Comparative Analysis for the Load Frequency Control Problem in Steam and Hydraulic Turbines using Integral Control, Pole Placement and Linear Quadratic Regulator

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Abstract– ***It is usual for electric power systems to include the following components: a load , an electrical generator (which will commonly be a synchronous machine), a turbine and a governor in order to control the turbine. The load frequency control problem is about the frequency control in power systems that are supplied with energy from a power plant (thermoelectric, hydroelectric). Therefore, the first step in order to study this problem is to model each one of the components of the system. Several different models will be presented, mostly for the modelling of the electric generator, load and the turbine. The response of three types of turbines under three different control techniques will be presented and compared: integral control, pole placement and linear quadratic regulator.***

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I. Introduction

Electrical frequency is one of the most important aspects to be considered in order to have a healthy power system operation. The inductive and capacitive loads are highly dependent on the frequency applied to theme. Electrical machines can suffer significant damage under inadequate frequency application.

Most of the electrical loads are fed from a power plant that will typically use a synchronous generator. Although loads have typical average values and tend to smoothly fluctuate around these typical power demands, there can eventually occur a load demand disturbance. A load disturbance can affect the speed of rotation of the electrical generator and a deviation from the nominal frequency will happen. What is desired is to guarantee that after the transient response, the system will always return to its nominal frequency.

There have been numerous studies regarding the behavior of the different type of power plants and their turbines. D.C Micea Duleau presented ad mathematical model of simple and reheat steam turbines, and also studied possible control techniques regarding this type of turbine [1], [2]. The have been comparisons made between the response of steam turbines and hydraulic turbines [3].

It is also of extreme importance in this problem to have an adequate model for the governor. Different models for hydraulic turbine governors were found in a review of the most common models used in America [4]. Mohit Kumar Pandey presented simulations of nuclear and hydroelectric systems using different governor models [5]. Sachin Kajuria explored the response of hydro-thermal systems [3]. There are available master degree thesis regarding the load frequency control problem. P.P.Pranab Patnaik analyzed the load frequency control for a single area and a single type of turbine using different control techniques [6], while P.K.R Sushmita Eeka presented a thesis about interconnected areas [7].

The layout of this paper is at follows: after a shot introduction, the modelling of the different elements in the system is presented in section II. The block diagrams and control techniques will be explored in section III, while the simulation results will be presented in section IV. The paper will be concluded in section V.

II. Mathematical Modelling of the system

In order to analyze the load frequency control problem, the following elements must be modelled: speed governor, turbine, power system and electrical generator. The output of the system will be the deviation of the frequency from the nominal one, as shown on (1):

$$\Delta f = f_{actual} - f_{nominal} \tag{1}$$

A sudden change in the power demanded from the load will result in a variation of the frequency in the generator. The relationship between a sudden change in frequency and power change in the generator is the regulation parameter R. Considering that there will be no change in the typical power of the load and all the load changes will be treated as disturbances, the change in the generated power will be:

$$\Delta P_g = -{}^{\Delta f}/_{R} \tag{2}$$

This generated power change signal will be received by an actuator and it is the actuator that will change the valve positions at the entrance of the turbine. For steam turbines, the model of this actuator will typically be [2], [6]:

$$G_{actuator}(s) = {}^{K}/_{1 + T_g s} \tag{3}$$

The actuator models used for the control of hydraulic turbines are fairly different from the simple first order model used in the case of the steam turbine. One common model for a hydraulic actuator is the HYGOV model [4]. It is a fairly precise model, but as it is modelled considering even massive load changes, it takes into account many non-linear effects. The HYGOV model requires the knowledge of several parameters and its complexity is not justifiable if the load changes will be fairly small, as in this case. IEEE has created several models for hydraulic actuators. Even though some of the models are fairly limited (such as the IEEE2), they are commonly used in the analysis of the load frequency problem. This is because they are simpler than the HYGOV family models and are still valid if the load changes are relatively small. In the governor models review, the percent of use for many models in the United States of America is presented, as we can appreciate.

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TABLE I

USE OF DIFFERENT HYDRAULIC GOVERNOR MODELS

Model	Total percent of use (%)
HYGOV	64.3
HYGOV 2	3.2
IEEEG2	15.4
IEEEG3	2.4
PIDGOV	2.2
WEHGOV	7.2
WPIDHIY	4.3

The selected model for this paper is IEEEG2, as it is simpler than HYGOV models and as it is the second most used governor model.

The next step is to model the turbines. The simple steam turbine is commonly modelled as a first order system [1]:

$$G_{turbine}(s) = {K}/{1 + T_t s} \quad (4)$$

The typical time constant for simple steam turbines oscillates around 0.2 and 2 seconds [8] and its gain is usually taken as unitary [7]. When taking into account a single reheat stage, the model becomes:

$$G_{turbine-reheat}(s) = {1 + sk_r T_r}/{1 + sT_r} \quad (4)$$

The typical model of the hydraulic turbine is given by [9]:

$$G_{turbine-hydro}(s) = {1 - sT_w}/{1 + 0.5sT_w} \quad (5)$$

The hydraulic time constant may vary between 0.5 and 4 seconds, but the most typical values are around 1 second [10], [11].

Finally, the electric generator and power system model can be derived by using the swing equation for the synchronous generator [12]:

$$G_{power-system}(s) = {K_{ps}}/{1 + T_{PS} s} \quad (6)$$

III. BLOCK DIAGRAMS AND CONTROL TECHNIQUES

Once all the single elements of the system have been modeled, the interaction between all of them must be represented as a block diagram in order to analyze the complete dynamic behavior of the system. The two inputs will be a change in the typical power consumption (which for this type of problems will be zero, as all the load fluctuations will be considered as disturbances in the system) and the load disturbance.

Fig.1 is the general block diagram for a single area:

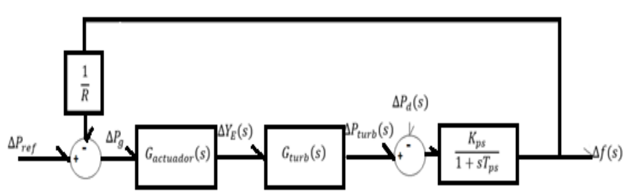

Fig. 1 Block diagram of a single area.

The equivalent transfer function for a simple steam turbine is given by:

$$G_{steam}(s) = \frac{K_{ps}R(1 + T_g s) * (1 + T_t s)}{K_{act}K_{turb}K_{ps} + R(1 + T_g s) * (1 + T_t s) * (1 + T_p s)} \quad (7)$$

The steady state for the frequency deviation applying a step load disturbance in power is:

$$\Delta f_{ss}(s) = -\frac{K_{ps}R\Delta P_d}{K_{actuador}K_{turb}K_{ps} + R} \quad (8)$$

The steady state frequency deviation for a hydraulic turbine area:

$$\Delta f_{ss} = -\frac{K_{ps}R\Delta P_d}{K_{gh}K_{ps} + R} \quad (9)$$

This means that without the use of any controller, both types of turbine will exhibit a steady state frequency deviation.

The most common type of control used in order to solve this type of situations is an integral loop. Fig.2 shows the block diagram modified including this control:

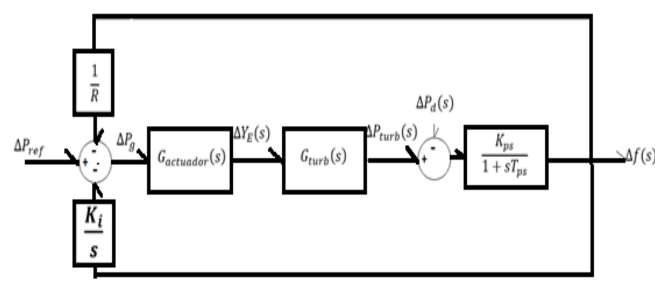

Fig. 2 Block diagram of a single area including integral control

The other proposed control techniques imply the analysis of the system using its representation in space state form:

$$\dot{x} = Ax + Bu \quad (10)$$
$$y = Cx + Du \quad (11)$$

It is desired to establish a control law with the following form:

$$u = -Kx \quad (12)$$

The pole placement technique consists in choosing the desired closed loop poles for the systems and looking for a vector K that will satisfy these conditions.

The linear quadratic regulator technique is based upon the minimization of a performance index:

$$J = \int_0^\infty [x^T(t)Qx(t) + u^T(t)Ru(t)]dt \quad (13)$$

IV. SIMULATIONS AND RESULTS

The first step in order to simulate systems is to select the parameters. The following table includes the values of the simulation parameters, as well as their meaning:

TABLE II
SIMULATION PARAMETERS

R	Regulation parameter	$3.60\ \frac{Hz}{p.uMW}$
T_g	Steam turbine governor time constant	$0.30\ s$
T_t	Steam turbine time constant	$0.35\ s$
T_w	Hydraulic turbine time constant	$1.50\ s$
K_p	Power system gain	$80\ \frac{Hz}{p.uMW}$
T_p	Power system time constant	$13.33\ s$
k_r	Reheat parameter	0.5
T_r	Reheat time constant	$10\ s$
T_{gh2}	Hydraulic time constant	$0.2\ s$
T_{k2}	Hydraulic time constant	$5\ s$
T_{k4}	Hydraulic time constant	$30\ s$

The block diagram in SIMULINK for the simple steam turbine is:

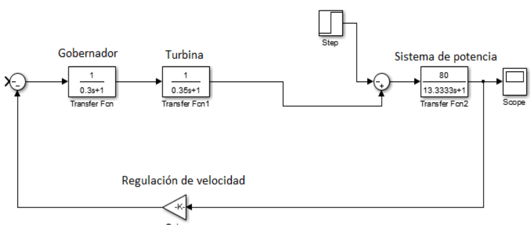

Fig. 3 SIMULINK block diagram for a simple steam turbine

The time response for the system after applying a 0.01 p.u disturbance is:

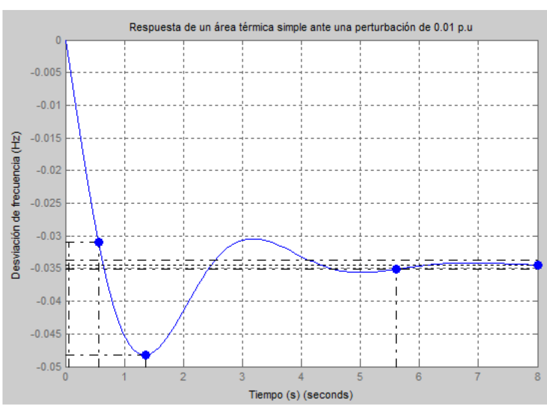

Fig. 4 Time response for the uncompensated simple steam turbine

The characteristics of the uncompensated transient response for the steam turbine system is presented in the following table:

TABLE III
TIME RESPONSE CHARACTERISTICS FOR UNCOMPENSATED STEAM TURBINE SYSTEM

Rise time	0.505 s
Peak amplitude	-0.0483 Hz
Settling time	5.600 s
Steady state value	-0.0344 Hz

As it was expected, the frequency of operation does not return to its nominal value after the transient response. The integral loop was included and the system response was simulated for different integral actions. Fig. 5 shows the response of the system by using different amounts of integral action.

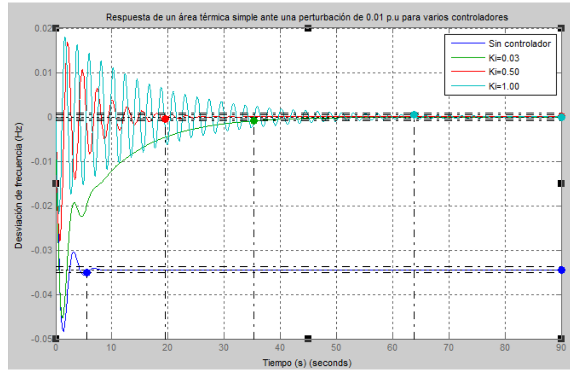

Fig. 5 Time response for the simple steam system with integral control

The steady state frequency deviation is eliminated by using any amount of integral action. However, there can be severe changes in the transient response that can be undesirable. There is an increase in oscillatory behavior, as well as a significant increment in the settling time when the integration constant is too large. Table IV presents the main characteristics of the transient response for all of the studied cases:

TABLE IV
TIME RESPONSE CHARACTERISTICS FOR COMPENSATED STEAM TURBINE SYSTEM

K_i	Rise time	Peak (Hz)	Settling time	Steady state
0.03	0	-0.04530	35.30	0
0.50	0	-0.0279	19.50	0
1.00	0	-0.0217	63.80	0

The state space representation of the system was calculated in order to apply pole placement and LQR.

$$\begin{bmatrix} \dot{x_1} \\ \dot{x_2} \\ \dot{x_3} \end{bmatrix} = \begin{bmatrix} -0.0750 & 6 & 0 \\ 0 & -2.8571 & 2.8571 \\ -0.9259 & 0 & -3.3333 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} + \begin{bmatrix} -6 \\ 0 \\ 0 \end{bmatrix} w \tag{14}$$

It is desired to reduce the settling time rather than making it larger as it happened by using the integral control. By using the pole placement technique and approximating the system to a second order one based on the closed loop pole locations, it is found for a 4 second settling time and a 5 rad/s natural frequency, the controller is:

$$K = [-0.1224\ \ 2.7059\ \ 3.6838] \tag{15}$$

The transient response by using the pole placement technique is shown in Fig 6. It is possible to appreciate a fairly smaller steady state frequency deviation, as well as a faster response and a smaller peak amplitude.

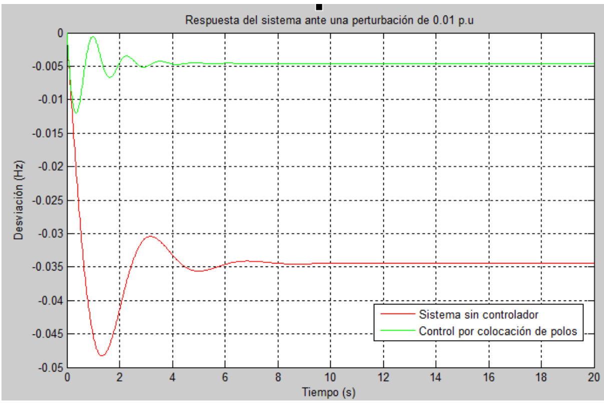

Fig. 6 Time response for the simple steam system with pole placement

Even though this type of control does not fully eliminate the steady state frequency deviation, it reduces it and offers a more desirable transient response.

In order to implement a linear quadratic regulator, the matrices Q and R must be chosen. The chosen matrices are:

$$Q = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}; R = 1 \tag{16}$$

The transient response in this case is given by Fig. 7.

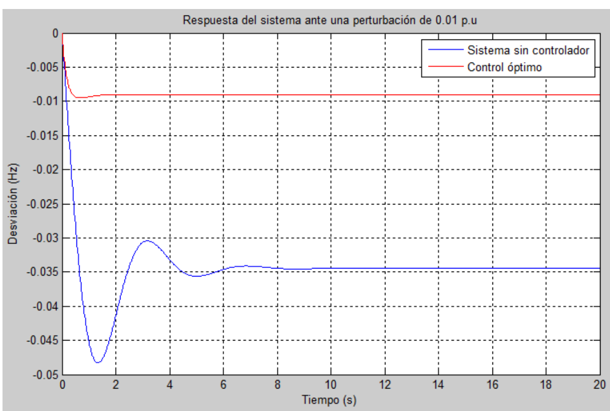

Fig. 7 Time response for the simple steam system with LQR; R=1

The settling time, as well as the peak response are much smaller than the ones the system had without any compensation. There is even more reduction in the steady state frequency deviation than what was achieved by using the pole placement technique. However, it is not possible to fully eliminate the steady state frequency deviation.

The R matrix can be seen as an indicative on how much control effort is been used to control the system. A smaller R indicates more control effort. Therefore, a simulation using R=0,10 (which indicates ten times more control effort than in the first simulation) was done in order to analyze the improvement in the response. The transient response by using a LQR with this new R matriz is shown in Fig 8.

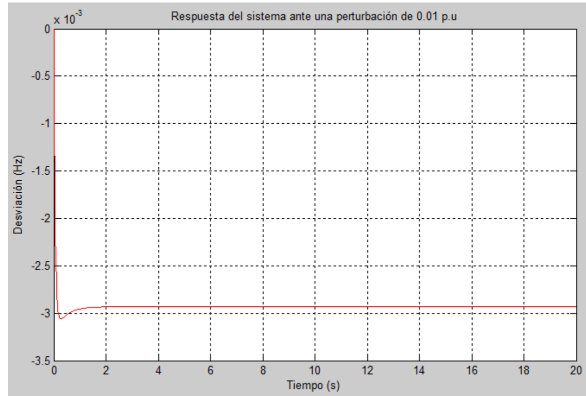

Fig. 8 Time response for the simple steam system with LQR; R=0,1

Even though the settling time is slightly larger than the one the system had with less control effort, the steady state frequency deviation is even smaller and closer to zero.

The next step was to reanalyze the case for the steam turbine, but now considering how a single stage of reheat can affect its dynamics. A block in series with the turbine is added in order to model the reheat stage, as is shown in Fig 9:

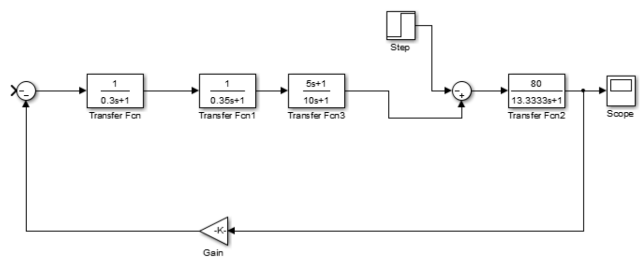

Fig. 9 SIMULINK block diagram for a reheat steam turbine

By using a 0,01 p.u load disturbance, the time response of the system is:

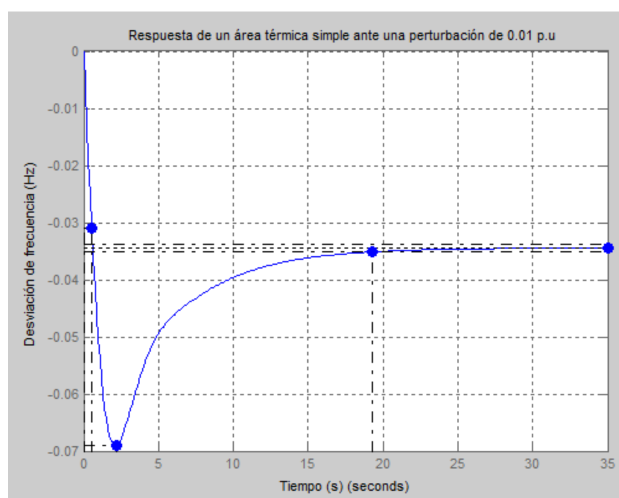

Fig. 10 Time response for the uncompensated reheat steam turbine

The characteristics of the uncompensated transient response for the reheat steam turbine system is presented in the following table:

TABLE V
TIME RESPONSE CHARACTERISTICS FOR UNCOMPENSATED STEAM REHEAT TURBINE SYSTEM

Rise time	0.485 s
Peak amplitude	-0.069 Hz
Settling time	19.200 s
Steady state value	-0.0344 Hz

There is no significant variation in the rise time because of the reheat stage, but the peak amplitude is substantially larger than in the simple steam turbine. The system exhibits a slower behavior, having a fairly larger settling time, but the reheat stage has no impact on the steady state operation, as the system has the same steady state frequency deviation.

The time response of the system by using the integral loop and varying the integral constant is presented in Fig.11:

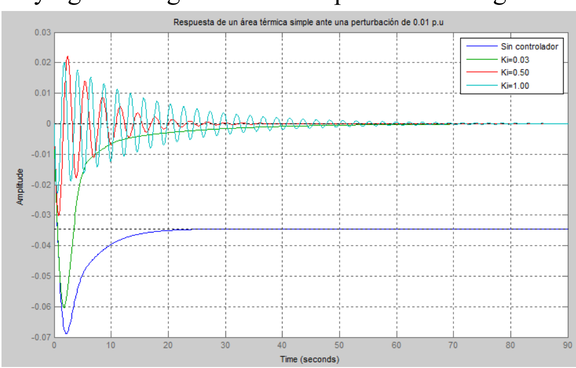

Fig. 11 Time response for the reheat steam turbine with integral control

The characteristics of the time response by using the different integral constants are:

TABLE VI
TIME RESPONSE CHARACTERISTICS FOR COMPENSATED STEAM TURBINE SYSTEM

K_i	Rise time	Peak (Hz)	Settling time	Steady state
0.03	0	-0.0603	34.90	0
0.50	0	-0.0302	25.60	0
1.00	0	-0.0227	63.50	0

As expected, the steady state frequency deviation is eliminated. The integral action tends to make the time response slower. It can be seen that the settling time relative to the uncompensated system is always larger, but the behavior is not linear or continuously increasing. There is a minima in the relationship between the settling time and the integral constant for this case. It can also be appreciated that the settling times and peak amplitudes for the compensated reheat steam turbine are fairly similar for the simple steam turbine.

The state-space representation for the reheat steam turbine system by using the parameters chosen is:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} -0.0750 & 6.0000 & 0.0000 & 0.0000 \\ 0.0000 & -0.100 & -1.3286 & 1.4286 \\ 0.0000 & 0.0000 & -2.8571 & 2.8571 \\ -0.9259 & 0.0000 & 0.0000 & -3.3333 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} -6 \\ 0 \\ 0 \\ 0 \end{bmatrix} w \qquad (16)$$

By putting the dominant poles of the system in the same position as in the simple steam turbine case and arbitrarily choosing two more non-dominant poles, the controller is found to be:

$$K = [-0.6058 \quad 443.2514 \quad -214.2693 \quad 4.0542] \qquad (17)$$

The system response by using the pole placement technique is:

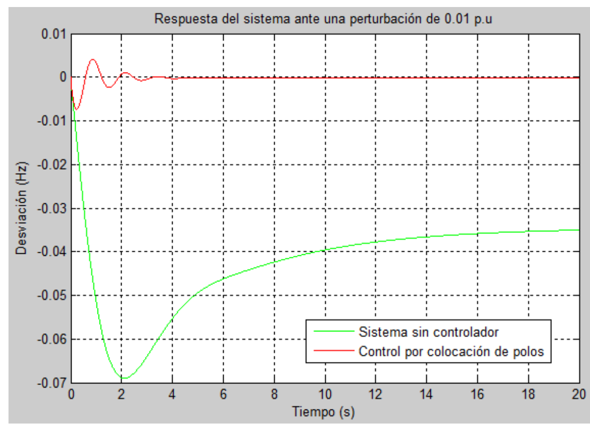

Fig. 12 Time response for the reheat steam system with pole placement

It can be seen that, in addition to the reduction in the settling time, there is an extremely small steady state frequency deviation, which was found to be -0.000152 Hz.

The LQR is now implemented in order to compare its performance with the previous controllers. The Q matrix is the identity and R is chosen to be 1.The controller in the state space is described as:

$$K = [-0.9982 \quad -0.6289 \quad 0.0505 \quad 0.0692] \qquad (18)$$

Fig. 13 shows the transient response for this case. Comparing with the pole placement technique, an even greater reduction in the speed of the system is seen. However, there is a larger steady state frequency deviation. The steady state frequency deviation is found to be -0.008901 Hz, which is one order of magnitude higher than what was obtained with the previous controller.

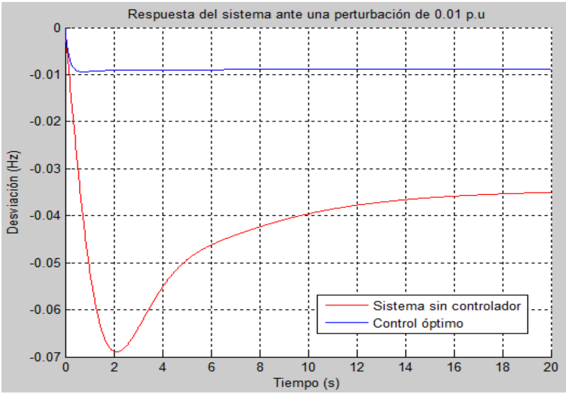

Fig. 13 Time response for the reheat steam system with LQR; R=1

The effort control was made larger by choosing $R = 0{,}1$ as a new value. The response of the reheat turbine under this circumstance is shown in Fig. 14.

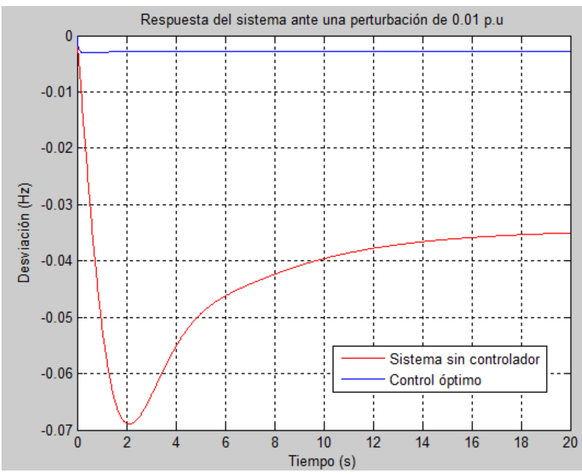

Fig. 14 Time response for the reheat steam system with LQR; R=0.1

Even though there is a significant reduction in the steady state frequency, it continues to be significantly larger than the one obtained by using the pole placement technique. In order to achieve a similar steady state value with the LQR, the control effort would need to be several orders of magnitude larger.

The last system that was analyzed was the hydraulic one. Hydraulic turbines characterize themselves for being more difficult to control as they have a zero in the right half-plane. The model for the governor is more complex. Fig. 15 shows the SIMULINK block diagram used in order to simulate this system.

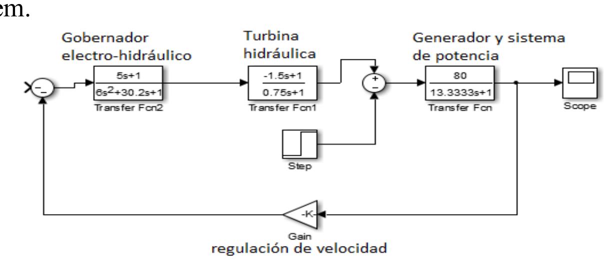

Fig. 15 SIMULINK block diagram for a hydraulic turbine

The time response of the uncompensated system is shown in Fig. 16.

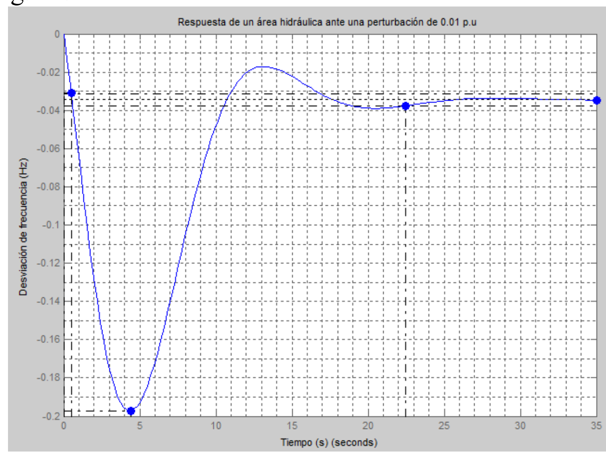

Fig. 16 Time response for the uncompensated hydraulic turbine

The characteristics of the uncompensated transient response for the hydraulic turbine system is presented in the following table:

TABLE VI
TIME RESPONSE CHARACTERISTICS FOR UNCOMPENSATED HYDRAULIC TURBINE SYSTEM

Rise time	0.441 s
Peak amplitude	-0.197 Hz
Settling time	22.500 s
Steady state value	-0.0344 Hz

The hydraulic system is slower than the steam turbine ones, as it is shown by comparing their settling times. There is a larger peak amplitude for this system. No change for the steady steate frequency deviation is seen when comparing the system to the steam ones.

The response by using different integral actions is shown in Fig. 17:

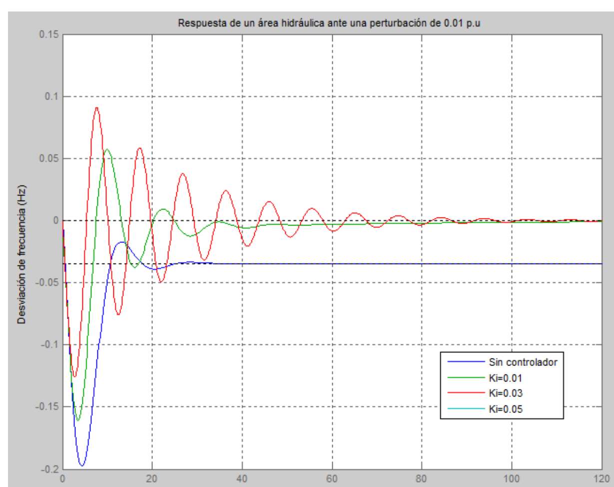

Fig. 17 Time response for the hydraulic turbine with integral control

The characteristics of the time response:

TABLE VII
TIME RESPONSE CHARACTERISTICS FOR COMPENSATED STEAM TURBINE SYSTEM

K_i	Rise time	Peak (Hz)	Settling time	Steady state
0.01	0	-0.161	56.60	0
0.03	0	-0.126	80.80	0
0.05	0	-0.107	458.00	0

The hydraulic system can easily become extremely slow and exhibit an oscillatory behavior. The peak values tend to be larger than in the previous cases.

The space-state representation for the hydraulic turbine system is given by:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} -0.075 & 6.000 & 0.000 & 0.000 \\ 0.463 & -1.33 & 1.399 & 1.600 \\ -0.231 & 0.000 & -0.033 & -0.80 \\ -1.389 & 0.000 & 0.000 & -5.00 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ -1.666 \\ 0 \\ 5.000 \end{bmatrix} u + \begin{bmatrix} -6 \\ 0 \\ 0 \\ 0 \end{bmatrix} w \qquad (19)$$

By choosing the same desired closed loop poles as in the previous case, the controller is calculated and given by:

$$K = [-0.6000 \quad -24.400 \quad 1243.2 \quad -211.9] \qquad (20)$$

The response of this turbine by using the pole placement technique is shown in Fig. 18.

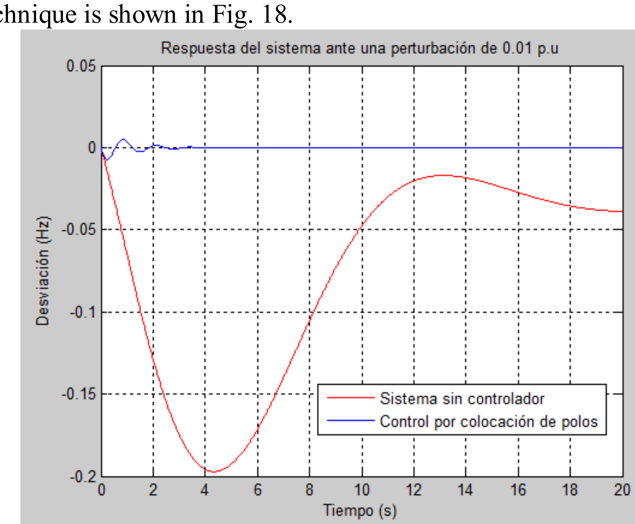

Fig. 18 Time response for the hydraulic turbine with pole placement

The settling time is reduced very drastically, being less than 4 seconds for the chosen poles. It can also be appreciated that the steady state frequency deviation, although non zero, can be considered negligible. The system does not exhibit oscillatory behavior, which means it does not loose robustness as it did by using integral control.

By using the same matrices as in the reheat steam turbine, the time response by using the LQR is shown in Fig. 19 and Fig.20.

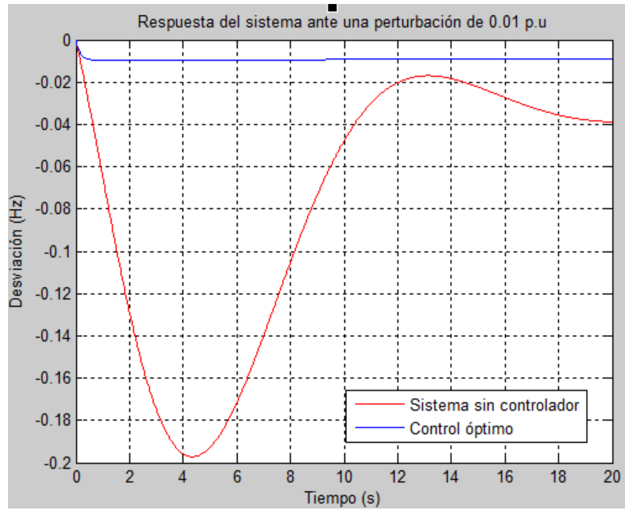

Fig. 19 Time response for the hydraulic turbine with LQR, R=1

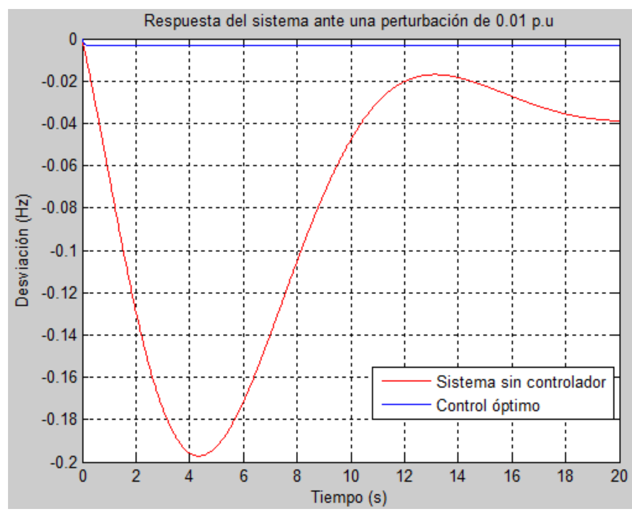

Fig. 20 Time response for the hydraulic turbine with LQR, R=0,1

The response has an even smaller settling time than with the pole placement. However, the steady state frequency deviation is considered larger. Even by increasing the effort control, the steady state frequency deviation obtained with the pole placement is superior. A fairly large control effort should be made in order to obtain a similar steady state frequency deviation with the LQR for the hydraulic system.

V. Conclusion

In this paper, three different control techniques were used for three different types of turbines. The uncompensated response of the systems shows that the simple steam turbine has the fastest response, while the hydraulic turbine is the slowest. There is no influence of the specific turbine in the steady state frequency deviation for the uncompensated systems.

Although steady state frequency deviation is eliminated in the three cases by using integral action, it tends to impoverish the transient response. All the systems tend to behave in an oscillatory way when integral action is used. The robustness of the simple steam turbine is larger than in the other turbines and it can be concluded that the hydraulic system is not robust, as a very small integral action can produce instability.

The pole placement technique improved the transient response for the three systems and for the reheat and hydraulic turbine exhibited a negligible steady state frequency deviation. However, the steady state frequency deviation for the simple steam turbine is larger and cannot be considered negligible.

When using the LQR, the improvement of the transient response was appreciated in the three systems. The steady state deviation for the simple steam turbine in this case was negligible, while in the other systems the control effort required to achieve a negligible steady state value was large.

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