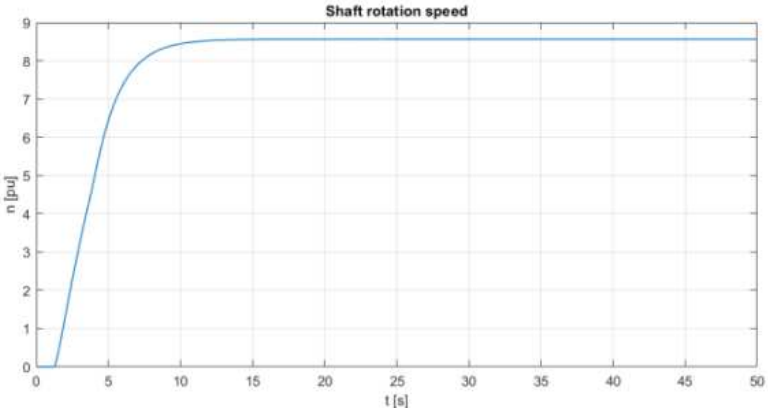

Fig. 10. Stabilized output

To reach the desired output value it is requered to change the control structure. The localization method allows to use a cascade controllers [4]. This will split the control into two loops. The internal loop governs the model of the servo and the external loop which controls the turbine model and the shaft model. It is shown in fig 11. This structure devides the model into the linear part with restriction of the servo and the non-linear of the turbine and the shaft. The cascade structure also simplifies the regulator and differentiating filters design and removes the potential problems with stability of these components.

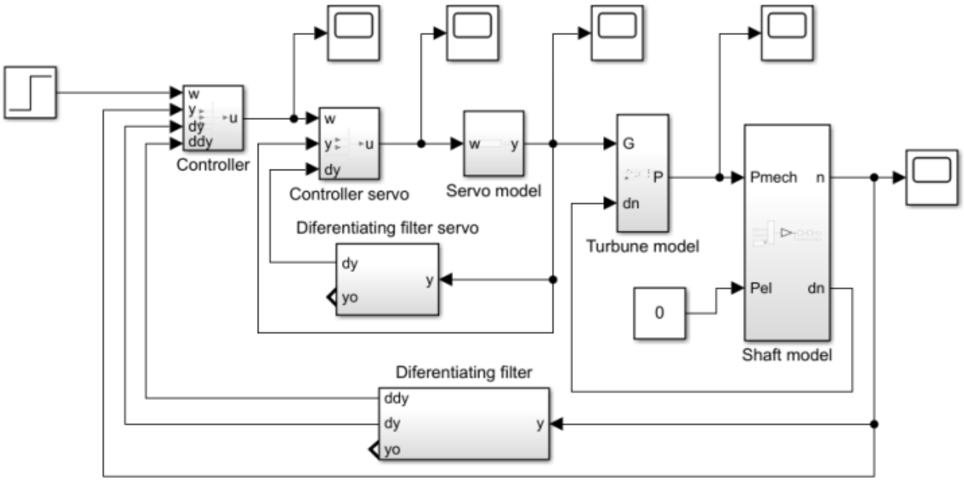

Fig. 11. Cascade feedback control diagram

The first step is to tune the internal loop to minimize its influence to the external loop. The requirements are set:

$$\Delta_{\max} = 0.01, t_{\max} = 4s, \sigma_{\max} = 0$$

The desired equation is set according to the dynamics requirements:

$$\dot{y} = 2u - 2y\,, \tag{14}$$

The static requirement is achived by proportional gain $k = 100$. For the differentiating filter is chosen parameter $\mu_i = 10^{-4}$. The differentiating filter for internal loop equation has form:

$$\mu_i\,\dot{\hat{y}} + \hat{y} = y\,, \tag{15}$$

The controller of the internal loop is characterized by equation:

$$u = 100\cdot(2w - 2y - \dot{\hat{y}})\,, \tag{16}$$

The external loop requirements are defined as follows:

$$\Delta_{\max} = 0.01, t_{\max} = 40s, \sigma_{\max} = 0$$

The desired equation has been chosen on basis of the experiment:

$$\dddot{y} = \frac{u}{21} - \frac{\dot{y}}{21} - \frac{9y}{21}\,, \tag{17}$$

During the experimentation it was found out that faster dynamics causes the transient process to lose monotonic motion. The slower dynamics causes the system's output overshoot.

The differentiating filter parameter is chosen as:

$$\mu_0 = 10^{-3}, a = 5$$

and the differentiating filter equation has following form:

$$\mu_0^2\,\ddot{\hat{y}} + a\mu_0\,\dot{\hat{y}} + \hat{y} = y\,, \tag{18}$$

The proportional gain of the controller has been set on basis of experiments to: $k = 9.3$. Through the experimentation it was found out that the gain k has direct influence on the steady state of output. The controller of the outer loop has is described by equation:

$$\dddot{y} = 9.3\cdot\left(\frac{u}{21} - \frac{9y}{21} - \frac{\dot{\hat{y}}}{21} - \ddot{\hat{y}}\right), \tag{19}$$

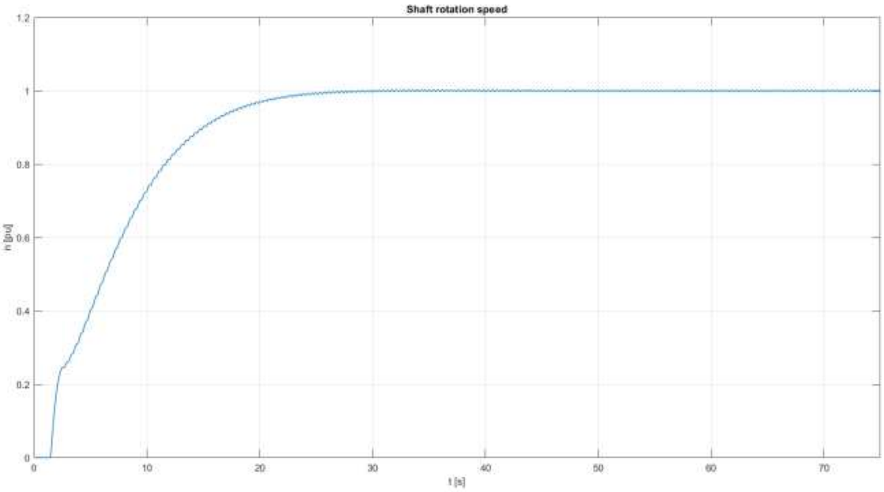

Fig. 12. Transient process of cascade control

In figure 12 the system reaches the desired value and is stable with small oscillations. The defined requirements of the transient process are met. The setting time is reduced to $\sim 30\ s$ due to mentioned influence of desired function on transient process.

5. Analysis of robustness and disturbance influence

The robustness of the control system is defined as capability of control system to achieve the regulation requirement even though the parameters of controlled system changes. The greater changes of the controlled system's parameters the control system can withstand without losing regulation requirement the more robust the control system is [10–12]. The robustness of the introduced control system will be evaluated on basis of changing parameters affecting the system dynamics. The system will be modified using the coefficient of dynamics change c which will multiply specific model parameter to test the robustness of the control system.

Firstly will be considered change in servo dynamics:

$$F(s)_{servo} = \frac{b_1 s + b_0}{a_2 s^2 + c \cdot a_1 s + a_0}, \qquad (20)$$

where c – coefficient of dynamics change.

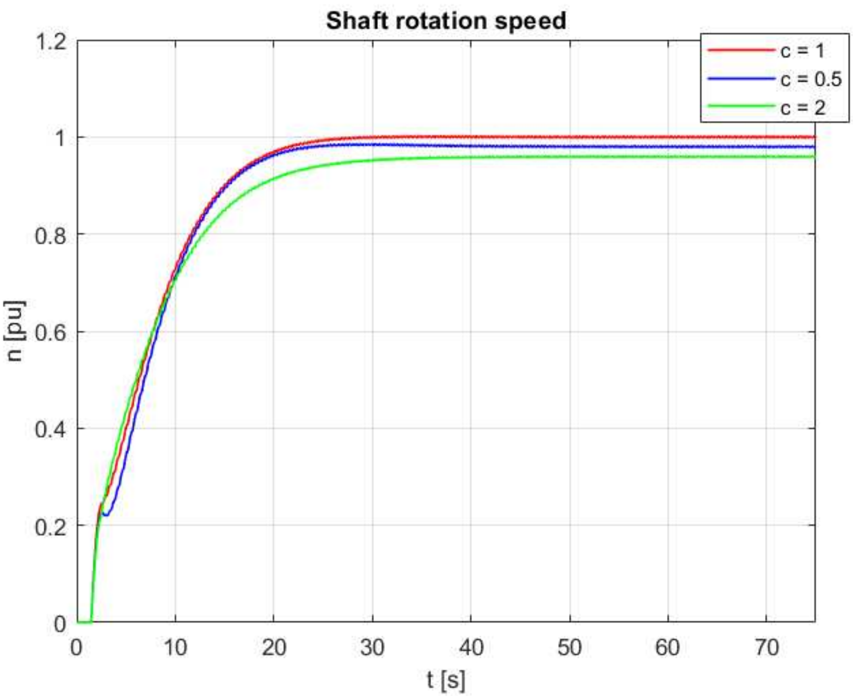

Fig. 13. Transient process variable servo dynamics

Both changes in parameters have change the dynamics at the beginning of the transient process but also affected the steady state of the system as it is shown in fig 13. The effect of dynamics change is relatively small because the servo model is placed in the internal cascade loop.

Secondly the water constant of the turbine will be changed as follows:

$$T_w = cT_{w0}, \tag{21}$$

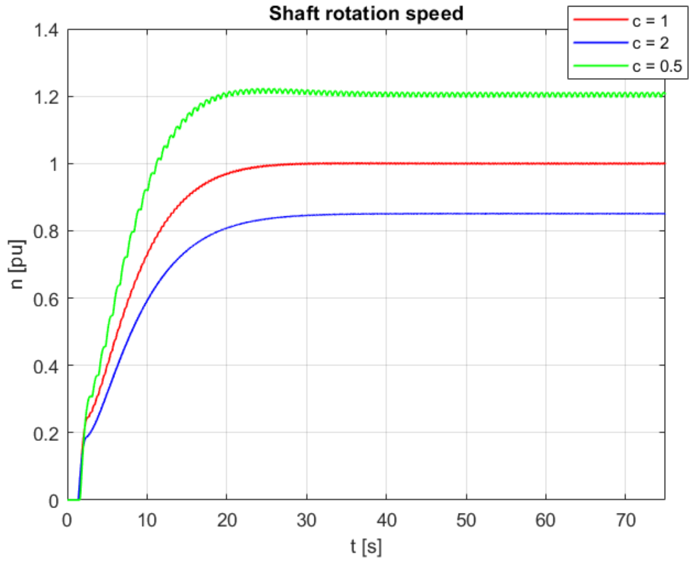

Fig. 14. Transient process variable turbine dynamics

In case of variable turbine water turbine constant we can see more expected results as is shown in fig 14. If the dynamics of the turbine is increased an overshoot shows up and the internal oscillations are highlighted. On the other hand reducing the dynamics hand further suppresses the internal oscillations. This is possibly because the turbine works increasingly as low pass opposed to the increasing the dynamics.

Thirdly the moment of inertia of the system will be changed in order to change the dynamics of the shaft subsystem:

$$J = cJ_0, \tag{22}$$

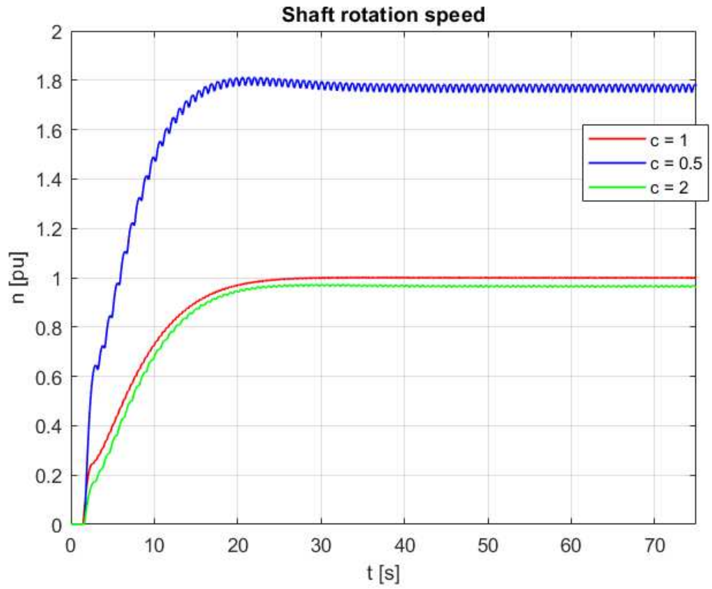

Fig. 15. Transient process variable shaft dynamics

From the result we can see that the oscillations are increased as the dynamics of the system is getting faster and the low pass characteristic of the decrease as is demonstrated in fig 15.

To explain the differences in the steady state of the system could probably be explained on basis of varying system's internal frequencies which change along with the dynamics. The frequencies may partially be caused by the internal constants of the model and the internal model feedback which is actually an algebraic loop. These internal frequencies may interfere with the control signals and cause the steady state inaccuracy.

From the experiments, it can be concluded that the nonlinear element in the general model of the shaft and turbine has the greatest influence on the output variable.

Next part is focused on the suppression of the disturbance. The disturbance is in the model introduced as the electric power of the generator which is connected to the water turbine via the shaft.

The model of the generator for the purposes of the experiment will be simplified to source of ramp signal with saturation of output value which is delayed till the transient process ends. The implementation is shown in fig 16.

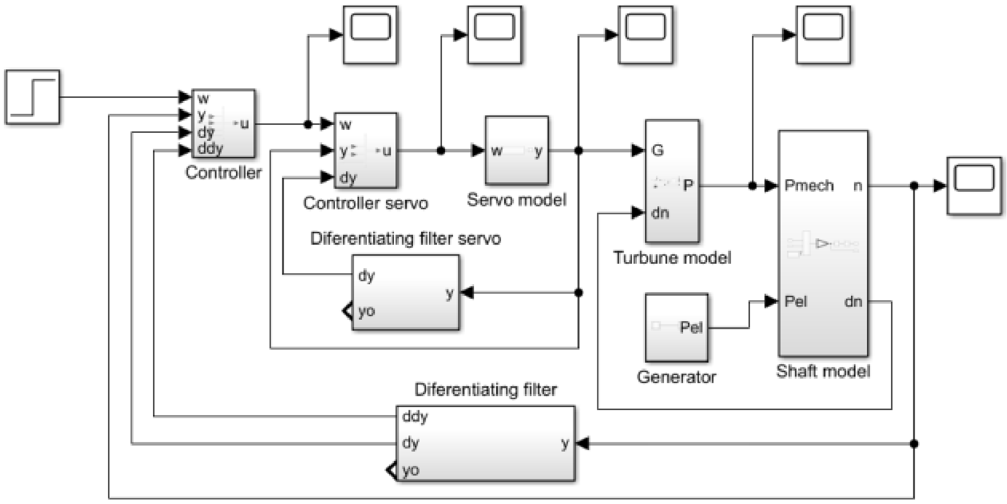

Fig. 16. Feedback control with generator

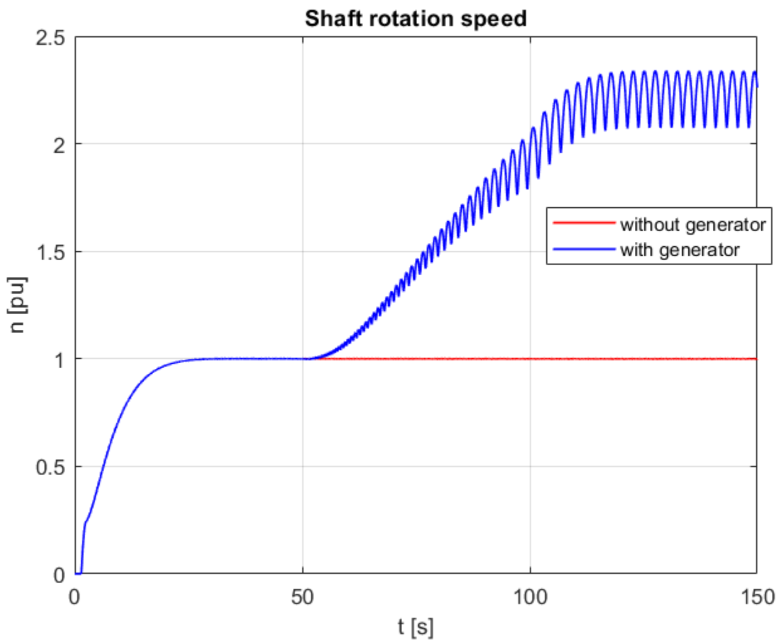

Fig. 17. Transient process with disturbance influence

The response to the disturbance is much more influencing the output than the changing the dynamics of the model in fig 17. The internal oscillations seems to be also mostly affected by the constants closest to the output of the system which can be verified by changing the friction constant

$$P_{fr} = cP_{fr0}, \tag{23}$$

The experiment confirms the hypothesis of the influence of the output constant. The friction constant strongly influences the internal oscillations of the output value and the stead state as well which is demonstrated in fig 18. At the beginning of the transient process can be seen the influence of the constant on smoothness of the curve.

Since the most problematic part of the model seems to be the shaft model constant so the next experiment will be focused on the regulation of the output power of the turbine instead of the speed control. The diagram is shown in fig 19. The shaft model is left in structure due to the systems internal feedback loop but it is considered:

$$P_{el} = 0$$
$$P_{fr} = 0 .$$

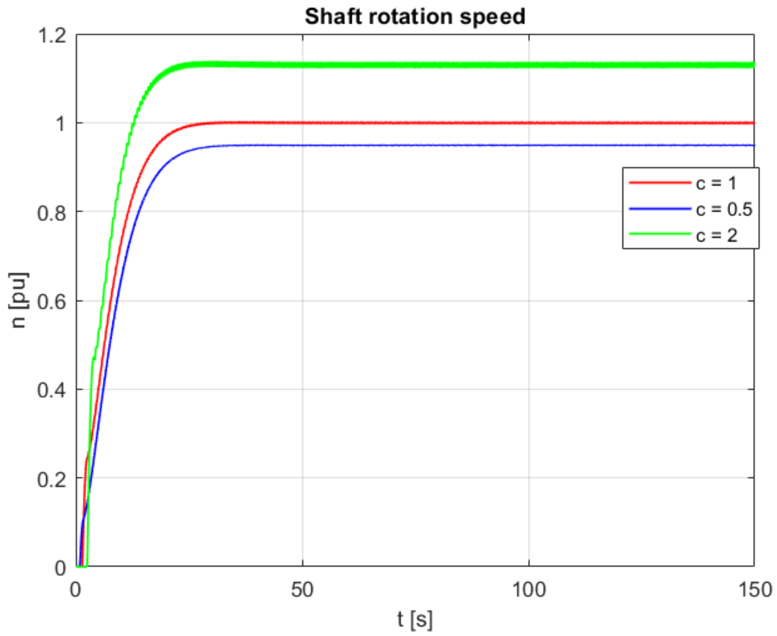

Fig. 18. Transient process with variable friction constants

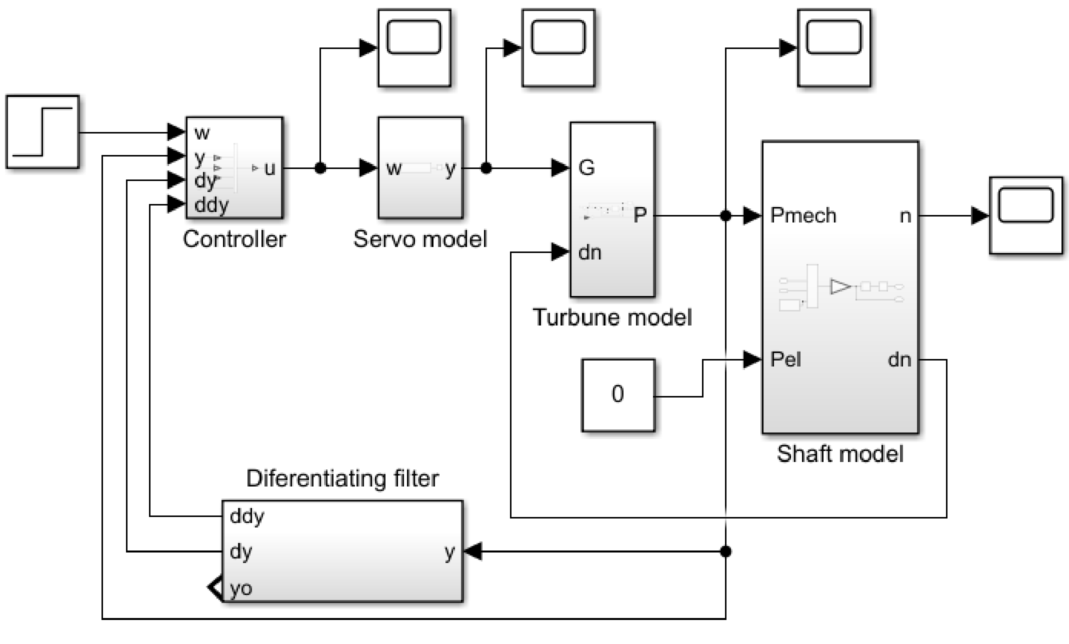

Fig. 19. Turbine power control diagram

Through the experimentation the parameters of the differentiating filter has been set to:

$$\mu_0 = 10^{-4}, a = 1{,}7$$

The regulator is described by this equation:

$$u = 23 \cdot (w - y - 2\dot{y} - \ddot{y}), \qquad (24)$$

Because the power output of the turbine is about 20-30 % during the setting speed regulation process the desired value is set to *w = 0.2*.

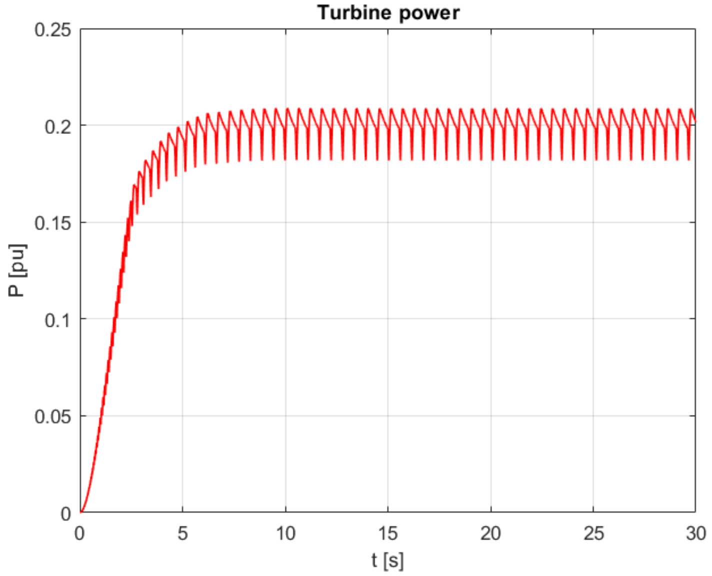

Fig. 20. Transient process power control

The output stabilizes in oscillations around the desired value in fig 20. The oscillations amplitude increases with output value while the frequency decreases with the output value.

The water constant of the turbine was once more changed due to inspect the robustness of the control loop, i.e.

$$T_w = cT_{w0}, \tag{21}$$

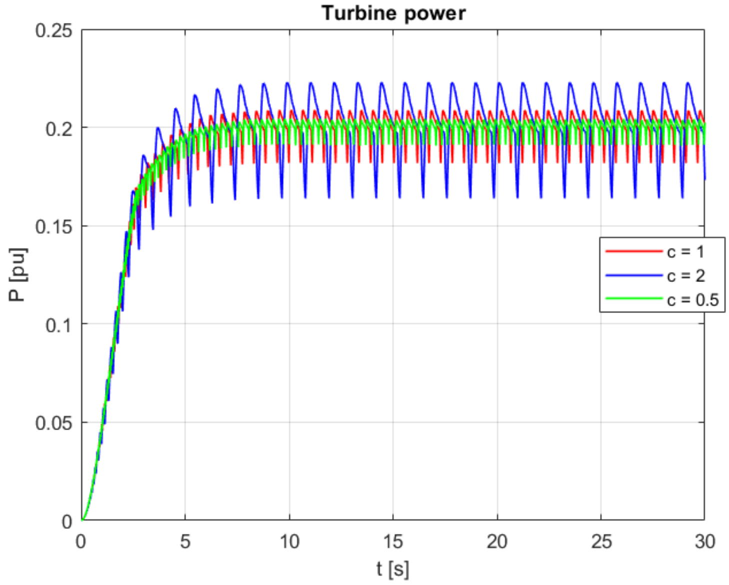

Fig. 21. Process power control variable turbine dynamics

The parameter changes in the model causes the changes in systems oscillations but the steady value around which the system is oscillating is constant in fig 21.

The oscillations generated by localization method control make impossible to use it in real system because the fast oscillations of turbine output could lead to its serious damage.

Conclusion

The dynamical model of water turbine implemented in Matlab Simulink is introduced. The coefficients of third order control system based on localization method are calculated on the basis of the model. Due to the system's specific behaviour the calculated coefficients cannot stabilize the output.

The internal non-linearity of the system is too influential to use analytical approach to calculate the control parameters so they are set based on experimental basis. The system output is firstly stabilized but the steady state output value cannot be controlled due the model's autonomy behaviour. The steady state control is acquired after splitting the control in two loops cascade structure.

The robustness analysis results show that changing of outer loop dynamics has higher influence on output signal. The steady states are also influenced by dynamics change.

Disturbance influence has demonstrated the major relation between the constant and disturbance in the shaft model and the oscillations of the system and dc gain as well.

The last experiment focused on controlling the turbine power output shows the higher robustness than the speed control but the oscillations of output makes impossible to use such control system on real system.

The results show that the localization method does not seems to be right governing system for the water turbine. The oscillations in the control loop in interferes with the systems internal frequencies. This phenomenon of localization method control in interaction with system's internal frequencies can be another step in research of localization method. The other way might be the analysis of the correlation between the order of localization method based control system and the autonomy behaviour of the controlled system with internal feedback.

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