Simulation of the vibration signal of cycloidal drives: preliminary results

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ABSTRACT

Introduction

Due to the extensive use of gearboxes in industrial applications, especially in the fields of robotics and automation, their study for maintenance optimization has a clear interest. In particular, such devices can be analyzed in terms of their vibration behavior, which may provide useful information for condition monitoring. This work focuses on cycloidal drives, which are widely adopted, thanks to their compactness and ruggedness; however, with respect to standard gear trains, cycloidal drives have a more complex architecture. We have developed an analytical model of the vibrational behavior of cycloidal drives, with the goal of obtaining a basic simulation tool, with which different faults (occurring due to the natural wear of components) can be considered. The model is based on the contact patterns of the rotating elements, which are defined as functions of the angular position: this way, any input speed profile can be taken into account. The internal load distribution is also considered in the model, since we need to estimate the contact forces in our analysis. We thus obtain an effective simulation tool, that may be easily generalized to other epicyclic gear trains.

Cycloidal drives: their role and applications

Cycloidal drives are a family of gear mechanisms used as speed reducers: due to their attractive advantages, such as high reduction ratios and near-zero backlash, they are applied in a number of fields, especially where high-precision motion transmission is sought, such as in robotics.

As shown in Fig. 1, a typical cycloidal drive has the following main components:

1. an *input crank shaft*, supported at its ends by frame-mounted bearings;
2. a (fixed) *ring gear*, on which cylindrical pins are mounted on the inside;
3. *planet discs* with an *epitrochoid* external profile, meshing with the pins on the ring gear (epitrochoids are curves related to cycloids, from which these drives derive their name);
4. an *output flange* with (larger) pins, engaging with eyelets on the planet discs.

As the shaft rotates, the planet discs, carried by eccentric bearings on the shaft, rotate on moving axes and transmit the motion to the output flange through its pins, which engage the discs through hollow rollers (to reduce friction): this way, the transmission ratio τ of the system is given by

$$\tau = \frac{\omega_D}{\omega_C} = -\frac{Z_{HP} - Z_L}{Z_L} \tag{1}$$

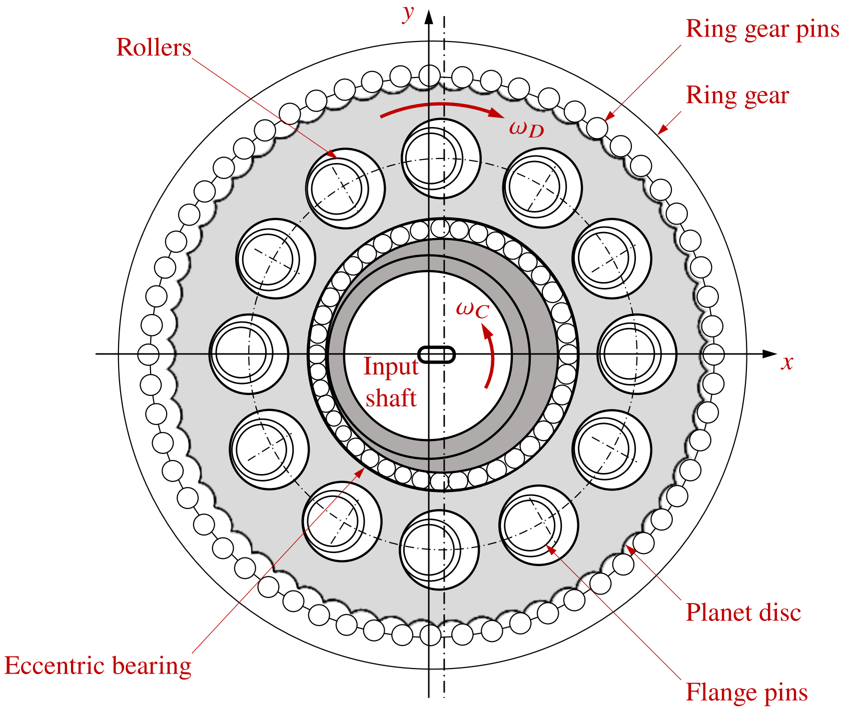

Figure 1. Schematic of a cycloidal drive (section view, normal to the shaft axis).

where Z_{HP} is the number of housing pins and Z_L the number of lobes on each disc; the ratio τ is negative, since the output flange and the input shaft rotate in opposite directions.

While several works have been written on the kinematic and static analysis of these mechanisms or on the technological process for generating the epitrochoidal disc profiles, very little information is available on the vibration profiles of the drives. These are especially interesting from the maintenance point of view: indeed, the analysis of vibration signals can provide essential information for condition monitoring on the health status of machines or of their components. In particular, some components have characteristic frequencies that can be clearly identified in the vibration spectrum and which allow us to recognize an incipient fault [1]. Regarding cycloidal drives in particular, one author [2] analyzed the effectiveness of applying standard coefficients that had already been developed for vibration-based condition monitoring of gears and bearings; others [3] presented the relevant frequencies to be monitored and the corresponding acceptable thresholds, as derived from experimental tests.

In our work [4], we study in particular the cycloidal drive *Fine Cyclo FCA 25G 29* by Sumitomo Drive Technologies [5]. We find the characteristic frequencies of vibration, similarly to what has already been done for simpler elements such as bearings [6]; also, we propose an analysis of the internal load distribution, adapting previous results [7], and estimate the global transmission efficiency. Finally, we develop a simulation tool that computes the characteristic vibration spectrum: the experimental data can then be used to calibrate the simulated model, in particular by setting the amplitudes of the peaks. In our approach, we define the contact pattern as a function of the angular coordinate of the input shaft, instead of the time: this way, any motion profile, including those at non-constant velocity, can be taken into account.

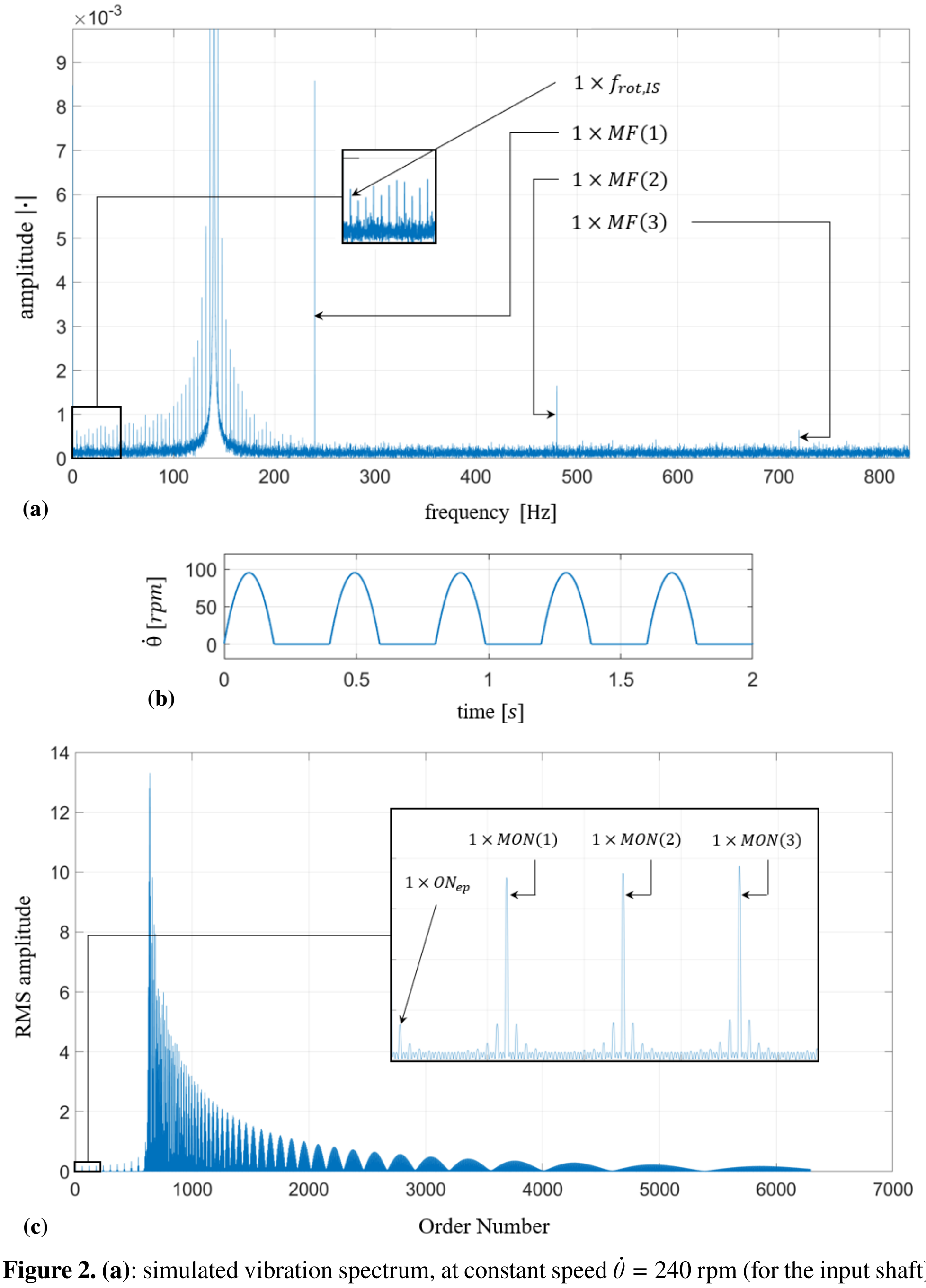

Figure 2. **(a)**: simulated vibration spectrum, at constant speed $\dot{\theta}$ = 240 rpm (for the input shaft). **(b)**: an example velocity profile for the input shaft, as a function of time. **(c)**: the resulting vibration spectrum; in this case, since the velocity is no longer constant, it is more meaningful to plot the amplitude as a function of the order. Here, $f_{rot,IS}$ is the rotation frequency of the input shaft, $MF(\lambda)$, for $\lambda = 1, 2, 3, \ldots$, are the meshing frequency (and its harmonics) of the planet discs with the ring gear pins, and $MON(\lambda)$ is the corresponding order number; here, $\lambda = 1$ corresponds to the effect of a single disc on the vibration spectrum. Finally, ON_{ep} is the number of repetitions of the motion law profile for a complete rotation of the input shaft.

Simulation results

Example results from the simulations are reported in Fig. 2. The model depends on parameters ω_n, k and ζ, which define the impulse response of a 1-Degree-of-Freedom (1-DoF) system: these parameters correspond, respectively, to the natural frequency, stiffness and damping coefficient of a mass-spring-damper system. We propose how these parameters may be derived numerically by a linear-least-squares optimization on the measured signal. The impulse response of the 1-DoF system is then used to filter the raw analytical signal, which depends on the angular positions where impact events (due to faults) occur, weighted by the contact loads acting at the corresponding rotation angles. Finally, a random Gaussian noise signal is added.

Figure 2a shows the vibration spectrum for the drive at hand, for a constant rotation speed. The peaks corresponding to the input shaft rotation frequency $f_{rot,IS}$ and to the meshing frequency $MF(\lambda) = \lambda \times Z_{HP} \times f_{rot,IS}$ are clearly visible; similar patterns have been observed in experimental tests currently underway in our laboratory, which supports the theoretical model. The model can also take into account a variable-speed motion law, such as the one shown in Fig. 2b; however, in this case it becomes necessary to use a order-based spectrum, as shown in Fig. 2c. Again, the peaks corresponding to the meshing orders $MON(\lambda)$ are visible. In a real test, results as those in Figs. 2a and 2c suggest the presence of a distributed damage on the ring gear pins: this model can then be used as a reference for diagnostic purposes.

Conclusions

We developed a computationally efficient and easy to use software model for the signal measured by an accelerometer mounted on the frame of a cycloidal gear, as preliminary work towards a general diagnostic tool for vibration-based condition monitoring of these systems, widely used in industrial settings. The results suggest that our model is simple enough to be practical for machine diagnostics and sufficiently flexible to be of interest for maintenance professionals.

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