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Drivetrain Flexibility Effect on Control Performance of a Horizontal Axis Wind Turbine

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Abstract

Wind turbine is a complex machine, several elements has height flexibility such as blade, tower and drive train, this phenomena have an important influence on the behavior of system it can make damage in system structure, Torsional vibrations in the drive train of wind turbine can cause large mechanical stress and reduce the life cycle of the components, it can affect the stability of the system and reduce the quality of produced energy, the aim of this work is to show the effect the flexibility of drivetrain on the control performance, in the first we apply a control law on a rigid model and we keep the same parameters to control a flexible model which is presented by a tow mass model , we note the flexibility effect on the performance system.

Keywords: Wind turbine, Control, PID, flexibility, vibration, Stability, drivetrain.

INTRODUCTION

In the last decade, wind energy has been developed considerably, generating a significant increase in installing capacity in the world This trend is being driven, first, by limiting and depleting the fossil fuels (diesel oil, coal, ...), fissile material (uranium) and other emissions of greenhouse gas emissions that has been caused by hydrocarbons, that doesn't mention the production difficulty to treat the nuclear radioactive waste case.

The mathematical modeling of a wind turbine is a critical task; the model complexity will change depending on the purpose of the study, research that aims to optimize the structures and analyze fatigue requires more complex models with a large number of degrees of freedom[1][3], the introduction of the flexibility of the different component is required; for work control, we find that most research develop control laws based on rigid models[14][15][17], which ignore the flexibility of different components.

The ignorance of this phenomenon reduces the mathematical model, but can cause serious problems that may affect the stability of the systems, degraded The ignorance of this phenomenon reduces the mathematical model, but can cause serious problems that may affect the stability of the systems, degraded the quality of energy produced and reduce the operational life of the machine; in this work we have shown the effect of this phenomenon on the control performance, in the first part of the paper, we present the different models configuration of a large wind turbine, beginning with a rigid model, and a flexible model with tow masses, finally a flexible model with three masses. We compare the behavior of these three models to show the effect of the flexibility of the drivetrain on the system behavior. In the second part we apply a control law on the rigid model by using a PID controlled, and we keep the same parameters to control a flexible model which is presented by a tow mass model, we note the flexibility effect on the performance system.

WIND TURBINE MODELING

The wind turbines are complex machines, with several subsystems (blades, actuators, generator, reducer ...); the modeling of all the system in one model is a difficult task.

The wind turbine model describe the dynamic's behavior of the machine, it‘s built generally from several sub-models, the first sub-model describe the interaction between the wind and the rotor, it's called aerodynamic model, it calculate the aerodynamic loads applied on blades, which are converted to aerodynamic torque, it's converted to electrical power through the gearbox and the Induction generator, measured power is feedback through the pitch controller to the pitch System that changes this actual pitch angle, these models are interconnected to form the total model, the following figure present the interaction between its sub-models.

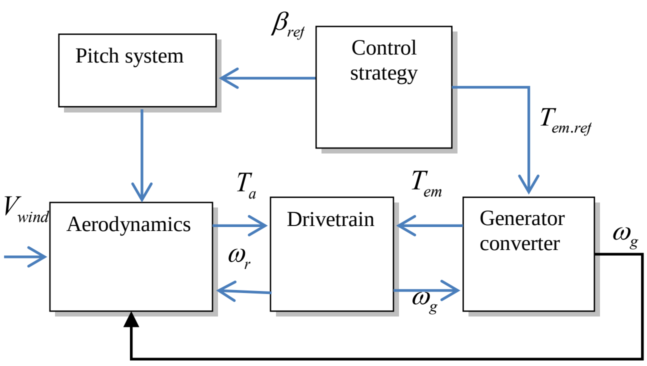

Figure 1: Wind turbine model

A. Aerodynamic modeling

The aerodynamic model is that which calculates the aerodynamic force applied to the system, several methods exist to determine these efforts that we find in the bibliography [10], in this work we used an Analytical approach to determine the aerodynamic power and the aerodynamic torque, which are presented by the following equations:

$$P_a = \frac{1}{2}\rho R^2 C_p(\lambda,\beta)V^3 \quad (1)$$

$$T_a = \frac{1}{2}\rho R^2 C_p(\lambda,\beta)V^2 \quad (2)$$

Where:

$$C_p = \frac{P_a}{P_v} \quad (3)$$

Is the so-called power coefficient, β is the blade pitch angle, and λ is the tip speed ratio.

The power coefficient can be calculated by an analytical approximation, the approach is defined as follow:

$$c_p(\lambda,\beta) = \left(\frac{c1}{\tilde{\lambda}} - c_2\beta - c_3\right)e^{-\frac{c_4}{\tilde{\lambda}}} + c_5\lambda \quad (4)$$

with

$$\frac{1}{\tilde{\lambda}} = \frac{1}{\lambda + c_6} + \frac{c_7}{\beta_3 + c_8} \quad (5)$$

$$\lambda = \frac{R\omega}{v} \quad (6)$$

Thus any change in the rotor speed or the wind speed induces change in the tip speed ratio leading to power coefficient variation. In this way, the generated power is affected [5] [6]

Figure 1 Shows a group of typical $C_p(\lambda,\beta)$ curves where optimum values of tip speed ratio, λ_{opt} correspond to the maximum power coefficient $C_{p\max}$.

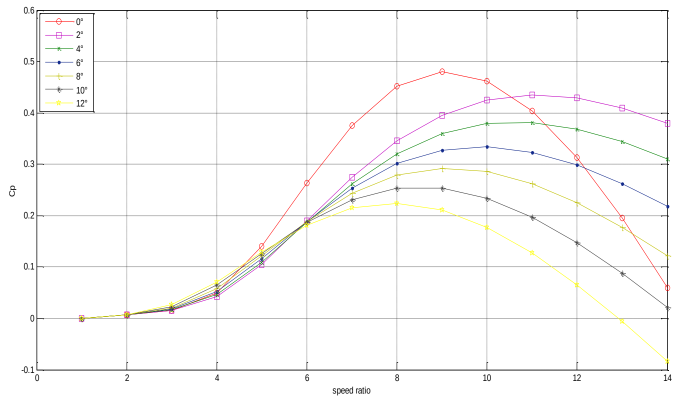

Figure 2: Power Coefficient Curve for Different Pitch Angles

B. Structural models of different configuration of drivetrain

1) One Mass Model

This configuration is used in different works which are based on classical laws of control [8] [9]. This is the simplest configuration of a wind turbine drivetrain. In this last, the system is reduced the maximum possible, considering all of the blades and the shaft as a one inertia and one damper that brings all the friction and amortization system, the transmission shaft is assumed rigid. The figure 2 presents the configuration of one masse model.

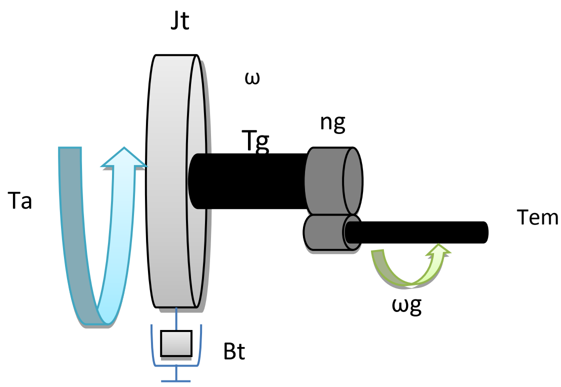

Figure 3: One mass model

The Dynamic equation of this model is given by:

$$B_t = B_a + B_g n_g^2 \quad (7)$$

With B_t is the total damper, B_a and B_g is respectively the low speed and height speed shaft dampers.

$$J_t = J_a + J_g n_g \tag{8}$$

Where J_r is the total inertia, J_a is the rotor inertia and J_g is the generator inertia, n_g is the gear ratio.

By applying the second law of newton we obtained

$$J_t \dot{\omega}_g = T_a - B_t \omega_t - T_g \tag{9}$$

$$\dot{\omega}_g = -\frac{B_T}{J_T}\omega_g + \frac{n_g}{J_T}T_a - \frac{n_g^2}{J_T}T_g \tag{10}$$

Where $\dot{\omega}_g$ is the generator speed, T_a and T_g is respectively the aerodynamic torque and electromagnetic torque of generator.

2) Two masses Model

in this model we include the deformation of the low speed shaft, we conceder each shaft by its mass, the first mass include the blade and hub masses and the second mass include the high speed shaft mass, the latter describes the vibrational behavior of the low speed shaft; it's used generally in control and most of control works are based on this model [8] [9]. The figure 4 presents the structure of tow masses model.

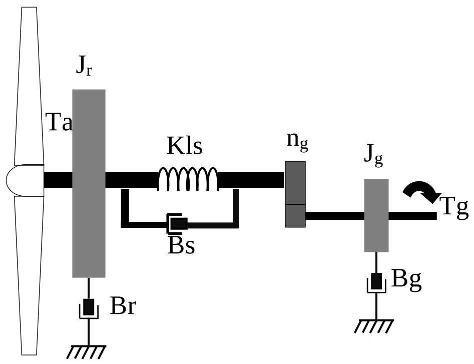

Figure 4: two masses model

By the application of the Lagrange method we obtained.

$$\begin{cases} J_r \dfrac{d\omega_r}{dt} = T_a - K_s(\theta_r - \dfrac{1}{ng}\theta_g) - B_s(\omega_r - \dfrac{1}{ng}\omega_g) - B_r\omega_r \\ J_g \dfrac{d\omega_g}{dt} = \dfrac{K_s}{ng}(\theta_r - \dfrac{1}{ng}\theta_g) - \dfrac{B_s}{ng}(\omega_r - \dfrac{1}{ng}\omega_g) - B_g\omega_g - T_g \\ T_{ls} = K_s(\theta_r - \dfrac{1}{ng}\theta_g) - B_s(\omega_r - \dfrac{1}{ng}\omega_g) \end{cases} \tag{11}$$

The model of the system can be written as following stat space.

$$\begin{bmatrix} \dot{\omega}_r \\ \dot{\omega}_g \\ \dot{T}_{ls} \end{bmatrix} = \begin{bmatrix} \dfrac{-Br}{J_r} & 0 & \dfrac{-1}{J} \\ 0 & \dfrac{-B_g}{J_g} & \dfrac{1}{n_g J_g} \\ \dfrac{k_s - BsBr}{J_r} & \dfrac{1}{n_g}(\dfrac{B_s B_g}{J_g} - K_s) & -K_s(\dfrac{J_r + n_g J_g}{n_g^2 J_g J_r}) \end{bmatrix} \begin{bmatrix} \omega_r \\ \omega_g \\ T_{ls} \end{bmatrix} + \begin{bmatrix} \dfrac{1}{J_r} \\ 0 \\ \dfrac{Bs}{J_r} \end{bmatrix} T_a + \begin{bmatrix} 0 \\ -\dfrac{1}{J_r} \\ \dfrac{Bs}{n_g J_g} \end{bmatrix} T_g \tag{12}$$

3) Three masses model

The model has three masses is a model that describes in more detail the system, in the latter it shall consider parameters that are ignored in other models, the mass of gearbox was added to the model, it is presented by its inertia (JG1, JG2), deformation of the high-speed shaft is included, it is presented by a couple of spring-damper noted (Khs,Bhs) in the following figure.

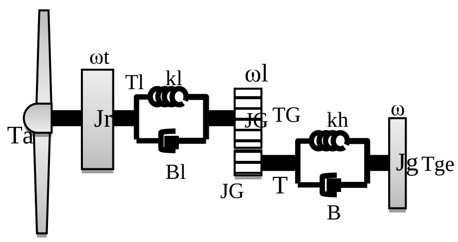

Figure 5: Three masses model

This model can be used in vibration analysis to determine the frequency components of system and found the unwanted resonance frequencies; it can be also used to the control of the power, speed and vibration of the system.

The dynamics equation of system is given by:

$$T_{ls} = J_r \dot{\omega}_t + B_{ls}\omega_t + K_{ls}(\theta_t - \theta_{tls}) \tag{13}$$

$$T_{hs} = J_G 2\dot{\omega}_{hs} + B_g \omega_{hs} + K_{hs}(\theta_{hs} - \theta_g) \tag{14}$$

$$T_{G1} = J_{G1}\dot{\omega}_{ls} + B_{ls}\omega_{ls} + kl_s(\theta_{ls} - \theta_{hs}) \tag{15}$$

$$-T_{gen} = J_{gen}\dot{\omega}_g + B_{hs}\omega_g + K_{hs}(\theta_g - \theta_{hs}) \tag{16}$$

The model of the system can be written as following stat space.

$$\begin{bmatrix} \dot{\omega}_t \\ \dot{\omega}_{ls} \\ \dot{\omega}_{hs} \\ \dot{\omega}_g \\ \dot{T}_{ls} \\ \dot{T}_{G1} \\ \dot{T}_{hs} \end{bmatrix} = \begin{bmatrix} -\frac{B_{ls}}{J_r} & 0 & 0 & 0 & -\frac{1}{J_r} & 0 & 0 \\ 0 & -\frac{B_{ls}}{J_g} & 0 & 0 & 0 & -\frac{1}{J_{g1}} & 0 \\ 0 & 0 & \frac{B_g}{J_{g2}} & 0 & 0 & 0 & -\frac{1}{J_{g2}} \\ 0 & 0 & 0 & \frac{B_g}{J_{gen}} & 0 & 0 & 0 \\ K_{ls} - \frac{B^2_{ls}}{J_r} & -K_{ls} & 0 & 0 & 0 & \frac{-1}{J_{g1}} & 0 \\ 0 & K_{ls} - \frac{B_{ls}}{J_{g1}} & -K_{ls} & 0 & 0 & \frac{-1}{J_{g1}} & 0 \\ 0 & 0 & K_{hs} - \frac{B_g}{J_{g1}} & -k_{ls} & 0 & 0 & \frac{-1}{J_{g2}} \end{bmatrix} \begin{bmatrix} \omega_t \\ \omega_{ls} \\ \omega_{hs} \\ \omega_g \\ T_{ls} \\ T_{G1} \\ T_{hs} \end{bmatrix} + \begin{bmatrix} \frac{-1}{J_r} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} T_a + \begin{bmatrix} 0 \\ \frac{-1}{J_r} \\ 0 \\ \frac{-1}{J_{gen}} \\ \frac{-1}{J_r} \\ 0 \\ 0 \end{bmatrix} T_g \tag{17}$$

OPEN LOOP BEHAVIOR ANALYSIS FREQUENCY ANALYSIS

Design, optimization and control of dynamic systems require knowledge of his frequency properties. In this section we present a frequency analysis of each model of power train comparing thereafter the difference between the proposed models.

In the last part we analyze the frequency properties for each model using bode presentation.

The figure 6 present bode diagram of one mass model; we note that it is similar to a low-pass filter, with transfer function of first order. , it's a rigid rotational system with natural frequency equal zero.

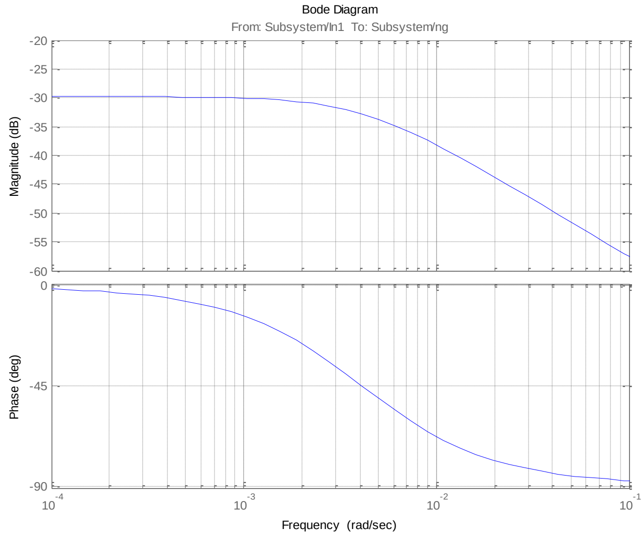

Figure 6: Bode diagram for one masse model

Figure 7 shows the frequency representation of tow masses model. We note the appearance of a peak resonant frequency at 1.2 Hz, the resonance vibration of drivetrain can be excited in this frequency field.

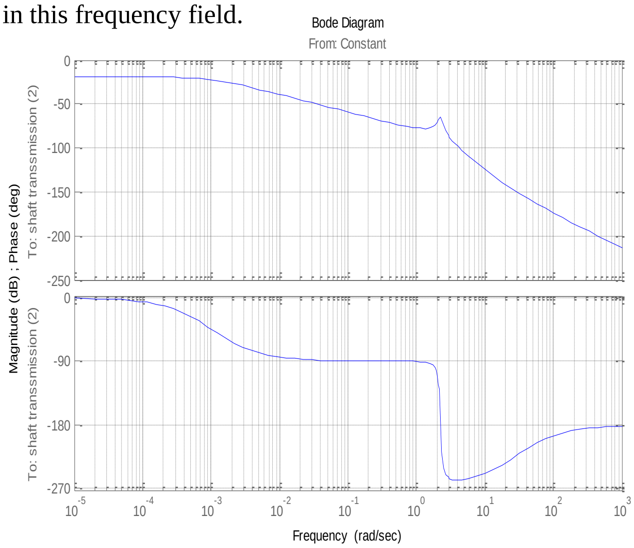

Figure 7: bode diagram for two masses model

From this diagram it Notes that the three masses model is similar to that of two masses, which mean that the influence of the mass of multiplier is almost negligible, note that the frequency of resonance in both graph is almost the same, its value in this diagram is 0.5 Hz.

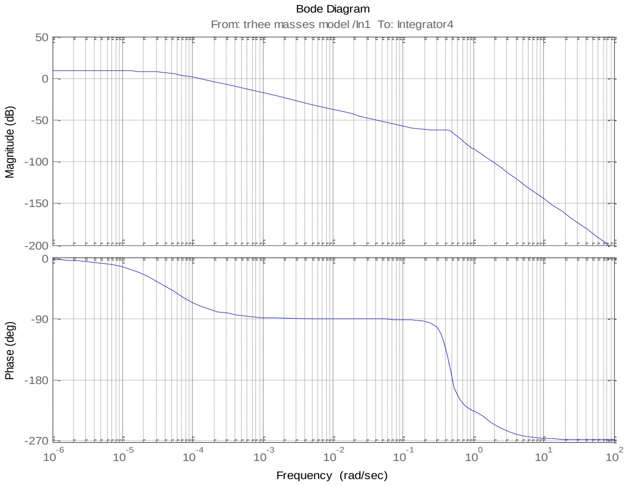

Figure 8: Bode diagram for three masse model

The resonance peak is related to viscous damping of system, high peak amplitude of this implies presence of a low damping.

Damping in the system of three mass was increased by the addition of damping of the high speed shaft and friction's generator and the gearbox. This explains the decrease of peak resonance.

Neglecting inertia multiplier, a two-mass model seems sufficient to describe the vibration behavior of the system.

Systems behaviors of different wind speed nature

In this part of this work, the cases of study focus on the response of different configurations of wind turbine models for different wind speed conditions, while keeping the pitch angle fixed, for this aim two simulation has done, firstly we analyze the behaviors of our models for different constant values of wind speed.

In the second part of the simulation we applied a stochastic wind speed on our wind turbine model and compare its behavior.

The table. 2, table 3 and table 4 present the parameters of wind turbine simulation for each model.

RESPONSE FOR CONSTANT WIND SPEED

In this section we applied a constant wind speed (12m/s) on each model; we note the appearance of the fluctuation on the generator speed caused by the deformation of slow shaft.

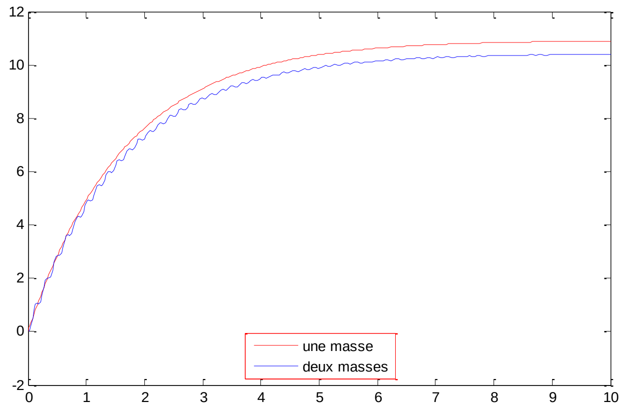

Figure 9: Generator speed of one and two masses model for constant wind speed

RESPONSE FOR VARIABLE WIND

In this part the wind turbine operation is simulated in time domain, for the typical wind speed time series as shown in the following figure, it represents the variable wind speed profile with medium speed equal 12m/s in 10s.

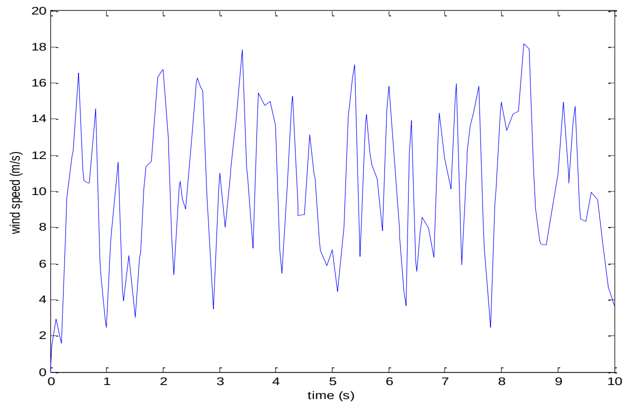

Figure 10: Wind profile

The following figure shows the progression of the generator speed for one mass model and two-mass model, as a side note the appearance of vibrations due to the effect of the drivetrain flexibility.

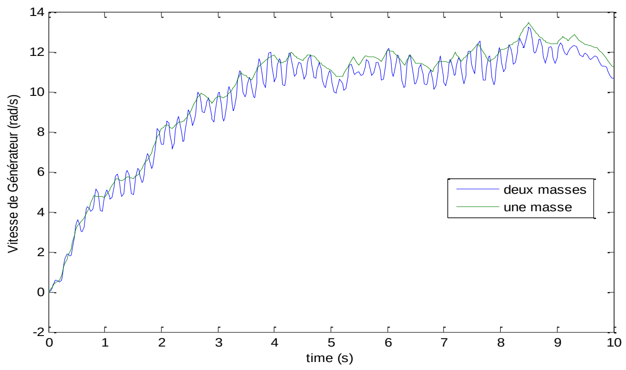

Figure 11: Generator speed of one and two masses model for variable wind speed

FLEXIBILITY EFFECT ON CONTROL SYSTEM

A. PID CONTROL OF WIND TURBINE

PID control is a conventional method of control, but it is still used by the majority of wind turbine around the world, the control structure varies between turbines to another [15] [16] [18], the following figure presents the chosen architecture of the control system.

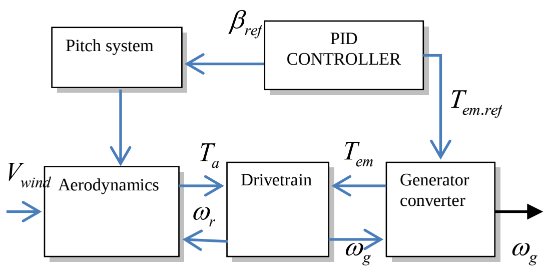

Figure 12: Control structure design

We conceder in system inputs the aerodynamic torque and the electromagnetic torque, and the output is the generator speed.

1) PID controller

The classical PID control law of generator speed is given by:

$$\omega_g = K_{p1}\varepsilon_1(t) + K_{d1}\frac{d\varepsilon_1(t)}{dt} + \frac{1}{K_{i1}}\int \varepsilon_1(t)dt$$

(18)

Where: $\varepsilon = \omega_{gref} - \omega_g$ is the generator speed error, ω_{gref} is the input reference signal and the terms K_p, K_i and K_d define:

The proportional term: providing an overall control action proportional to the error signal through the all pass gain factor.

The integral term: reducing steady state errors through low frequency compensation by an integrator.

The derivative term: improving transient response through high frequency compensation by a differentiator.

Control parameters: by using the graphical methods, we find the control parameters presented in following table.

TABLE I

CONTROL PARAMETERS

Kp	Ti	Td
13467.257	94567.661	-1123.899

2) Result and simulation

The aim object of this simulation is to show the flexibility effect on the control performance under variable wind speed, for this we are applied a PID control on a rigid model of wind turbine drive train, the control parameters are calculated for this model, thereafter, we apply the same controller on a flexible model, we note that the flexibility of low speed shaft we note the appearance of the high frequency fluctuation in generator speed, due to the torsional damping of the LSS, the figue 13 shows the evolution of the generator speed curve for each model.

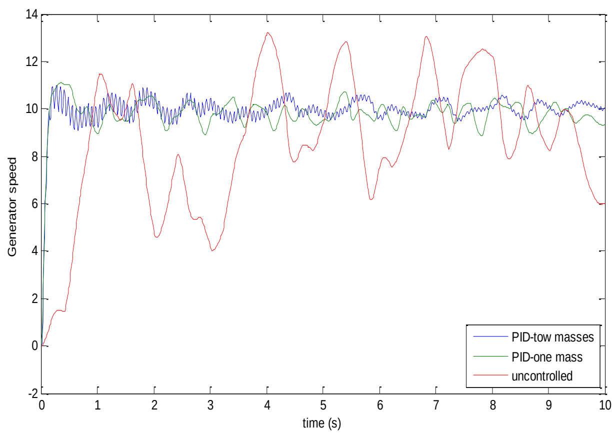

Figure 13: Generator speed for controlled one mass model/tow masse model/uncontrolled model

The following figure shows the PSD presentation of flexible controlled, uncontrolled model and the rigid model, it can be observed the appearance of vibration in the high frequency.

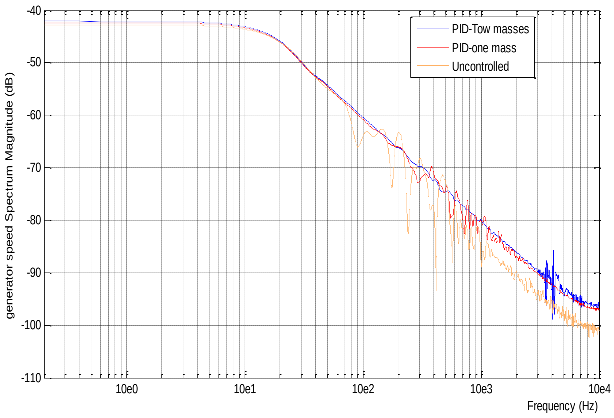

Figure 14: PSD graph speed for controlled one mass model/tow masse model/uncontrolled model

The characteristics of the loads in the drive train in time field and frequency field are shown in Figure 15 and Figure 16 respectively. So that the controller can stabilize the generator speed by acting on the load torque, the rotational speed is vary abruptly due to the stochastic nature of the wind, so naturally, the load torque of the shaft also suddenly changes to compensate this variation, that can appear a damping torsional torque.

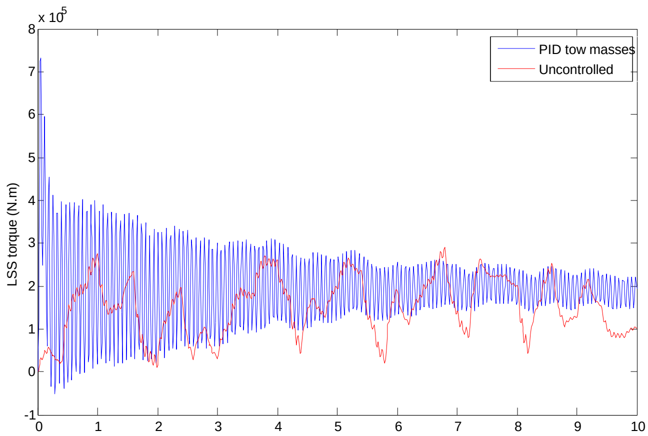

Figure 15: Torque of low speed shaft

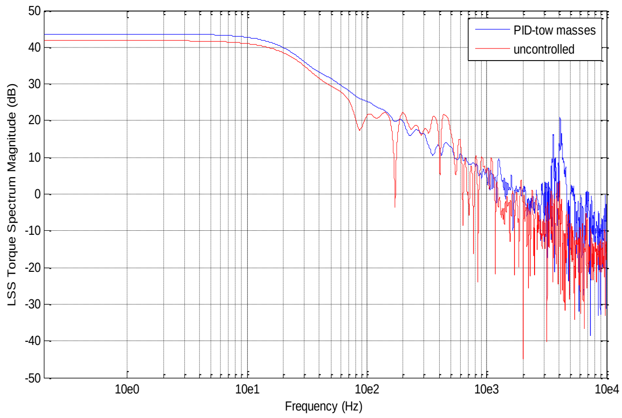

Figure 16: PSD spectrum of low speed shaft Torque

CONCLUSION

In this paper we have performed a comparative study of several drivetrain structures of wind turbine, in order to show the effect of flexibility on the behavior of the machine.

In the 1st part of this labor we proposed three configurations of drivetrain model, a rigid model (one mass model), a flexible model with two mass, and a three masses model masses, we conclude that the two masses model is similar to the three masses model and the addition of the mass of gearbox and the flexibility of the high speed shaft does not change the vibrational nature of model, we conclude that the two masses model seem sufficient to describe the oscillatory behavior of a wind turbine drivetrain.

In the second part we apply a control law on the rigid model by using a PID controlled, and we keep the same parameters to control a flexible model which is presented by two masses model, we note that the flexibility of the LSS can affect the stability of the system if the control parameters are close to the critical point of stability.

We note also the evolution of the torsional damping in the LSS, which can cause damage of shaft by fatigue. To remedy these problems we have proposed as perspectives an adaptive PID control to cancel the stability problem. For the torsional vibration of the LSS we propose a vibration control to suppress this vibration.

APPENDIX

This labor are perform on a large dimension wind turbine, we have chosen the HWAT wind turbine 1.5 MW, the next tables present the mechanicals parameters of each mode

TABLE II

SIMULATION PARAMETERS FOR ONE MASS MODEL

Parameter	Value
Jt	325840 kg.m²
Bt	30.9600 N.m/(rd/s)
Ng	43.165

TABLE III

SIMULATION PARAMETERS FOR TOW MASSES MODELS

parameter	Value
Jr	3.25 .105 kg.m²
Jg	440 kg.m²
Kls	2.691 .105 N.m/rd
Bls	9500 N.m/(rd/s)
Br	27.36 N.m/(rd/s)
Bg	2 N.m/(rd/s)
ng	43.165

TABLE IV

SIMULATION PARAMETERS FOR THREE MASSES MODELS

Parameter	Value
Jr	3,23.105kg.m²
Jg	400 kg.m²
JG	39 kg.m²
Kls	2.69 .105 N.m/rd
Bls	9500 N.m/(rd/s)
Khs	1.87 104N.m/rd
Bhs	87.2 N.m/(rd/s)
ng	43.165

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