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RESEARCH ARTICLE

DYNAMIC ANALYSIS OF WIND TURBINE GEARBOX PLANET GEAR CARRIER USING FINITE ELEMENTS APPROACH

Prof. Shawki Abouel-seoud, Dr. Ibrahim Lotfy and Dr. Mohamed Elmorsy

Helwan University, Cairo, Egypt

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ABSTRACT

Planetary gears are effective power transmission elements in wind turbine where high torque to weight ratios, large speed increase in compact volumes, co-axial shaft arrangements, high reliability and superior efficiency are required. The present work examines the complex dynamic behavior of planetary gearbox gear carrier using finite element models. The finite element model (FEM) is developed from a unique finite element-contact analysis solver specialized for gear dynamics. The natural frequencies are the most important parameter in the design of planet gears carrier for dynamic loading conditions, so that the modal analysis had been utilized to determine the natural frequencies of the planet gears carrier system. The simulated results show that the planet gears are taking responsibility of the instability depending on the mode thus unfitting the different approaches to planet gears carrier dynamic and enabling the most appropriate component to be targeted dynamic behavior control. The natural frequencies and of the wind turbine gearbox planetary gear carrier are classified as rotational and translational or out-of-phase modes. A confirmation was made with the experimental results taken from a lab-scale wind turbine gearbox and found to be good. The gained information can be used for diagnosis and prognosis planet gears carrier.

INTRODUCTION

Planetary gears are effective power transmission elements where high torque to weight ratios, large speed ratio in compact volumes, co-axial shaft arrangements, high reliability and superior efficiency are required. Example applications are automotive transmissions, tractors, wind turbines, helicopters, and aircraft engines. Gear vibrations are primary concerns in most planetary gear transmission applications, where the manifest problem may be noise or dynamic forces. Noise levels exceeding 110 dB observed in a helicopter cabin are attributed largely to vibration of the planetary gear. Large dynamic forces increase the risk of gear tooth or bearing failure. Experiments on a spur gear pair and observed various nonlinear phenomena including gear tooth contact loss, period-doubling and chaos are performed. Tooth separations at large vibrations, which are common in spur–gear pairs, occur even in planetary gears as evident from the experiments. Planetary gear researchers have developed lumped-parameter models and deformable gear models to analyze gear dynamics. The literature mainly addresses static analysis, natural frequencies and vibration modes, modeling to estimate dynamic forces and responses, and cancellation of mesh forces using the planetary gear symmetry through mesh phasing. Accurate analytical modeling, including proper mesh phasing relations and detailed characterization of the nonlinear dynamics of planetary gears, is needed to estimate relative gear noise and predict dynamic forces in industrial applications. Little work has been done to characterize the nonlinear effects of tooth separation on planetary gear dynamics. The lack of experimental studies to understand the complex dynamics of planetary gears and the availability of finite element software specialized for gear dynamics motivated the present study [1, 2].

Knowledge of the modal properties of planetary gears is crucial for developing strategies to reduce vibration. Planetary gear dynamic models are developed in [3, 4]. These show that two-dimensional, spur planetary gears with equally spaced and diametrically opposed planets possess well-defined modal properties. They report all vibration modes belong to one of the three categories: (1) Rotational modes where the central members (sun, carrier, and ring) rotate but do not translate. The planet motions are identical. (2) Translation all modes with degenerate natural frequencies, where the central members translate but do not rotate. There are well-defined relations between the two independent vibration modes at each natural frequency. (3) Planet modes where only the planets move, and their motions are scalar multiples of the arbitrarily chosen first planet's motion and a similar categorization applies to compound planetary gears. Furthermore, the modal properties of spur planetary gears having elastically deformable ring gears. These vibration mode characteristics

***Corresponding author:** metwalley1965@hotmail.com

are crucial in vibration suppression strategies using mesh phasing and eigen-sensitivity analysis of planetary gears.

Extensive experimental and numerical analyses on the dynamic behavior of planetary trains are conducted. From a 2D finite element model, ring-gear deflections were found to significantly influence tooth loads and load sharing properties. The analyses also pointed out that ring-gear deformations can reduce the distance between adjacent teeth leading to higher actual contact ratios and possible interferences. Furthermore, the influence of the planet angular phasing on dynamic loads induced by planet run-out errors is studied. Significant differences were found depending on planet indexing with certain configurations being less sensitive to manufacturing errors. The profile and run-out deviations in a 2D model for investigating planet load sharing is introduced and found that the phases between the planets could modify sun-gear trajectories, and several studies confirmed the contribution of mesh phasing as well as the possibility of cancelling some frequency components of the global translational or torsional excitations on sun-gears and ring-gears [5-7].

Modeling of planetary gear set dynamics received significant attention for the last 30 years. A number of studies proposed lumped-parameter models to predict free and forced vibration characteristics of planetary gear sets. These models assumed rigid gear wheels, connected to each other by springs representing the flexibility of the meshing teeth. In these studies, nonlinear effects due to gear backlash and time-varying parameters due to gear mesh stiffness fluctuations were neglected. The corresponding Eigenvalue solution of the linear equations of motion resulted in natural modes. Modal summation technique was typically used to find the forced response due to external gear mesh displacement excitations defined to represent motion transmission errors. These lumped-parameter models vary in degrees of freedom included, from purely torsional models to two or three-dimensional transverse-torsional models. While these models served well in describing the dynamic behavior of planetary gear sets qualitatively, they lacked certain critical features. First, the gear mesh models were quite simplistic with a critical assumption that complex gear mesh contact interaction can be represented by a simple model formed by a linear spring and a damper. These models demand that the values of the gear mesh stiffness and damping, as well as the kinematics motion transmission error excitation, must be known in advance. It is also assumed that these parameter values determined quasi-statically remain unchanged under dynamic conditions. In addition, gear rim deflections and Hertzian contact deformations are also neglected. Another group of recent models used more sophisticated finite element-based gear contact mechanics models. These computational models address all of the shortcomings of the lumped-parameter models since the gear mesh conditions are modeled as individual nonlinear contact problems. The need for externally defined gear mesh parameters is eliminated with these models. In addition, rim deflection and spline support conditions are modeled accurately. These models are also capable of including the influence of the tooth profile variations in the form of intentional profile modifications, manufacturing errors or wear on the dynamic behavior of the system [8-10]. The transmission of the UH-60A helicopter has an epicyclic, or planetary, gear train in the final stage of the main rotor gearbox. Torque is transmitted from the central sun gear through the planets to the planet carrier and from the planet carrier to the main rotor shaft. It is important to have tools to assess the condition of these components as they form a non-redundant critical part of the drive to the main rotor. One such tool is vibration analysis. However, the vibration of epicyclic gear trains is difficult to analyze. Not only are there multiple planet gears producing similar vibrations, but there are also multiple and time-varying transmission paths from the gear mesh points to the transducers, which are typically mounted on the gearbox housing. These factors combine to reduce the effectiveness of conventional fault detection algorithms when they are applied to epicyclic gear trains. These data sets were using several standard diagnostic parameters that were modified for the special case of an epicyclic gearbox and applied to the time synchronous averages of the planet carrier vibration. The vibration data were also analyzed where a number of different metrics for the detection of faults in fixed-axis gears were modified and applied to the time synchronous averages of the planet carrier vibration [11].

The objective of this study are to characterize the complex, nonlinear dynamics of planet gear carrier system using a unique finite element model across the range of complicated nonlinear dynamics occurring in the system. The finite element formulation of static and dynamic contact/ impact problems for planet gear carrier is introduced. To verify the method and the program developed, experimental results are presented.

MODELING OF PLANET GEARS CARRIER DYNAMICS

The description of the planet gears carrier dynamics

In wind turbines, the overall influence of the structure flexibility, consisting of the tower, the nacelle and, of course, the rotor has become very significant. On top of that, due to the larger forces and moments, the gearbox and other drive train flexibilities have also become of larger influence on the global turbine behavior. Within this framework, more detailed gearbox loading simulation models with explicit focus on the influence of flexibility on the model behavior and practical implementation of flexibility within the model are needed.

This work will focus on the wind turbines planet gears dynamics and do this by means of a generic gearbox, consisting of three gear stages: one planetary stage and two helical stages, shown in Fig. 1. The planetary gear stage consists of three planets and fixed ring wheel, indicated by number 2 in Fig. 1. On the low speed shaft (LSS), indicated by number 3 in Fig. 1, the slow wheel is pressed. This is in contact with the teeth on the intermediate shaft (ISS), marked by number 4 in Fig. 1. On the intermediate shaft a high speed wheel is mounted, which establishes contact with the teeth on the high speed shaft (HSS), indicated by number 5 in Fig. 1.

The description of the gearbox dynamics is performed in two methods. The first method is a state-of-the-art six degree of freedom (DOF) multibody model (MB model) with discrete flexibility. In this model, the different gearbox components: shafts, gears, planets, planet carrier and housing are considered to be rigid. Bearings and gear contact are modeled by means of discrete flexibilities. This type of model will be referred to by the term: rigid multibody model with discrete flexibility. The second method is to let the component flexibilities to be included in multibody (MB) model. These are introduced, using finite element models reduced by means of the component mode synthesis technique (CMS) . The term flexible multibody model will be used for these CMS based type of multibody models. Important in this reduction, is the use of appropriate coupling structures at the interfaces between the Finite element (FE) and multibody model. In this work only second method is used and applied on planet carrier element.

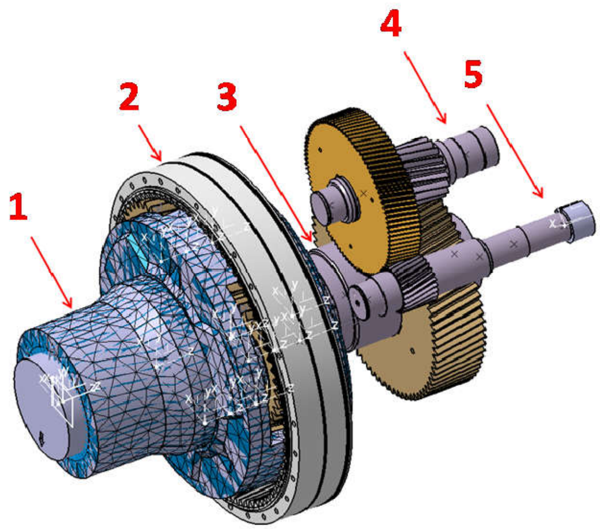

Fig. 1: General view of investigated gearbox

For the planet carrier, the coupling between the multibody and FE model is more complex. The following coupling structures need to be introduced:

- The two bearing connections, supporting the planet carrier, are modeled using flexible multipoint constraints, as indicated by number 1 in Fig. 2 in order to have a more accurate introduction of the bearing forces over all significant nodes.
- Flexible multipoint constraints are introduced to represent the planet bearing connections between each of the planets and the planet shafts. These are marked by the arrows with number 2 in Fig. 2.
- The coupling with the main turbine shaft is realized, using a rigid multipoint constraint, indicated with number 3 in Fig. 2. A rigid multipoint constraint is used to represent the interface, because the main turbine shaft, which in the turbine is inserted at this location in the planet carrier flange, is assessed to be very stiff. The components of the planet carrier are shown in Fig. 3.

Finite element model formulation

The finite element method is a technique for mathematically modeling complicated shapes (feature) as an assembly of a simpler shape (elements) that is more easily defined [12]. Linear and non-linear problem in engineering field are of the great importance to be studied in this work. Therefore, the Finite Element package called Ansys 5.7 has been chosen to solve this problem. The meshed carrier and planet gears are examples of 3-D finite elements models (FEM). FEM here is used to characterize the dynamics of the carrier and planet gears efficiently and then these characteristics are split into elements. These elements are

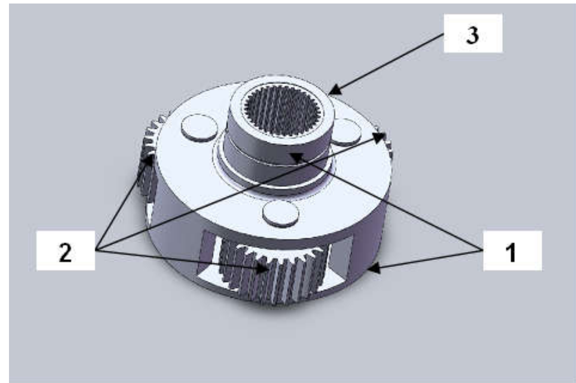

Fig. 2: Planet gear carrier multipoint locations

Fig. 3 Planet carrier components

connected with each other through points called nodes. The complete collection of the elements is called mesh. Restraints and loads are added after this to the meshed part and whole thing then is called model. One advantages of FEM is thus, many different design concepts can be tested via computer, and shape can often be finalized before any prototype design. The finite element technique has been used in this paper to study the modal analysis of planet gears carries. The modal analysis has been used to determine the natural frequencies and associated mode shapes of the carrier system [13].

MODAL ANALYSIS OF PLANET GEARS CARRIER

The modal analysis of planet gears carrier perhaps the most important process to understand in order to find solutions for the wind turbine gearbox problem. Some properties like loss factor, natural frequencies and mode shapes of planet carrier are crucial in defining which type of planetary gearbox problem may occur. The planet gears carrier was supported by two slender cables in order to simulate a free-free boundary condition [14]. The free-free condition allows the structure to vibration without interference from other parts, making the visualization easier of mode shapes associated with each natural frequency. Moreover, in this case, the rigid body oscillation frequency of the assembly is much lower than the first natural frequency of the structure

ANALYSIS DESCRIPTION AND SUBSTRUCTURE CONCEPT

General

In order to get sufficient answers to the occurring vibration behavior within the wind turbines gearboxes, the different gearbox components, such as shafts, gear wheels, planet carrier, etc. are modeled by representing them using finite element models. In order to keep the calculation times of the full model reasonable, the finite element (FE) models need to be reduced using the component mode synthesis technique.

Component Mode Synthesis (CMS)

An appropriate solution for achieving a model reduction is the use of the component mode synthesis (CMS) technique [15]. This is a form of substructure coupling analysis in which the dynamic behavior of the substructure is formulated as a superposition of modal contributions of pre-selected component modes of the following types: normal modes, rigid body modes, constrained modes, attachment modes, inertia relief modes and/or inertia attachment modes. There are several combinations of these sets that generate a superposition of the modes, sufficient for determining the exact static and dynamic response of the component submitted to external forces applied at boundary nodes. The CMS technique starts from the second order differential equations describing the linear dynamic behavior of the considered component:

$$M\ddot{q} + Cq + K\dot{q} = F \qquad (1)$$

In any of the CMS techniques the classical modal transformation:

$$q = \boldsymbol{\psi}.\boldsymbol{\eta} \qquad (2)$$

is performed to represent the physical coordinates q in terms of component coordinatesη. The transformation matrix consists of pre-selected component modes. In this research, the reduced householder method is used. In this, the modal transformation matrix ψ is defined b

$$\boldsymbol{\psi} = [\boldsymbol{\psi}_C \; \boldsymbol{\psi}_K] \qquad (3)$$

where ψ_C are the constrained modes and ψ_K the normal modes. After performing the modal transformation, the equations of motion are written in terms of the component coordinates .

ANALYSIS METHOD

Planetary Stage Modes

The considered planetary stage, consists of a cage planet carrier with three planets, a sun and a planet-ring wheel. The planet-ring is non-rotating and can be considered as rigid multibody with discrete flexibilities of full gearbox with free boundaries. The modes of such a planetary system can be classified into rotational and translation modes.

- ***Rotational Modes:*** These modes have pure rotation of the carrier, ring and sun. All planets have the same motion and move in phase. The multiplicity of the corresponding eigenfrequencies is one for these modes.
- ***Translational Modes*:** In these modes, the carrier, ring and sun have pure translational movement. The multiplicity of the corresponding natural frequencies is two.

Flexible Model of Planetary Stage

The second important gear stage type in the overall gearbox is the planetary stage. Because this system is far more complex than the helical system, the flexibility investigation is performed in several steps. To this end, the eigen frequencies are calculated based on free-free flexible planet carrier conditions, where the technical parameters for the planetary gearbox comprises 3-planet gears are tabulated in Table 1.

Table 1 Technical data for planetary gearbox (3 planet gears, 1 sun gear,1 rig gear)

No.	Model parameter	Notation	Unit	Value	Remarks
1	Number of plant teeth	Z_p	--	26	
2	Number of sun teeth	Z_s	--	16	
1	Number of ring teeth	Z_R	-	68	
2	Mass of planet gear	m_p	kg	2.746	
3	Mass of sun gear	m_s	kg	1.446	
4	Mass of ring gear	m_R	kg	0.677	
5	Mass of carrier	m_c	kg	13.153	
6	Number of planet gear	s	-	3.0	
7	Young's modulus	E	N/m^2	2.068×10^{11}	
8	Poisson's ratio	ν	-	0.3	
9	Pressure angle	α	degree	20	
10	Planet diameter	D_p	mm	110	
11	Sun diameter	D_s	mm	84.87	
12	Ring diameter	D_R	mm	360.68	
13	Face width	W	mm	60.0	
14	Transmission ratio	R_P	-	5.25	

RESULTS AND DISCUSSION

Theoretical Results

A detailed FE model of the planet carrier, consisting of 105564DOFs and using multipoint constraint connecting structures located as shown in Fig. 2, was constructed. Different mesh sizes were compared until stabilization of the first 15 flexible modes occurred within a range below 1%. Fig. 4 shows the meshed structural model of planetary gears carrier. The free-free Eigen frequencies are listed in Table 2. It can be seen that the planet carrier itself has already quite some eigen frequencies within the range of interest. Therefore it is advisable to further investigate the planet carrier influence. The modal analysis of the gear carrier showed 15 modes in the frequency range 1000-15000 Hz by using the technique of reduced householder method in the Ansys 5.7 package. Due to the flexibility of the planet carrier a large number of modes is introduced, which are dominated by the flexible displacement/deformation of the planet carrier. These are listed in Table 2. Examples of such mode shapes are shown in Figs. 5 and 6. Furthermore, due to the frequency shift and the extra frequencies, it can be concluded that it is advisable to model the planet carrier as a flexible component within the full gearbox model.

Table 2. Eigenfrequencies (Hz) for Free-free planetary stage modes

No.	Parameter	Values							
1	Mode Number	1	2	3	4	5	6	7	8
2	Frequency, Hz	1097	1461	1982	2055	2941	3100	3822	4622
No.	Parameter	Values							
1	Mode Number	9	10	11	12	13	14	15	
2	Frequency, Hz	5320	6050	6911	8010	9985	12001	13088	

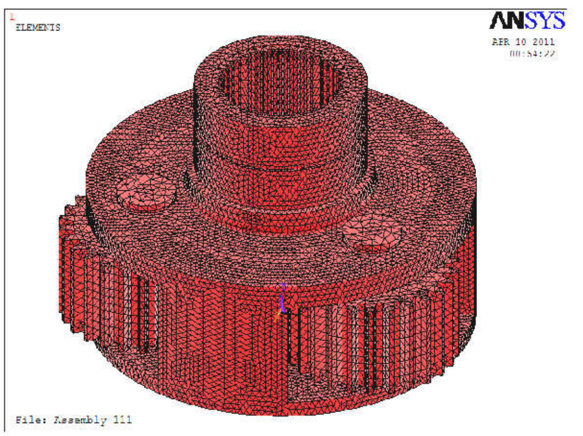

Fig. 4. 3-D Meshed structural model of planetary gears carrier

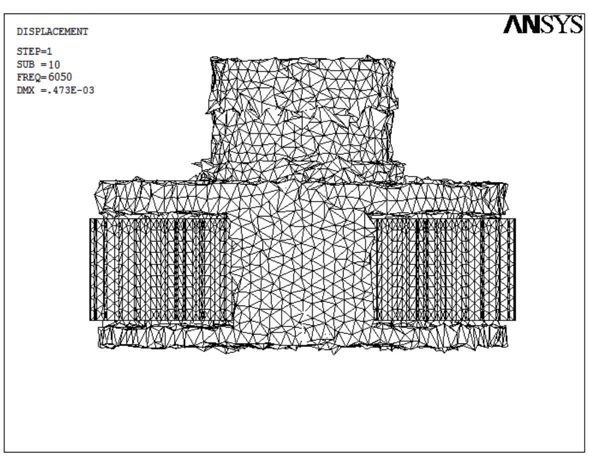

Fig. 5. Planet carrier mode at 6050Hz

Experimental confirmation

Figures 7, 8 and 9 depict the locations of the calculated modes on the measured vibration velocity spectra at different speeds and torque loads. The details of the method of measurement is presented in Ref. [16]. The whole calculated modes are divided into four groups based on the high amplitude existed in the spectra. A confirmation for the calculated modes with the experimental results has been gained.

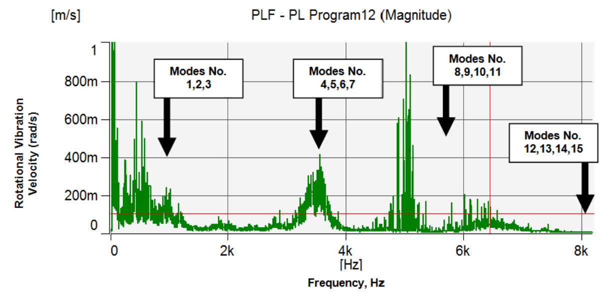

Fig. 7. Locations of the calculated modes over the entire spectrum at 20 rpm and 40 Nm

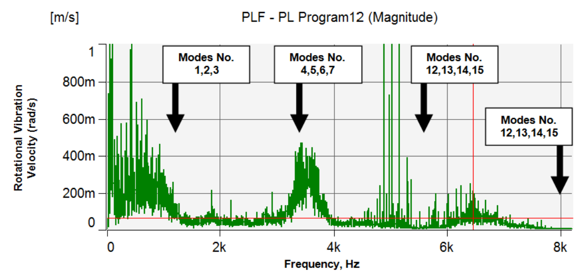

Fig. 8. Locations of the calculated modes over the entire spectrum at 30 rpm and 30 Nm

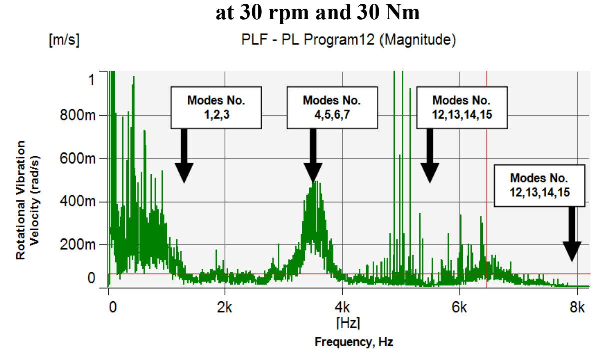

Fig. 9 Locations of the calculated modes over the entire spectrum at 30 rpm and 40 Nm

CONCLUSIONS

1. Component flexibility was introduced into the wind turbine planet gear carrier model, using the householder method reduced finite element models. A first main focus was laid on the coupling between the FE and MB model. In order to get an accurate description of the dynamics of the real system under investigation, the coupling structure is found to be of relevance, as they both influence the system constraint modes as well as its normal modes.

2. The planetary stage flexibility was investigated in detail. Two main conclusions were drawn. First of all, the rigid MB model with discrete flexibility locates the eigenfrequencies quite accurately. Furthermore, a good confirmation for the calculated modes with those measured has been gained. The gained information can be used for diagnosis and prognosis planet gears carrier.

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