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Compact Analytical Models for Vibration Analysis in Modelica/Dymola: Application to the Wind Turbine Drive Train System

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ABSTRACT

Modeling mechatronic multibody systems in the preconception phase is an extremely important step in their design process. This paper presents a pre-designing method applied to a mechatronic system with regarding the vibrational aspects without going through costly design techniques. Currently, modeling of flexible structures by systems approaches is a relevant issue. In this regard, flexible elements relatively simple based on analytical method have been developed in Modelica/Dymola, creating compact analytical models. Due to the objected oriented structure of Modelica, these elements can be subsequently connected to other components (components from Modelica Standard Library) in order to study the vibrational behavior of such a system, in an early design phase. Our approach is illustrated for the drive train system of a wind turbine. Several parameters having significant influence on the vibration propagation have been analyzed. This new method will allows us performing a representative and robust modeling and simulation.

INTRODUCTION

The design of a mechatronic system deals with the integrated design of a mechanical system with its embedded control system (Hammadi et Al., 2014; Lu et Gong, 2014, Guizani et Al, 2014).

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Mechatronic has changed the design, different methodologies and design tools have been developed for modeling mechatronic systems taking into account the multidisciplinary aspect (Hammadi, 2012; Hammadi et Al, 2011), (Lisandrin and van Tooren., 2006; Lv et Al, 2015). These methodologies are designed to improve the performance of mechatronic products and reduce the cost and the time needed for delivery to the customer. To achieve these objectives, it is important to take a great effort from the early phases of the design cycle (Hamza et al., 2015a; Hamza et al., 2015b; Hammadi et Al., 2012, Hammadi et Al., 2012b).

The pre-designing phase is an extremely important step in the design process. It takes place between the needs analysis phase and the detailed design phase (Raka, 2011). It is performed from different specifications and should define the system architecture and its setting. To better predict the behavior of a mechatronic system, the pre-dimensioning phase requires the realization of representative and robust modeling and simulations of the overall system in order to optimize the search for an optimal design (Plateaux et Al., 2011), (Mandil G.,2011). The design of a mechatronic system presents some difficulties which can be related to various reasons. Firstly, it may be related to the model topology such as the hybrid dynamic aspect of the system (Khalfaoui S., 2003). In fact, a mechatronic system is usually described by a dynamic multi-physics system, whose states are time functions must agree to the environmental simulations. To simulate a mechatronic system, a mathematical model describing the system must first of all be established. The mathematical models of physical phenomena are usually based on differential equations. Contrary to a static system, a dynamic system is a system that evolves with time for causal and deterministic manner. The deterministic evolution of a dynamic system allows differentiating models and classified them into two groups which are continuous models and discrete models (Khalfaoui S., 2003). For a continuous model (Gaaloul Chouikh, 2012), the corresponding mathematical approach is when the time is the variable that describes the evolution of the system (Cellier et Al., 1996; Cellie et Greifeneder, 2013). Based on the differential equations, there are two types of time continuous descriptions: Ordinary differential equations (ODE) and algebraic differential equations (ADE). Unlike the continuous system, the discrete system is governed by state variables that change their values instantly at different times such as the sampled systems (Gaaloul Chouikh, 2012). It is necessary to distinguish between continuous/discrete models in time and continuous /discrete models in space. For example, the mass-spring system is the most famous case. It is a continuous model in time but discrete in space. Modeling with a system of masses and springs is interesting as its implementation is easy, its resolution is not complex, and also its ability to represent the physical reality of a structure which allows simulating a variety of mechanical behavior such as deflection, torsion and traction. However, modeling with mass spring presents some difficulties. Indeed, a real system is formed by an infinity system which is virtually impossible to be considered for a computer, it is also difficult to set all parameters in all bodies and all springs in order to model such a phenomenon. Therefore, continuous approaches are preferred more than the discrete approaches. Although they are more complex, they are typically more reliable and accurate. Evenly, the source of complexity in mechatronic system modeling can also be related to its multi-physic aspect. The multi-physical systems and especially systems that operate in severe conditions such as high temperature and vibration can cause many constraints in their design. The integration between disciplines results in multiple functional flow through components (Breedveld, 2004). The multi-physical interactions between mechanical, electrical and thermal are those that determine the dynamic performance of a mechatronic system, and this implies that it is important that the system be designed as together as much as possible (Fisette et Al., 2006). The methods for mechatronic systems design are increasingly studied and many models of process design with a highly integrated development strategy have been proposed in order to address the issue based on the multi-domain integration (Hammadi, 2012). For example, Moncef and Jean-Yves (Hammadi et Choley, 2015) have developed a compact approach using meshless methods for the simulation of multiphysics systems. This method combines the functions radial bases with the quadrature différentielle method (DQM) for solving problems modeled with partial differential equations in Modelica environment. Currently most systems are regarded as mechatronic. Wind turbine can be considered as a typical example. In recent years, much attention has been given to model the wind turbine as a mechatronic system. For example, Modelica library is in development at the Fraunhofer Institute for Wind Energy and Energy System Technology (IWES). This library comprises all major components of a wind turbine also models for aerodynamic and hydrodynamic load calculations. The in-house tool One Wind proves how coupled analyses of a wind turbine are realized based on the object oriented modeling language Modelica (Strobel et Al, 2011; Brommundt et Al, 2012; Samlaus et Al, 2012; Rickert, 2012). Among the limitations of this project, all components are considered rigid and therefore the material elasticity is ignored as well as the resulting vibration. The aim of this paper is to present a pre-dimensioning method, based on analytical approach, taking into consideration the effect of vibration without using techniques of detailed design such as 3D CAD and the finite element method. In order to predict the behavior of a mechatronic system, robust modeling and simulations with Modelica/Dymola are carried out. We investigate the sensitivity of the system for several parameters and different architectures in order to develop accurate and reliable design. Our contribution is structured as follows: the first section deals with this introduction, in section 2 we introduce the analytical formulation of the problem. Implementation in Modelica is presented in section 3 and application to wind turbine drive train system is given in section 4. The conclusion of our paper is presented in section 5.

THEORETICAL APPROACH

The analytical methods agree well with the object oriented structure (GAALOUL CHOUIKH S., 2012; Hadj-Amor, 2008). In this section, the theoretical basis of the compact models development has been presented (Singiresu, 2007). A rigorous mathematical formulation of the models allows knowing the analytic expression of the variables that evolve in time. These models will be implemented in the next section in Modelica\Dymola in order to build a library of compact analytical models.The equation of motion of a uniform shaft subjected to distributed external torque $m_t(x,t)$ is given by (Singiresu, 2007):

$$GI_P \frac{\partial^2 \alpha(x,t)}{\partial x^2} + m_t(x,t) = I_0 \frac{\partial^2 \alpha(x,t)}{\partial t^2} \tag{1}$$

with, $\alpha(x,t)$ is the angular displacement of a shaft in torsional vibration , G is the shear modulus, I_P is the polar moment of inertia of cross section, I_0 is the mass polar moment of inertia per unit length, $m_t(x,t)$ is the external torque applied per unit length and l is the beam length.

The homogeneous solution of eq. (1) can be obtained by the separation of variables method (Singiresu, 2007) i.e. decomposing $\alpha(x,t)$ into two independent variables $f_n(x)$ and $\eta_n(t)$.

$$\alpha(x,t)=\sum_{n=1}^{\infty} f_n(x)\eta_n(t) \quad (2)$$

With, $f_n(x)$ is the n[th] normal mode and $\eta_n(t)$ is the generalized coordinates. The mode shapes $f_n(x)$ are determined by solving the eigenvalues problem and applying the boundary conditions of the shaft (Singiresu, 2007).

$$GI_p \frac{d^2 f_n(x)}{dx} + I_0\omega_n^2 f_n(x)=0 \quad (3)$$

Substituting equation (2) in the equation (3), then:

$$\sum_{n=1}^{\infty} GI_P f^{"}(x)\eta_n(t)+m_t(x,t)=\sum_{n=1}^{\infty} I_0 f_n(x)\ddot{\eta}_n(t)$$

(4)

$$\text{With,} \quad f_n^{"}(x)=\frac{d^2 f(x)}{dx^2}, \quad \ddot{\eta}_n(t)=\frac{d^2\eta_n(t)}{dt^2} \quad (5)$$

The eq. (1) becomes (Singiresu, 2007):

$$-\sum_{n=1}^{\infty} I_0\omega_n^2 f_n(x)\eta_n(t)+m_t(x,t)=\sum_{n=1}^{\infty} I_0 f_n(x)\ddot{\eta}_n(t)$$

(6)

Multiplying eq. (6) by $f_m(x)$ and integrates it between 0 and l gives (Singiresu, 2007):

$$I_0\omega_n^2\eta_n(t)\sum_{n=1}^{\infty}\int_0^l f_n(x) f_m(x)dx+\int_0^l m_t(x,t) f_n(x)dx=$$

$$I_0\ddot{\eta}_n(t)\int_0^l f_n(x) f_m(x)dx$$

(7)

For two distinct eigenvalues $\omega_i \neq \omega_j$, the orthogonality relation of the normal modes of the shaft is as follows: $\int_0^l f_i(x) f_j(x)dx=0, \quad i\neq j$ (8)

The normal modes must satisfy the following normalization condition:

$$\int_0^l f_i^2(x)dx=1, \quad i=1,2... \quad (9)$$

Taking into account the orthogonality relationship and the normalization condition, we obtained:

$$\ddot{\eta}_n(t)+\omega_n^2\eta_n(t)=Q_n(t), \quad n=1,2,... \quad (10)$$

With, the generalized force $Q_n(t)$ can be calculated as (Singiresu, 2007):

$$Q_n(t)=\frac{1}{I_0}\int_0^l m_t(x,t) f_n(x)dx \quad (11)$$

The generalized coordinate can be written as (Singiresu, 2007):

$$\eta_n(t)=A_n\cos(\omega_n t)+B_n\sin(\omega_n t)+ \frac{1}{\omega_n}\int_0^t Q_n(\tau)\sin\omega_n(t-\tau)d\tau \quad (12)$$

In steady state, the angular displacement (Singiresu, 2007):

$$\alpha(x,t)=\sum_{n=1}^{\infty}\frac{f_n(x)}{\omega_n}\int_0^t Q_n(\tau)\sin\omega_n(t-\tau)d\tau \quad (13)$$

We consider that the shaft is subjected to a concentrated torque at the end (x = 0).
The torque applied to the shaft is:

$$m_t(x,t)=M.e^{j\omega t}\delta(x) \quad (14)$$

The generalized force is then:

$$Q_n(t)=\frac{1}{I_0}\int_0^l Me^{j\omega t}\delta(x) f_n(x)dx=\frac{Me^{j\omega t}}{I_0} f_n(0)$$

(15)

For a free -free shaft (Fig.1), the natural frequency and modes shapes are given by these equations (Singiresu, 2007):

$$\omega_n=\frac{n\pi c}{l}=\frac{n\pi}{l}\sqrt{\frac{G}{\rho}} \quad (16)$$

$$\text{And,} \quad f_n(x)=\sqrt{\frac{2}{l}}\cos\left(\frac{n\pi x}{l}\right), \quad n=1,2.. \quad (17)$$

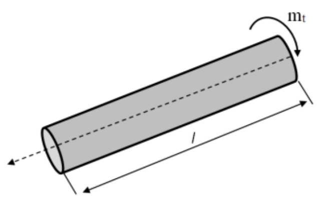

Fig.1. Free- free shaft subjected to external torque

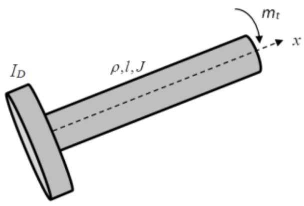

Fig. 2. Free shaft with a rigid disk subjected to external torque

For a free shaft with a rigid disc (Fig.2), the frequency equation is (Singiresu, 2007):

$$\alpha_n.\cot(\alpha_n)=-\beta \quad (18)$$

$$\text{With,} \quad \alpha_n=\frac{\omega_n l}{c}, \quad \beta=\frac{\rho Jl}{I_d} \quad (19)$$

And, J is the polar moment of inertia of cross section, I_d is the disk inertia.

$$c = \sqrt{\frac{G}{\rho}} \qquad (20)$$

It is impossible to found an analytical expression of the natural frequency (ω_n) for this type of boundary conditions (free at one end and provided with a rigid disk at the other end). For this reason, we traced the approximate curves (linear functions) of α_n according to β (Singiresu, 2007), for n varies from 1 to 5. The following polynomials were found:

$$\alpha_1 = 1.7\beta + 1.4,\ \alpha_2 = 1.7\beta + 3, \alpha_3 = 1.7\beta + 4.6,$$
$$\alpha_4 = 1.5\beta + 6.2,\ \alpha_5 = 1.7\beta + 7.7$$

(21)

The angular displacement can be written as follows:

$$\alpha(x,t) = \frac{-1}{I_0\omega^2} Me^{j\omega t} + \sum_{n=1}^{\infty} \frac{2}{l} \cos\left(\frac{\omega_n x}{c}\right)\left(\frac{MA_n}{I_0}\right)$$
$$\left(\frac{\omega_n}{\omega_n^2 - \omega^2} e^{j\omega t} - \frac{j\omega \sin(\omega_n t) + \omega_n \cos(\omega_n t)}{\omega_n^2 - \omega^2}\right)$$

(22)

DEVELOPMENT OF PRELIMINARY DESIGN LIBRARY

The models of shafts in torsional vibration have been implemented into usable objects in Modelica/Dymola by the system of equations presented in section 2. The two shafts have different boundary conditions. The first shaft is free-free and the second one is free at one side and provided with a rigid disk at the other side. Their icons developed in Modelica/Dymola are shown in Figures .3 and 4.

Fig.3. Model icon of the free-free shaft in Modelica/Dymola

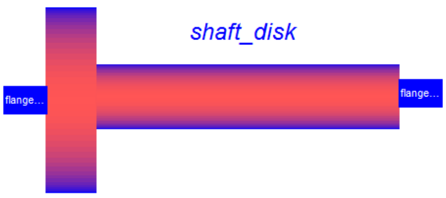

Fig.4. Model icon of the free shaft with a rigid disk in Modelica/Dymola

For these shafts, quantities such as the torque, the angular frequency and the angular displacement are required to describe the interaction between shafts submitted to torsional torque. For this reason, a new connector *flange1* has been developed in Modelica /Dymola. The connectors are represented by small blue rectangles in the shaft icons. The code of the connector *flange1* in Modelica is shown is:

```
connector flange1
 Modelica.SIunits.Torque T;
  Modelica.SIunits.Angle A;
 Modelica.SIunits.AngularFrequency omega;
end flange1;
```

MODELS VALIDATION

Validation of the free-free shaft model

The numerical validation of the free-free shaft is performed by comparing the dynamic response obtained by Modelica/Dymola (analytical model) with those obtained by the finite element method using ANSYS and those obtained by the model based on the Beam component which belongs to the Flexible Body library of Modelica/Dymola. The beam properties are shown in Table. 1.

Table 1. Geometric and material properties of the free-free shaft

Paramaters	Values
Shaft length	1 m
Shear modulus (G)	81000×10^6 Pa
Materiel density	7850 Kg/m^3
Poisson ratio (υ)	0.3
Young Modulus (E)	21000 MPA
Shaft radius	0.02 m

Table 2. Natural frequency (Hz) of the free-free shaft

Mode number (n)	Dymola (Analytic model)	Dymola (Flex Body)	ANSYS
1	1565.18	1565.96	1567.8
2	3130.35	3131.94	3135.6
3	4695.53	4697.91	4703
4	6260.7	6293.88	6272
5	7825	7829.80	7843

Table. 2 shows the first five natural frequencies obtained by Dymola (analytic model and beam based model) and ANSYS. It can be seen that the error between these models is less than 1%. Excellent results convergence is thus observed. A modal analysis was performed using ANSYS. The first three eigenmodes (torsional modes) are shown in Figures 5, 6 and 7.

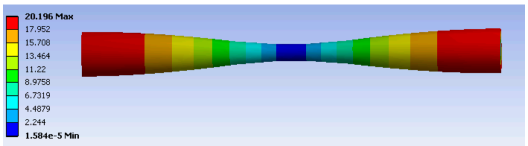

Fig. 5. The first eigenmode (F2 = 3135.6 Hz)

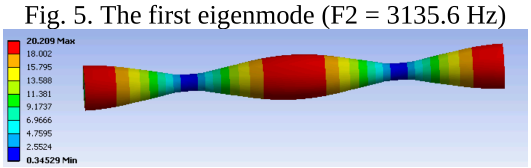

Fig. 6. The second eigenmode (F2 = 3135.6 Hz)

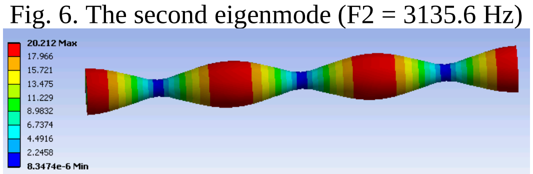

Fig. 7. The third eigenmode (F3 = 4703 Hz)

To validate the analytical model of the free-free shaft in torsional vibration, we have developed the setup shown in Fig. 8. We consider that the shaft is subjected to an external torque concentrated at its free end (x=0). $T = M.\cos(\omega t)$ with M= 20 N.m and $\omega = 10\ rad/\sec$.

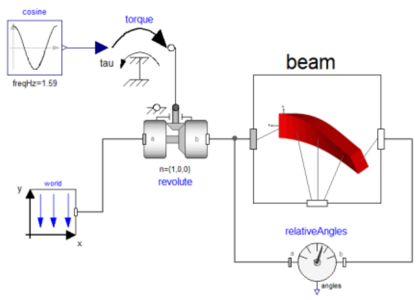

Fig.8. Configuration of the model validation the of the free-free shaft inModelica/Dymola

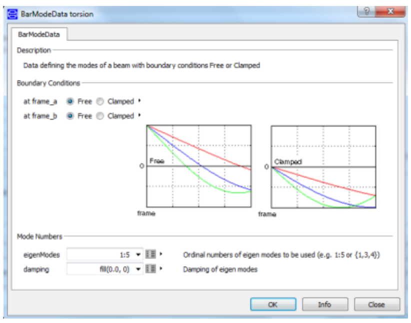

Fig. 9. Parameters configuration of free-free shaft in Modelica/Dymola

The setup shown in Fig.8 serve to find the angular displacement at the end of the beam (x = L) regardless the rigid body motion. In fact, the flexible beam element "*beam*" is connected at "*Frame a*" to a revolute and a periodic movement.

The input "*cos*" has a frequency of 1. 59 Hz and 20 N. m as amplitude. The vector "n" in the "revolute" object defines the beam direction movement in the test. A"*relative angles*" sensor is used for measuring the angle of twist at the end of the beam (*Frame b*) relative to the other end (*Frame a*). The "*Word*" component is used to define the gravity direction. In the graphical interface of the beam element, all the geometric and material properties are defined (diameter, length, Young's modulus, Poison ratio, etc.) as well as the boundary conditions (Figure.9) necessary to perform the simulation of the shaft. The torsion amplitude response has been compared between the two models. The first model is based on our analytical method developed in Modelica/Dymola, and the second one is based on the Beam component which belongs to the Flexible Bodies library of Modelica/Dymola (Fig.10). The result shows that the two curves have substantially the same amplitude of vibration and the same oscillation frequency as that of the excitation. In addition, the two curves have significant fluctuations due to the shafts flexibility. Good results convergences are then observed.

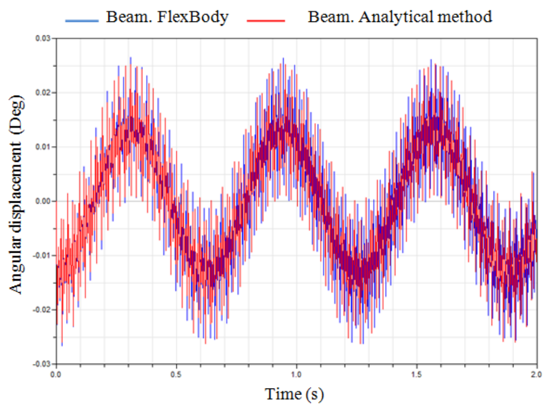

Fig.10. Relative angular displacement of the free- free shaft

Validation of the free shaft with a rigid disk model

To validate the model of the free shaft with a rigid disk, a model based on the *Beam* component belonging to *the flexible Bodies library* has been developed in Modelica/Dymola. The geometrical and material characteristics are presented in Table.3.

Table. 3 Geometric and material properties of the free shaft with a rigid disk

Paramaters	Values
Shaft lenght	1 m
Shear modulus	G= 81000×10^6 Pa
Materiel density	7850 Kg/m^3
Poisson ratio (υ)	0.3
Young Modulus (E)	21000 MPA
Shaft radius	0.02 m
Disk inertia (I_d)	0.005 Kg.m^2

The developed setup is shown in Figure.11. It is the same model used for the validation of the free-free shaft by adding to it a Body component representing inertia at the end of the beam.

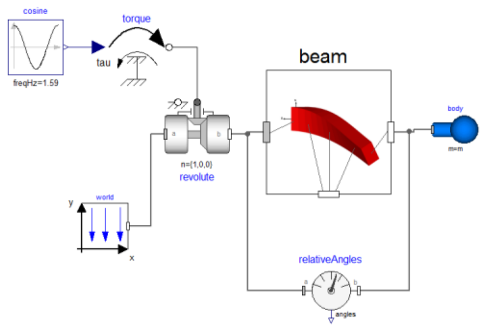

Fig.11. Model validation of the free shaft with a rigid disk in Modelica/Dymola

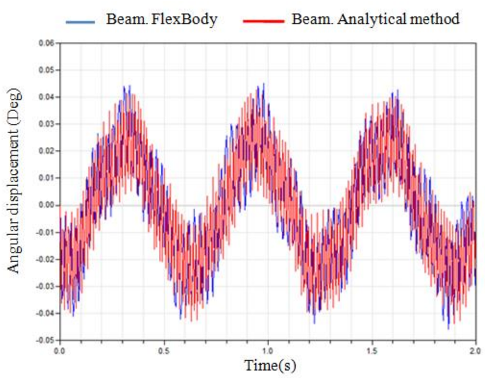

Fig. 12. Relative angular displacement of the free shaft with a rigid disk

Figure.12 shows a comparison between the angular displacements derived from the two models. The first model is based on an analytical method and the second one is based on the *Beam* component belonging to the *Flexible Body library* of Modelica/Dymola. It can be seen that the two curves have the same oscillation frequency which is the same as that of the excitation, and they have amplitudes of vibration very similar, except, there is a difference in the fluctuations which result from the shafts flexibility. The difference in fluctuations is due to the approximations made to find the natural frequencies.

APPLICATION TO THE WIND TURBINE DRIVE TRAIN SYSTEM

Case study presentation

The proposed pre-dimensioning methodology is applied to a typical example of a mechatronic system which is the wind turbine. In fact, all the flexible models developed in Modelica\Dymola are equipped with interfaces allow the exchange of data between them. This property allows investigating a complex mechatronic system such as a wind turbine, as precise as possible by simply using the best suitable model behavior. The drive train system is adopted to transform the slow speed of the rotor side to a fast speed of the generator side (Figure.13). This training consists mainly of the following components: a rotor which converts the wind energy into mechanical energy, a slow shaft of the rotor side, a gearbox, speed shaft of the generator side and a generator.

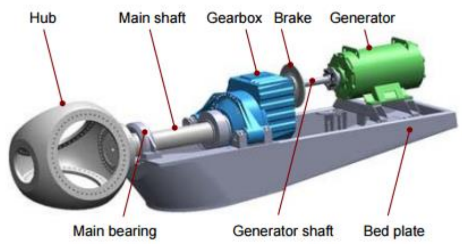

Fig.13. Drive train system

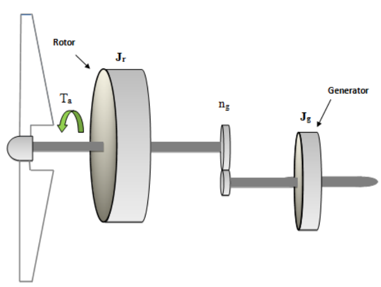

Fig.14. Wind turbine drive train system with two masses

The wind turbine drive system can be represented by a three masses model, two masses or a single mass model (Boukhezzar, 2006). The choice depends on the objectives and the complexity of the mechanical part. In order to predict the dynamic characteristics of the torsional vibration of the drive train system, a mechanical model consisting of two masses linked by flexible shafts has been adopted (Fig.14).
The developed model contains rigid and flexible components (Boukhezzar, 2006). The slow shaft and the fast shaft are considered flexible. The flexible modes of the blades are assumed to be very high to be neglected and all the flexible modes are located in the flexible element of the slow shaft (Boukhezzar, 2006). The proposed system is characterized by the sum of all the mechanical characteristics. The inertia of the rotor side is represented by Jr and the inertia of the generator side is represented by Jg. The damping is considered negligible.

The wind power available through a surface s of the blades is defined by:

$$P_v = \frac{1}{2}\rho.s.v^2 \qquad (23)$$

With, ρ is the air density and v is the wind speed
The mechanical power of the turbine is then:

$$P_{tur} = C_p \frac{\rho.s.v^3}{2} \qquad (24)$$

The power coefficient C_p is defined by the ratio of the power received by the turbine on the wind power:

$$C_p = \frac{P_{tur}}{P_v} \qquad (25)$$

The aerodynamic torque T_a is given by the following ratio: $T_a = \frac{P_{tur}}{\omega_t} = \frac{C_p}{\omega_t}.\frac{\rho.s.v^3}{2}$ (26)

With, ω_t is the rotor speed.
The speed multiplier is the connection between the turbine and the generator; it seeks to regulate the rotational speed of the generator. In this application, we consider that the multiplier is ideal and then, friction and energy losses are neglected.
For an ideal multiplier: $T_{ls} = n_g.T_{hc}$ (27)

With, n_g is the multiplier ratio of the transmission, n_g is the torque of the slow shaft and T_{hc} is the torque of the fast shaft.

Modelica modeling of the drive train system

The drive train model (two masses model) has been implemented in Modelica/Dymola as well as a control part (Fig.15). In fact, In order to adjust the output power, a PID controller is used. When the power reaches a threshold, the regulator sends a set point to adjust the input torque. The PID controller generates a maximum torque of 15000 N. m and the desired electric power is 5. 106 Watt

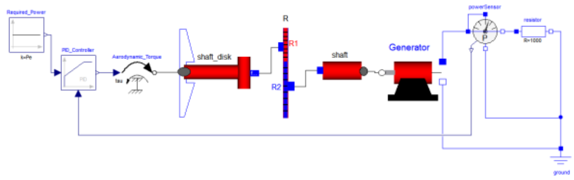

Fig.15. Modeling of the drive train system in Modelica /Dymola.

The setup of the drive train system consists of a set of connected objects. PID controller (proportional integral derivative), Aerodynamic torque, an electric current generator *Generator*, the main shaft and the inertia of the rotor side are represented by the element *shaft-disk*, the fast shaft *Shaft*, the inertia of the generator side is included in the component *Generator*, an ideal multiplier *R*, an electrical ground *ground*, a resistance *resistor* and a power sensor for measuring the voltage at the generator terminals. The initial conditions of the system are assumed null.
Modeling of the electrical generator:

The electrical model which defines the electrical current generator is given by the following equation:

$$V_g(t) = L\frac{di(t)}{dt} + R.i(t) + e(t) \qquad (28)$$

With, i (t) is the generated current (A), $V_g(t)$ is the terminal voltage of the generator (V), L is the inductance of the generator and R is the internal electrical resistance of the generator (Ohm).
Figure.16 show the graphical model of the generator developed in Modelica/Dymola. The generator is modeled by the connection of set of components from Modelica standard library: a resistance *Ra*, an inductance *La*, electromotive force *emf*, inertia of the generator side *Jm*, a positive terminal *p1*, a negative terminal *n1* and a connector *Flange_b*. Figure.17 illustrates the generator icon developed in Modelica/Dymola.

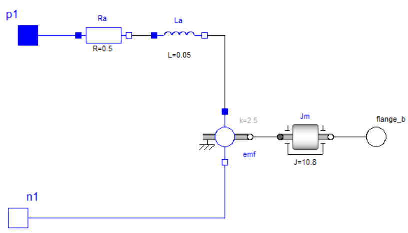

Fig.16. Equivalent circuit of the developed in Modelica\Dymola

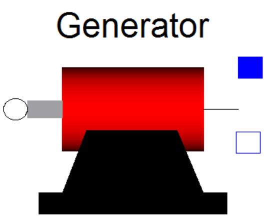

Fig.17. Generator icon developed in in Modelica\Dymola

Table. 4 Simulation parameters of the drive train model

Parameters	Values
Diameter of the slow shaft	0.172 m
Length of the slow shaft	2 m
Diameter of the fast shaft	0.0473 m
Length of the fast shaft	1 m
Inertia mass on the rotor side (J_r)	130,000 $Kg.m^2$
Inertia mass on the generator side (J_g)	10.8 $Kg.m^2$
Transmission ratio (n_g)	20
Shear modulus (G)	81.10^9 Pa
Internal electrical resistance of the generator (R)	0.05 Ohm
Inductance of the generator (L)	0.05 H
Transformation coefficient (K)	2.5 N. m /A
PID integration time constant t_i	0.5 s
PID derivation time constant t_d	0.5 s

Figs.18 and 19 show, respectively, the input torque and the output generated power. The results show that when the measured power is very low, a maximum torque is developed (the rotor turns with a very fast speed) until the power reaches the threshold limit (nominal value). A set point is sent to the PID controller and then the magnitude of the torque decreases and stabilizes at a lower value compared to the starting torque. So, the PID controller tends to provide and maintain a nominal value of the generated power.

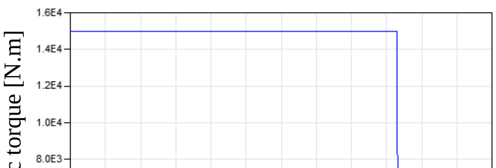

Fig.18. Aerodynamic torque [N.m]

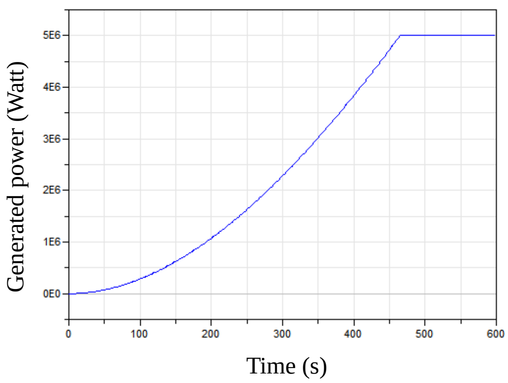

Fig. 19. Generated power (Watt)

Figure. 20 shows the angular displacement at the end of the slow shaft (at the disk level) with respect to the other end (x = 0) and Figure. 21 illustrates the twist at the end of the fast shaft (generator side). It can be observed that the angular displacement is important when the maximum torque is applied then the angular displacement decreases with the decrease of the input torque. In addition, we can notice that the fluctuations in the torque curve of the fast shaft are smaller than those in the slow shaft and this is due to the increasing of the rotation speed.

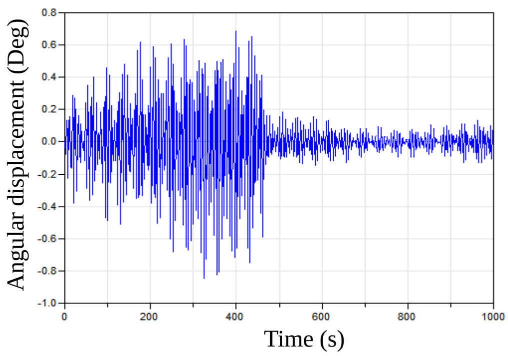

Fig.20. Relative angular displacement of the slow shaft

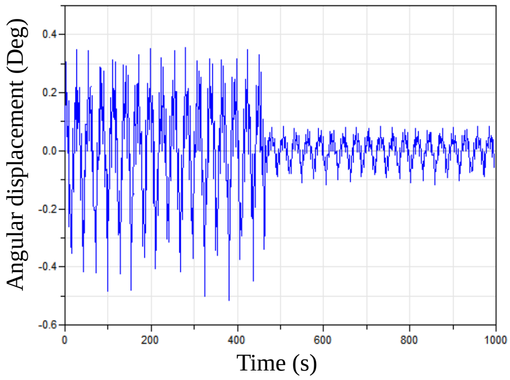

Fig.21. Angular displacement at the end of the fast shaft

Figure. 22 illustrates the variation of the relative twisting angle of the fast shaft (Angle of twist of the generator side with respect to the angle of twist of the multiplier side) as a function of the transmission ratio. It may be observed that the torsion of the fast shaft decreases with the increasing of the transmission ratio.

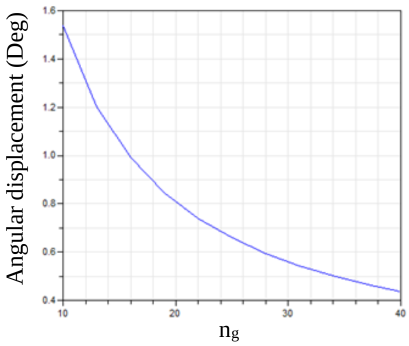

Fig.22. Relative torsional angle of the fast shaft according to the gear ratio

CONCLUSION

In this paper, an analytical pre-design approach has been developed, to study the vibrational behavior of a mechatronic system based on the object oriented modeling using Modelica. Compact flexible elements have been created in Modelica/Dymola for modeling a mechatronic system taking into account the vibrational aspect. The analytical models have been collected from a detailed exploration of the literature to extract and test the compact models used in pre-dimensioning. Based on these models, the vibration response of the drive train system is studied. The load applied to the drive system is considered harmonic but really the nature of wind is very unpredictable which means that the loads transferred to the drive train system are variables. In the pre-sizing phase, a model of low complexity is important to give an understanding of the basic physical effects and analyze the impact of different design parameters taking into account the material elasticity.

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