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Effects of variable loading conditions on the dynamic behaviour of planetary gear with power recirculation

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ABSTRACT

Variable loads to which gearboxes are subjected are considered as one of the main sources of non-stationarity in these transmissions. In order to characterise their dynamic behaviour in such conditions, a torsional lumped parameter model of a planetary gear with power recirculation was developed. The model included time varying loading conditions and took into account the non-linearity of contact between teeth. The meshing stiffness functions were modelled using Finite Element Method and Hertzian contact theory in these conditions. Series of numerical simulations was conducted in stationary conditions, with different loading conditions. Equation of motion was solved using Newmark algorithm. Numerical results agreed with experimental results obtained from a planetary gear test bench. This test bench is composed of two similar planetary gears called test planetary gear set and reaction planetary gear set which are mounted back-to-back so that the power recirculates through the transmission. The external load was applied through an arm attached to the free reaction ring. Data Acquisition System acquired signals from accelerometers mounted on the rings and tachometer which measured instantaneous angular velocity of the carrier's shaft. The signal processing was achieved using LMS Test.Lab software. Modulation sidebands were obtained from the ring acceleration measurements as well as a non-linear behaviour in case of variable loading resulted by a transfer of the spectral density from the fundamental mesh stiffness to its second harmonic.

1. Introduction

The use of planetary gears in industrial applications is justified by their ability to transmit high torques with substantial ranges of speed reduction. Many industrial applications use this kind of transmissions such as aircraft engines, and wind turbines.

Several studies were devoted mostly to planetary gears running under constant speed and load i.e. in stationary conditions [1–5]. However, it is possible to find industrial applications involving planetary gears where both load and speed are time varying. Severe non-stationary conditions may lead to excessive vibrations and instability [6].

McFadden and Smith [1] focused on the modulating sidebands around meshing frequency and harmonics caused by non-stationary conditions. Al-shyyab and Kahraman [2] implemented multi-term harmonic balance methodology on a nonlinear torsional model of a single stage planetary gear in order to solve the equations of motion. Inalpolat and Kahraman [3] developed a numerical model to predict modulation sidebands, he described the mechanisms causing sidebands and he validated experimentally the obtained numerical results. He also studied amplitude and frequency modulations caused by manufacturing errors of gears and estimated dynamic loads on sun-planet and ring-planet gear meshes [4]. Liu et al. [5] took into account in their dynamic model variable transmission path of vibration in planetary gears and validated their work experimentally.

Concerning studies devoted to variable loading conditions, it seems that Randall [7] was the first to relate load fluctuation to vibration level. Chaari et al. [8] proposed a bi-dimensional model of a spur planetary gear subjected to both time varying load and speed and highlighted corresponding amplitude and frequency

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modulations. Mark and Hines [9] was interested in the variability of load transmission from one planet to the other caused by imperfections in the transmission. He also focused on vibration spectra shapes of signals registered by fixed-accelerometer for the case of a cracked planet-carrier plate that caused variability in load transfer through planets. Mark [10] extended his studies to the influence of planet–carrier vibration signals modulations on the computed spectra. Feng and Zuo [11] considered, in his dynamic model of planetary gear, the effect of distributed and localized gear faults on amplitude and frequency modulation effects observed in vibration signals. Lei et al. [12] implemented the adaptive stochastic resonance method for a planetary gearbox having a chipped and missing sun gear tooth. Bartelmus and Zimroz [13] showed that planetary gearbox running in bad conditions is more susceptible to external load than planetary gearbox running in good conditions. They introduced a new diagnostic feature which is used for monitoring the condition of planetary gearboxes under varying external load conditions [14]. Kim et al. [15] was interested in the influence of time varying pressure angles and contact ratios on the dynamic behaviour of planetary gear. In all these cited woks, the varying external load caused a variation of speed in the studied systems.

Complex planetary gearboxes received great interest by scientists. Ligata et al. [16] studied on a back-to-back planetary gear system the influence of manufacturing defects on stresses in the tooth roots and on the load sharing between planets. Singh et al. [17] focused on the impact of changing parameters of multistage planetary gearbox on stresses in gear teeth and on load sharing through numerical and experimental investigations. Hammami et al. [18] was interested in the modal characteristics of back-to-back planetary gear set by computing modal kinetic and strain energy distributions. In addition, they [19] achieved a series of experimental tests for run up and run down regimes of the same gearbox in order to validate the modal analysis and to study its dynamic behaviour in non-stationary conditions.

This paper is dedicated to the study of the dynamic behaviour of a complex planetary gearbox running in another non-stationary operation which is the time varying loading condition with the specifications of an imposed constant speed. To achieve this target, a back-to-back planetary gearbox with mechanical power recirculation set up was characterised as the load was applied by the external arm. First, a dynamic model of planetary gear set with power recirculation will be developed. Modulation sidebands will be highlighted in stationary condition in the case of equally-spaced planets and sequentially phased gear meshes. Then, different loading conditions will be considered to show the non linear behaviour of the studied gearbox and to explain its behaviour in the variable loading conditions which is presented in the last section. Simulation and experimental studies for this gear system will be presented and correlated in all studied conditions.

2. Material and methods

The test bench used in this research work is composed of two similar planetary gears mounted back-to-back so that the power recirculates through the transmission. This special configuration is selected in order to minimize costs and improve energy efficiency. Fig. 1 shows a general scheme of the studied transmission.

The two planetary gear sets are named respectively test gear set and reaction gear set. The main planetary gear set is the test gear set where its output power from its carrier is reintroduced to its sun through the reaction planetary gear set. They are connected in back-to-back configuration. Sun gears are mounted on the same shaft and the carriers are connected by a hollow shaft.

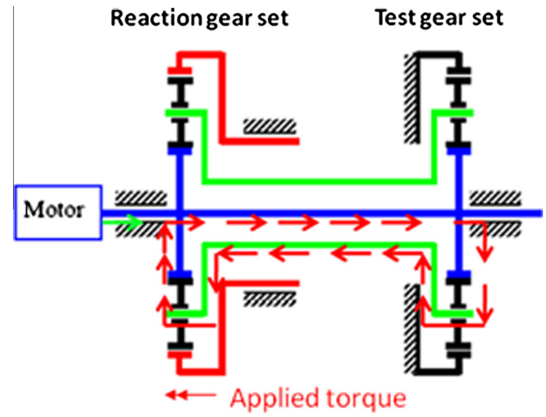

Fig. 1. Test bench scheme.

In order to introduce external load, mass is added on an arm attached to reaction gear. The test gear ring is clamped (Fig. 3). The direction of rotation is such that the friction torque always adds to the reaction ring gear applied torque.

The transmission is driven by an asynchronous motor to which a speed controller is added in order to impose desired speed evolution and values. Accelerations on rings are measured by two triaxial accelerometers (Figs. 2 and 3).

An optical tachometer combined with pulse tapes is mounted on the hollow shaft in order to measure instantaneous angular velocity.

LMS SCADAS 316 Data Acquisition System acquires signals from tachometer and accelerometers and the signal processing is achieved using LMS Test.Lab software.

International standard "ISO 6336" [20–23] allows to determine the minimum tangential torque on the reaction ring that will produce the mechanical failure in case of tooth bending and pitting for the different gear components of the system. These limits are presented in Table 1.

In order to avoid bending and pitting defect, applied torques on the test rig will not exceed 1100 N m.

3. Numerical model

In this section, a dynamic model of the back-to-back planetary gear test rig presented in Section 2 will be developed. It is a torsional model based on previous work of Lin and Parker [24]. Fig. 4 shows the main components of this model.

Mesh stiffness for test ring-planet and test sun-planet are represented respectively by the linear springs $K_{tr,i}$ and $K_{ts,i}$ where $i = 1\ldots n$ and n is the number of planets ($n = 3$). For the reaction gear set, mesh stiffness is represented by the linear springs $K_{rr,i}$ and $K_{rs,i}$ which are related to reaction ring-planet and reaction-sun planet. Since the reaction ring is free, its torsional stiffness K_{rru} is zero, however, the test ring is clamped and modelled by a torsional stiffness K_{tru} with high value. The connecting shafts are modelled by torsional stiffness K_s and K_c. Table 2 shows the parameters of the model.

Non floating suns and planets are considered and only rotational motions of the gear bodies are considered [25]. The equation of motion of the system with 3 planets can be written as:

$$M\ddot{X} + C\dot{X} + (K(t) + K_c)X = F(t) \quad (1)$$

where M is the mass matrix. $K(t)$ is the stiffness matrix, K_c is the coupling shaft stiffness matrix and $F(t)$ is the external force vector applied to the system.