

## **ADVANCED DRIVE TRAIN MODELING IN A VIRTUAL WIND TURBINE USING THE MULTIBODY SIMULATION CODE SIMPACK**

Stefan Hauptmann<sup>1</sup>, Steve Mulski<sup>2</sup>, Martin Kühn<sup>1</sup>, Lutz Mauer<sup>2</sup>

<sup>1</sup> Endowed Chair of Wind Energy (SWE), University of Stuttgart  
Allmandring 5B, 70550 Stuttgart, Germany  
Phone: +49(0)711/685 60339, Fax: +49(0)711/685 68293  
E: hauptmann@ifb.uni-stuttgart.de

<sup>2</sup> INTEC GmbH  
Argelsrieder Feld 13, 82234 Wessling, Germany  
Phone: +49(0)8153/9288 41, Fax: +49(0)8153 9288 11  
E: steve.mulski@simpack.de

### **Summary**

In order not to overlook important loads or dynamic effects caused by drive train components when simulating wind turbines, a wind turbine environment is built up in the multibody simulation code SIMPACK. The simulation model consists of detailed aerodynamics, controllers and wind turbine components. The virtual gearbox includes gearbox shafts, bearings, and, at all stages, detailed gear wheels. These gearbox models are used within a complete virtual wind turbine for combined linear and non-linear analyses. The most important analysis methods are also described.

### **1. Introduction**

Presently, wind turbine simulation is achieved in two stages. In the first stage, the attributes of the overall model are studied with wind turbine simulation software codes that often use a modal approach and model the drive train with very few rotational modes only. In the second stage, components are analysed in detail with other software codes that neglect the entire wind turbine system and environment characteristics.

The newly developed entire wind turbine model has been tested with a model of a typical variable-speed pitch-regulated 1,5MW wind turbine at the University of Stuttgart. Intec GmbH has assisted different wind turbine and component manufacturers with detailed resonance analyses of different wind turbine drive train configurations. In the near future, the SIMPACK capabilities will be beyond the current status described in this paper.

### **2. Design Process**

In the early design stages only a very small amount of data is available, therefore only simple models, with low levels of detail, can be used to carry out initial concept analyses.

Because these simpler models only require short computation times, and produce easily manageable and understandable result data, a designer may quickly fence in the ideal design configuration by carrying out a multitude of parameter studies.

By subsequently adding levels of detail to the already existing models, designers can then home in on final configurations.

Similarly to choosing the appropriate level of model detail, the suitable analysis method should also be selected for each specific design stage. Broad-reaching analyses with simple low-detail models, e.g. with only a few torsional drive train modes, are used for the initial design stages. Once areas have been identified which require more focused simulation analyses, and the required model parameters are available, the models can be appropriately enhanced to include the necessary level of detail, e.g. taking into account tooth contact at the different gear stages. Fig. 1 shows how the level of model detail and focus of analyses can continually change throughout the design process.

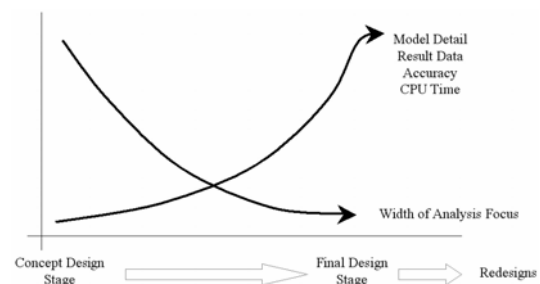


Fig. 1: Models and analysis methods throughout the design process

### **3. SIMPACK Virtual Wind Turbine**

The simulation code SIMPACK is enhanced for the consideration of wind turbine characteristics. The extensions include parameterised component models and modules for the generation of external loads in the SIMPACK virtual wind turbine.

This virtual wind turbine consists of models with an arbitrary number of degrees of freedom and modules that simulate external loadings in different level of detail. The degrees of freedom can be adjusted according to the user's requirements, allowing rough but fast as well as more detailed but time consuming simulations.

The SIMPACK extensions dealing with the virtual wind turbine are introduced in the following sections.

### 3.1 Aerodynamic Module

The aerodynamic loads resulting from turbulent wind fields have a dominant influence on the dynamic behaviour of wind turbines. The wind fields and the rotor aerodynamics can be taken into account for dynamical simulations in different ways. A rough but fast way to consider rotor aerodynamics is realised with the Blade-Element-Momentum method (BEM) [1] which is the preferred method at present. SIMPACK has been extended with such an aerodynamic module based on the blade-element-momentum theory. This module called AeroDyn [2] uses empirical corrections to calculate the rotor aerodynamics. These empirical corrections are needed to overcome the simplicity of the BEM theory which is based on the assumption of uniform induction on radial annuli, steady 2D aerodynamics and for example neglects the interdependence of the airflows at adjacent radial blade sections.

The empirical corrections account for the losses caused by the airflow around the blade tip and at the rotor hub. The effect of turbulent-wake state that occurs if the rotor strongly decelerates the axial airflow is also considered as well as unsteady airfoil aerodynamics and wake inertia or 3D-effects such as stall delay. An alternative approach to the BEM method also available in AeroDyn is called the Generalized-Dynamic-Wake theory (GDW) [2]. This theory is used to take the significance of dynamic inflow into account.

AeroDyn is capable of generating a wind field with deterministic wind shear, tower shadow and yawed flow and a stochastic representation of the temporal and spatial structure of 3D atmospheric turbulence by using a pre-processor [3].

The aerodynamic module AeroDyn is coupled to SIMPACK by an interface which controls the calculation of the aerodynamic forces in the AeroDyn module and the exchange of data between SIMPACK and AeroDyn. [4]

For the future it is planned to also have more detailed aerodynamic modules, e.g. based on the free vortex wake model, available in SIMPACK to create the possibility to include aerodynamic calculations in a level of detail that suits the requirements in the best way.

### 3.2 Control System

Another significant influence on the dynamic behaviour of the entire system originates from the wind turbine control system. A parameterised basic model of the control system is currently implemented in SIMPACK. This model of the control system consists of a variable-speed controller that gains the optimal power coefficient in partial load and a closed loop proportional plus integral plus derivative (PID) pitch controller that limits the rotational speed of the rotor above rated wind speed.

For the near future there are plans to create an interface to include model specific control algorithms in SIMPACK using a DLL approach.

### 3.3 Generator Model

Currently, the generator is modelled in the SIMPACK virtual wind turbine with a simple characteristic curve of an asynchronous generator. This generator model is sufficient for analyses that neglect generator dynamics but more detailed models are needed for in-depth analyses of components considering the overall dynamics. Since this will be the key capability of the SIMPACK virtual turbine, more detailed models are going to be implemented in SIMPACK.

### 3.4 Component Models

Furthermore, flexible wind turbine rotor blade and tower models are needed. Without restriction of the modelling capacities of SIMPACK users, a pre-processor is being developed to create rotor blade and tower models based on the parameters that are commonly used in the wind turbine industry. This pre-processor creates rotor blades and towers as beam models using an internal FE code.

The rotor blade models have a user-defined level of detail. The rotor blades can either be defined as stiff models that only consider the blade mass and inertia but can also include bending, torsional and even tensional stiffnesses. The number of considered mode shapes can also be specified by the users.

The component requiring the highest level of detail is the drive train with its subcomponents. The parameterised gearbox models implemented in SIMPACK for wind turbine simulations are presented in the next chapter.

### 4 Gearbox Model

The level of detail of the virtual gearbox model should be chosen according to the phenomena under investigation. SIMPACK enables gearbox models in a large range of diversity and complexity to be easily interchanged to suit the current analyses [5].

In the simplest form, a gearbox model may include only one or two natural modes of rotation. Detail can be enhanced by including separate bodies for the input

and output shafts and the gearbox housing which may either be rigidly or freely mounted with appropriate force elements depicting the mounting bushings. Multiple variations can be easily generated by changing the degrees of freedom of the housing and detail of the support arm mounts, for example by including the non-linear and frequency-dependent stiffness. A large range of detail for torque conversion also exists.

A library of kinematics converters range between simple direct torque to torque conversion and planetary gear layouts. Other elements may be used to include the effects of gear wheel meshing with backlash, stiffness and damping. For highly detailed simulation a gear wheel force element, developed initially for Formula1 engine applications, may be used.

An analytical force element built to the DIN 3990 standard is used to build up complex drive trains. Helical gears, both internal and external, can be modelled. Backlash, dynamic separation distance, multiple tooth contact and material properties are all taken into account. Automatic graphical representation of the geometrical gear parameters enables easy visualisation and inspection (Fig. 2).

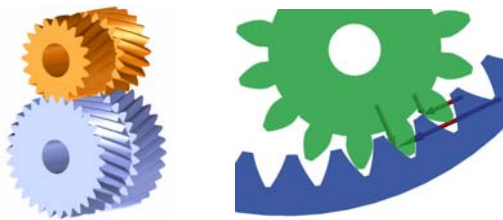


Fig. 2 Graphical representation of gear wheels

The non-linear jump phenomenon in the frequency response according to G. W. Blankenship and A. Kahman [6] is easily verified. Fig. 3 depicts the frequency curves of a gear pair between a constant run-up and run-down simulation.

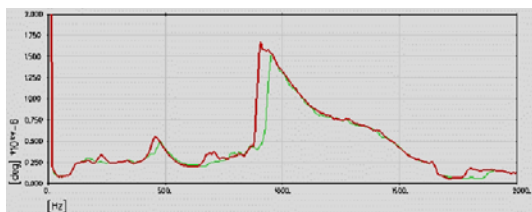
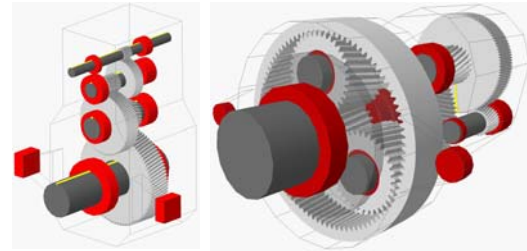


Fig. 3 Gear-pair frequency sweep between a constant run-up and run-down simulation

The detail of the gearbox may be further enhanced by incorporating the flexibility of selected bodies. Structures comprising straight beam elements, ideal for the shafts, are created directly in SIMPACK using an internal FE code. More complex flexible structures, such as the gearbox housing, may be imported from FEA programs.

Completely parameterised models of detailed gearboxes have been built up in SIMPACK. The separation of the parameters from the model enables easy access and modification of the data by inexperienced SIMPACK users. Pre-defined batch jobs of analyses can then also be carried out with virtually no SIMPACK experience. The individual parameters within the files are clearly documented. The origin of the data is also documented and can be easily updated, which facilitates project work. Variations of substructures, e.g. neglected axial and radial shaft degrees of freedom, are easily generated.



Jahnel-Kestermann (used by Komai) SIMPACK Database

Fig. 4 Models of detailed gearboxes. Bearings and bushings highlighted in red.

Before a model is imported into a complete virtual wind turbine, model verification is carried out using virtual test rigs. Linear and non-linear analyses are used to inspect and validate input data. Resonance analyses, described in the next chapter, can also be used.

## 5 Analysis Methods

During the initial phases of design, when many variations are under consideration and only preliminary data is available, the focus of analysis methods is primarily on giving designers an overview of system behavior for a wide range of operating conditions, as apposed to concentrating on accuracy.

Methods are required which can quickly highlight any possible design flaws, which designers want to avoid or further investigate. As the design process progresses and the model data begins to coalesce, more focused methods are required which also give a higher priority to accuracy.

### 5.1 Resonance Analysis

For a quick overview of possible excitations of undesirable vibrations, resonance analyses are carried out on the entire virtual wind turbine (Fig. 5). For the first step eigenfrequencies at various rotational speeds of the 'settled' system are plotted in a Campbell diagram to show the possible resonance frequencies of eigenmodes excited by the rotating shafts and meshing gears and higher orders thereof.

Before the eigenfrequencies can be calculated for a defined rotational speed the model must be brought into a state of quasi-static equilibrium. All transient

states such as the gear wheel chatter must first be eliminated.

Initial rotational speeds of the shafts are easily set by making all initial rotational speeds dependent upon one shaft-speed input, by means of parameterisation. A simple non-linear input function is used to model the reaction torque of the generator which is dependent upon the rotor's rotational speed. Knowing the reaction torque of the generator and the rotational speed of the main shaft enables the input torque on the system, exerted on the hub, to be automatically calculated and applied by means of a function expression.

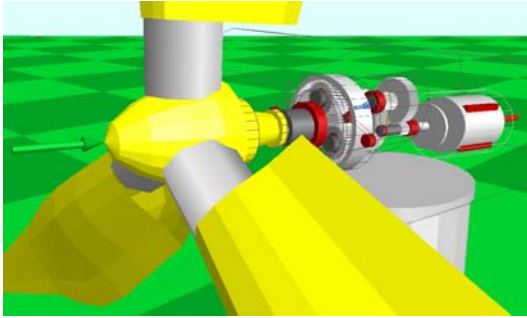


Fig. 5 Complete virtual wind turbine

For a given rotational speed a non-linear time integration is carried out until all transients have dissipated. The end states of the time integration process are copied to the linearisation state at which point an eigenfrequency analysis is performed. All frequencies are calculated for the specified rotation speed and the process is repeated for a different speed. This process can be completely automated using batch commands and scripting. For initial design investigations of drive train dynamics the coupling of the rotor blades with the aerodynamic model is normally not considered.

Generally, the rotor blades and main shafts all include flexibility for resonance analyses. Addition shafts and other components (e.g. tower, base plate and gearbox housing) may also be modelled as flexible if deemed necessary.

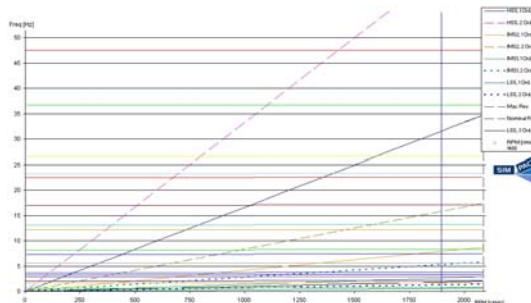


Fig. 6 Campbell Diagram

An indication of the magnitude of the resonance can be acquired by carrying out order analyses. A run-up is simulated in the time domain. The turbine is started at still stand and then driven through the complete

range of velocities by increasing the torque applied at the hub. The order analysis can then be plotted for any output signal (e.g. force, acceleration, etc). All orders may be plotted together forming a Campbell diagram (Fig. 6). Other excitations, in addition to gear meshing, can be applied to the system (e.g. unbalanced shafts, square wave force signals, etc.) in order to excite various vibration modes.

## 5.2 Linear System Response

Having already linearised the system at a number of discrete quasi-static rotational speeds the designer can put this data to further use by applying linear system response to obtain an appreciation for the magnitude of the resonance. Multiple inputs can be used in order to excite the various components. To incorporate the loading caused by unbalanced shafts, excitations can be made dependent upon the circular frequency. The multiple-input single-output transfer function (MiSo-functions) is:

$$\hat{y}_i = \mathbf{F}_i(j\omega) \hat{\mathbf{u}} \quad \text{where} \quad \mathbf{u}(t) = \hat{\mathbf{u}} e^{j\omega t} \quad 1$$

$$y_i = \hat{y}_i e^{j\omega t}$$

With the input function being:

$$\hat{\mathbf{u}} = \sum_k \hat{\mathbf{u}}_k \quad 2$$

$$\hat{\mathbf{u}}_k = A_k e^{j\varphi} = A_k (\cos(\varphi) + j \sin(\varphi)) = \text{Re}_k + j \text{Im}_k$$

Stochastic excitations may also be imparted to the system using Power Spectral Density (PSD) and polynomial input functions.

Although this linear method can achieve results almost instantaneously, all results are dependent upon the chosen linearisation point and excitations. Non-linearities arising from kinematics and force elements are obviously not included. However, if this method is used wisely, it is an extremely powerful analysis tool since numerous variations can be computed extremely quickly, Fig. 7.

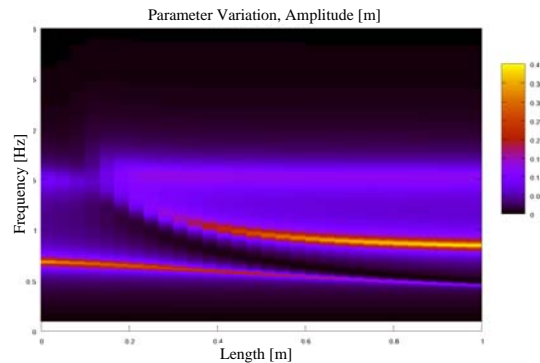


Fig. 7: Parameter variation of a vibration absorber using linear systems response

## 5.3 Run-Up Fast Fourier Transformation Analysis

In order to include non-linear effects, the simulations must be solved in the time-domain. Unfortunately,

simulation times increase substantially but more realistic operating conditions can be achieved and less effort has to be given towards approximating reality. Excitations caused by an unbalanced shaft, for example, are simply included by employing off-centered masses. All non-linearities within the model are now considered.

Starting from a stand still, a drive train can be run-up through a wide range of operating speeds. If the run-up is gradual enough the complete data can be divided into discreet segments along the time line. Performing an FFT on each segment and then combining all the FFT together results in generating a waterfall plot that can be seen in Fig. 8.

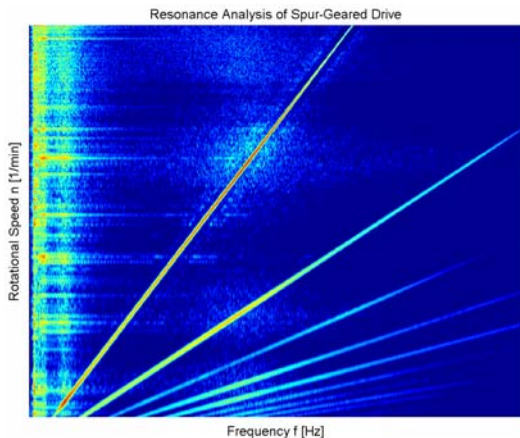


Fig. 8: Run-up Fast Fourier Transformation (FFT) analysis

In addition to the areas of resonance, as with the Campbell diagram, an indication of amplification magnitude is given.

#### 5.4 Order Analysis

Another method commonly used in analyzing rotating machinery, for example drive trains, is order analysis. Unbalanced shafts and gear teeth meshing are the major source of internal excitation in rotating machinery.

These excitations increase proportionally with the rotating speed and can be seen clearly in Fig. 8. Order analysis enables designers to analyze data along these diagonals. Similar to a single FFT analysis, amplitude is plotted as a function of frequency with order analysis, but rather than plotting at a fixed rotational speed, the amplitude is plotted for a fixed order (in Fig. 9), or in other words, along one diagonal line.

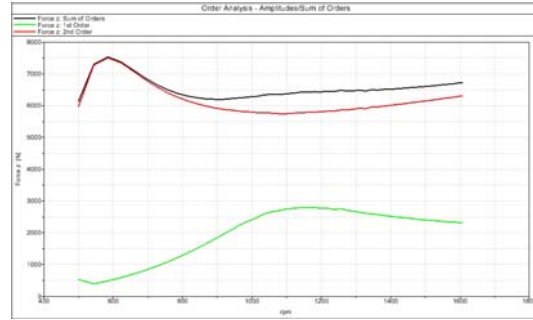


Fig. 9: Order analysis showing 1st and 2nd orders and total

Because each order occupies a range of rotational speeds a run-up simulation is required for an order analysis. The velocity gradient of the run-up can be much steeper than that of an FFT analysis, and therefore result in shorter CPU times. Because the results of the order analysis are dependent upon the known rotational speeds, relatively few rotations are required in order to assure accurate analysis. The mechanism, or order, which is primarily responsible for a particular excitation peak, can often be easily identified by simply plotting all relevant orders in one diagram along with the sum of all orders (Fig. 9).

Once a particular order is identified as being the primary cause for vibrations at a certain frequency (Fig. 9), the support mount or bushing through which the force is being transferred can be identified. Not only can the particular element be identified but also which component is primarily responsible. For each force component a diagram showing the normalized amplitudes of all coupling elements is generated, Fig. 10.

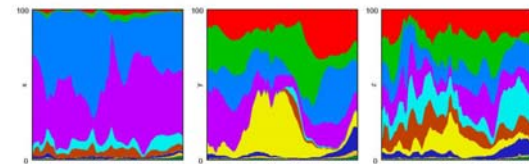


Fig. 10: Normalized amplitudes of coupling elements

Plotting all orders in one diagram can also produce a waterfall plot similar to the run-up FFT in Fig. 8. The frequency resolution can be made finer by lowering the velocity gradient and increasing simulation times, but the result data will still appear to radiate from the origin.

#### 5.5 Durability Analysis

With the design process nearing completion and components entering into the expensive test phase, full fatigue analyses using data from simulations can vastly reduce time and costs. The models built up through the design process can now be used in parallel with real fatigue testing. After a run-up analysis in the time domain all coupling loads and modal model co-ordinates i.e. bending of each mode of the flexible body to be investigated are exported to a durability



program, e.g. FEMFAT. These signals are then assigned to the associated stresses from the normal and inertia relief modes generated by a FE software. The stress contributions are superimposed and durability predictions can be made. Fig. 11 shows the usage of durability analysis in the automotive industry.

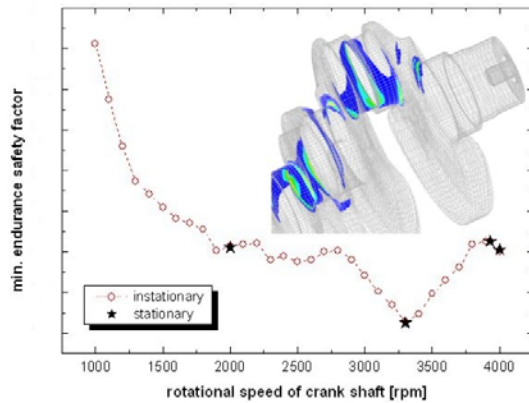


Fig. 11: Durability analysis of crankshaft

## 6. Conclusion

To overcome the shortcomings of the actual simulation approach, Intec GmbH together with the Endowed Chair of Wind Energy at the University of Stuttgart is currently developing an add-on module for the non-linear multibody system (MBS) modelling software SIMPACK. This module extends the capability of SIMPACK for the simulation of wind turbines.

Additional functionalities have been added to SIMPACK to create a wind turbine environment that allows detailed drive train analyses while taking the dynamics of the entire wind turbine into account. This virtual wind turbine considers the most important characteristics of a wind turbine.

Component analyses considering wind turbine aero-elasticity and detailed resonance analysis for all rotational speeds (Campbell diagrams) with adapted degrees of freedom can be performed. Throughout the design process a constant balance must be reached between the level of model detail, type of simulation methods, and the particular phenomena under investigation. Too much detail leads to time-consuming data collection, long CPU times and difficulties in interpreting the results. On the other hand an insufficient amount of modelling detail will fail to show the desired phenomena. A virtual wind turbine model has been created in the multibody simulation code SIMPACK. Among others, parametric gearbox models, rotor blade and tower models as well as rotor aerodynamics and turbulent wind field input are offered. Further capabilities are under development.

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