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A PRELIMINARY NUMERICAL ANALYSIS OF THE NOISE RADIATED BY A GEARBOX

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ABSTRACT

The noise spectrum generated by a gearbox is characterised by harmonics which depend on the geometry of the gears, their rotational speed, possible axis eccentricity and imperfections of the teeth surface. The experimental investigation of the gearbox noise emission is a complex task, especially due to the difficulties in separating its contribution from the overall noise generated by other machineries of the testing train. Conventional laboratory tests evaluate the condition of full speed and no load, which is not representative of a realistic scenario, where the gearbox is subjected to a torque provided by the machinery it is connected to. This paper presents a numerical approach to predict the noise radiated by the gearboxes, through a multibody analysis in the time domain combined with a vibroacoustic analysis in the frequency domain using the boundary element method (BEM). The study investigates the influence of the rotational speed and the resisting torque on the radiated sound power. The results show that an increment in the rotational speed affects the position and magnitude of harmonic components, while an increment in the applied torque has a significant impact only on the magnitude of the radiated noise.

Keywords: *Gearbox, radiated noise, BEM simulations.*

1. INTRODUCTION

The gearboxes consist of one or more pairs of gears enclosed in a casing with the aim of transmitting a mechanical motion, typically to drive compressors, pumps and fans. Their noise spectrum is characterised by harmonics which depend on the geometry of the gears, their rotational speed, possible axis eccentricity and imperfections on the teeth surface [1]. In particular, the main tonal component is given by the meshing frequency, defined as the rate at which gear teeth mate together and expressed as the number of teeth of one gear times its rotational frequency. At this frequency, the amplitude of vibration is determined by the meshing stiffness periodic function, depending mainly on the transmitted load, the rotational angle and the geometry of the gears [2, 3]. The presence and the entity of the defects on the gears contact surface determine the magnitude of its multiple and submultiples harmonics. Even though the evaluation of the frequency of such components is a simple task, the experimental determination of the actual noise level emitted in-situ is not trivial. Gearboxes are used in transmission trains together with other several noise sources affecting the sound filed in the test facility. Thus, the recognition and the separation of the individual contributions is not always possible. The emitted noise is normally estimated in the laboratory, without an applied load, such as the one required by the element to drive. However, this operating condition is not representative of the in-situ working condition, since the torque provided by the machinery connected to the gearbox may significantly affect the vibrational level induced on the casing surfaces and thus the radiated noise. For this scope, some semi-empirical models can be found in the literature [4, 5], however they determine only the overall sound pressure level without considering the effect of the casing geometry.
This paper belongs to a preliminary work to determine an analytical formula for estimating in the design stage the noise produced in-situ by gearboxes. In particular, this article analyses the effect of the rotational speed and the resisting torque on the sound power emitted by a gearbox with a numerical approach coupling a time domain analysis of the mechanical system and a frequency domain analysis of the noise radiation into the free field, applying the previously calculated acceleration on an acoustic domain

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modeled with the boundary element method (BEM) [6, 7]. This approach has been chosen for saving computational load with respect to a finite element model (FEM), which would require the discretization of an air volume of adequate size with respect to the gearbox dimension [8]. Section 2 describes the model and the methodology used for the numerical analysis, while the results of the parametric studies are reported in Section 3. Finally, Section 4 illustrates the conclusions and the next steps of this study.

2. METHODOLOGY

The numerical analysis is composed of two distinguished steps: a time domain analysis of the mechanical system and a frequency domain analysis of the noise radiation.
The first step is aimed to evaluate the vibratory behaviour of the multibody model shown in Fig. 1. This system consists of a pair of spur gears, modeled as rigid bodies and with the geometric characteristics reported in Tab. 1, enclosed in an elastic casing with size 250 mm x 150 mm x 120 mm and thickness 3 mm. All the elements are made of steel with density $\rho = 7850$ kg/m^3, elastic modulus E = 200 GPa, Poisson's ratio $\nu = 0.3$ and damping factor $\eta = 5\%$. In terms of constraints, the two shafts are supported by the walls normal to the y-axis with two pairs of hinges which allow the rotation around their axes, while the case is fixed to the ground in four localized points. The contact between the two gears is described by means of a meshing stiffness periodic function, evaluated in a preliminary stage, taking into account the elasticity of the teeth. This function is shown in Fig. 2 and is computed from the application of a rotation to the pinion and a rotation plus a twist to the wheel through a well-known formulation valid for spur gears [2].
The vibrational behaviour is analysed for different rotational frequencies assigned to the pinion shaft and different torques applied to the wheel shaft, representing the resistance provided by the machinery connected to the gearbox. The duration and the time step of the time domain simulations are equal to 0.021 s and $3.5 \cdot 10^{-5}$ s respectively.
For the second step, the normal acceleration of the casing, responsible of the noise radiation into the free field, is converted into the frequency domain through Fourier transform and mapped onto the BEM model of the acoustic domain, represented by the external surface of the casing. The acoustic model is discretized with a maximum elements size equal to one fifth of the minimum wavelength (in this study associated with the maximum analysed frequency of 5000 Hz). The determination of the sound power radiated by the gearbox is done by integrating the sound pressure field on the upper hemisphere of a sphere with radius 2 m, since the lower hemisphere is not very meaningful in a practical application.

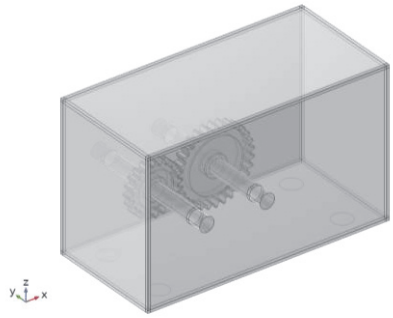

Figure 1. Geometry of the gearbox.

Table 1. Geometric characteristics of the gears.

	Pinion	Wheel
Teeth Number	20	30
Pitch Diameter	50 mm	75 mm
Pressure Angle	25°	25°
Width	10 mm	10 mm
Shaft Diameter	15 mm	15 mm
Teeth Height	4.5 mm	4.5 mm

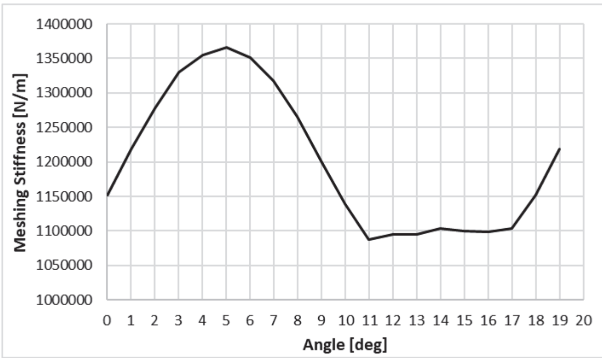

Figure 2. Meshing stiffness computed on one period.

3. RESULTS

3.1 Modal analysis

A modal analysis of the system is preliminarily carried out, aimed to know the frequencies of the modes which could interact with the meshing frequency. The modal frequencies are reported in Tab. 2, showing the first modes up to maximum analysed meshing frequency (2000 Hz in this study). This preliminary analysis allows to justify the presence of some peaks in the spectra of the successive parametric analyses.

Table 2. Frequencies of the first vibrational modes.

Mode	1	2	3	4	5	6
Frequency [Hz]	614	1202	1206	1493	1873	2059

3.2 Variation of the resisting torque

The sound power spectra computed for different resisting torques are reported in Fig. 3 for a rotational frequency of the pinion equal to 100 Hz (6000 rpm). In the graph, the tonal component related to the meshing frequency (2000 Hz for a rotational frequency of 100 Hz) emerges from the spectrum and is the main responsible of the global sound power level. As it can be observed, the increment of the torque corresponds to a proportional increment of the sound power amplitude on the whole spectrum. Similar results can be found for different rotational frequencies.

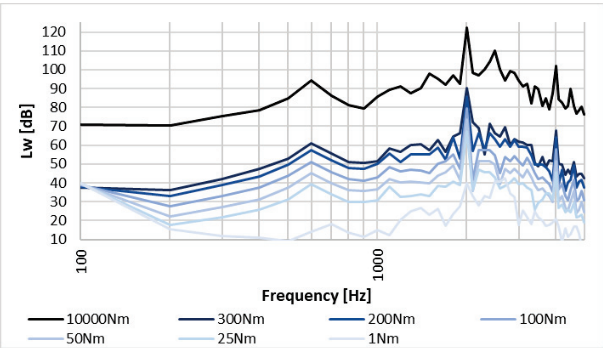

Figure 3. Sound power spectrum for different torques.

3.3 Variation of the rotational speed

In Fig. 4, the sound power spectra for different rotational frequencies of the pinion are shown for a torque equal to 300 Nm. In this case, the peak associated with the meshing frequency moves along the spectrum and, with respect to the previous analysis, its amplitude has not a direct proportion with the rotational frequency, due to the interaction with the casing modes. In particular, comparing the meshing frequencies with the frequencies of the first vibrational modes reported in Tab. 2, it can be observed that the meshing frequency associated with the rotational frequency of 60 Hz excites the second and the third modes, while the one associated with the frequency of 100 Hz excites the sixth mode. As a consequence, also the relationship between the global sound power level and the rotational frequency is not directly proportional, since the meshing frequency represents the main contribution to the global level. Similar conclusions can be drawn for different torques.

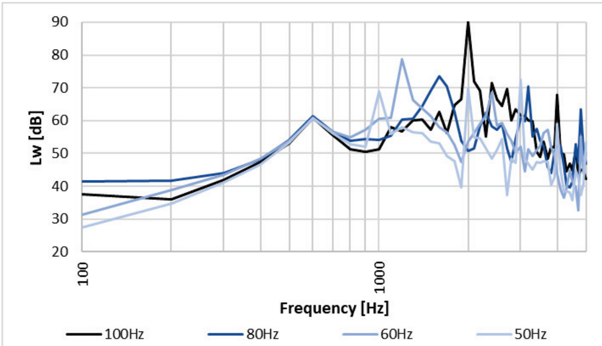

Figure 4. Sound power spectrum for different rotational speeds.

4. CONCLUSIONS

In this article, a numerical analysis of the effect of the resisting torque and the rotational speed on the noise radiated by a gearbox is illustrated. In particular, a direct relationship between the sound power level and the resisting torque has been found, for which an increment of the torque corresponds to a proportional increment on the whole power spectrum. Moreover, it has been demonstrated that the sound power level depends on the interaction between the meshing frequency and the modal frequencies, causing a not directly proportional relationship. This type of approach allows to overcome the problems related to the lab testing, in which a condition of no load is typically analysed, neglecting the real in-situ conditions with the torque provided by the element to drive. In the next steps of this development, parametric analyses varying the geometry of the gears and the casing will be carried out, aiming to establish an analytical formulation for the prevision of the sound power level emitted in-situ by the gearboxes. Finally, some experimental measurements will be carried out for the validation of the analytical formulation.

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