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Simulation and Modeling of the turbo alternator 301 vibrations impact on the production of electrical energy of the power plant C3 of the Senegal National Electricity Company

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Abstract In a power plant, the turbine drives the rotor of the turbo-generator group (TG), which is at the origin of the production of electrical energy. However, this continuous rotation of the rotor generates mechanical vibrations (sources of rotor imbalance) which, in the long run, can damage the machine and reduce the production of energy. It is therefore necessary to reduce these vibrations in accordance with the principles and standards of noise and vibration of industrial machines in general and in particular in the TG. In this article, it is a question of finding ways of balancing of the rotor allowing the improvement of electrical energy production conditions. The study concerns the turbo-generator 301 of theC3 of the Senegalese National Electricity Company (Senelec) power plant. The first step is to carry out measurements and vibratory calculations in order to compare these values with the reference values by simulation. Then, in the second step, in order to optimize the dynamic behavior of the turbo alternator in motion, a 3D modeling of the rotor is carried out using the Matlab programming software. The obtained results show that from the computer and by variation of input data, we can obtain the best performances of the machine by decreasing the vibrations (mainly characterized by the displacement and speed of the GT) by rotor balancing. This makes it possible to increase the lifetime of the machine as well as its efficiency and increase the production of electrical energy of C3 plant at lower cost.

Keywords rotor, turbo alternator, vibration, balancing, efficiency, production

Introduction

In the systems of electrical energy production, to a better starting of a turbo alternator group, it is necessary to balance its rotor [1,2]. The latter is flexible with several rigid elements that are: - the exciter - the alternator and the turbine (low, medium and high pressure). During the production of electrical energy, the operation of the turbo alternators induces vibratory effects at the level of the rotor.

To balance the inadequacies of these effects, several vibratory tests have to be made, some of which may be replaced by acoustic tests. These aspects of TG functioning impact on the lifetime, as well as on the cost of the production of power plant as that of C3.

The objective fixed in this present article is to propose methods of turbo alternator 301rotor balancing for improving production conditions of the C3power plant.

Methodology

Schema diagram of the turbo generator system

The rotor studied is rigid and carries a low and high-pressure (LP, HP) turbine, an alternator and an exciter as shows in figure1.

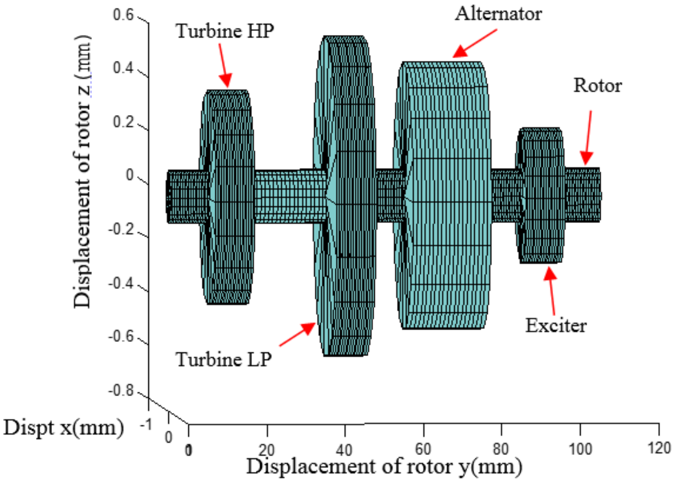


Figure 1: Schema *diagram of the turbo-generator 301*

Technical data and vibration measurements of the rotor 301

Using the information in tables 1, 2 and 3, a comparative study between the vibration measurements and the reference values is carried out for a vibratory analysis of the system.

This will allow to make a 3D simulation of our system in order to optimize the dynamic behavior of the rotor by adjusting physical parameters [3]. For the simulation of the operation of the turbo-generator system, we use the technical data listed in table 1.

Table 1: Characteristics of the various compartments of the turbo-alternator system 301 [3].

Elements considered of the rotor	Length (mm)	Mass (kg)	Diameter (mm)
Alternator	7617	19000	250-840-250
Exciter	1300	1800	250-330-250
HP Turbine Shaft	3632	3985	145-300-552-500-650-300-450-330
LP turbine shaft	3982,3	7700	480-310-330-250-330-310-734-1131-1189-1159-310-280-225-145

The vibration measurements according to the displacement (vertical and horizontal) of the rotor, shown in table 1, are taken in the form of turbine velocity readings during its operation under load. They are effected according to the power for different given periods.

Table 2: Daily reading of the characteristics of the steam turbine 301 of the power C3

Applied load and dates read for the turbine 301		First stage of the turbine		Second stage of the turbine		Third stage of the turbine	
		Bearing speed 01 turbine (mm / s)		Bearing speed 02 turbine (mm / s)		Bearing speed 03 turbine (mm / s)	
Load	Dates	Vertical	Horizontal	Vertical	Horizontal	Vertical	Horizontal
20 MW	23 January	2.5	4.4	1.2	1.3	1.4	2.1
15 MW	27 January	1.9	2.4	4.8	1.9	2.1	2.3
18 MW	19 February	2.9	2.1	4.8	1.6	2.5	2.7
20 MW	11 March	1.6	3.1	4.3	1.6	2	2.1
20 MW	09 April	1.3	2.8	4.3	1.3	2	1.8
20 MW	20 April	1.2	2.7	4.1	1.1	2.3	2
20 MW	31 July	1.7	3.5	3.9	4.2	1.7	1.6
24 MW	08 August	2.7	6.1	5	1.2	6	1.7

Table 3 gives the daily reading of the vibration speed measurements as a function of the alternator power, following the displacement of the rotor (vertical or horizontal).

Table 3: Record of vibratory measurements of 301 alternator speeds as a function of the displacement (vertical or horizontal) of the rotor [3]

Applied load and dates read alternator 301		**Bearing speed alternator 301**			
		Bearing speed 01 (mm/s)		**Bearing speed 02 (mm/s)**	
Load	**Dates**	**Vertical**	**Horizontal**	**Vertical**	**Horizontal**
20 MW	23 February	2.3	1.3	1.2	6.1
15 MW	27 February	2.4	1.9	0.7	0.6
20 MW	11 March	2.3	1.7	1	0.8
20 MW	09 April	2.5	1.4	1.2	0.7
20 MW	20 April	2.4	1.5	1.4	0.8
20 MW	31 July	2.5	1	1	1.1
24 MW	08 August	3.1	2.4	1.4	1.3

Table 4 gives the appreciation reference values of displacement and speed of rotating machines operation, such as that of TG 301.

These international vibratory references make it possible to compare the dynamic behavior of our machine with the displacement and speed values of references in order to optimize the efficiency of the machine.

Table 4: References for comparing 301rotor vibration measurements (rotor speed n = 3000 rpm) [3]

Dynamic appreciation of the machine	**Micron peak to peak displacement**	**Speed of displacement mm /s RMS**
Stop	Superior	Superior
Very bad	50	10
Bad	30	6
Acceptable	15	3
Good	inferior	inferior

Table 5 gives the displacement in micron peak to peak and the displacement speed mm/s RMS with:

RMS = rootmean square (RMS value = maximum valuex $\frac{\sqrt{2}}{2}$).

Table 5: Record of vibratory measures of the speeds of the exciter301 as a function of the vertical or horizontal displacement of the rotor

Applied load and dates read exciter 301		**Bearing speed exciter 301**			
		Bearing speed 01 (mm/s)		**Bearing speed 01 (mm/s)**	
Load	**Dates**	**Vertical**	**Horizontal**	**Vertical**	**Horizontal**
20 MW	23 January	1.2	1.3	1.8	3
15 MW	27 January	1.3	1.1	0.9	0.5
20 MW	11 March	0.9	1.5	0.8	0.7
20 MW	09 April	1	1.3	1.1	0.4
20 MW	20 April	1.1	1	1.2	0.7
20 MW	31 July	1.9	1.3	1	1.4
24 MW	08 August	3	2.4	2.5	2.2

Results and Discussions

Simulation of the vibration programming of the 301 turbo-generator system

This simulation is done with Matlab from ordinary differential equations (ODE) using the Runge Kuta method, with the turbo alternator at its maximum load of 24 MW. These forms of second order differential equations are given in the following general form:

$$M\ddot{x} + Kx = F \quad (1)$$

Where:
M = Mass matrix (kg);
x = Displacement (m);
$\ddot{x}$ = Acceleration (m / s^2);
K = Matrix of rigidity;
F = Matrix of forces.

With the method ODE and after variable change, the equation (1) becomes a differential equation of the first order following:

$$z = \dot{x} \text{ and } \dot{z} = \ddot{x} \quad (2)$$

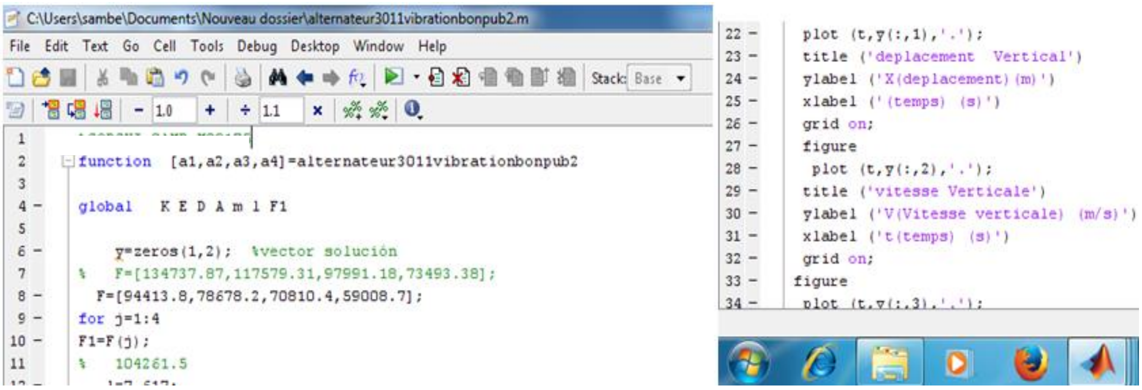
z = displacement in the axis oz (m);

$\dot{x}$ *et* $\dot{z}$ velocities (m / s).

Thus, equation (1) is then placed under the form of the first order:

$$\dot{z} + K = F \quad (3)$$

By the RungeKuta method, the numerical resolution of this equation (3) is in the following form:



From these results, the simulation gives the maximum displacement for each load applied to the rotor [4, 5, 6, 7, 9]. Figures 2 and 3 respectively show the results of the simulation of the vertical displacement x and that of the velocity as a function of the operating time of the alternator rotor and for a load of 24 MW applied.

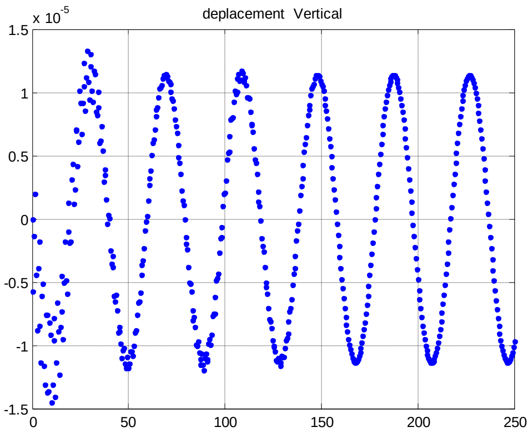


Figure 2: Vertical displacement x(m) of the alternator rotor as a function of the time (s) of its operation with its maximum load of 24 MW.

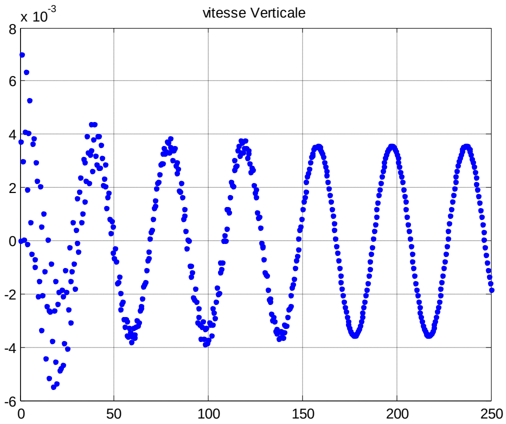


Figure 3: Alternator rotor vertical velocity (m / s) as a function of the time t (s) of its operation with its maximum load of 24 MW

These two figures show the difficult conditions of starting the system for a time less than 40 s. This explains a random fluctuation of the displacement x and the velocity V in this zone.
The second phase between 40 s and 130 s is the transient phase characterized by less solicitation of the rotor. Consequently, a more or less significant fluctuation of the displacement x and of the velocity V is noted here.
Beyond 130 seconds, a permanent regime is entered where absence of perturbations due to mechanical constraints on the rotor are noted. This zone constitutes the ideal part of the operation of the turbo alternator 301. However, in this article the study deals with the established regime of the system; namely time greater than 130 s.
The results of the simulation with the Matlab software of the rotor operation according to the different loads applied are given in table 6.

Table 6: Simulation results of the alternator 301 operation according to the different loads for vertical displacement

Load applied to the rotor (MW)	**Vertical displacement of the rotor x(m)**	**Vertical displacement velocity of the rotor V (mm/s)**
24	8.84×10^{-6}	2.75
20	8.83×10^{-6}	2.69
18	7.05×10^{-6}	2.26
15	7.02×10^{-6}	2.1

From this table and for a vertical displacement, the results obtained from the simulation of the dynamic behavior of the machine 301 show that the system is acceptability balanced. Indeed, the maximum displacement noted by the machine is less than 15 microns and the maximum speed is less than 3 mm/s. These results are in accordance with the international references on rotating machines as in table 4.
Following the simulation of the rotor movement during the alternator operating period and using the differential equations, the variation of its displacement is given in figure 4.
Figure 5 shows the results of the simulation of the speed for a horizontal displacement of the alternator 301 rotor as a function of the working time, for an applied load of 24 MW.

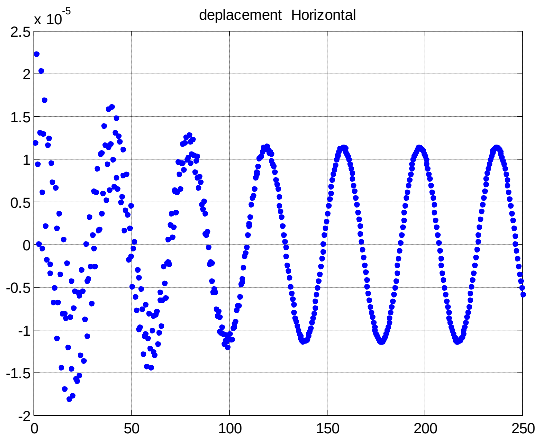


Figure 4: Horizontal displacement of the alternator 301 rotor as a function of time and for an applied load of 24 MW

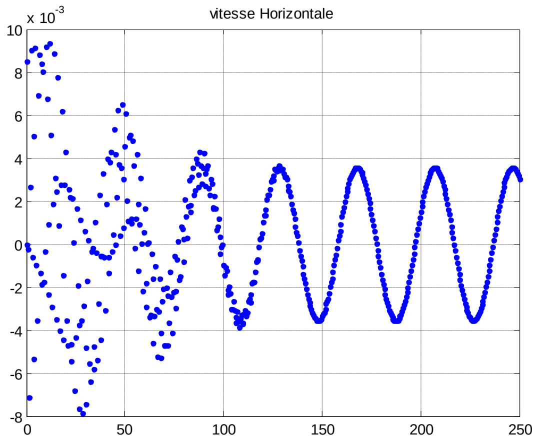


Figure 5: Horizontal velocity of the alternator 301 rotor as a function of time and for a load of 24 MW.

As for the vertical displacement, the two figures show three zones (start - transient and normal) operating.
The first is characterized by time less than 60 s corresponding to a strong perturbation of the rotor operation, which reflects the existence of random displacement and velocity.
The second zone (between 60 s and 110 s) is symbolized by an average reduction in the mechanical constraint of the rotor.
The third phase (time greater than 110 s) is the ideal situation for the turbo generator where the rotor undergoes fewer constraint.
Table 7 below gives the results of the simulation for different applied loads.

Table 7: Simulation results of the alternator 301 operation according to the different loads for horizontal displacement

Load applied to the rotor (MW)	Horizontal displacement of the rotor x(m)	Horizontal displacement velocity of the rotor V (mm/s)
24	$6.7x10^{-6}$	2.26
20	$6.3\ x10^{-6}$	2.1
18	$5.5\ x10^{-6}$	1.77
15	$5.3x10^{-6}$	1.5

This table shows that the displacements as well as the speeds do not exceed the reference values, which makes it possible to considerably improve the production of the C3plant.

However, for a good analysis of the 301turbo-alternator system, 3-D modeling during its operation is better suited than a simulation based on the differential equations of the motion. The rest of the work deals with the modeling in 3D with the Matlab system.

Modeling the 301rotor of the TG

3D modeling with Matlab of the system by the Finite Element Method (FEM)

The 3-D modeling of the system and the application of a maximum force on the rotor makes it possible to see the constraint and deformation reactions of the turbo-generator system. On the basis of the differential equation (1) and using the FEM, we use a number of parameters (mass m, stiffness K, force f, kinetic, potential and mechanical energy, constraint of deformation of Von Mises) for 3-D modeling of the system during its operation [9, 10].

Figures 7 and 8 illustrating the operation of the 301 turbo generator, will enable us to analyze and optimize the dynamic behavior of the entire system in order to increase the lifetime of the machine and the production of the C3 power plant.

3D modelling of the whole system of the 301turbo alternator with Matlab

By the FEM, the results of the modeling of the 301turbo-alternatoroperation, allowed obtaining the following geometric figure.

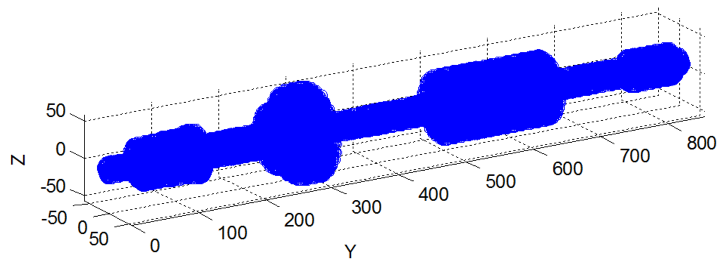


Figure 6:3D view of the whole system turbo alternator (turbine HP and LP, alternator and exciter)

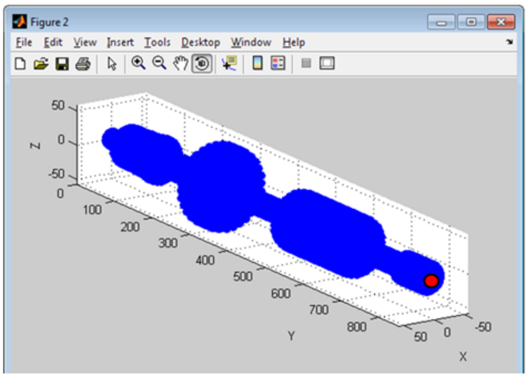


Figure 7: 3D view of the whole 301turbo alternator system with maximum force applied to the rotor

Here we have the real geometric figure approximated to the whole 301 turbo generator system in 3-D with Matlab. Knowing the geometry of the rotor of the301TG will make it possible to automatically calculate the inverse matrix of the coefficients of influence to directly propose the correcting unbalances to be implanted on the alternator and on the turbine.
Figure 7 makes it possible to check the behavior of the 301TG with the maximum force applied to the rotor symbolized here by the red dot in the center.
This figure which results from the 3D modeling of the machine makes it possible to have all the vibrations (elastic deformations) of the 301 turbo-generator system from the computer in order to compare the values with the reference values. This allows the operator to make decisions for production monitoring.
Figure 8 which illustrates the 3D modeling of the system gives the diffusion of the Von Mises constraint along the entire rotor as a result of the maximum force applied to it from the high pressure turbine to the exciter.

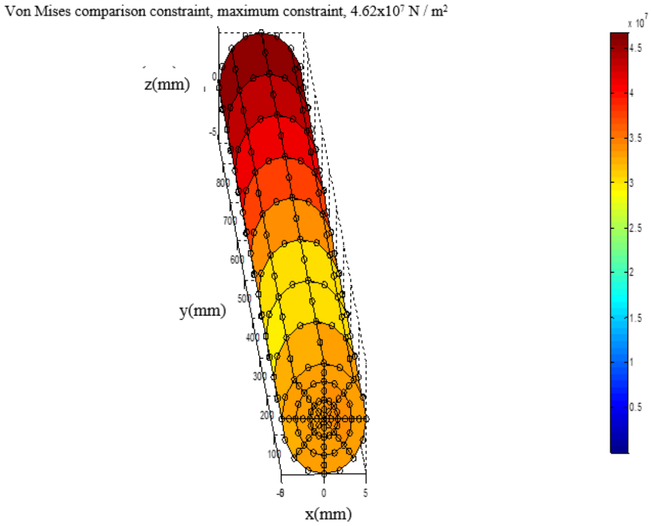


Figure 8: Representation of the Von Mises constraint in the center of the 301rotor, for a load of 27.5 MW

This figure, representing the deformation of Von Mises, shows that all along the center of the rotor is strongly *requested* as a result of the forces applied to the 301rotor.This leads to the propagation of the constraint of Von Mises towards the peripheries of the rotor, which induce fairly large mechanical vibrations. This figure also shows the parameters (level and value of solicitation constraint) of the deformation and propagation in the system.
The 3D modeling following the force applied to the rotor 301 made it possible to obtain the following results:

- Work of the external forces = 184564441,379 J;
- Elastic potential energy = 184564441,387 J;
- From the FEM, we can obtain the displacements of each element (here 2571 elements) constituting the turbo-alternator system 301.

These results make it possible to evaluate all the vibrations of the whole turbo-generator system, which gives the possibility of comparing them with the referential values in order to optimize the dynamics of the rotor and so increasing the production of theC3 plant.
With 3D modeling by Matlab, the following table gives some results of 301rotor displacements with the maximum force applied to each FEM element.

Table 8: Displacement of the rotor with 3D modeling, according to each element considered and maximum force applied

Element of the rotor from the turbine to the exciter	Displacement x(m)
(22.1)	$0.0212x10^{-4}$
(23.1)	$0.0009x10^{-4}$
……	……
(2571.1)	$-0.0316x10^{-4}$

This table gives all the mechanical vibrations of the entire turbo-generator system with 2571 elements; with maximum displacement of is 5018x10^{-5} m. Such a value is greater than the normal vibration threshold (15x10^{-6} m) and is due to unequal distributions of the various elements (turbines, alternator and exciter) as well as their masses throughout the rotor.
This requires balancing the entire turbo-alternator system by adjusting physical parameters.
The finite element method is of paramount importance in the study of vibration dynamics.Indeed, beyond the mechanical vibrations studied, we observe the finite field studied in real form (figure 6) under the solicitation submitted to it.
Being suspended at the level of the bearings, the rotor shaft undergoes elastic deformations and its elements (turbine, alternator and exciter) undergo mechanical vibrations by propagation of the vibration waves.
The results obtained show that the 301 turbo-alternator system will have to be better balanced in order to comply with the safety standards in force, that is to say to lower this maximum of 5.5x10^{-5}m to a value of less than 15x10^{-6} m. To do this, it is proposed for the next step to mount corrective unbalances on either side of the turbo-generator system to ensure balancing of the rotor.

Method of balancing from the coefficients of influence
In this part, the method of balancing chosen is a dynamic method requiring the rotation of the rotor. It is called the coefficients of influence method and has become universal. Indeed, it covers all cases of application (figure 9) [11-16]. This figure being a test bench is formed of the rotor and of two discs A and B mounted on this latter.

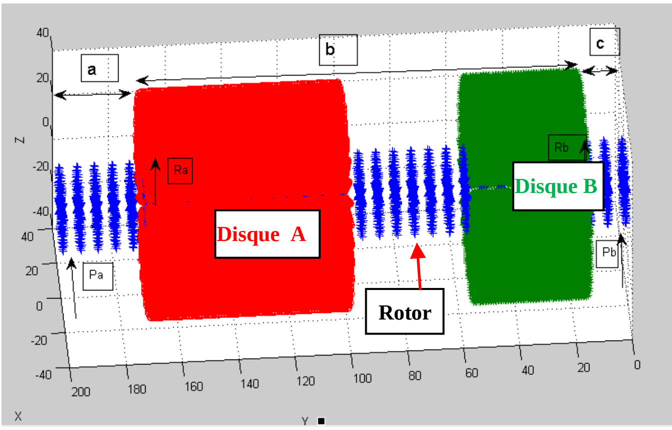


Figure 9: Test bench for two-disc balancing of rotor
This method is applicable for relatively short rotors or having fairly concentrated inertial characteristics and where the mass of the central portion extends over a length at least equal to the outside diameter.
We then compensate for the two eigen modes of solid body of translation and tilting. The rotor being rigid, we write with the relation (4) below its static equilibrium (force and torque).

These unbalances correctors $\overrightarrow{B_{lA}}$ and $\overrightarrow{B_{lB}}$ will compensate for the strong imbalance at the levels P_a and P_b due to the imbalance of the disks A and B on the rotor.

$$\begin{pmatrix} \vec{P}_1 \\ \vec{P}_2 \end{pmatrix} = \begin{pmatrix} 1-\alpha & \gamma \\ \alpha & 1-\gamma \end{pmatrix} \begin{pmatrix} \vec{B}_{lA} \\ \vec{B}_{lB} \end{pmatrix} \text{(4)with } C_1 = \begin{pmatrix} 1-\alpha & \gamma \\ \alpha & 1-\gamma \end{pmatrix} \quad (5)$$

In these equations, we have:

$$\alpha = \frac{a}{a+b+c} \quad (6)$$

$$\text{and } \gamma = \frac{c}{a+b+c} \quad (7)$$

From the initial distribution of the imbalances $\overrightarrow{B_{IA}}$ and $\overrightarrow{B_{IB}}$ the rotor of figure 9, directly brought back in the correction planes in terms of grams-millimeters and taking account of their angular positions, we have their corrective values to be implanted directly $\overrightarrow{B_{CA}}$ and $\overrightarrow{B_{CB}}$ with the aid of the following equation:

$$\begin{pmatrix} \overrightarrow{B_{IA}} \\ \overrightarrow{B_{IB}} \end{pmatrix} + \begin{pmatrix} \overrightarrow{B_{CA}} \\ \overrightarrow{B_{CB}} \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \end{pmatrix} \quad (8)$$

Is: $$\begin{pmatrix} \overrightarrow{B_{CA}} \\ \overrightarrow{B_{CB}} \end{pmatrix} = -C_1^{-1} \begin{pmatrix} \overrightarrow{P_1} \\ \overrightarrow{P_2} \end{pmatrix} \quad (9)$$

The determination of the balancers is based on the knowledge of the geometry of the rotor considered. These balancers can be masses, hollows and screws distributed on the circumference, the lateral surface, the section, etc.

Figures 10 and 11 respectively show the low pressure turbine and the 301 alternator respectively with the location (white holes on the circumference of the turbine section and on the lateral surface of the alternator) of the balancing balances on these elements.

Figures 10: Low pressure 301turbine with the location (empty in the form of white holes on the circumference of the turbine section) of balancing balances

Figure 11: Alternator with the location (empty in the form of small white holes on the side surface of the alternator) of balancing balances

Following Matlab's modeling, the balancing balances (white points) obtained are respectively presented for the alternator and for the turbine in figures 12 and 13 followed.

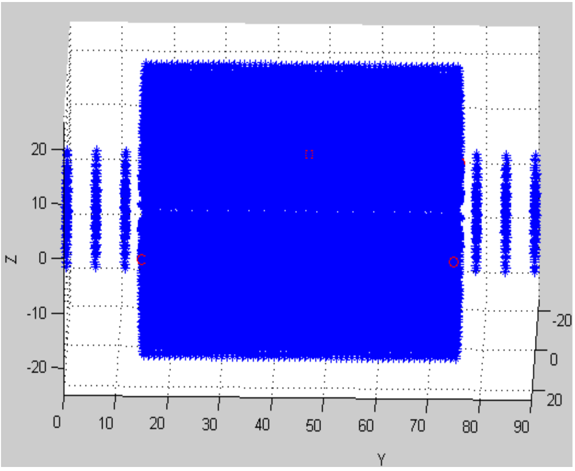


Figure 12: rotor avec la position latérale des emplacements des équilibreurs sur l'alternateur

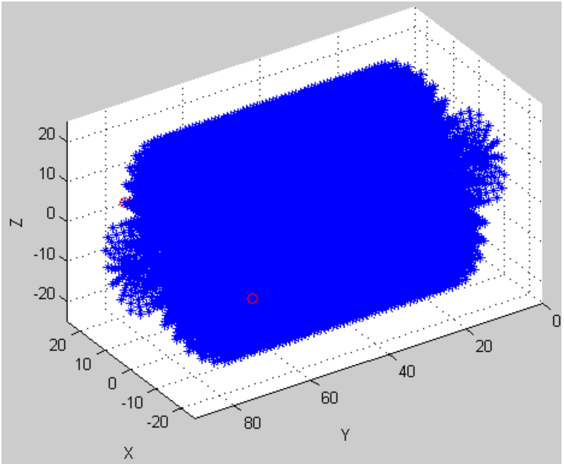


Figure 13: rotor avec la position des équilibreurs sur la circonférence de la section de la turbine

Balancing the 301turbo alternator rotor

Balancing of the 301 rotor of the alternator and the high pressure turbine

With influence coefficients method by Matlab, the resolution of equation (9) yields the values of p and p' of the corrector weights of the alternator and the HP turbine with:

A= [0.916 0.439; 0.084 0.561]' (10)

>> B= [19000; 5000]/20 (11)

>>p = A*B (12)

p = 891.2000

557.3000

>> 891.2/420

As alternator = 2.1219 (13)

>> 557.3/400

An turbine= 1.3932 (14)

With reference to the technical data of 301TG, we have small screws on the circumference and the area of the generator (450x5g = 2,250 kg).

For the turbine, the weight of the balancers in the form of 24 screws on the section of the HP rotor is 24 × 60 g = 1.44 kg.

This justifies the validity of the model proposed in this balancing study of 301 TG by the method of coefficients of influence. After correction of the proposed balancing, we obtain the displacements and speeds of the alternator rotor and for an unbalance force of 2.12kg.

Table 8 gives the values of the displacements and velocity before and after balancing by the proposed modeling.

Table 9: Values of displacements and velocity before and after modeling balancing

Load applied to the rotor (MW)	Horizontal displacement of the rotor x(m)		Horizontal displacement velocity of the rotor V (mm/s)	
	Before modeling	After modeling	Before modeling	After modeling
24	$8{,}84 x 10^{-6}$	$7{,}8 x 10^{-6}$	2.75	2.4
20	$8{,}83 \ x 10^{-6}$	$7{,}7 x 10^{-6}$	2.69	2.2
18	$7{,}05 \ x 10^{-6}$	$6{,}01 x 10^{-6}$	2.26	2.05
15	$7{,}02 x 10^{-6}$	$5{,}7 x 10^{-6}$	2.1	1.80

We again note a slight decrease in vibration in accordance with the standards of the reference table 4, hence the importance of unbalances in the balancing of the 301turbo-alternator system.
This situation constitutes the ideal operating conditions of our turbo alternator. This leads to an increase in the production of the C3 power plant with a better efficiency and a good lifetime of the machine.

Balancing the 301alternator rotor and the low pressure turbine

The resolution of equation (9) by the influence coefficients method gives the value of the compensators for balancing the alternator rotor and low pressure turbine.
Thus, we obtain:

>> A' = [0.887 0.425; 0.113 0.575]' (15)

>>b' = [19000; 10000]/20 (16)

>> p' =Axb (17)

p' =899.1500

691.2500

>> 899.15/420

Ans alternator = 2.1408 (18)

>> 691.25/415

Ans turbine = 1.6657 (19)

The results show that we find almost the same value for the alternator (2.14 kg almost equal to 2.12 kg for the alternator HP turbine).As for the LP turbine, we have 1.6657 kg as a mass or almost equal to 28 x 60 = 1.680 kg (mass of the 28 screws).
In view of the results obtained by simulation and modeling of the 301 TG system of C3 of Senelec, it is noted that:

- Mechanical vibrations control of the machine passes through the control of the balancing of its rotor with its various elements (turbine HP and LP, alternator and exciter);
- To achieve a better balancing of the rotor, the results obtained from the simulation and the modeling impose a reduction of mass on the alternator and an increase on the turbine; respectively 2.12 kg and 1.4 kg (HP turbine) and 1.6 kg (LP turbine) as the value of the balancing unbalances of these elements of the TG;
- The values obtained by calculations resulting from the influence coefficient method and from the technical data of the machine show that the proposed method of balancing the 301rotor is acceptable at 99% (2.12 kg / 2.14 kg);
- By the RungeKutta method, simulation could extend for lower loads (15 MW, 18 MW and 20 MW);
- Adjustment (balancing of the 301rotor) of parameters which induce the control of the vibrations increases the electrical energy production of the C3plant as well as the lifetime of the turbo generator. This gives the operator the possibility to make decisions to monitor the production of the C3 plant.

Conclusion

In power plants such as the C3 of Senelec, the effects of vibration have a great impact on the production of electrical energy via the turbo-generator groups.
In this work, emphasis has been placed mainly on the simulation and modeling methods of 301 TG with a view of controlling its vibrations and, consequently, the electrical energy production of the Senelec C3 power plant.
In the sense of balancing the 301 TG machine, in the first step, the resolution of ordinary differential equations by the RungeKuta method, from Matlab software allows us to obtain the vibrations characterized by displacements and velocity at each level of the rotor (HP and LP turbine, alternator and exciter).
Compared with the reference values, the results obtained are acceptable from the point of view of international standards on the noise and vibration of electromechanical machines.
In a second step, using the finite element method and by means of the Matlab software, a three-dimensional computer modeling of the 301 rotor was done. This modeling of the 301TG made it possible to optimize the dynamic behavior of the whole system (turbine HP and LP, alternator and exciter) by adjusting (balancing) physical input parameters such as mass and stiffness from a computer.

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