Analysis of nonlinear gear dynamics based on visualization of vibro-impact regimes

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Summary. Nonlinear gear dynamic response and rattle noise induced by vibroimpacts between gear teeth are investigated using a specific experimental set-up allowing the visualization of impacts thanks to a high-speed camera. The control parameters during the experiment are the drag torque, the mean drive gear rotational speed, and the velocity fluctuation amplitude and frequency. Most of the time, an almost 1*T*-periodic response is observed with 2 impacts per period. A contact phase between gear teeth is observed after each impact instead of an instantaneous rebound. The number of successive tooth pairs crossing the meshing zone without any contact between gear teeth varies according to the ratio of the excitation frequency to the rotation frequency. Analytical and numerical works performed using a gear rattle model show good agreement with the experiments. Finally, the sound pressure emitted from the gear pair is measured. The acoustic power imputable to gear rattle is found to be proportional to the total kinetic energy transferred per second to the system by the successive impacts.

Introduction

Many geared systems, are subjected to such external excitations that contact losses between gear teeth may occur under some particular operating conditions (e. g. manual automotive gearbox [1], roots vacuum pump [2]). The nonlinear gear dynamic response is then characterized by impacts between the active and/or the reverse tooth flanks, leading to a broadband noise emitted from the mechanical system known as gear rattle noise. In this study, a specific experimental set-up is designed to analyse the rattle behaviour of a spur gear. Most of the key parameters governing the nonlinear dynamics are controlled during the experiments, that is to say the velocity fluctuation amplitude and frequency of the drive gear, the inertia of the output gear, the drag torque, and the gear backlash. Unlike most systems and experimental studies, the vibratory level of the drive gear is controlled independently of its mean rotational speed. Operation is performed without oil lubrication in order to allow simple modelling of the elastic and damping characteristics during impacts and easy direct visualization of the meshing zone, thanks to a high-speed camera. The occurrence of successive impacts between gear teeth obtained from video post-processing are then coupled with the gear dynamic transmission error measurement thanks to high resolution optical encoders. Finally, the sound pressure generated by successive impacts between gear teeth is also measured thanks to a microphone.

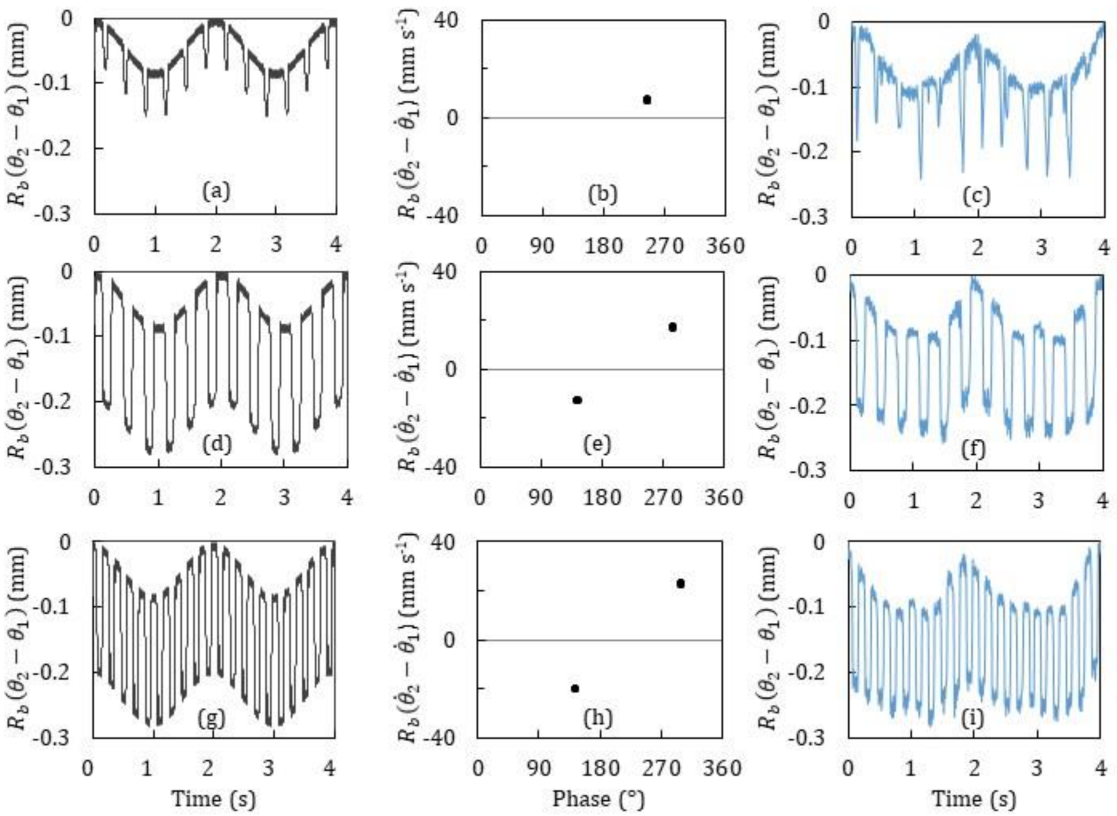

Figure 1: Impact response (numerical transmission error, Poincaré section, experimental transmission error) for several velocity fluctuation frequency and amplitude.

Nonlinear gear dynamic responses

The instantaneous drive gear velocity is:

$$\Omega(t) = \Omega_0 + \Delta\Omega \sin(\omega t) \quad (1)$$

The piloting allows increasing and decreasing sweeps of the mean rotational speed Ω_0, the velocity fluctuation amplitude $\Delta\Omega$ and frequency ω. Parameters are controlled independently. First, the effect of velocity fluctuation amplitude $\Delta\Omega$ is analysed for a chosen excitation frequency ω. Second, the effect of the excitation frequency ω is analysed for a chosen velocity fluctuation amplitude $\Delta\Omega$. For a very low excitation, an almost permanent contact between the active flanks is observed. When the amplitude is increased, the rattle threshold is reached. Video post-processing (see figure 2) and dynamic transmission error response (see figure 1) show noticeable contact losses and impacts. First, impacts only occur between the active flanks with a low impacting velocity, because the excitation amplitude is still too low to cross the gear backlash. Larger excitation amplitude leads to successive impacts alternating between the active and the reverse flanks. The output gear crosses the gear backlash forward and backward. Each impact is followed by a persistent contact phase between the gear teeth. The free flight period and the following persistent contact phase show a duration of the same order of magnitude. Considering the period of the excitation $T = 2\pi/\omega$, the gear dynamics corresponds to a $1T$-periodic response with 2 impacts per period. Poincaré sections show that impact phases and impacting velocities are almost constant for all the successive impacts between the active flanks, as well as for all the impacts between the reverse flanks. Visualization confirms that the number of successive tooth pairs in contact and the number of successive tooth pairs crossing the meshing zone without any contact between gear teeth vary according to the ratio of the excitation frequency to the rotation frequency.

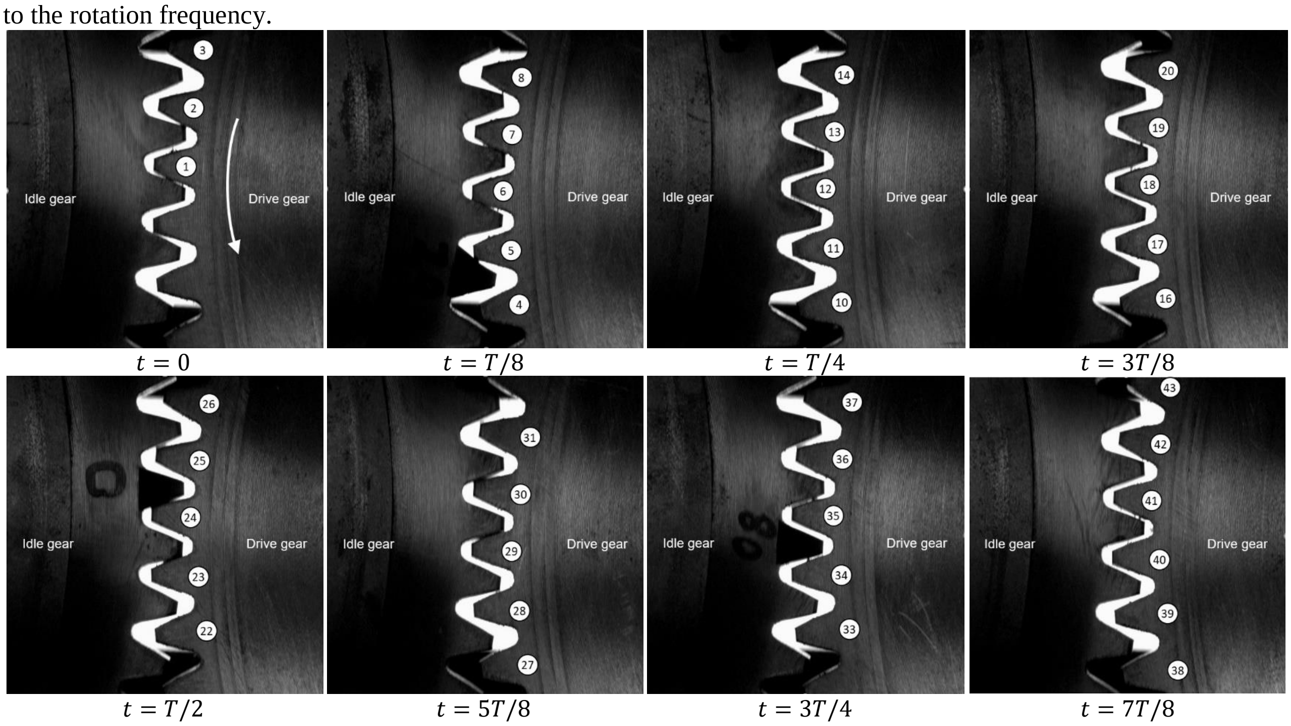

Figure 2: Sequence of images of the contact along a period of excitation.

The gear rattle dynamics is modelled with a SDOF bouncing ball with two moving walls excited by the velocity fluctuation. After adjustment of the restitution coefficient modelling the damping characteristics during impacts, a very good agreement between experimental and numerical results is observed. A slight difference is confirmed between active flanks impacts and reverse flanks impacts for which the impacting velocity is slightly lower. The squared impacting velocity transferred to the system, shows a linear relationship with the product $\omega\Delta\Omega$. It is proportional to the impacting kinetic energy which is entirely transferred to the system, because of the persistent contact observed after the impact. Taking account of the number of impacts per second proportional to ω, the assumption that the acoustic power generated by the successive impacts is proportional to the energy transferred to the system per second i.e. parameter $\omega^2\Delta\Omega$, is validated by rattle noise measurement. For low amplitude of parameter $\omega^2\Delta\Omega$, the successive impacts are clearly audible once they occur, but the sound pressure radiated from the system is mainly due to the gear whining noise. For larger amplitude of parameter $\omega^2\Delta\Omega$, the rattle noise induced by the successive impacts becomes the main source of acoustic nuisance.

References

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