DOI: 10.61782/fa.2025.0867

FORUM ACUSTICUM EURONOISE 2025

THE INFLUENCE OF HELIX ANGLE AND GEAR WIDTH ON THE NOISE RADIATED BY A GEARBOX

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ABSTRACT

The noise spectrum generated by a gearbox depends on multiple factors, such as gear design parameters, casing geometry, rotational speed of the shafts and the presence of imperfections. This numerical study analyses the influence of the helix angle and gear width on the noise radiated by a gearbox through parametric studies. The numerical investigation combines a time-domain analysis on a multibody model of the gearbox with a frequency-domain analysis on a boundary element model of the air to evaluate noise radiation in an open space in free-field conditions. The simulations assume a constant workload and a constant rotational speed of the shafts. The results indicate that increasing the helix angle and gear width leads to a smoother noise spectrum and modifies its shape. Compared to spur gears, helical gears generate lower sound power levels due to a reduced variation in the meshing stiffness.

Keywords: *gearbox, noise radiation, helical gears, numerical prediction.*

1. INTRODUCTION

Gearboxes, essential components of mechanical transmission systems, typically consist of one or more pairs of gears enclosed in a casing. They are widely employed in applications such as compressors, pumps and fans to regulate motion and torque. Acoustically, gearboxes are often part of complex noise generation systems, in which multiple noise sources interact, making it difficult to isolate and identify individual contributions. As a result, experimentally evaluating the noise generated by gears meshing and radiated in the surrounding fluid by the casing in real operating conditions is typically challenging.

The noise spectrum radiated by a gearbox depends on various factors, including gear geometry, rotational speed, axis misalignment, and surface imperfections [1,2]. In a previous study, the effect of the workload was assessed for a pair of spur gears [3]. These gears exhibit a distinct harmonic component at the meshing frequency, which is associated with significant variations in the meshing stiffness throughout the meshing cycle. This study employed a coupled numerical approach to evaluate the radiated noise: a time-domain multibody analysis on a finite element (FE) model of the gearbox to determine the dynamic response of the casing and a frequency-domain acoustic analysis on a boundary element (BE) model of the semi-infinite fluid domain to determine noise radiation in free-field conditions [4].

This paper investigates the influence of the helix angle and gear width on the noise radiated by a gearbox employing the same numerical method. Section 2 describes the numerical model used for the study and outlines the parametric analysis conducted to assess the effects of helix angle and gear width. Section 3 presents the results of this parametric analysis. First, the influence of these two parameters on the meshing stiffness function is analyzed. Then, their effects on the sound power spectrum and overall sound power level are examined. Finally, Section 4 discusses the key findings of this study and the potential follow-up for future research.

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2. METHODOLOGY

The parametric analysis to assess the influence of the helix angle and gear width is structured into three main stages. The first stage evaluates the meshing stiffness of helical gears. For this purpose, a helical gear of width L is assumed to be composed of a series of spur gears with an infinitesimal width dz, each rotated by a small angle determined by the helix angle β. Specifically, the meshing stiffness $K_{G,H}$ of the helical gears as a function of the rotational angle θ is determined as:

$$K_{G,H}(\theta) = \int_0^L K_{G,S}\left(\theta + \frac{z}{R}\tan\beta\right)\frac{dz}{L_0}, \quad (1)$$

where R represents the pinion pitch radius and $K_{G,S}(\theta)$ is the meshing stiffness of the spur gears with the same cross-section as the helical gears. This function is computed from a multibody simulation in which the spur gears of width L_0 are treated as elastic bodies in contact: the pinion rotates of a certain angle θ, while the wheel rotates of the same angle plus an additional twist $\theta + \mathrm{d}\theta$.

Once the meshing stiffness $K_{G,H}(\theta)$ is determined, the second stage consists of a time-domain analysis on the multibody model shown in Fig. 1. This model consists of two helical gear pairs, treated as rigid bodies with geometrical characteristics detailed in Tab. 1. The helix angle β and the gear width L are varied as part of the parametric study. The elasticity of these elements is incorporated into the meshing stiffness function $K_{G,H}(\theta)$ previously determined. The steel casing measures 250 mm x 150 mm x 120 mm, with a thickness of 3 mm and is modelled through isotropic linear elastic 3D elements, with an elastic modulus $E = 200\,\text{GPa}$, Poisson's ratio $\nu = 0.3$, damping factor $\eta = 5\%$ and density $\rho = 7850\,\text{kg/m}^3$. The casing is assumed to be fixed to the ground at four localized points, while the two shafts are supported by the walls perpendicular to the y-axis through two pairs of hinges, which allow rotation around their respective axes. The analysis considers a pinion rotational frequency of $f_p = 100\,\text{Hz}$ and a resisting torque of $T = 100\,\text{Nm}$ applied to the wheel. The duration and the time step imposed in the time-domain simulations are set to 0.021 s and $3.5 \cdot 10^{-5}$ s respectively.

In the final stage, the normal acceleration computed on the casing, which is responsible for the noise radiation in the free field, is transformed into the frequency domain using a Fourier transform. The resulting acceleration field is mapped onto a BE model representing the infinite acoustic domain, which, in this study, is bounded by the source boundaries, corresponding to the external surface of the casing. The acoustic model is discretized with a maximum element size equal to one-fifth of the shortest wavelength, which, in this study, is associated with the highest analyzed frequency of 5000 Hz. The radiated sound power is determined by integrating the sound intensity field over the upper hemisphere of a sphere with a radius of 2 m, enclosing the noise source.

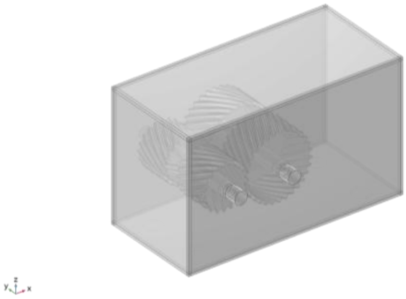

Figure 1. Geometry of the gearbox employed in numerical simulations.

Table 1. Geometric characteristics of the gears.

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

3. RESULTS

3.1 Meshing stiffness analysis

In the first stage, the meshing stiffness was evaluated for helix angles ranging from 0° (spur gears) to 40° and for gear widths varying between 10 mm and 50 mm. Fig. 2 shows the meshing stiffness function over the meshing cycle for different helix angles. The increase of the helix angle reduces the oscillation of this function. Similar conclusions can be deduced for the increase of the gear width. Fig. 3 illustrates the normalized amplitude difference Δ_n for different helix angles and gear widths, defined as:

$$\Delta_n = \frac{K_{G,H,max} - K_{G,H,min}}{K_{G,H,avg}} \cdot 100, \quad (2)$$

where the $K_{G,H,max}$, $K_{G,H,min}$ and $K_{G,H,avg}$ represent the maximum, minimum and average value of the meshing stiffness over the meshing cycle, respectively. For spur gears, the gear width acts as an identical scaling factor for all three values (maximum, minimum and average). Thus, the normalized amplitude difference Δ_n does not depend on the gear width. In contrast, for helical gears, the effect of helix angle and gear width are combined. As both parameters increase, the normalized amplitude difference Δ_n progressively decreases. This trend can be attributed to the fact that, compared to spur gears, helical gears have more teeth simultaneously engaged in meshing at any given instant, leading to a smoother and more continuous load distribution.

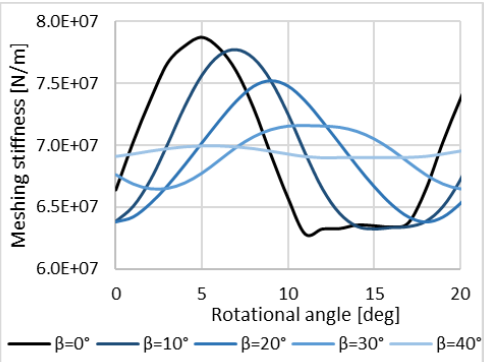

Figure 2. Meshing stiffness for different helix angles.

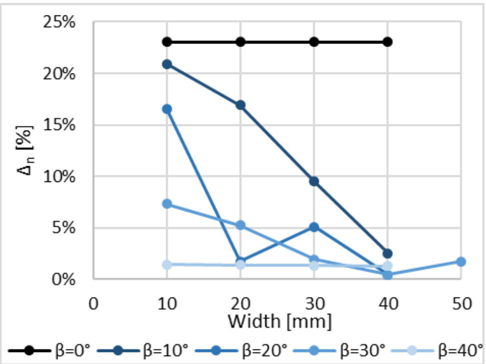

Figure 3. Normalized amplitude difference of meshing stiffness.

3.2 Effect of the parameters on the spectrum

Fig. 4 presents the sound power spectra for various helix angles, keeping the gear width constant at 30 mm. As the helix angle increases, the amplitude of the meshing frequency component decreases. This trend suggests that a larger helix angle results in smoother meshing between the gear teeth, reducing the impact of this tonal component. As a result, a significant reduction of the global level is observed, since the component associated with the meshing frequency represents the main contribution to the overall sound power level. On the other hand, there is a relevant increase in noise levels at lower frequencies, indicating a redistribution of energy across the spectrum. This shift may be due to changes in the dynamics of the gear meshing as the helix angle varies, affecting how vibrations are transmitted to the casing.

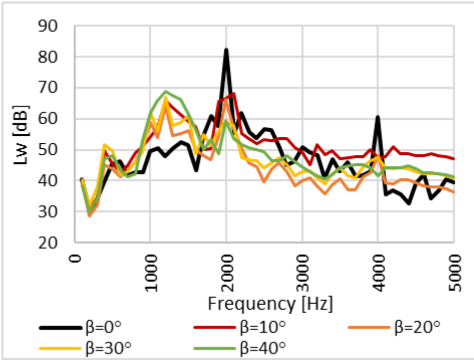

Figure 4. Sound power spectrum for different helix angles (width L = 30 mm).

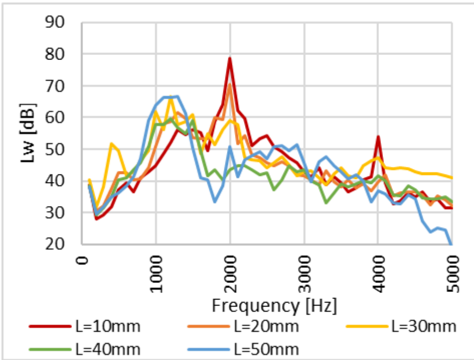

Figure 5. Sound power spectrum for different gear widths (helix angle $\beta = 30°$).

Fig. 5 shows the sound power spectra for different gear widths, while maintaining constant the helix angle value, $\beta = 30°$. Similar to the previous case, the increase in gear width provides a reduction in the amplitude of the meshing frequency component. This reduction can be attributed to a smaller amplitude difference in the meshing stiffness, leading to less pronounced fluctuations in the force transmitted between the gear teeth. Additionally, there is a noticeable increase in noise levels below the meshing frequency, suggesting that as the gear width increases, more energy is distributed into lower frequencies, possibly due to changes in the contact mechanics and in the overall stiffness of the system.

3.3 Effect of the parameters on the overall level

The overall noise level as a function of the normalized amplitude difference Δ_n is shown in Fig. 6. This parameter accounts for the combined effects of both helix angle and gear width, providing a comprehensive measure to assess their influence on noise levels radiated by the gearbox. As shown in this figure, an increase in the normalized amplitude difference results in a higher sound power level radiated by the gearbox. Thus, larger variations in the meshing stiffness lead to an amplification of the generated noise. A non-uniform gear meshing, for which the helix angle and the gear width play an important role, will thus lead to higher vibration amplitude or more significant impacts during the meshing cycle.

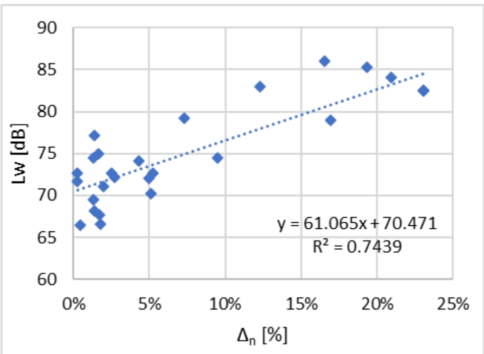

Figure 6. Correlation between the sound power level and the normalized amplitude difference of the meshing stiffness.

4. CONCLUSIONS

In this article, a numerical analysis is presented to investigate the influence of helix angle and gear width on noise radiated by a gearbox. In particular, the study focuses on how these two parameters influence the shape of the meshing stiffness function, which plays a crucial role in the acoustic behavior of the system. This function can be used as a control variable to assess and optimize the influence of helix angle and gear width on the overall noise level. A direct relationship between the sound power level and the normalized amplitude difference of the meshing stiffness was identified. This relationship shows that a larger variation in the amplitude of the meshing stiffness corresponds to a proportional increase in the overall sound power level. Since the meshing stiffness variation is significantly influenced by gear geometrical characteristics such as helix angle and gear width, an optimal design of these parameters can provide a reduction of noise radiated by the gearbox. Additionally, this study revealed that increasing either the gear width or the helix angle leads to a reduction in the amplitude of the meshing frequency component. However, this reduction also increases radiated noise at lower frequencies, suggesting a redistribution of the vibrational energy across the spectrum.

Through this analysis, a clear correlation between the key design parameters of the gears and the sound power level is established. Future research should aim at conducting experimental measurements to validate the trends observed in the numerical analysis. Furthermore, the effects of other geometrical factors and working parameters should be explored.

5. ACKNOWLEDGMENTS

This work was supported and financed by the collaboration with Nuovo Pignone Tecnologie (Baker Hughes).

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