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STEAM TURBINE Stability Study Case of Congo Power Plant

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ABSTRACT: In this article, we will present a simple model of steam turbine generator and generator in a thermal power plant to regulate the output frequency of the alternator coupled to this turbine to maintain it constant or evolving in an interval well defined.

The model will be assumed to be linear and studied around the nominal operating point. The modelling of the turbine will be done using the operating equations of each subset of the steam turbine generator system.

The model obtained will be translated as a transfer function and exported to the Matlab / Simulink software which will allow simulations of the operation of the assembly under normal operating conditions and subsequently under disturbed conditions.

The empirical method of Ziegler and Nichols is that adopted for the determination of the initial parameters of the PID (Proportional Integral Derivative) corrector which one proposes to use to carry out the regulation.

KEYWORDS:Stability, PID corrector, thermal power plants, Steam Turbine; modelization; regulation; simulation; frequency; Matlab / Simulink; Republic of Congo.

I. INTRODUCTION

Electric energy is an essential factor for the development and evolution of human societies, whether in terms of improving living conditions, or on the development of industrial activities.

65% of the electrical energy produced and consumed in the Republic of Congo is of thermal origin produced by Gas Turbines, number two in the COTEMATEVE power station, which are the most powerful in SNE's generating fleet. Thus, a disturbance of the electromagnetic charge can easily represent a large percentage of the power produced by a power plant and this can result in an imbalance of the network. Load variations cause fluctuations in the frequency.

The main objectives of this study is to ensure the stability of the electricity grid despite load variations (fluctuations in demand) that cause frequency variations; it is therefore necessary to maintain the latter at the constant value of 50 Hz or in a range of limit values dictated by the requirements of the network. It is $\pm 1\%$ for the network considered by the national electricity company (SNE).

Thus, the stability study of a production plant located in a developing country seems well suited to find regulatory models adapted to the special conditions of the electricity networks in these areas, most often characterized by demand is growing rapidly because of rampant urbanization.

II. WHY THE STABILITY OF STEAM TURBINE IS SO IMPORTANT

In highly industrialized countries, the sources of energy production are multiple. This is not always the case in developing countries. There are fewer production centers and therefore the networks are often not very interconnected. Thus, a disturbance of the electromagnetic charge can easily represent a large percentage of the power produced by a power plant. this can result in an imbalance of the network. Load variations cause fluctuations in the frequency of turbomachines in production plants. These frequency fluctuations must then be permanently adapted to their point of



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operation. The production sources are most often subjected to strong constraints to maintain the frequency at the set value. Hence the interest of the study the stability of the steam turbine, case of the Congo power station

III. MODELING OF THE ALTERNATOR STEAM TURBINE SYSTEM

In order to present the model of the regulation system based on the mathematical equations associated with block diagrams, the following main hypotheses will be considered:

The regulation will be processed around the nominal operating point of the turbine and we will be interested in low amplitude oscillations that will not exceed 10%. One can thus consider the modeled as linear (that approximation being actually possible only for amplitudes of the order of 10 to 20% for the phenomena concerning the turbines).

All systems will be linear. That is to say, the equations linking the variables will be assimilated in first approximation to linear differential equations with constant coefficients.

In addition, parameters such as the temperature and pressure of the inlet steam will be considered constant.

The turbine will be studied already in steady-state and non-transient operation that is to say after the start-up phase, after it has reached its nominal operating speed of 3000 rpm and after it has been coupled to the network.

The system we propose to study (FIG.1.) Will include:

- The corrector.
- The servovalve.
- The device of the control valves (4) for the flow of steam entering the turbine.
- The steam turbine.
- The alternator.



FIG. 1: Global diagram of the turbo-alternator group

The figure above shows the studied system connected to a load, composed of; The corrector, The servo valve, The device of the control valves (4) for the flow of steam entering the turbine, The steam turbine, The alternator.

A. Modeling of the steam turbine

The steam turbine is represented by the following first-order transfer function:

Where Y (p) is the angular velocity of the impeller shaft at the outlet and X (p) the flow rate of steam at the impeller inlet. This transfer function can be rewritten in another way showing the gain as well as the time constant.



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FIG. 2: Simulation transfer function of the turbine

The figure above shows the unit rate response of the transfer function governing the operation of the steam turbine.

B. Modeling the servovalve

We can now move on to the next step, which is modeling the servovalve that adjusts the pressure of the regulating oil circuit that controls the opening of the valves.

The technical specifications of the servovalve will be given in the appendix. The mathematical model, namely the transfer function of the servovalve is provided by the manufacturer and is in the form of the second order system below:

$$\frac{P(S)}{I(S)} = \frac{K}{1 + \left(\frac{2\xi}{\omega_n}\right)S + \left(\frac{S}{\omega_n}\right)^2}$$

With: **K**: Represents the static gain of the servovalve in bar / rn.A.

 $\omega n = 2\pi f n$ **fn:** The apparent natural frequency ξ : The damping coefficient.

The parameters of the transfer function of the servovalve will be calculated as follows: The apparent natural frequency is given by the manufacturer:

fn = 200 Hertz

We will then have $\omega n = 1256.637 \text{ rad} / \text{s}$

The static gain is equal to:

$$\mathbf{K} = \frac{\Delta \mathbf{P}}{\Delta \mathbf{I}} \tag{4}$$

K = 0.195bars/m.A

As for the damping coefficient, we have chosen in the characteristic table of a second-order system (Appendix A) a value giving the shortest response time (5.3 seconds), ie:

ic.

$$\xi = 0.5$$

(6)

(2)

(3)



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The transfer function of the servovalve linking the current to the inlet and the pressure at the outlet will therefore be:

C. Modélisation des soupapes

In other words, the minimum oil pressure in the regulating circuit for the opening of the first valve will be maintained at 0.6 bar and the maximum oil pressure at which the maximum vapor flow will be reached will be 3.72 bar as mentioned in diagram of FIG.3. (Valve adjustment diagram).

To establish the transfer function linking the flow and the pressure we will use the diagram of FIG.3 that we resume below.



FIG 3: Valve Timing Diagram

The figure above shows the valve adjustment chart, linking the flow rate to the steam entering the turbine at the set pressure.

We keep the extrema of pressure values (0.6 bar and 3.72 bar).

We will therefore approach the flow curve from these two points mainly to ensure the linearity of the flow of steam between these two extrema.

Thus at the pressure of 0.6 bar corresponds a zero flow and at 0.72 bar the maximum flow of 126.7 t / h. get The resulting line gives the following equation:

$$Q = 40P - 24.4$$

With: Q: being the flow P: the pressure.

D. Alternatormodeling

Indeed, it is a question here of representing by a gain the relation which links the speed of rotation of the shaft of the turbine to the frequency of exit at the level of the alternator namely:



(9)

(8)



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Since it is the angular velocity (in radish) that represents the direct output of the model of our turbine. We will need to perform the correct conversions to be in compliance with the previous formula.

As a result, the relationship between the angular velocity of the turbine shaft and the frequency will be:

$f = \omega / 2\pi$	(10)
Indeed	
$N = 60\omega / 2\pi$	(11)
With ω in rad / s. N in rpm. P = 2.	
We will have:	
$f=\frac{60\omega*2}{2\pi*120}$	(12)

The output gain of the turbine will be:

$$\mathbf{K} = \frac{1}{2\pi} \tag{13}$$

E. Modeling of the PID corrector

One thus obtains a proportional integral and derivative corrector (P.I.D.). There are several ways to associate these three actions.

We distinguish essentially. The parallel PID corrector and the mixed PID corrector. The parallel PID is one of the most moderate because the gain does not act on the integral and derivative equities.

However, the most energetic is the mixed PID corrector, because the gain affects the other two actions. Its use is probably the most common. The transfer function of the mixed PID corrector is as follows:

$$\boldsymbol{\mathcal{C}}(\boldsymbol{s}) = \boldsymbol{G}_r \times \left[1 + \frac{1}{T_i \boldsymbol{s}} + T_d \boldsymbol{s} \right] = \boldsymbol{G}_r \frac{1 + T_i \boldsymbol{s} + T_i T_d \boldsymbol{s}^2}{T_i \boldsymbol{s}}$$
(14)

This transfer function is only theoretical. For the degree of the numerator is greater than that of the denominator. It is therefore necessary to filter the derivative action, ie add a term of the first order: to the denominator.

However, this filtering will not fundamentally change the parameters of the answer of the corrector, since the filtering will only intervene on the high frequencies. Low and intermediate frequencies will not be affected.

F. Advantages of the PID corrector.

- It delays in phase the low frequencies while bringing them a high gain,
- It acts almost only on intermediate frequencies such as



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(15)

- It acts on the high frequencies by amplifying them and bringing them a phase advance.
- It makes it possible to increase the damping coefficient ξ : It is the stabilizing effect of the derivative correction. It also increases the bandwidth since $\omega_R^t < \omega_R$ and rounds up the system faster.
- The integral action modifies the low frequencies (infinite gain in open loop) and ensures a static gain equal to 0dB. So offers a positional gap is zero.

G. Conclusion:

The PID corrector, because it combines the three actions, proportional, integral and derivative seems to be the most effective. This is why we will choose a mixed PID corrector for our regulation. Thus, we will write the transfer function of the PID corrector as follows:

$$\mathcal{C}(s) = \mathcal{G}_r \times \left[\mathbf{1} + \frac{1}{\mathsf{T}_i s} + \frac{\alpha \mathsf{T}_d s}{1 + \mathsf{T}_d s} \right]$$
(16)

Or, if one filters at the output of the PID, the transfer function of the corrector is written:

$$\boldsymbol{C}(\boldsymbol{s}) = \boldsymbol{G}_{r} \times \left[1 + \frac{1}{T_{i}\boldsymbol{s}} + \alpha T_{d}\boldsymbol{s}\right] \left[\frac{1}{1 + T_{d}\boldsymbol{s}}\right]$$
(17)

With:

The initial parameters of the PID corrector, to be determined by simulation.

IV. EXPERIMENTAL RESULTS

A. Determination of the initial parameters of the PID corrector

The purpose of open-loop simulation is to find the initial parameters of the PID corrector, namely Gr, Ti and Td. We will use the method of Ziegler and Nichols. This method is based on observation of the process response and knowledge of the regulator structure used.

The procedure is as follows:

- Apply an amplitude step E0 to the input of the system and observe the output S.
- Draw the tangent to the inflection point (FIG 4) and measure the delay T and the time τ set to reach E0.



The figure above shows the waveform of the result allowing the determination of the initial parameters of the PID corrector obtained by sending an amplitude step Eo to the input of the PID corrector.



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The proposed PID is a mixed PID. The values of the parameters are given in Table .1.

noculator	Loop open
regulator	
PID	$G_r = 1.27 \frac{\tau}{T}$ $T_i = 2T$ $T_d = 0.5T$

Table.1: Values of the PID corrector parameters

The figure above shows the initial parameters of the PID corrector, obtained by the empirical method of Ziegler and Nichols. These values are given below in Table 1.

1. Parameters of Open Loop System Simulation:

The simulation of the open-loop system is illustrated in FIG.2;

An amplitude step of 1mA (18mA at 19mA) is applied to the input of the system and will be maintained at 18mA for 50 seconds before passing 19 mA. The duration of the simulation will be one minute and 40 seconds or 100 seconds.

Once the simulation is done, the graph of the system response at the step will be displayed so that the Gr, Ti and Td values can be measured as shown in FIG.1.



FIG 5: Open System Simulation of the System

The figure above shows the functional diagram of the entire open loop system.

Index response of the system at a step



The figure above shows the waveform of the response of the whole system in open loop at a unit level.



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2. Results of calculation of the parameters of the PID corrector

The results are given in the following table:

Measurements	Calculated initial PID
made(Second)	values
T = 8.088s	$G_r = 1.478s$
$\tau = 9.412 s$	$T_i = 16.176s$
	$T_d = 4.044s$
	-

Table...2: Determination of the values of the PID corrector parameters

The figure above shows the results of all the parameters of the PID corrector, obtained by calculation and by measurements. These values are given above in Table 2

B. Closed-loop simulation in the absence of disturbances

Once the initial parameters of the PID corrector have been obtained, we can now proceed to the closed-loop simulation which will initially consist in observing the behavior of our system in the absence of a disturbance and in a second time to ensure the proper functioning of this system when subjected to disturbances:

We will then see how to improve its response in terms of speed, stability and accuracy by modifying the initial parameters of the P1D corrector.

Note that by system response, we mean in fact the behavior of some of its variables (ie the frequency, the speed of rotation, the input current of the servovalve and its output pressure).

1. Parameters of the simulation in the absence of disturbances

Indeed the input current of the servovalve must varies between 4 and 20mA. The outlet pressure of the servovalve must remain between 0.6 and 3.72 bars while the steam flow in the turbine must be between 0 (excluded except at start-up) and 35 kg / s; the speed of rotation when it must be between 351.8 rad / s and 282 rad / s.

2. Results of the simulation in the absence of disturbances



FIG. 7: frequency of the system in the absence of disturbances

FIG. 8: Steam flow in the turbine in the absence of disturbances

The figures above show the waveforms of the frequency and flow rate of steam in the turbineof the whole system in open loop, subject to a unit level in the absence of disturbances.



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FIG. 9: system speed in the absence of disturbances

The figures above show the waveforms of the speed and output pressure of the servovalve of the whole system in open loop, subject to a unit level in the absence of disturbances.



FIG. 10: Servovalve input current in the absence of disturbances

The figure above shows the waveform of the input current of the servovalve of the open-loop system, subjected to a unitary step without disturbance.

3. Analysis of results in the absence of disturbances

In the absence of disturbances we observe that the system is maintained at the frequency of 50 Hertz as shown in FIG 3, we can also observe the evolution of the rotational speed curve in the vicinity of 314.2 rad / s for a frequency of 50 Hertz. This corresponds to the optimal operating values provided by the turbine manufacturer.

The other parameters such as, the input current of the servo valve, its outlet pressure and the flow rate of steam entering the turbine evolve between specified limits thus respecting the technical constraints related to the optimal operation of the alternator turbine group recommended by the manufacturers.



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C. Closed loop simulation in the presence of the first disturbance (a)



FIG. 11: Servovalve output pressure in the absence of disturbances

1. Parameters of the simulation in the presence of disturbances (a)

We will subject the system to a unit level echelon of amplitude of 5 Hertz, ie 10% of the nominal frequency for 5 seconds and starting 70 seconds after the start of the simulation (FIG 12).



FIG. 12: Signal (a) frequency disturbance (5 Hz, over 5s)

The figure above shows the unit level echelonof amplitude of 5 Hertz, ie 10% of the nominal frequencywaveform used in simulating the system in the presence of first disturbances.

2. Simulation results in the presence of disturbances (a)



FIG. 13: frequency of the system following the disturbance (a)







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The figures above show the waveforms of turbine frequency and rotational speed of the entire closed-loop system, subjected to a unitary scale during a first disturbance.



FIG. 15: Steam flow entering the turbine following the disturbance (a)

FIG. 16: Input current to the servo valve following the disturbance

The figures above show the waveforms of the steam flow entering the turbine and that of the input current to the servo valve of the entire closed-loop system, subjected to a unitary step when a first disturbance.



FIG. 17: Output pressure of the servo valve following the disturbance (a)

The figure above shows the output waveform of the servovalve output of the entire closed-loop system, subjected to a unit step after the observed the first disturbance.

3. Analyse des résultats suite à la perturbation (a)

In FIG11, we observe that the system first detects the disturbance before correcting it. Subsequently we notice that the frequency returns within acceptable limits (between 49 and 51 Hertz) after 0.34 seconds.

Then, when the disturbance ceases 5 seconds later, it also causes the reaction of the system which tends to lower the frequency up to the value of 45 Hertz, but which is immediately brought back into the range of tolerable variations after 1.7 seconds. (After the disturbance has ceased).

D. Closed Loop Simulation in the Second Disturbance (b)

1. Parameters of the simulation in the presence of the disturbance (b)

Let us submit this time the system to a perturbation of amplitude 2.5 Hertz starting at the instant t = 70 seconds and during 2.5 seconds,



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The figures above show the waveforms of turbine frequency and rotational speed of the entire closed-loop system, subjected to a unitary scale during a second disturbance.

3. Analysis of the results in the presence of the disturbance (b)

We find that the system response time (FIG-16 frequency curve) is shorter than before. It is only after 0.08 seconds that the system reacts and reduces the frequency to acceptable limit values. Subsequently, when the disturbance ceases, the system also reacts in a very short time, 0.4 seconds and stabilizes permanently.

These results are identical to that of the first case of disturbance. The system restores all the parameters in their nominal values and all the constraints related to the regulating bodies. This confirms once again the effectiveness of the PID corrector.

E. Closed Loop Simulation in the presence of the Third Disturbance (c)

1. Parameters of the simulation in the presence of the perturbation (C)

This disturbance is differentiated by its duration, rather short, which is 1 second, with an amplitude of 5 hertz. It appears at the same time as the other disturbances at t = 70 seconds and makes it possible to test once again the stability of the system.



The figure above shows the unit level echelonof amplitude of 5 Hertz, ie 10% of the nominal frequency waveform used in simulating the system in the presence of the third disturbances tt = 70 seconds.

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2. Results of the simulation in the presence of the disturbance (c)



The figures above show the waveforms of turbine frequency and rotational speed of the entire closed-loop system, subjected to a unitary scale during thethird disturbance.

3. Analysis of the results in the presence of the disturbance (c)

FIG.19 indicates that in this case, the system reduces the frequency in the tolerated range (between 49 and 51 Hertz) 0.34 seconds after the appearance of the disturbance and when it disappears it takes 2.09 seconds to the system to bring back the frequency, before it goes out of limits 3.7 seconds later to stabilize definitively after 1.8 seconds.

Once again the control system gives good results, especially as the parameters such as the rotational speed of the turbine are restored to their nominal values after the disturbances as shown in FIG.20.

V. CONCLUSION

In conclusion we can say that the control system as it was designed (with the parameters of the corrector PID determined) meets the requirements of the functional parameters of performance and safety required by the National Society of Electricity.

Indeed the maximum response time we have been able to take is 2.09 seconds. Which is quite short. In addition the frequency of the network is always reduced to its nominal value of 50 Hertz for a rotation speed of the turbine identical to that provided by the manufacturer in nominal operation of the system. Other parameters such as current, pressure, flow rate and rotational speed of the turbine evolves in the optimum safety and efficiency intervals.

The results of the different simulations allow us to choose a PID corrector to perform frequency regulation of the turbo-alternator group. We therefore note for the PID corrector based on the model we studied and the data collected at the SNE, the following parameters:

Gr=15, Td= 4.044, Ti= 16.176.

REFERENCES

- [1] J. Ricard "Équipement Thermique des Usines Génératrices d'Énergie Électrique, " vol. 3, pp. 14–55, 1962.
- [2] Guy de Livois "Régulation des Générateurs à vapeur et des Centrales Thermiques, ". vol. 1, pp. 10–15, 1979.
- [3] J.L. Jezequel"International Symposium MV1, Active Control in Mechanical Engineering, ".

^[4] Y. Takahasihi, M.J. Rabins and D.M. Auslander "Control and Dynamics Systems, ". vol. 2, pp. 10–15, 1972.



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- J.C. Hsu and A.U. MEYER "Modern Control Principles and Applications, ". vol. 1, pp. 22-27, 1968. [5]
- R. Lonchamp "Commande Numérique des Systèmes Dynamiques, ". vol. 1, pp. 17-19, 1995. [6]
- [7] H. Constantine, J.J D'azzo and Houpis"Linear Control System Analysis and Design Conventional and Modern, ".vol. 2, pp. 11–29, 1975.
- M. Rivoire and J-L. Ferrier " Cours d'Automatique, tome l, ". vol. 3, pp. 33-40, 1997. [8]
- R.Vichniesky "Thermodynamique appliquée aux Machines, ". vol. 3, pp. 26–30, 1967. [9]
- [10] B. Szezeniowski "Turbines à Vapeur et à Gaz, "vol. 2, pp. 16–19, 1987.
 [11] S. Sawadogo "Notes de cours d'Automatique, ". vol. 4, pp. 7–11, 2010.
- [12] S. Tamba, "Notes de cours de Machines Hydrauliques," vol. 5, pp. 20–23, 2012.
 [13] P. Demba, "Notes de cours de Moteurs Turbines et Compresseurs,". vol. 1, pp. 54–60, 2007.

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