ACCURATE CALIBRATION OF STEAM TURBINE SPEED CONTROL SYSTEM AND ITS INFLUENCE ON PRIMARY REGULATION AT ELECTRIC GRID

Washington Orlando Irrazabal Bohorquez, wirraz@yahoo.com
João Roberto Barbosa, barbosa@ita.br
Center for Reference on Gas Turbine and Energy, Technological Institute of Aeronautics, São José dos Campos, São Paulo, Brazil.

Abstract. *In an interconennected electric system there are two very important parameters: the field voltage and the frequency system. The frequency system is very important for the primary regulation of the electric grid. Each turbomachine actuating as generator interconennected to the grid has an automatic speed regulator to keep the rotational speed and mechanical power of the prime machine operating at the set conditions and stable frequency. The electric grid is a dynamical system and in every moment the power units are exposed to several types of disturbances, which cause unbalance of the mechanical power developed by prime machine and the consumed electric power at the grid. The steam turbine speed control system controls the turbine speed to support the electric grid primary frequency at the same time it controls the frequency of the prime machine. Using a mathematical model for the speed control system, the transfer functions were calculated, as well as the proportionality constants of each element of the steam turbine automatic speed regulator. Among other parameters, the droop characteristic of steam turbine and the dynamic characteristics of the automatic speed regulator elements were calculated. Another important result was the determination of the behavior of the speed control when disturbances occur with the improvement of the calibration precision of the control system.*

Keywords*: thermal power plants, steam turbine, speed control system, electric grid*

1. INTRODUCTION

The steam and power conversion system is designed to convert the thermal energy generated by the boiler into electricity in the turbine generator. To accomplish that propose all turbomachinery is equipped with a speed control system with two main tasks:

a) Maintain the desired turn speed and second;
b) Over speed protection control, which operates if the normal turn, speed is modified due to a load rejection.

The control system provides the functions of speed control, load control and automatic turbine control by regulating the steam flow through the turbine. The main task of such control system consists of improving the static and dynamic characteristics of turbine as controlled plant to obtain a better response to frequency deviations and therefore, to make them more maneuverable.

Disturbances influence the static and dynamic stability and may produce emergency conditions in the power system. This risk can be reduced by adopting a frequency recovery scheme as fast as possible. The speed governor model includes a static droop parameter that is used as feedback control (Chen, 2005).

Any facility participating in the primary frequency control of an interconnected electric grid is changing its power output continuously of a required value accordingly to the command of an electric grid frequency controller. The new power output must be achieved in a minimal time and the new load must satisfy a maximum efficiency concept. This implies a turboset control to minimize the thermal losses of the turbo generator.

Normally, most of the studies related to turbogenerator speed control system show electrical and control characteristics leaving aside the mechanical behavior. A contribution of this research is to show that these three main characteristics are very important and they must be jointly taken into account to obtain a real analysis of a speed control system.

2. CHARACTERISTICS OF THE SPEED CONTROL SYSTEM STUDIED

The turbine governing system used in this study is of a mechanical-hydraulic type and it uses the Watt centrifugal mechanism as the speed governor. Figure 1 shows a schematic of governor-turbine-generator system. The speed control system controls the amount of steam using the Main Stop Valve (MSV) by controlling the position of the valve according to the shaft speed of the prime machine, and then controlling the output mechanical power of the steam turbine.

This power station is part of Ecuadorian electrical grid and the data were taken from operational parameters when the turbogenerator was running on idle and peak load (Bohorquez and Barbosa, 2010).

The mechanical power applied on the prime machine should always be the same as the electric power output to the grid to keep the generator rotating at synchronous speed. The speed control system is formed by the following elements:

- Regulator;

- Speed Changer;
- Servomechanism;
- Tuning valves;
- Turbogenerator.

The speed governor model is a first order system with a time constant which represents the time constant of servomechanism and steam tuning valves position dynamics (Wang, 2003).

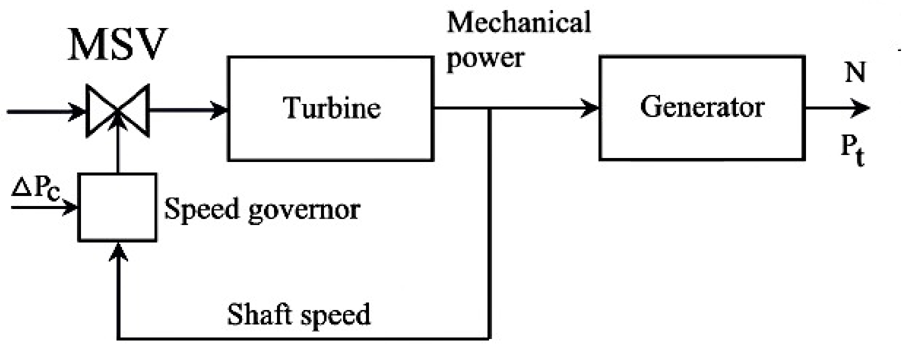

Figure 1. Governor-turbine-generator scheme

A scheme more detailed of that model is shown in Fig. 2. It is possible to observe blocks representing all elements of the speed control system studied.

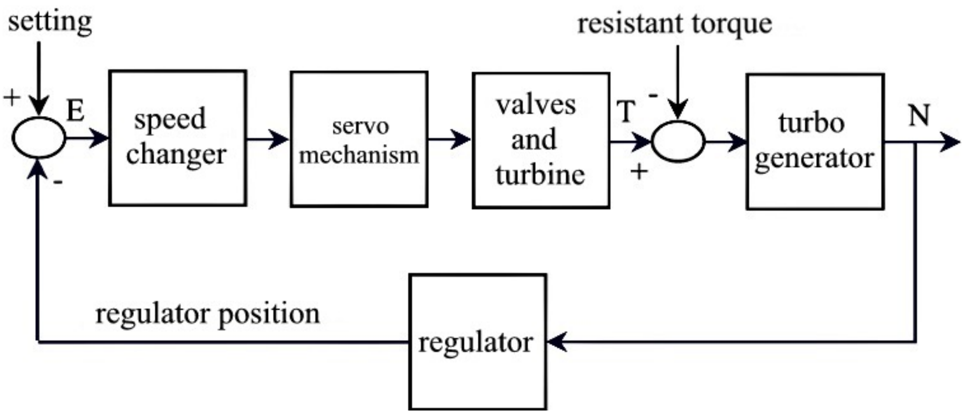

Figure 2. Block diagram of the mechanical-hydraulic type speed control system

The characteristic transfer functions have been used to build the block diagram of the control system. The block diagram with the respective functions is shown in Fig. 3.

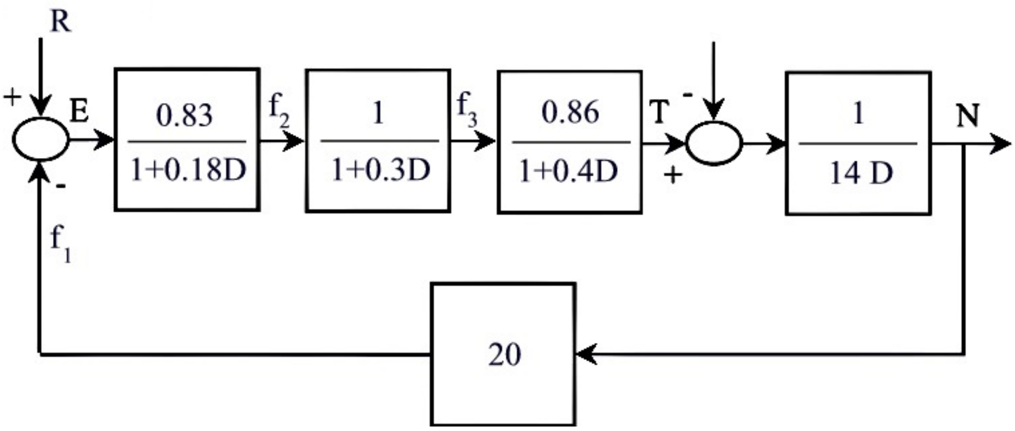

Figure 3. Transfer functions of the mechanical-hydraulic type speed control system

3. GENERAL EQUATIONS FOR THE DYNAMIC BEHAVIOR OF A SPEED CONTROL SYSTEM

1.1. Regulator

This device measures the change of speed and responds with a displacement proportional sign. The turbine turn speed is detected using a mechanical device or proximity sensor each other fixed to the turbine rotor. The speed change in the regulator inlet will produce a position change in the exit sign. For this analysis was chosen a simplified model and it is showed in Fig. 4, it has two masses rotating respect of an axis, where the centripetal force is an important variable in the operational analysis of the regulator and this is expressed by Eq. (1) (Lapem, 200).

$$m\frac{\partial^2 r}{\partial t^2} = m\omega^2 r - K(r + r_0) - C\frac{\partial r}{\partial t} \quad (1)$$

where:

$\frac{\partial}{\partial t}$ - First order partial derivative;
$\frac{\partial^2}{\partial t^2}$ - Second order partial derivative;
C - Damping coefficient of regulator chamber (N.s/m);
K - Constant that relates velocity and displacement (s);
m - Mass balance (kg);
r_0 - Spring pre-compression (m);
r - Turning radius (m);
ω - Angular speed (radians/s).

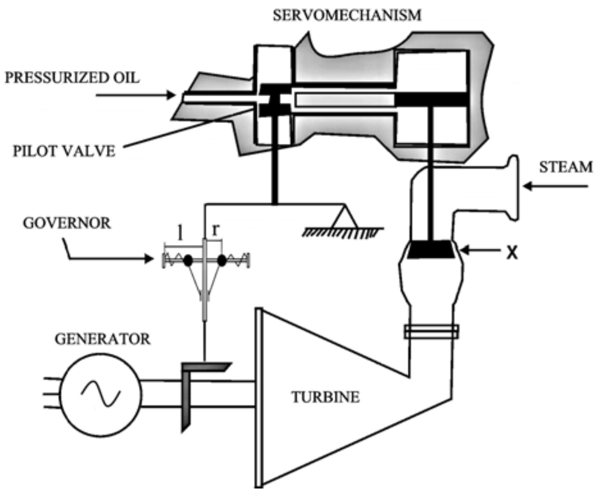

Figure 4. Basic scheme of the mechanic-hydraulic regulator

1.2. Speed Changer

This mechanical device makes two important actions: to detect and after amplify the displacement sign produced by regulator. Here are adjusted simultaneously turbogenerator speed and load. Usually, the set of the speed tuning on the steam turbines varies between 2 to 7% of the nominal speed. The main characteristic of the speed changer is the amplification of the mistake sign detected at the regulator. The dynamic behavior of speed changer is shows by Eq. (2).

$$C_1(y_1 - K_1 y_2) + A\cdot\frac{d^2 y_2}{dt^2} = 0 \quad (2)$$

where:

A - Cross-sectional area of the lower chamber (m^2);
C_1 - Discharge damping coefficient of the speed changer chamber (N.s/m);

K_1 - Constant that relates velocity and displacement (s);
y_1 - Governor displacement (m);
y_2 - Pilot valve displacement (m).

1.3. Servomechanism

To open or close the steam tuning valves is necessary a second amplification stage and it is realized with hydraulic servomechanism compost by a pilot valve and a servomotor. The speed changer sign controls the movement of the servomechanism to operate the tuning valves (Lapem, 2000).

The servomechanism uses pressurized oil to produce a displacement of the vertical axis when a speed change is realized. Following the speed changer sign, the pilot valve is displaced and this produces a servomotor displacement.

Equation (3) shows the servomechanism dynamic behavior and Fig. 5 shows the basic scheme of this equipment.

$$A_1 \cdot \frac{dy_3}{dt} + C_2 \left(\frac{x_1 \cdot x_3}{x_2 \cdot x_4} y_3 - y_2 \right) = 0 \tag{3}$$

where:
A_1 - Cross-sectional area of servomotor (m^2);
C_2 - Discharge damping coefficient of pilot valve chamber (N.s/m);
x_1 - Distance between feedback lever and tuning valves (m);
x_2 - Distance between servomotor lever and tuning valves (m);
x_3 - Distance between speed changer lever and servomotor lever (m);
x_4 - Distance between servomotor lever and feedback lever (m);
y_3 - Servomotor displacement (m).

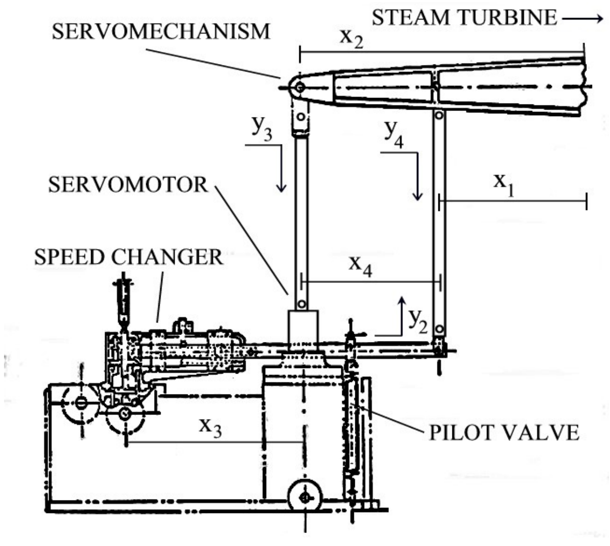

Figure 5. Basic configuration of the second amplification stage

1.4. Tuning valves

The objective of the tuning valves is controlling the steam flow inlet in the turbine. This mechanism has inner bars assembled steam admission chamber inside and it opens sequentially depending on the steam requirements that it is directly proportional to the requirements of grid load (Domachowski, 2000).

The servomotor can be moved ascendingly or descendingly and this produces a rotative movement of the spindle valves producing an upright displacement of the pin valves. The movement of the pin valves is transmitting to the internal rod and it regulates the tuning valves moving, obtaining to control the prime mover speed.

The steam flow and servomotor displacement have a lineal relationship and the tuning valves operates sequentially to control the steam flow. The system variable controlled of the tuning valves and rotor turbine is the servomotor position. This control system is assembly with following elements: tuning valves, inner bar and nozzles.

Equation (4) describes the tuning valves dynamic behavior.

$$T + \frac{V}{m_s v} \cdot \frac{dT}{dt} - C_4 y_3 = 0 \quad (4)$$

where:

C_4	- Ratio between displacement and tuning valves steam flow (m/ton);
$\dot{m}_s$	- Steam flow to peak load (ton);
T	- Torque (N.m);
V	- Volume of the admission chamber (m^3);
v	- Steam specific volume (m^3/kg).

1.5. Turbogenerator

The output torque, turn speed and turbogenerator inertia are main variables of this system and one can say a torque change accelerates the rotor turbine with rate proportional to the rotor inertia. The turbine mechanical torque, counteracting generator electromagnetic torque and the turbine-generator turn speed are related using Eq. (5).

$$I\omega \frac{d\omega}{dt} = \dot{W}_M - \dot{W}_R \quad (5)$$

where:

$\frac{\partial \omega}{\partial t}$	- Angular acceleration (radians/s^2);
I	- Moment of Inertia of turbogenerator (kg.m^2);
$\dot{W}_M$	- Power output of turbogenerator (MWe);
$\dot{W}_R$	- Consumed load of the grid (MWe).

The Equation (5) can be integrated giving initial conditions so that the speed behavior can be studied on the time. For the study of speed control systems is necessary calculate the peak values, response time and stabilization time for the fast change between idle and full load. These initial conditions are necessary to calculate the characteristic curves of the speed control system.

4. PRIMARY AND SECONDARY CONTROL

1.6. Primary regulation

When the total electricity produced is equal to the total grid demand then the electrical system is in equilibrium and the generation characteristic is described by Eq. (6). Several generation units are joined to the grid and each one of them using a droop to represent the frequency change with respect to individual power change.

$$\frac{\Delta\left(\sum_{i=1}^{N_G} P_{m_i}\right)}{\sum_{i=1}^{N_G} P_{m_i}} = K_L \cdot \frac{\Delta f}{f_n} \quad (6)$$

where:

f_n	- Grid frequency (Hz);
Δf	- Frequency change (Hz);
K_L	- Frequency sensitive coefficient (dimensionless);
Δ	- Total power output change (MWe);
$\sum_{i=1}^{N_G} P_{m_i}$	- Total power output produced by several generator units (MWe).

If there is not a speed automatic control system the difference between the generated and the consumed power would cause prohibitive variations of frequency. Power plants participating at the primary frequency control of an interconnected electrical system are changing continuously its power output due to the action of a system frequency controller (Moya e Ramos, 2008).

The characteristic of a turbine speed control system is expressed in terms of the load change rate and its correspondent frequency change. The frequency of a turbogenerator is modified in function of the prime mover power and a characteristic curve describes the operation of the primary regulation.

Equation (7) shows this characteristic parameter commonly denominated turbine droop characteristic value.

$$S = \frac{\Delta f \cdot P_n}{f_n \cdot \Delta P} \tag{7}$$

where:

Δf - Frequency change after disturbance (Hz);
P_n - Nominal power (MWe);
ΔP - Power output change (MWe);
S - Droop characteristic (%).

The generation response characteristic on actual electric power system is more frequency dependent than demand response characteristic. Then to a frequency decrease corresponds a generation drop and an electrical load increase. Each one generating units of an interconnected electrical power system participate with a droop characteristic value that represents the frequency change proportion with respect to individual power change of the system.

The droop characteristic value in the speed control system of each power unit is an important static and dynamic aspect to be considered into primary frequency regulation.

1.7. Secondary regulation

If the turbine speed control system accomplished only primary regulation will exist differences between the real consumption and power generation and this will cause a deviation of the operational frequency with regard to the nominal frequency. If this situation is not corrected, the frequency error can reach intolerable values. An expression used to quantify this error is described by Eq. (8) (Chernomzov and Nefedov, 2008).

$$E = \frac{\Delta P_n}{\Delta f_n} \tag{8}$$

where:

E - Frequency error factor (MWe/Hz);
Δf_n - Frequency change of the grid (Hz);
ΔP_n - Real power consumption change (MWe).

Secondary regulation has as purpose reduced to zero the frequency error after a disturbance. Then the power level of primary regulating units is re-established. To get this objective an integral signal is introduced in the frequency control loop of the prime mover and consequently its reference real power is changed with gain K_i and a time delay T_s. The secondary control regulation is given by Eq. (9).

$$\Delta P = \frac{K_i \cdot (f - f_n)}{s \cdot (1 + T_s)} \tag{9}$$

where:

f - Frequency after disturbance (Hz);
f_n - Nominal frequency of the grid (Hz);
K_i - Constant integral gain;
ΔP - Real power output of turbogenerator (MWe);
s - Laplace operator;
T_s - Time delay (s).

5. RESULTS FOR DROOP CHARACTERISTIC AND SPEED CONTROL DYNAMIC CHARACTERISTICS

A small power plant has been select for case study in this paper, total load demand of which is 71.6 MWe. One generation unit with 73 MWe capacity is installed (steam turbine generator). The characteristics of the prime mover and its speed control system are described in "Tab. 1".

Table 1. Technical characteristics of prime mover and speed control system.

Characteristic		Value
Prime mover	Type	Steam turbine
	Rotational speed	3,600 rpm
	Nominal power output	73 MWe
	Steam temperature inlet	510 ºC
	Steam pressure inlet	8,630 kPa
	Condenser pressure	63.5 mm Hg (abs.)
Speed control system	Type	Mechanical-hydraulic
	Maximum speed	3,850 rpm
	Minimum speed	3,400 rpm
	Maximum speed rate	190 rpm
	Minimum speed rate	180 rpm
	Delay time	1.0 s
	Response time	2.0 s
	Peak time	2.5 s
	Stabilization time	3.8 s
	Deadband	2 %

In this study were calculated the operational ratio of the speed control system, between those we can say the quick, accuracy and stability of the real response system. All these operational ratios define the static and dynamic characteristics of the turbine governing system.

To find the operational characteristics of the turbine governing system are necessaries tests on the speed control system in steady state conditions. The tests realized were:

a) Range and hysteresis of the speed changer;
b) Deadband of the speed control system.

For the first test was calculated the unit operational range of speed changer opening from 0 to 100 % and it was possible to determine the linearity and curve hysteresis. The results are shown in "Tab. 2" and Fig. 6.

For the second test was calculated the insensibility range of the speed control system in function of tuning valves position. The results are shown in "Tab. 3".

Table 2. Speed changer characteristics parameters.

Point	Speed changer position (mm)		Power (MWe)	Range (%)	Hysteresis (%)
	Upward	Downward			
1	9.5	9.5	0	65	2
2	11.2	11.2	8		
3	13.2	13.2	16		
4	14.2	14.2	23		
5	16.2	16.2	33		
6	17.7	17.7	40		
7	19.0	19.0	48		
8	20.5	20.5	56		
9	22.0	22.0	64		
10	23.7	23.7	73		
11	25	25	80		

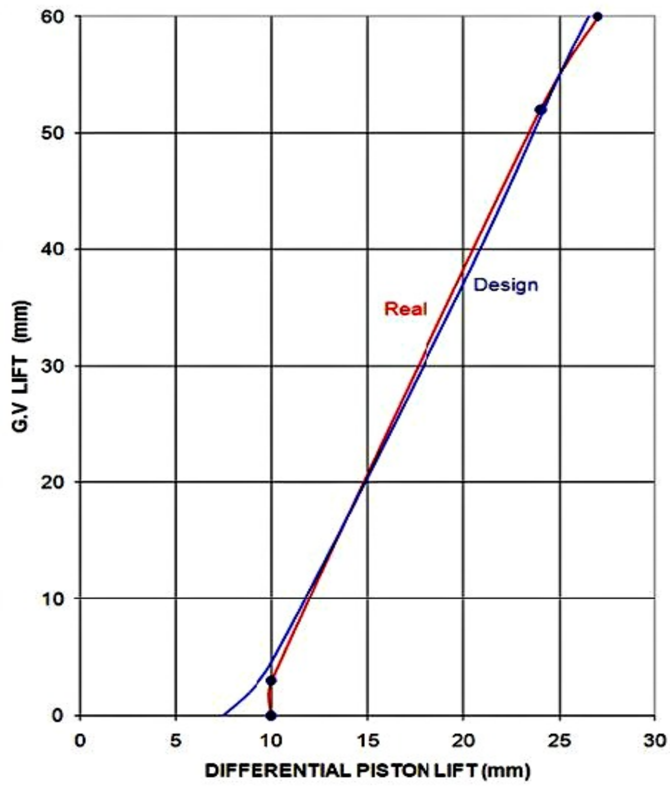

Figure 6. Characteristic curve of the speed changer

Table 3. Deadband of the speed control system.

Point	Maximum speed (rpm)	Minimum speed (rpm)	Power (MWe)
1	3,603	3,601	73
2	3,601	3,600	64
3	3,603	3,600	56
4	3,601	3,600	48
Average	3,602	3,600.3	-
Deadband (%)	0.05		

One important result is the called “droop characteristic” of a turbo generator. The term “droop” is related to the primary frequency because when the system load increases its initial value, this will produce a deceleration of the prime machine and the primary control will increase the mechanical power to regain the balance with setting frequency.

This parameter is very important because when droop is high the turbo generator is very sensible to the disturbances and when droop is low it does not participate on the primary frequency of the electric grid. Droop of the studied mechanical-hydraulic speed control system is shown in “Tab. 4” and Fig. 7.

Table 4. Droop characteristics value for the steam power plant studied.

Frequency (Hz)	Speed (rpm)	Power (MWe)	Tuning valves position (mm)						Steam flow mass (ton/h)	Steam pressure (kg/cm^2)	Droop (%)
			1	2	3	4	5	6			
60	3,600	17.93	17	9	0	0	0	0	75.87	88	-4.75
60.9	3,654	0	5	0	0	0	0	0	15	94	
60.7	3,642	0	0	0	0	0	0	0	15.14	91	

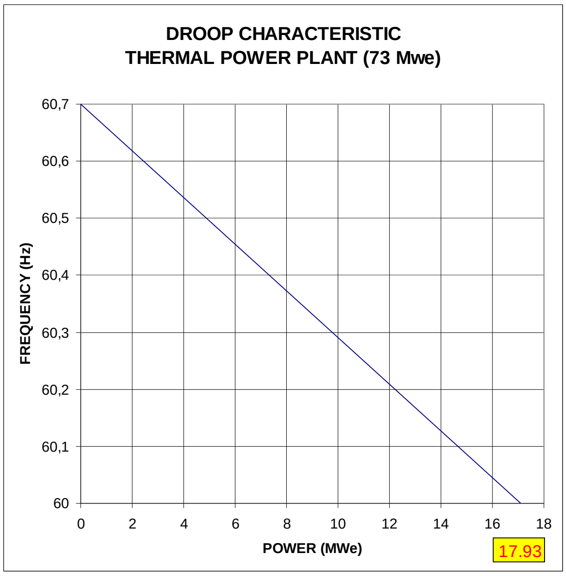

Figure 7. Droop characteristic of 73 MWe, thermal power plant

Other important parameters related with the speed control behavior are the dynamic characteristics. They can be obtained doing two tests: First, prime mover response to a step speed and second, machine response to a load rejection.

When the prime mover is running with idle and nominal speed is applied a step speed to the speed changer (100 % variation of the speed changer) and turbine changes its nominal speed from 3,600 to 3,780 rpm. Normally, the speed control system responds to the disorder and system response is used to calculate the damping constant, response time, peak time, exceed speed and steady-state error. These results are shown in "Tab. 5" and Fig. 8.

Table 5. Speed control system response to a step speed.

Parameter	Test 1	Test 2
Maximum speed rate (rpm)	220	220
Minimum speed rate (rpm)	164	168
Terminal speed rate (rpm)	180	180
Time delay (s)	0.64	0.62
Response time (s)	1.4	1.5
Peak time (s)	1.7	1.7
Stabilization time (s)	8.2	8.0

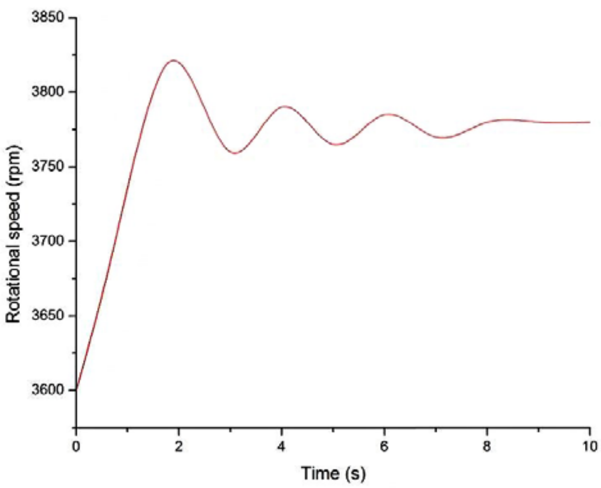

Figure 8. Dynamic response for a step speed applied over a speed changer

To obtain the prime mover response to a load rejection is necessary apply a step load on the turbine and then it is possible calculates the response time, stabilization time and participation value in the primary regulation. When the load is raised the turbine speed value falls and it is very important to know the rate of this change.

A load rejection test can be made when the power plant is synchronized to the grid. The Load Management Control changes the grid frequency. Every unit in the grid responds to the frequency change based on the droop characteristic. It is possible to take measurements of speed, frequency and tuning valves position before, during and subsequent to the frequency change. With these measured data it is possible to determine the response time, stabilization time and other turbogenerator characteristic parameters.

The results obtained for load rejection response are shown in “Tab. 6” and Fig. 9, 10.

Table 6. Speed control system response for a step load.

Parameter	Load rejection		
	Before	During	After
Load (MWe)	71.6	76.4	75.25
Turbine speed (rpm)	3,600	3,572	3,586
Tuning valves position (mm)	47	60	50.5
Response time (s)	0.99		
Stabilization time (s)	15		

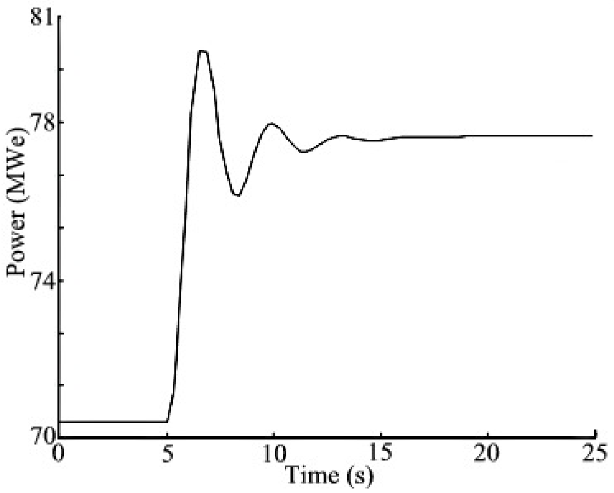

Figure 9. Dynamic response for a step load applied over a speed control system

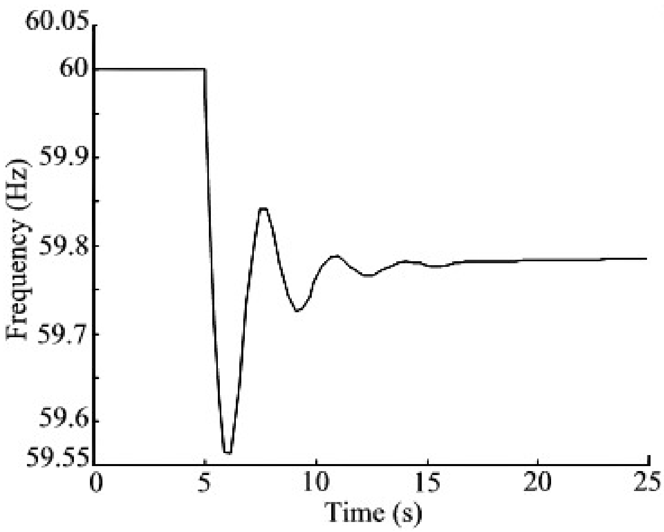

Figure 10. Frequency response for a step load rejection

6. FINAL REMARKS

A speed control system of a steam turbine determines a steam necessary for a specified power output. Then, the steam mass flow is adapted to the required power output. For this purpose, there are several tuning valves opening sequentially.

The tuning valves opening sequence are driven by servomechanism. A sequence of the tuning valves opening is characterized by one curve representing the tuning valves position versus differential piston lift.

Turbine control valves are individually or collectively governed and their loading characteristic can be linearized to find the main characteristics of the speed control system. When a disturbance occur, the error with the prime mover power output is changed does not exceed 2% of the rated power.

The tuning valves are driven by servomechanism (pilot valve and servomotor) and it is specially adjusted to obtain a linear loading characteristic. All tuning valves must also be adjusted to obtain a linear loading characteristic.

The speed control system controls the amount of steam through the tuning valves governing the position of them to the servomechanism position. To keep the turbogenerator rotates at synchronous speed the mechanical torque applying on the rotor should always be same as the electric power output to the grid.

The term “droop” means with primary regulation when system load increases to do a deceleration of the turbogenerator then the primary control will do an increase in mechanical power to regain the balance with lower frequency. Droop characteristic value is the slope of the curve frequency versus power output.

The speed control system has two basic functions: The first function is maintaining the desired nominal speed; and the second function is the overspeed protection control which operates if the nominal frequency is overpass or when occur a load rejection. The turbine governing system provides the functions of speed control, load control and automatic turbine control.

The methodology described in this paper permits the identification and comparison of several static and dynamic characteristic parameters when a power unit is linked o the electric grid. It may be observed that accurate calibration of the speed control system is very important to guarantee the proper behavior of the turbogenerator and its participation in the primary and secondary control of an electric grid.

7. ACKNOWLEDGEMENTS

The authors acknowledge the support from the CNPq (National Council for Scientific and Technological Development) and the Center for Reference on Gas Turbines and Energy from the Technological Institute of Aeronautics (ITA).

8. REFERENCES

Anderson, P.M., 1972, “Modeling Thermal Power Plants for Dynamic Stability Studies”, Pacific Gas and Electric Company, San Francisco, United States of America.

Anderson, P.M., Fouad, A.A., 2003, “Power Control System and Stability”, Second Edition, John Wiley & Sons Publication, Vol. 1, United States of America, pp. 402-481.

Bohorquez, W.I., Barbosa, J.R., 2010, “Modelagem Matemática para Regulagem Automática de Velocidade de Turbinas a Vapor”, Procedings do Congresso Nacional de Engenharia Mecânica, Campina Grande, Paraíba, Brazil, pp. 1-10.

Chen, Q.Z., 2005, “Study on a System Frequency Response Model for a Large Industrial Area Load Shedding”, Elsevier, Electrical Power & Energy systems, pp. 1-5.

Chernomzov, I.Z., Nefedov, K.A., 2008, “Improvement of Automatic Control Systems for Large-Capacity Steam Turbines”, Pleiades Publishing, Thermal Engineering, Vol. 53, No. 10, pp. 838-845.

Domachowski, Z., 2000, “Partial Loading Optimal Control as Steam Turbine Efficiency Improvement”, Proceedings of 2000 International Joint Power Generation Conference, Miami Beach, Florida, USA, pp. 1-6.

Kure-Jensen, J., Barker, W., 2009, “SPEEDTRONIC Mark V Steam Turbine Control System”, GE Power Systems, Schenectady, New York, USA, pp. 1-16.

Machowski, J., Bialek, J., Bumby, J., 2008, “POWER SYSTEM DINAMICS Stability and Control”, Second Edition, John Wiley & Sons Ltd, Vol. 1, West Sussex, United Kingdom, pp. 335-481.

Moya, O., Ramos, A., 2008, “Primary and Secondary Frequency Regulation in a Longitudinal Hydrothermal System by Liapunov Method”, ScienceDirect, Electric Power Systems Research 78 (2008) 137-146, pp. 1-10.

Wang, X., 2003, “Automatic Generation Control Model Using Matlab Simulink”, Department of electrical and Computer Engineering, Penn State University, United States of America, pp. 1-12.

5. RESPONSIBILITY NOTICE