24th INTERNATIONAL CONGRESS ON SOUND AND VIBRATION
23–27 July 2017, London

SYSTEM-LEVEL MULTI-BODY SIMULATIONS OF A WIND TURBINE GEARBOX

Shadi Shweiki[1,2], Domenico Mundo[1], Antonio Palermo[1], Francesco Gagliardi[1]

[1] *Deptartment of Mechanical, Energy and Management Eng., Università della Calabria, Rende, Italy*

[2] *Siemens Industry Software, SISW NV, Interleuvenlaan 68, B-3001 Leuven, Belgium*

email: domenico.mundo@unical.it

Modern industry is largely using computational models to support the design of mechanical systems and to reduce duration and cost of the development process. Multi-Body (MB) simulations allow to simulate large and complex models in dynamic conditions in affordable simulation time thanks to the computational efficiency of the MB solvers. These capabilities can be exploited to simulate complex mechanical systems, such as mechanical transmissions. The complexity of non-linear dynamic phenomena generated during gear meshing, coupled with other mechanical components (e.g., bearings and flexible supporting shafts), make the system-level dynamic simulation of geared transmissions a very difficult task, where a proper balance between computational efficiency and reliability of predictive models is to be searched for. To achieve the wished accuracy without requiring prohibitively large computational time, critical phenomena, such as time-varying mesh stiffness of engaging gears and bearing compliance, must be modeled efficiently, but with sufficient detail. In this work, such a goal is achieved by combining non-linear Finite Element (FE) static simulations of gear meshing with MB simulations of the entire gearbox. In order to provide of proof-of-concept of the proposed approach, the gearbox of a wind turbine is analyzed through dynamic MB simulations.

Keywords: Multi-Body simulation, Transmission Error, wind turbine gearbox.

1. Introduction

Mechanical transmissions are extensively used in many industrial fields, thanks to their design flexibility and reliability. Multiple and interconnected phenomena that are typical in mechanical transmissions play a crucial role in determining the overall dynamic performances of the systems where they are used. From the simulation viewpoint, predictive analyses on mechanical transmissions are still a complex task, requiring consistent computational power and deep knowledge of the physical phenomena for a proper modelling. Planetary gear sets are the industrial standard in several applications related to automotive, aerospace and wind turbine industries, especially where high transmission ratios are required. The possibility to generate a large set of speed ratios in an extremely compact design in comparison with fixed-axis gearboxes is one of the advantages that contributed to their diffusion. The presence of multiple planets sharing the loading torque allows also to reach high power-to-weight ratios, making planetary gears particular suited for applications where lightweight design is crucial, such as in the aviation field. Additional advantages of their axi-symmetric and compact configuration are the wider manufacturing tolerances allowed [1] and, typically, the lower levels of noise and vibrations (N&V) [2] with respect to fixed-axis gearboxes.

In the last decades, many modelling strategies have been developed to simulate meshing gears, differing from each other for the level of detail and consequently for the computational effort required for the analyses. An efficient analytical formulation has been proposed by Blankenship and Kahraman [3]. This formulation consists in representing the meshing gears as a single degree-of-freedom (DOF) dynamic system. The position-dependent mesh stiffness is modelled as a spring element parallel to the direction of the line of action of the gears. Power losses are modelled with a damper element acting in parallel with the spring. Inertia moments represent the rotating masses.

Analytical formulations enable also investigations on the typical dynamic phenomena exhibited by planetary transmissions. As an example, Inapolat and Kahraman [4] use an analytical model to investigate the occurrence of modulation sidebands that appear in the proximity of the gear meshing orders at the gear mesh frequency in planetary gear sets. Additional dynamic effects, which are typical in planetary transmissions, are analysed by Cooley and Parker in [5].

Further DOFs can be added to the model in order to take into account the relative displacement between the gears due to static or dynamic deflection of other components in the transmission. Ongoing research activities still focus on analytical models, since the computational efficiency guaranteed by the lumped parameter formulation makes these models still attractive from the scientific viewpoint. The increased level of detail in the simulation models reached by these more complex formulations can be found in the methodology developed by Cai [6], which allows for taking into account the component flexibility. The large use of analytical formulations for the basic analysis of whine and rattle problems can be explained by their computational efficiency.

When a high level of detail is required or when cross-coupling effects between the meshing gears and other flexible structural components of the transmission have to be investigated, analytical formulations are not the ideal simulation approach. Moreover, tailoring the analytical formulations to the specific application by tuning all the parameters in the model, in such a way that multiple phenomena are properly represented at the same time, requires a high level of experience.

Examples of semi-analytical formulations based on linear FE and analytical formulations were developed by Parker et al. [7] with the aim of increasing the computational efficiency in comparison to full dynamic FE simulations. In this case, the non-linear load-displacement relationship due to contact loads is modelled analytically based on contact rolling cylinders. Once the non-linear contact loads are computed based on such analytical formulation, they are applied onto the nodes of a coarse FE mesh. A higher level of accuracy with respect to purely analytical formulations is ensured by the gears discretized with finite elements.

More detailed 3D approaches, such as the non-linear FE simulation, provide a high level of predictive accuracy at the cost of higher computational time. When stress patterns in the contact zones or load paths in static conditions have to be investigated this class of simulation tools are still a valid option for researchers and engineers. However, in case of system-level dynamic analysis, the computational time required by such simulations is prohibitively large.

Multibody (MB) simulations represent a trade-off between the computational efficiency of the analytical formulations and simulation accuracy of FE approaches. Through the flexible representation of structural components, MB simulations allow also for taking into account detailed deformation phenomena of the system components.

In this paper, an efficient MB approach, aimed at including three-dimensional gear contact in the dynamic simulation of meshing gears and proposed by Palermo et al. in [8] is used in combination with FE simulations in order to investigate the coupled effects of time-varying mesh stiffness of the gears and bearings compliance. The proposed approach consists of three main steps: (i) generation of the FE models of the meshing gears; (ii) derivation of the static transmission error (STE) curves for each gear pair through a series of non-linear FE simulations; (iii) dynamic simulation of the mechanical transmission in a MB environment at an assembly level, where the instantaneous values of the mesh stiffness are calculated by interpolation of the pre-computed STE curves. In order to provide of proof-of-concept of the proposed approach, the gearbox of a wind turbine is analyzed through dynamic MB simulations.

2. Multibody dynamic simulation of meshing gears

The first step of the proposed approach is a series of non-linear FE static simulations that are executed to compute the static transmission error (STE) for a discrete number of relative angular positions between two engaging gears along the meshing cycle, from which the angle-dependent meshing stiffness is derived during the dynamic simulation. The computed STE values are stored into look-up tables. During the MB dynamic simulations, the look-up tables are interpolated in order to extract the proper value of the STE for the actual configuration of the gears.

The time-demanding but accurate FE simulations are limited to a pre-processing phase, while the computational efficiency of the dynamic simulations, carried out in the MB environment, is preserved. This approach allows to perform computationally affordable MB system-level simulations with still an accurate representation of the gear meshing process. A flowchart of the proposed simulation approach is shown in Fig. 1.

Contact forces in in each gear pair are considered in the MB simulation through dedicated User Defined Force (UDF) elements implemented in LMS Virtual.Lab Motion software. The latter formulation is based on a spring-damper element, where the actual value of the mesh stiffness is computed starting from the STE values interpolated from a dataset for a possible discretized range of operating conditions and stored in multivariate look-up tables. The STE calculations have to be made for each different pinion-gear pair in the model.

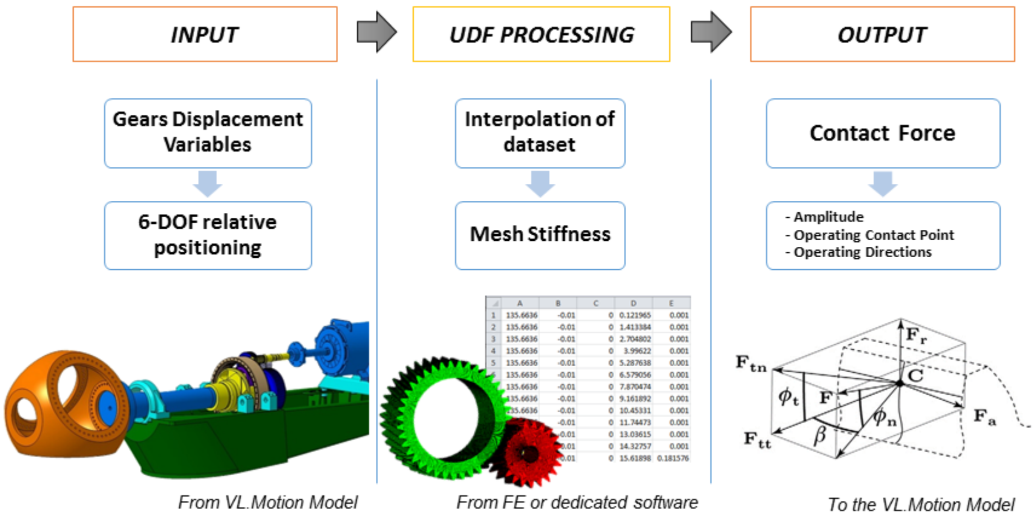


Figure 1: Main steps in the proposed simulation approach.

2.1 Multibody formulation of gear meshing

The UDF element used for MB simulation of gear meshing has to be defined for each gear pair in the model. It computes the gear meshing forces derived from the instantaneous conditions and determines the proper load location on the teeth flanks at the subsequent step of the dynamic simulation. The process of contact force determination uses the computed dynamic transmission error (DTE) between the gears at a given time step of the dynamic simulation by multiplying it for the actual value of the mesh stiffness.

The position of two reference systems fixed to the gear bodies is compared with the position of a third reference system for each time step of the simulation in order to calculate the DTE value, computed in a transverse plane tangential to the pitch circles of the meshing gears. DTE value is given as the sum of two different contributions: the relative rotations of the two meshing gears (DTE_r) and the contribution coming from the relative translation in the tangential direction (DTE_t), as described by the following equation:

$$DTE = DTE_r + DTE_t \tag{1}$$

In the UDF element the gear mesh force is derived from the DTE value and its time derivative as the sum of the elastic and viscous contributions:

$$\begin{cases} F_{tt}\boldsymbol{T} = \boldsymbol{F} \\ F_{tt} = F_{tt_k} + F_{tt_c} \\ F_{tt_k} = kDTE \\ F_{tt_c} = c\frac{dDTE}{dt} \end{cases} \quad (2)$$

where $\boldsymbol{F}$ is the normal contact force, F_{tt} is the magnitude of the tangential contact force and $\boldsymbol{T}$ is the transformation vector from the tangential direction to the direction normal to the tooth surface in the force application point. F_{ttk} and F_{ttc} represent the elastic and the damping components of the tangential contact force. The constants k and c are the instantaneous mesh stiffness and the viscous damping coefficient respectively. The transformation vector, which allows to compute the radial, the tangential and the axial components of the contact force for a pair of helical gears, is defined as:

$$\boldsymbol{T} = \begin{Bmatrix} tan\varphi_n/cos\beta \\ 1 \\ -tan\beta \end{Bmatrix} \quad (3)$$

where β and φ_n are the helix and the normal pressure angles respectively.

The meshing force is applied in the operating contact point, which is always located at the operating pitch point in the transversal plane according to the proposed formulation. Moreover, in the case study analysed in this work, the operating contact point is considered to be always located in the middle plane of each gear, which implies neglecting shuttling phenomena.

Under the assumption that the external load is constant, the time-varying nature of the mesh stiffness is taken into account by calculating it, at each time step in the simulation, as:

$$k(PMC, CD, M) = \frac{F_{tt_n}}{STE(PME,CD,M)/\cos\varphi_t} \quad (4)$$

where PMC, CD and M are the actual values of the position along the meshing cycle, of the gears centre distance and of the angular misalignment in the plane of action respectively. In the case study analysed in this paper, the latter two parameters are neglected, while PMC, which assumes values in the range between 0 and 1, is calculated at each time step as:

$$PMC = \frac{1}{\theta_{P1}}\left[|\theta_1 + \theta_{1s}| - floor\left(\frac{|\theta_1 + \theta_{1s}|}{\theta_{P1}}\right)\theta_{P1}\right] \quad (5)$$

where θ_1 is the actual rotational angle, θ_{P1} is the angular pitch and θ_{1s} is the initial angular position. By setting proper values for the latter parameter, the ring-planet and sun-planet mesh phases can be taken into account. In the approach proposed here, the STE curves for each pair, to be used for the estimation of the mesh stiffness according to Eq. (4), are calculated through static FE simulations, as described in the following sub-section.

2.2 Estimation of STE curves by non-linear FE simulations

The transmission error (TE) is considered as one of the main dynamic excitation sources in a system where meshing gears are present. The correct representation of TE in the dynamic system has a dramatic influence on the results of the dynamic simulations. In the approach described in this work numerically-generated STE curves are used in the MB simulations. The generation process can be achieved by using dedicated tools or general purpose FE solvers, as described in [9]. In this work, implicit static analyses with geometrical nonlinearities were performed. In order to illustrate the proposed approach, the FE model of a pair of meshing gears used in the model is shown in Fig 2.

TE can be defined as the difference between the angular position that the output shaft of a drive would occupy if the drive were perfect and the actual position of the output [10]. Such a difference in angles can converted into a linear displacement as:

$$TE = r_{bG}\theta_G - r_{bP}\theta_P \tag{6}$$

where θ_P and θ_G are the rotation of the pinion and of the gear respectively, while r_{bP} and r_{bG} are their base radii.

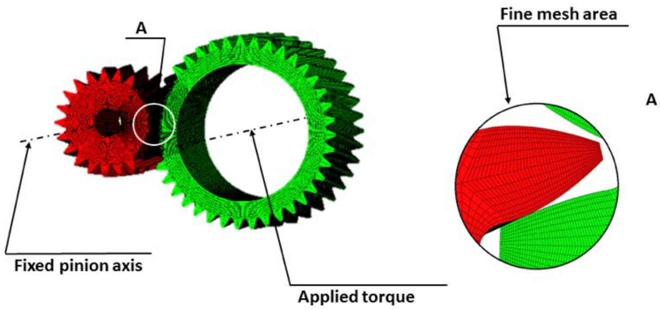


Figure 2: Schematic of the FE-based approach used to estimate STE curves.

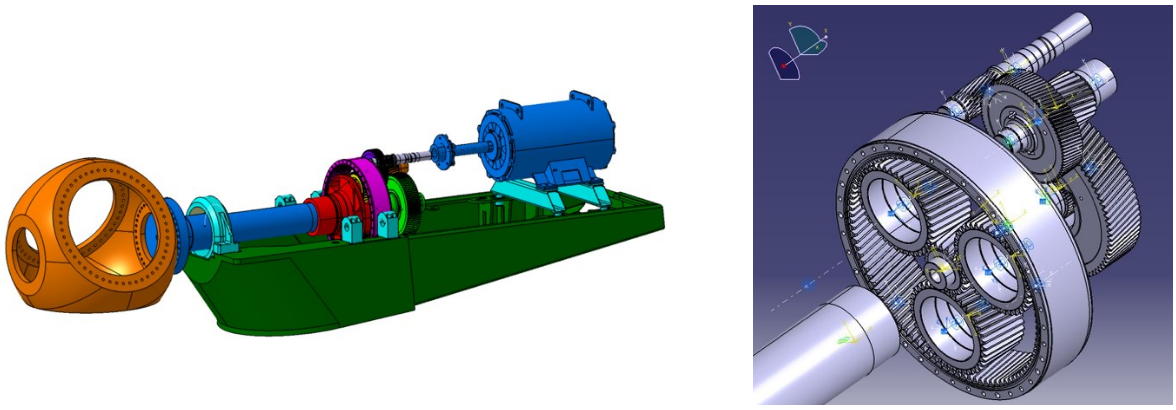
Each FE simulation models consists of a gear, which is supported by an infinitely rigid shaft and is allowed to rotate around the rotational axis, and a pinion, in which the rotational degree of freedom of the central node is constrained. A loading torque is applied to the gear shaft and by estimating the rotational angle after the load application θ_G, the STE value for the gear pair in a given angular position is calculated according to Eq. (6). Here, θ_P is set to zero, since the pinion axis is held stationary. The value θ_G has to be computed for all the different angular position of the gear pair, in which the meshing cycle is discretized.

3. Case-study description

In order to provide a proof-of-concept of the proposed simulation methodology, the dynamic behavior of a wind turbine gearbox is analyzed. The MB model of the entire transmission and of the gearbox are shown in Fig. 3. These models were built starting from the GRC 750 kW gearbox, used by the National Renewable Energy Laboratory (NREL) in the Gearbox Reliability Collaborative [11]. This gearbox is used as a speed increaser coupled with a generator, and the total gear ratio is 81.49. A description of the gearbox, consisting of a planetary set (low speed stage) and two parallel axes sets (intermediate and high-speed stages) is been given in [11], while the main macro-geometry parameters used to generate the gears models are reported in Table 1.

Table 1: Key macro-geometric parameters of the gears in the gearbox

Parameter	Planetary stage			Intermediate stage		High-speed stage		Units
	Ring	Planets	Sun	Gear	Pinion	Gear	Pinion	
Teeth number	99	39	21	82	23	88	22	N/A
Normal module	10.0	10.0	10.0	8.25	8.25	5.0	5.0	mm
Normal pressure angle	20	20	20	20	20	20	20	deg
Helix angle	7.495	7.495	7.495	14	14	14	14	deg
Centre distance	308.0	308.0	308.0	450.0	450.0	285.0	285.0	mm



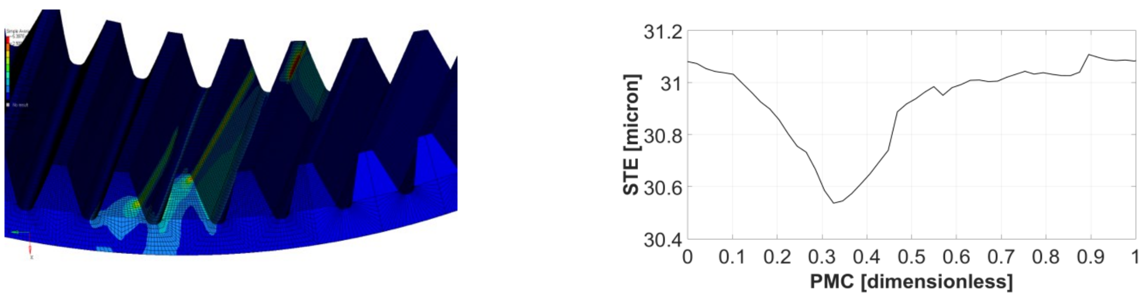
a) b)

Figure 3: MB model of the analysed wind turbine gearbox: full transmission (a) and gearbox (b).

In the MB model of the gearbox different UDF elements are present, which are intended to take into account the periodic, time-varying mesh stiffnesses acting between the ring and the three planets, between the latter and the sun and between the engaging gears in the intermediate and high-speed parallel stages.

In order to show the capabilities of the proposed approach for gear contact modelling in system level simulations, two models of the gearbox have built and analysed, one with ideal kinematic pairs supporting the gears, and a second one, in which compliant bushing elements are introduced in the model to take into account the flexibility of the bearings. The inertia proprieties of the different components have been derived directly from their CAD models.

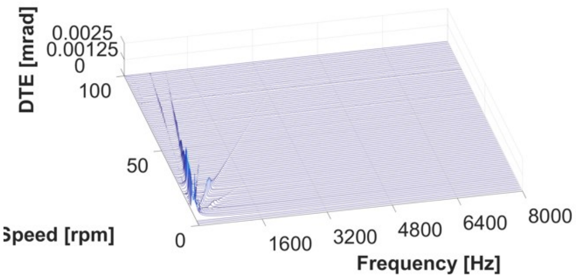
With the aim of deriving the mesh stiffness curves to be used in the MB simulation, the FE models of each gear pair were built by using 8-node hexahedral elements and analysed through static non-linear FE simulations that allowed to estimate the STE curves. Figure 4 (a) shows an example of contact stress distribution on the ring of the planetary gear in a given configuration, while the STE curve estimated for the ring-planet pair is shown in Fig. 4 (b) as a function of the gears' angular position along the meshing cycle. Each STE curve is the results of 50 FE analysis of the gear pair, being the meshing cycle discretized in a set of 50 equally distributed angular positions. For each gear pair, the static simulations were executed by considering the supporting shafts as infinitely rigid.



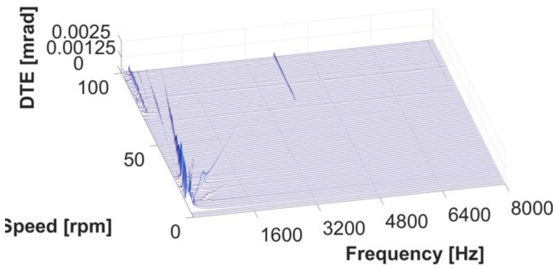
a) b)

Figure 4: Example of contact stress distribution on the ring in a given configuration (a) and STE curves estimated for the ring-planet pair (b).

The dynamic behaviour of the gearbox was analysed by means of multibody simulations in LMS Virtual.Lab environment, based on the contact formulation illustrated in the previous section. Simulations were performed by applying a constant torque value of 240 kNm and 3000 Nm on the input and output shafts respectively. A run-up in velocity from 0 to 215 rpm was thus achieved for the carrier in 30 sec. The results of the simulations are illustrated in Fig. 5 in terms of waterfall diagrams of the overall DTE of the transmission. From the result sets, multiple dynamic amplifications can be observed, occurring at those rotational speeds where the order of the meshing frequency or of its harmonics cross a natural frequency of the system.



a)



b)

Figure 5: Waterfall diagrams of the estimated overall DTE for the model with ideal joints (a) and the one with compliant bushing elements (b).

By comparing Fig. 5 (b) with Fig. 5 (a), it is possible to notice additional resonances in the waterfall diagram of the overall DTE, corresponding to the additional eigen-frequencies of the model with bushing elements. In order to further clarify these results, the dynamic amplification of the DTE shown at around 3200 Hz of Fig. 5 (b), which is not present when the model with perfectly rigid joints is analysed, is explained by a global mode shape of the transmission, shown in Fig. 6.

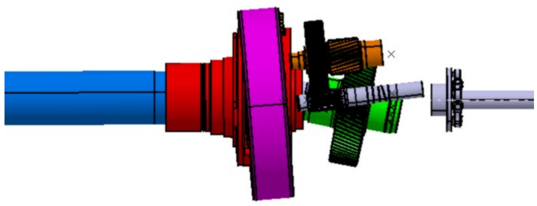


Figure 6: Eigenvector of the transmission corresponding to the system resonance frequency at 3215 Hz in the model with compliant bearings.

4. Conclusions

In this paper, the coupled effects of gears' transmission error and bushing compliance on the dynamic behaviour of a wind turbine gearbox has been investigated by combining FE and MB simulation methodologies. Two modelling strategies have been analysed, considering ideal joints and compliant bushing elements. In line with the findings of other research works, the numerical results show that the dynamic behaviour of the gearbox significantly depends on a proper modelling technique, since the excitation of vibration modes in the system can be missed if the compliance of the bearings is neglected.

The main advantage of the proposed approach is the exploitation of time-consuming FE simulations in the pre-processing phase to accurate model of gear contact at the level of individual meshing pairs, while during the system-level simulations the mesh stiffness is quickly derived from the already stored STE values. The latter analyses are executed in a MB environment, which enables efficient, but still accurate predictions. To exploit such an advantage, this simulation approach is employed in system-level dynamic simulations with the aim analysing the effects of local phenomena occurring at each gear mesh. Possible applications of the proposed approach include investigations on the effects of planet phasing in the planetary stage or bushing stiffness and location on the behaviour of the full mechanical transmission. Finally, non-rigid MB modelling could be also exploited to assess the impact of the structural flexibility (e.g., flexible shafts, planet arm, housing) on the N&V behaviour of the entire mechanical transmission in an efficient but still accurate way.

Acknowledgments

The research leading to these results has received funding from the People Programme (Marie Curie Actions) of the European Unions Seventh Framework Programme FP7/2007-2013/ under REA grant agreement no. 324336 DEMETRA: Design of Mechanical Transmissions: Efficiency, Noise and Durability Optimization. The authors also thank NREL (the National Renewable Energy Laboratory) for providing the gearbox design specification and manufacturing drawings that are used in the Gearbox Reliability Collaborative (GRC) project and also used in this work.

REFERENCES

1 Ligata, H., Kahraman, A., Singh, A. An experimental study of the influence of manufacturing errors on the planetary gear stresses and planet load sharing, *ASME Journal of Mechanical Design*, 130 (4) (2008)

2 Lynwander, P. *Gear Drive Systems*, Marcel Dekker Inc., New York, 1983.

3 Blankenship, G. W., Kahraman, A. Steady state forced response of a mechanical oscillator with combined parametric excitation and clearance type non-linearity, *Journal of Sound and Vibration*, vol. 185 (1995)

4 Inalpolat, M, Kahraman, A. A theoretical and experimental investigation of modulation sidebands of planetary gears, *Journal of Sound and Vibration,* Vol. *323* (2009)

5 Cooley, C.G., Parker, R.G. A review of planetary and epicyclic gear dynamics and vibrations research, *Applied Mechanics Reviews*, 66 (4) (2014)

6 Cai, Y. Simulation on the rotational vibration of helical gears in consideration of the tooth separation phenomenon (a new stiffness function of helical involute tooth pair), *Journal of Mechanical Design*, vol. 117 (1995)

7 Parker, R. G., Vijayakar, S. M., and Imajo T. Non-linear dynamic response of a spur gear pair: modelling and experimental comparisons, *Journal of Sound and Vibration*, vol. 237, no. 3 (2000)

8 Palermo, A., Mundo, D., Hadjit, R. and W. Desmet, W. Multibody element for spur and helical gear meshing based on detailed three-dimensional contact calculations. *Mechanism and Machine Theory*, 62, 13–30 (2013)

9 Korta, J.A., and Mundo, D. Multi-objective micro-geometry optimization of gear tooth supported by response surface methodology, *Mechanism and Machine Theory*, 109, 278–295 (2017)

10 Smith, J. D. *Gear noise and vibration*. Marcel Dekker, Cambridge (2003).

11 Oyague, F., *Gearbox Reliability Collaborative (GRC) Description and Loading*, NREL Technical Report, 2011