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Progressive Dynamical Drive Train Modeling as Part of NREL Gearbox Reliability Collaborative

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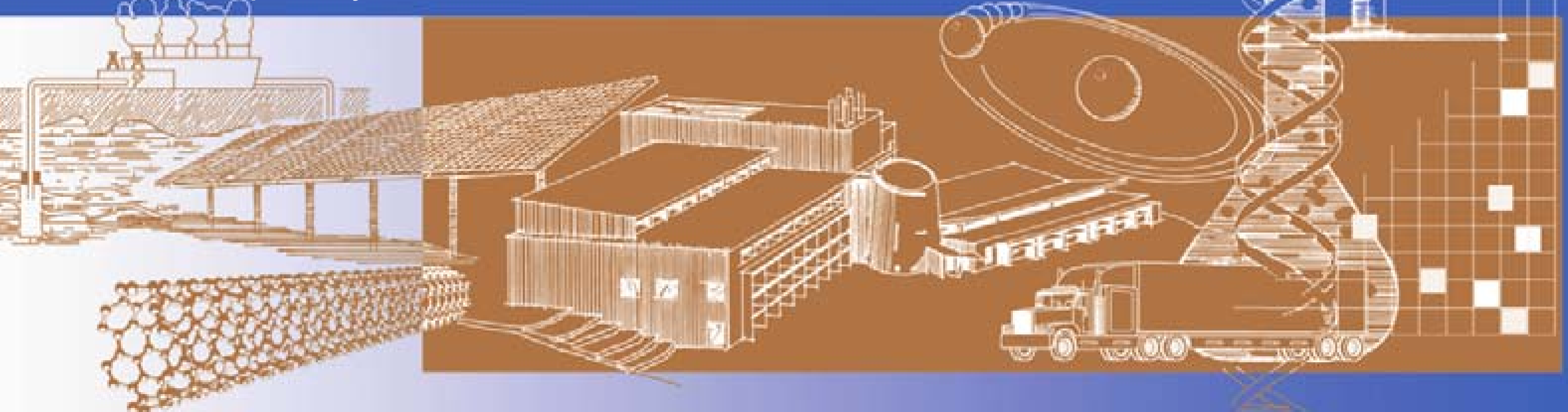

Abstract

The Gearbox Reliability Collaborative (GRC) is an effort by the National Renewable Energy Laboratory that seeks to develop a numerical model representative of the current standards in the industry. The intent of the model is that it can be extrapolated to a large number of turbines with different sizes and dimensions but with the same configuration. The models are also correlated with experimental data collected from a carefully selected, preexisting machine with a significant operating history. This approach ensures that the information revealed by the testing and analysis will be valuable and relevant to the current industry.

The GRC analysis also seeks to integrate into the drive train design process several numerical models that capture the dynamical nature of the drive train. This document briefly describes these numerical models, which are both progressively complex, and aim to reveal new insight into the internal forces inherent to the dynamical behavior of the drive train. Additionally, the progressive nature of these models allows for their validation through a comparison of models of less complexity to models of higher complexity, thus eliminating error in the model development. These models additionally can allow for the filtration of sensitive information between various parties of the design process; therefore, ultimately increasing transparency.

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Introduction

The National Renewable Energy Laboratory's (NREL) National Wind Technology Center (NWTC) has embarked on the difficult task of revealing the causes and loading conditions that result in the premature failure of wind turbine gearboxes.

The NWTC approached the problem by forming a Gearbox Reliability Collaborative (GRC) of the different parties involved in the gearbox design process. The goal of the GRC is to improve the performance and increase the lifetime of gearboxes.

In addition to facilitating cooperation, the GRC seeks to achieve its goal by exploring three avenues of research. These include drive train numerical analysis and modeling, full-scale dynamometer testing, and field testing. These avenues of research will be correlated and iterated to obtain the closest representation of actual load behavior in a controlled environment.

This document gives a brief overview of the GRC analysis effort and briefly describes the models without giving details on the topology. In addition, it describes one of the numerical approaches pursued by the GRC. However, only data interpretations and subsequent conclusions are presented here, actual experimental data and results are left out. The intent is to simply provide a synopsis of the aims of the GRC analysis along with a description of its general approach.

The numerical analysis follows a progressive approach, developing models that gradually increase in complexity. The models range from basic, two degrees of freedom (DOF) dynamical models, up to models that are capable of capturing gear tooth interaction and bearing compliances. Additionally, a complete finite element model of the gearbox is developed to obtain better insight into housing deflections and tooth load distribution. The models are supplied relevant loading conditions generated from aeroelastic simulations. Each step of the numerical analysis will be validated with data obtained from the dynamometer and field tests, resulting in a comprehensive numerical simulation tool. The numerical approach in collaboration with the GRC experimental effort should provide great insight into the flaws embedded in the gearbox design process.

Analysis

The following section describes the progressive dynamical models utilized by the GRC.

The analysis preformed seeks to capture the dynamical behaviour of the drive train, and more specifically, the dynamic interaction of the internal components of the gearbox. Multibody system simulation is a technique that is capable of predicting the dynamics of the system and is computationally light; thus, it was chosen for the analysis.

The models presented in this document were generated with the commercial code named Simpack. Simpack is a multipurpose, multibody simulation code operated by means of a graphical user interface. It allows the user to input parameters describing the simulated system such as mass and inertias of each body. The user can then create kinematic loops by applying the pertinent joints, constraints, and forces. The software creates the equations of motion internally and allows the user to choose different options for the time integration. The user is also capable of inputting three dimensional primitives, which describe the geometries of the interacting bodies to finally obtain visual animations of the interacting system. Finally, Simpack has a force element library that contains a number of specific force elements tailored for different areas of the industry. Among these are specific elements designed for the drive train simulation that have proved to be valuable in the GRC analysis.

Model Description

Stage I: Two-mass Oscillator

In the first stage of the modeling, the entire drive train was modeled in a very simple form. The system is composed of two rigid bodies: the first represents the rotor of the turbine and the second represents the generator. These two bodies and their respective torsional inertias are connected to each other with a torsional spring damper joint. With respect to the generator inertia, the influence of the gear ratio of the drive train is taken into consideration by calculating an effective inertia proportional to the square of the gear ratio [4].

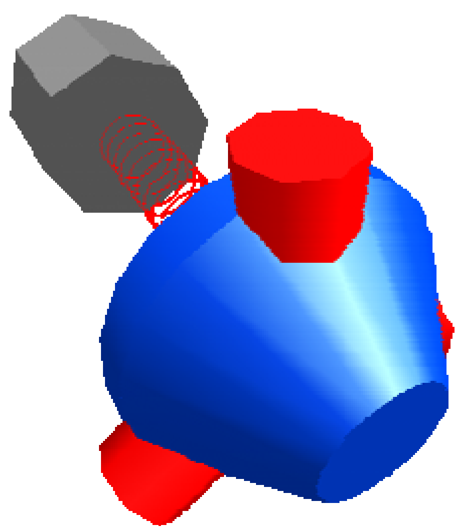

Figure 1: Stage I graphical representation

The bodies are connected to the reference frame with only one degree of freedom (rotation), giving the overall system only one DOF per body. This connection has unknown properties such as stiffness and damping coefficients, and is assumed to be massless. These parameters are determined by the use of experimental data.

Although this is the simplest model that was built, it is of greatest importance because it serves as the basis for the input and validation of the subsequent stages. In addition, this simplified drive train model is used by many aeroelastic codes that are used to generate the loading conditions of the drive train. This configuration serves as further validation for the aeroelastic models that have already been validated.

Stage II: Torsional Multibody

This stage implements the same approach for the rotor and the generator as in stage one. However, every major component of the gearbox such as gears and shafts were modeled. The system has one DOF for each rotating body.

This approach accounts for the torsional compliances resulting from the bending and contact deflection of the gear teeth, as well as torsional deflections of the shafts. The model ignores the added torsional compliance from bending of shafts and from bearing deflection [1].

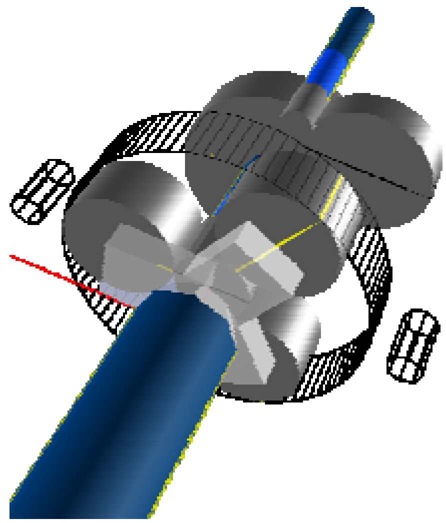

Figure 2: Stage II graphical representation

Spring dampers joining both gears were used to simulate gear interaction. These joints lie over the line of action and will be subjected to the tangential forces of the gears [6]. The overall torsional response of the system was obtained, as well as the response from the internal components in a dynamical and coupled manner. The force element FE-14 from Simpack was used as the previously mentioned tangential spring interaction [2].

The gear bodies were represented by rigid primitives used in the 3D visualization and do not have any influence in the calculation outcome. The shafts were simulated by torsional spring dampers or FE_12 in Simpack, giving the insight of the torsional shaft deflection as a separate parameter. All the respective torsional inertias from each individual component have to be calculated from the masses and geometries as inputs for the model. It is assumed that there is no change in direction, and the connecting springs between gears will always be under tension (no backlash).

Stage III: Gear Element Implementation

This stage recreated the physical representation of stage two, excluding the rigid bodies for the gears. The primitives were created from standard gear wheel parameters. These were utilized to create contact elements that are capable of reproducing multi-tooth contact, as well as backlash and changes in direction of the gear rotation. The shafts are represented as torsional spring dampers as in the previous stage. Additionally, the connection between the gears and the housing only has one DOF.

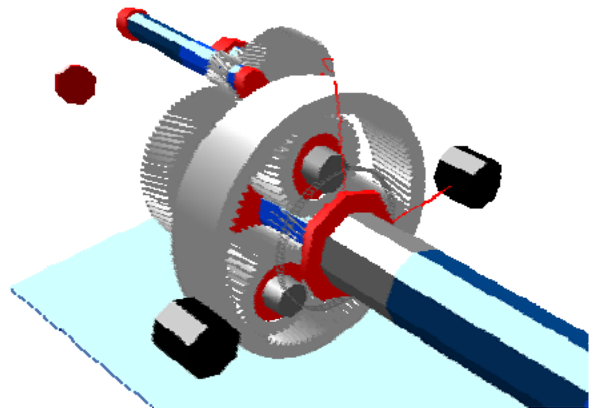

Figure 3: Stage III graphical representation

State 3 uses a built-in Simpack force element. Its only requirement is that a body fixed marker in the center of each gear be defined. This marker is required to establish the relations between the two interacting gears. [3]

To account for the force interaction of all components, the ring gear has to be taken into consideration. The ring gear was fixed to the gearbox housing with zero degrees of freedom. The force interactions of the system included planet to ring, as well as planet to sun. In this manner, the two forces and their direction ns were taken into consideration for the analysis. The direction and magnitude of the forces are of great importance, especially for the planets because the force on the tooth changes direction with each cycle, which increases the fatigue damage. The Simpack force element FE-225 is capable of accounting for changes in the gear center distance and backlash [3]. For this model, as the joints of the system have only one degree of freedom, the center distances will not change; nevertheless, backlash is taken into consideration.

Stage IV: Bearing Compliance Implementation

This stage adds three DOF to the joints between the rotating bodies and the housing of the gearbox. These extra DOF were constrained by the use of the Simpack force element FE_43. This force element has the capability of representing six independent stiffness and damping coefficients (three translations and three rotations). This force element can implement linear or nonlinear stiffness, which is of great use due to the nonlinear stiffness and clearance that characterize bearing behavior. Although Simpack FE_43 is capable of constraining six DOF, only three stiffness and damping coefficients are implemented in this stage (translation x,y,z).

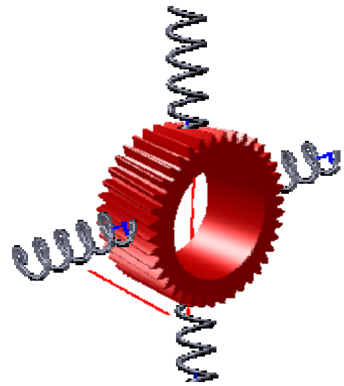

Figure 4: Bearing stiffness representation

The new addition of the force element required the addition of new markers that describe the attachment points of the bearings. This is of great relevance because the bearings are represented with three stiffnesses, and each shaft needs a minimum of two constraining forces. This required a more precise description of the housing and the exact positions of the bearings. Additionally, the length of the shafts and the position of the bearing on the shaft were also of great importance because they needed to match the housing to prevent the undesired prestress conditions. This was not required on the previous models, as they were purely torsional models with rigid bodies.

Constraints were also added to the connection between the input shaft and the gearbox. Originally, only rotation was constrained, now all six DOF are constrained. This simulates the rigid coupling between the shaft and the gearbox, and is of great importance as it reveals how nontorsional loads affect the internal components of the system.

Stage V: High Fidelity Semistatic Model

This model is intended to reveal information that the dynamic models would not be able to reveal. It uses the finite element approach to reveal deflections and load distributions in a more detailed manner than the previous models. With the implementation of high fidelity finite elements, gear tooth load distribution and contact stresses can be visualized and quantified. In addition, the loading distribution of the bearing internal components, such as roller load distribution, as well as contact stresses in individual rollers can be revealed. This high fidelity modeling reveals the influence of housing deflection and misalignment, as well as planet deformation in the load distribution and stresses.

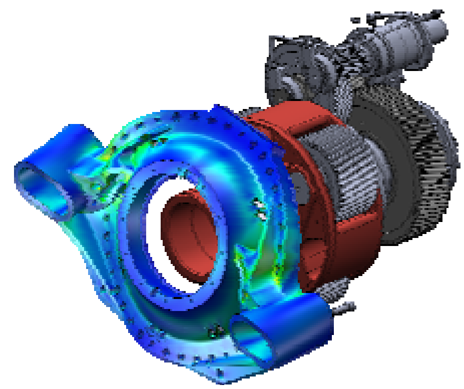

Figure 5: High fidelity semistatic model

The main downside to the high fidelity model is that it is computationally expensive, thus making the dynamical approach inconvenient. As a result, the semistatic approach is followed where only very short time series are simulated. Time series more relevant to the study were chosen by performing longer simulations on the previously mentioned dynamic models.

General Gearbox Design Process

The initial stage of the gearbox design process defines the basic requirements that the gearbox will have to fulfil. Although the main requirement is the loads document, the outcome of this initial stage will be the definition of the general configuration of the gearbox. This configuration will include: number of stages, epicylic or helical, gear ratio, and general structure.

From this original configuration, the gear design is carried through. This includes: calculation of number of gear teeth, module, and the center distances. This is followed by rating calculations that include the allowable surface and bending stresses and safety factors. Shaft dimensions are also defined in this stage, which allows for the initial bearing selection that is usually obtained from a sizing catalogue. This is followed by the cooperation of the bearing manufacturers that use in-house rating capabilities to corroborate the initial selection.

At this point, a life analysis is performed on the previously selected gears. These fatigue calculations are typically based on the Miner Palmgren method, which calculates the cumulative fatigue damage caused by the variable load spectrum. Common practice maintains constant ratios between pitch diameter and face width. If the desired life is not met, dimensions are increased proportionally to meet the desired life. This is closely followed by the introduction of micro geometry to reduce local loading peaks caused by elastic deformation of the bearings, shafts, and gear bodies.

Following this, the interfaces have to be satisfied. These typically include connections to the main shaft and the high speed shaft, as well as design and interaction of the torque arms with the bed plate. Several iterations may be required to satisfy the design objectives.

Aspects of the manufacturing process must also be discussed as part of the iterative process. This includes the final refinements of the design, material characteristics and heat treating, and the outsourcing requirements for casting and forging. Finally, details such as oil cooling, sensors and data interfaces, corrosion protection, and noise are taken into consideration.

The manufacture of two prototype gearboxes is the final stage of the design process. The prototypes are used for testing under specified loading conditions using the in-house dynamometer. The tested elements are dissected, and assessment of the wear is performed to fine-tune any possible changes needed to the design before it is released for limited series "0" production and field-testing in operating turbines at various test sites. On completion of field-testing, any modifications required by field-test results are implemented. This is followed by the release of the design for large-scale production and sale.

The Design Process in Today's Industry

The general design process can be executed in a number of ways; however, the most common are the vertically integrated design or the non-vertically integrated design processes. The main difference between these drive train design processes is the nature of the iterative process. The vertical design process not only iterates its individual subsystems, such as gearbox and bearing designs, but also integrates the rest of the components of the design into the overall iteration, creating a more complete process. This is achieved by a continuous update of the loading definitions. Because all the different subsystems of the design depend on the loading conditions, the process is valid throughout. This type of integration can only be performed when the wind turbine manufacturer internally manufactures the gearbox or when the manufacturer purchases the gearbox designs. This can cause a dependence on a provider or the need for the wind turbine manufacturer to fabricate their gearboxes in-house. Consequently, the most popular approach for the drive train design is the decoupled (non-vertical) design process.

The non-vertically integrated design process does not include a great deal of iteration, other than the iteration of the independent subsystems. Hence, the overall calculations and ratings are based on the initial assumptions for masses and stiffness, which define the load document. The reason for this is not the definition or layout of the process as much as the interaction between the wind turbine manufacturers and the gearbox vendors involved in the process.

To update the load document and integrate the iterative process into this design scheme, a large amount of information must be shared so that the wind turbine designer can create a comprehensive dynamic model. Because this design process involves a number of competing gearbox vendors, very little information is willingly shared. On the other hand, the providers are most likely also working for other wind turbine manufacturers, thus, information from the particular wind turbine design is also not shared. This informational barrier is the disengaging factor of the feedback process.

Multistage MBS and the Design Process

A detailed model of the gearbox components is needed to truly capture the dynamical behaviour of the drive train and properly predict the loading conditions of the system. The multistage approach to drive train design presented in this paper allows for the reinstatement of the iterative nature of the design process back into the non-vertical design process. The model's progressive nature can be used to sanitize the data shared among vendors and wind turbine manufactures. With this process, the level of useful modelling data shared could be increased, thus improving the level of accuracy of the overall model without increasing the level of sensitive and proprietary data shared.

Commonly, the simple model used by the aeroelastic simulation codes resembles the model in stage one. Hence, the entire drive train was represented by a single DOF. The advantage of this model is the simplicity of its input parameters, and the minimal amount of data given to the wind turbine manufacturer.

Stage two, which already represents all moving components, provides a more accurate behavioural response of the system, which is required to calculate the loading conditions. Although this stage requires more gearbox geometry detail, the information given (general masses and gear ratios) is not of much value to competitors. The stiffness coefficient required would be a trivial task for the gearbox manufacturer as the manufacturer generally performs a complete finite element analyses of the system.

Stage three requires a greater level of trust between parties. It requires the overall configuration of the gearbox, as well as the number of teeth and module. If the tooth geometry is sensitive information, the standard geometry ratios described by the American Gear Manufacturers Association (AGMA) standard could be used. Regardless of the tooth geometry, the tooth micro-geometry, which is a current field of research for noise reduction and efficiency, will not be shared.

Stage four involves the aforementioned parameters for stage three with the addition of the required bearing stiffnesses. Although the bearing stiffnesses do not have a large influence on the torsional behaviour of the system, they are quite important for vibration analysis. In this case, generalized stiffnesses from the manufactures are required. The stiffness response can be provided without revealing any of the geometrical characteristics of the bearing. The fact that the analysis methods used by the bearing manufacturer to calculate this stiffness would not be revealed can also protect the data integrity. In addition, the study does not seek to find the sensitivity of a particular bearing stiffness; a general stiffness for all bearings could be used, therefore, reducing the amount of data shared.

The advantage of the progressive model approach is not only the enabling of increasing complexity among models but also enabling the possibility of a complexity reduction. By implementing complex stiffness parameters for a simpler model, the response of a more complicated model can be mimicked. For example, the stage three model can be reduced to the stage two model by the implementation of more complicated stiffness responses. Nonlinearities can be implemented to the stiffness and damping coefficients, and prescribed excitation can account for tooth interaction vibrations. This creates a model with a closer response to a more complicated model, even though all the sensitive parameters are hidden under the nonlinear stiffnesses. It is evident that the response of a system with less DOF would not be as accurate and will not mimic all the desired parameters. Nevertheless, with this approach, models closer to reality can be created without the compromises generally involved, and consequently, the loading parameters can be redefined to a more accurate level.

The detailed multibody system simulation (MBS) truly captures the dynamical behaviour of all moving components in a minute manner. Unfortunately, the approach does not predict the minute mechanical and structural behaviour. This necessitates the cooperation of different disciplines. For example, the creation of load cases for particular components of the drive train can be created from the MBS simulation. These load cases can be used in the finite element model, which can reveal in greater detail the mechanical behaviour of the internal components such as tooth load distribution and tooth bending characteristics. The load conditions cover a large spectrum of loading and load variations, which are narrowed to the load conditions of most relevance.

Conclusions and Final Remarks

Well known standards, such as the AGMA standard 6006 and Germanischer Lloyd standards, discuss the importance of implementing multibody system dynamics into the design process of wind turbine drive trains. Nevertheless, these guidelines do not specify the level of detail required for the fully-coupled model. They also fail to specify how these models can be integrated into the design process.

The premise of this work is to create a baseline model for a wind turbine drive train. These models were developed with increasing complexity and correlated in a progressive manner. The models were also validated to experimental data collected from a representative wind turbine from the field. The progressive models created represent the gearbox and drive train industry standards for turbines ranging from 750 kW to 5 MW.

Different branches of the design process for wind turbine drive trains were also discussed, including the fully integrated design and the coupled design. This document briefly discusses the challenges in the data sharing between the wind turbine designers and the gear box providers with respect to the design process. These barriers to information accessibility produced by the fierce competition and inherent proprietary data prevent the iterative nature of the design process.

This document suggests the use of the progressive multibody system dynamic approach as a data-sharing sanitation tool. Thus, with a minimal amount of sensitive data, comprehensive and detailed models can be created to enhance the load prediction methods of the design process and return iterative characteristics to the non-vertical design process.

Current and Future Work

In addition to the modeling presented, the GRC is sponsoring an analysis Round-Robin. This exercise will bring together many participants from the manufacturing industry and a diversity of simulation tools to establish which approach is best suited for the different areas of the design process. It also intends to broaden the range of analytical tools, as well as the number of models evaluated by the GRC, thus making the GRC analytical effort more robust and comprehensive.

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