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VIBRATIONAL STUDY OF EPICYCLIC GEAR TRAINS –A REVIEW

[1*]Brij Narayan Babeja, [2]Jiyaul Mustafa

[1]M.Tech Student, School of Mechanical Engg, Galgotias University, Greater Noida, India

[2]Assistant Professor, School of Mechanical Engg, Galgotias University, Greater Noida, India

ABSTRACT

The authors summarize published journal articles on planetary and epicyclic gear dynamics and vibration. Research in this field has increased dramatically over the past three decades. The wide range of research topics demonstrates the technical challenges of understanding and predicting planetary gear dynamics and vibration. The research in this review includes mathematical models, vibration mode properties and gyroscopic effects, among other topics. Practical aspects are also included, for example, planet load sharing and characteristics of measured vibration response.

Keywords: Epicyclic Gear Trains, Vibrational Analysis, Dynamics analysis, Isomorphism.

I INTRODUCTION

Epicyclic gears are used to transmit power in a wide range of industrial applications. A schematic of a epicyclic gear is shown in Fig. 1. All epicyclic and epicyclic gears have three bodies that we refer to as central members: the carrier, the ring gear, and the sun gear. The planet gears are connected by bearings to the carrier and are simultaneously in mesh with the sun gear and ring gear. The number of planet gears varies depending on the design load of the system. Having multiple planets gives epicyclic gears multiple load paths and compact packaging. All central members kinematically rotate about the same axis, although one central member is typically stationary. The planet gears rotate about axes that are fixed to the carrier, which might also rotate. In a star configuration, the carrier does not rotate, and all gears, including the planets, rotate about fixed axes.

Despite their long history and wide use, epicyclic gears still experience noise and vibration problems. In automotive applications vibration from the transmission, which contains several epicyclic gears, leads to noise that is perceived negatively as a measure of vehicle quality. Epicyclic gears are the main noise source in helicopter cabins [1]. Measured sound levels can exceed 100 dB. This causes problems with communication and creates a noise hazard for the pilot and passengers. Noise is also an issue in wind turbines, which contain one or more epicyclic gears, when they are located near populated areas. Dynamic tooth and bearing loads affect the lifetimes of epicyclic gear components. In aircraft engines, vibration can cause structural failure. Planet bearings fail in wind turbines as a result of dynamic loads and vibration from the gear mesh excitation [2,3]. Such vibration, along with excitation from wind loading on the blades, is a dominant vibration source [4]. Epicyclic gear failures in wind turbine applications are costly because of the large amount of downtime necessary to make repairs [5].

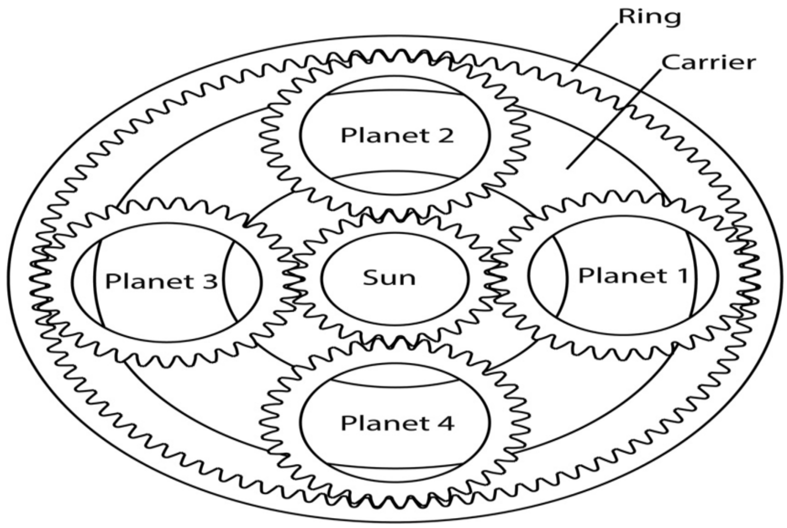

Fig. 1.Schematic of a planetary gear with four planets

Engineers in industry have multiple software tools available toanalyze epicyclic gear vibration. Several of these analysis toolsare based on multibody dynamics codes and can model completetransmission systems, not just epicyclic gears. They can incorporatefinite element models of the housing and carrier built usingconventional finite element software. These methods historicallyhave had only basic tooth mesh modeling, although more recentlythe focus has been on better mesh modeling. Multibody dynamicsmethods can provide quick answers that can be used to assess differentdesigns, but they are less effective for root cause failureanalyses because of simplifications of the gear mesh. There arealso finite element/contact mechanics tools, for example, Calyx[6] which can be used for high fidelity contactmodeling and analysis. Although accurate contact between bodiesis captured, this solution often requires long computation times.Interestingly, lumped-parameters models (i.e., models based onmass-spring representations for rigid body gears) are not widelyused in industry, and underutilized when they are. In contrast totheir use in applications, these models have been widely used inepicyclic gear research, where they have been shown to comparewell with experiments and detailed finite element/contact mechanicsmodels. Lumped-parameter models can be put togetherquickly, require relatively few input quantities, and are solvedeasily. They likely perform similarly to multibody dynamicscodes for evaluating the natural frequencies and vibration modesof different design variations. Of course, they cannot be used tocalculate stresses and strains. For these reasons, lumpedparametermodels should be incorporated into gear design

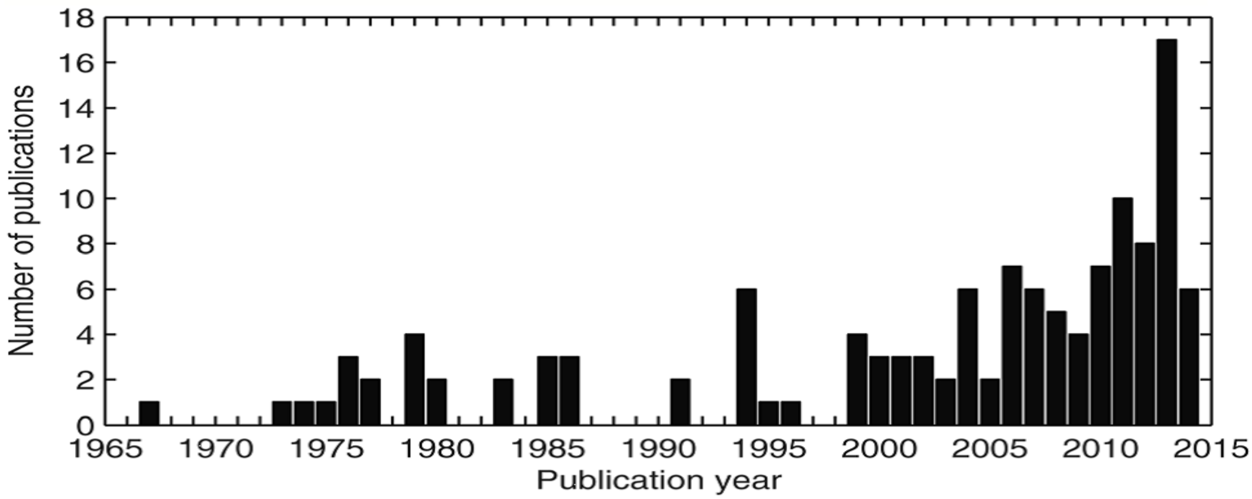

Fig. 2. Histogram of research papers on planetary gear dynamics and vibration

II VIBRATION MODE PROPERTIES

Epicyclic gears have distinct vibration mode structure because of their cyclic symmetry. This was first noticed in lumped parameter models [7-16], and later validated with finite element/contact mechanics models [23,24] and experiments [25]. Cunliffe et al. [8] determined the vibration modes of a particular 13 DOF epicyclic gear.

They grouped modes into low frequency "bearing modes" and high frequency "tooth modes," but did not identify the vibration structure. Botman [9] determined the vibration of a DOF epicyclic gear model, where each gear has two translational and one rotational DOF. Botman [9]first noticed the distinct vibration structure. When analyzing athree-planet system, he discovered two types of vibration modes and grouped them as "axisymmetric" and "non-axisymmetric" because of the planets' motion. Botman [9] states that in axis ymmetric modes "all planets perform the same motion with respect to the sun and the other components have only rotational motion."Botman also noticed that the central members have "lateral motion" in non axis ymmetric modes. Kahraman [10] noticed a third mode type when analyzing a four-planet system. This modewas not observed by Botman [9] in their analysis of a three planet system. The third mode type was called a "counter-phased mode" [10]. Both Botman [9] and Kahraman [10] only identified the aforementioned vibration structure on example epicyclic gears; no general conclusions were given.

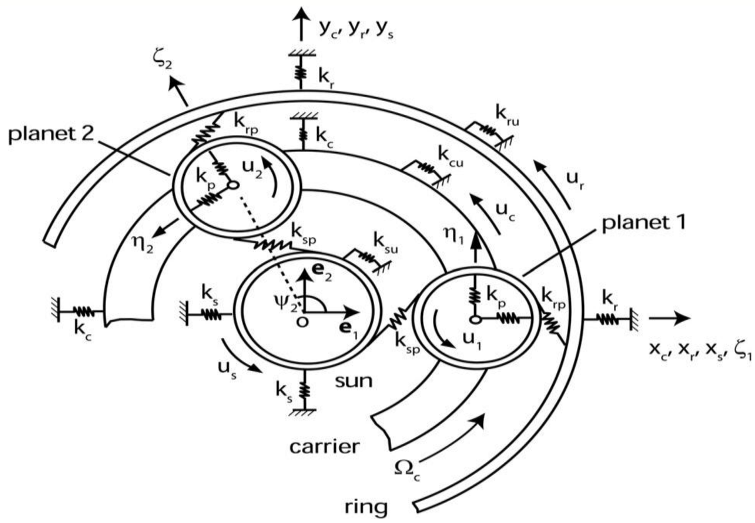

Fig. 3. Lumped-parameter planetary gear model from Ref. [7]

The vibration structure was formally identified and proved byLin and Parker [7] for epicyclic gears with equal planet spacing. Their lumped-parameter model included radial and tangential planet deflections for each component. They proved that epicyclic gears have exactly three types of vibration modes, independent ofthe number of planet gears and the system parameters. They called these planet, rotational, and translational modes. Planet modes occuronly for epicyclic gears with four or more planets. There are exactly three planet mode eigen values with degeneracy $N>3$.

The central members (the sun, carrier, and ring) have no motion. Only the planets have motion. There are exactly six rotational modes and these have distinct eigenvalues. Rotational modes have central members with only rotational motion but no translation. All planets have identical motion in rotational modes.

There are exactly six translational mode eigenvalues that have degeneracy two. In these modes the central members have only translation and no rotation. This structure persists for some unequally spaced planet configurations, specifically diametrically opposed spacing [21]. For general spacing, the rotational and translational modes couple and lose their distinct structure. In contrast, planet modes occur for any planet spacing.

Ericson and Parker [25] found that certain epicyclic gear eigen values group into clusters. Each cluster consists of a translational,rotational, and planet mode (when $N>3$) eigenvalues. These clustersremain for a wide range of parameter variations.

It is common in applications to have two or more epicyclicgears transmit the engine power. Systems that consist of two or more epicyclic gear stages or epicyclic gears with more than oneplanet in each load path are called compound (or sometimes complex)epicyclic gears. Kahraman [13] investigated the vibrationmodes of compound epicyclic gears based on a model with only rotational DOFs. He observed, but did not analyze, three modetypes called a rigid body mode, asymmetric planet modes, and axisymmetric overall modes. Kiracofe and Parker [14] derived ageneral set of governing equations for compound epicyclic gear sincluding translational and rotational DOFs for each body. Their formulation includes all configurations, including multistage, meshed-planets, and stepped-planets. The distinct vibration modestructure identified by Lin and Parker [1] also occurs for compound epicyclic gears [18]. Guo and Parker [20] derived a general rotational DOF compound epicyclic gear model that clarified previously published rotational models. They showed that there are exactly two vibration mode types, called overall modes and planetmodes, in contrast to the three mode types found in Ref. [17]. The rigid body mode in Ref. [17] is an overall mode.

The vibration of helical epicyclic gears was investigated in Refs. [14] and [21]. These gears have axial forces and tilting moments that lead to 3D vibrations. Eritenel and Parker [21]determined the vibration structure of helical epicyclic gears.T here remain exactly three mode types called planet, rotational–axial, and translational–tilting modes. Interestingly, these mode types are not disrupted by asymmetry about the gear plane; onlycyclic symmetry is necessary for these mode types to occur. In the absence of axial and tilting motions, these mode types converge tothose of Ref. [10].

At higher frequencies, for lightweight epicyclic gears, or forthin ring gears designed to improve load sharing, gear compliancemay cause elastic deformation. Wu and Parker [24] expanded the lumped-parameter epicyclic gear model from Ref. [10] to includethe effect of elastic ring deflection, which they captured by a partial differential equation using thin ring theory. Their model gives four mode types called planet, rotational, translational, and purely ring modes. The planet, rotational, and translational mode shave the same properties as the lumped-parameter modes in Ref.[10] with specific nodal diameter elastic ring deflections for each mode type. Similar structure was found for epicyclic gears with an elastic ring gear and some unequally spaced planet arrangements[26]. There have been studies on the dynamic response ofepicyclic gears with compliant ring gears that do not focus on vibration mode structure, for example, Refs. [17], [18], and [27].

These works will be discussed in Secs. 5 and 7.Bu et al. [7] investigated the vibration mode structure of aspecific herringbone, or double helical, epicyclic gear. They include only in-plane displacements for the sun, carrier, and planet herringbone gears. In mesh with the planets are two helical ring gears modeled as 3D rigid bodies. This system has rotational–axial, translational, and planet modes, as expected from Refs. [7] and [16]. For this system there are two additional mode types, called “rotational and axial ring modes” and “translational ring modes.” There are exactly two rotational and axial ring modes and one degenerate (with degeneracy two) translational ring mode. These modes have vibration confined to the two helical ring gears. Qian et al. [29] determined the vibration modes for a specific epicyclic gear used in coal mining applications. Sondkar and Kahraman [20] derived a general model for the vibration of double helical epicyclic gears. Their model considers each gear of a double helical set as a rigid body. This allows for investigation of different relative orientations of the helical gears in each set, called “staggering.” Example vibration modes are shown and characterized according to their planar features. No general conclusions are made on the 3Dfeatures of these vibration modes.

The vibration modes of epicyclic gears operating at highcarrier speeds were investigated by Cooley and Parker [12]. Theydetermined the vibration modes when gyroscopic effects caused by constant carrier rotation are modeled. Past analyses haveneglected gyroscopic effects. Their study showed that the rotational, translational, and planet mode types in Ref. [7] persist,but the modes are complex-valued with phase differences betweendifferent DOFs. Additionally, the system natural frequencies andvibration modes change with carrier speed. The degenerate translationalmode natural frequencies at zero speed become distinct atnonzero speeds. In contrast, degenerate planet mode natural frequenciesremain degenerate for all carrier speeds.

In the authors' work with industrial partners, other systems containingepicyclic gears have been analyzed using the epicyclicgear model in Ref. [7]. Of particular interest is the case of a epicyclicgear in an aerospace transmission connected by a shaft to anoutput parallel-axis gear pair. A model was developed to determinethe natural frequencies and vibration modes. The addition ofthe gear pair disrupts the system cyclic symmetry, and the rotationaland translational modes couple to become "overall modes."Planet modes, in contrast, occur for this system and have the sameproperties as given in Ref. [7]. Figure 3 shows representativemode shapes for this system. Similar mode types are observedin wind turbine applications, where one or sometimes moreepicyclic gears are connected to a parallel-axis gear pair. For anysystem involving a epicyclic gear with four or more planets,one should expect planet modes regardless of the connection toother transmission components or the epicyclic gear mountingstructure.The vibration mode structure given above has been seen inmultibody dynamics models of wind turbine gearboxes [30].

Helsen et al. [30] also report additional structural modes of thecomponents, for example, elastic carrier modes.Epicyclic gear cyclic symmetry is fundamental to their structuredmode types. This leads us to hypothesize, with confidence,the mode types for other epicyclic gears that have not yet beenstudied. For example, high-speed compound epicyclic gears arelikely to have rotational, translational, and planet mode types asin Ref. [14] with phase relationships in Ref. [12]. Similarly, high speed helical epicyclic gears likely have rotational–axial, translational–tilting, and planet modes types as in Ref. [16] with phase relationships as in Ref. [12].Often in applications it is only necessary to know the operating speeds with potential resonances. Design modifications are used to tune natural frequencies away from potential excitation frequencies. In these cases only the natural frequencies and vibration modes are needed; no calculations of forced response are necessary. Frater et al. [31] investigated the effect of unequal mesh stiffnesses on epicyclic gear natural frequencies and vibration modes. Saada and Velex [32] investigated the influence of tooth mesh stiffness, planet bearing stiffness, and ringsupport stiffness on the natural frequencies of epicyclic gears.

Kahraman [11] investigated epicyclic gear natural frequencies as support stiffness is varied. These have been numerical parameter studies.The effects of design parameter changes on natural frequencies and vibration modes can be analyzed mathematically. The well-defined vibration mode properties of epicyclic gears lead to simplified expressions for eigen value sensitivity calculations to model parameter variations. Lin and Parker [33] derive eigen value sensitivity to system parameters for the general epicyclic gear model in Ref. [7]. Guo and Parker [34] derive eigen value sensitivity expressions to model parameters for the general compound epicyclic gear model in Ref. [14].

They include eigen value sensitivity expressions for parameters specific to compound epicyclic gears, for example, the coupling stiffness between epicyclic gear stages. Gyroscopic effects on the natural frequencies of epicyclic gears near zero speed were investigatedby Lin and Parker [33].

Eigenvalue veering [35] occurs when two nearby eigenvaluel oci, when plotted for a changing system parameter, approacheach other then repel (or veer away) as a model parameter isvaried. It is accompanied by a mixing of the two mode shapes inthe vicinity of the veering zone and an exchange of modal propertiesbetween eigenvalue loci on opposite sides of the veeringzone. Because of its dramatic effects on the vibration modes, veeringcan strongly impact the gear vibration in the veering zone.

Veering of epicyclic gear eigenvalues was investigated by Linand Parker [36]. They used the structured vibration modes toshow that eigenvalue loci of different mode types always crossand those of the same mode type veer as a model parameter isvaried. Ericson and Parker [25] showed that clusters of natural frequenciesveer away when the natural frequencies of two clustersapproach as a parameter is varied.Cooley and Parker [37] showed epicyclic gear single-modevibration has well-defined geometry and frequency content due itsvibration mode structure. They explain how measurements fromfixed and rotating sensors have different frequency content.

III SUMMARY AND CONCLUSIONS

The dynamics and vibration of planetary and epicyclic gearsis a practically important engineering problem with numeroustechnical challenges. Planetary gears have well-defined naturalfrequency and vibration mode structure because of their cyclicsymmetry. This structure persists with increasing model complexity,such as 3D effects, elastic continuum vibration, and gyroscopiceffects from carrier rotation. Models for the forcedresponse of planetary gears show resonances from mesh stiffnessfluctuations and nonlinearity due to tooth separation, which alsooccurs in measurements of practical planetary gear systems.

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