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Gear sound model for an approach of a Mechanical Acoustic Vehicle Alerting System (MAVAS) to increase EV's detectability

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ABSTRACT

Hybrid-electric and electric vehicles significantly reduce noise road emissions. This noise mitigation also causes a reduction in the sound detectability and therefore it increases the potential of causing accidents. A suitable solution arises with the Acoustic Vehicle Alerting Systems (AVAS) emitting a warning sound to alert pedestrians about the presence of a silent vehicle. This paper details an acoustic prediction model capable of simulating the sound produced by a pair of spur dry gears used as a Mechanical Acoustic Vehicle Alerting System (MAVAS). This proposal that tries to reproduce a sound closer to the mechanical sound of a conventional vehicle would be used as an alternative to existing systems. The prediction model developed is validated and consists in two consecutive parts: first, a dynamic model studies the rattle of the gears, then, an analytical model reproduces the sound of each impact of the gear teeth. This sound model makes it possible to characterize a proposed gear combination of the MAVAS, verifying its compliance with the European legislation.

1. Introduction

Road traffic is the main cause of urban acoustic pollution in our cities and the noise source that affects the largest number of people [1]. Hybrid-electric (HEVs) and electric (EVs) vehicles would not only avoid annoying emissions of polluting gases in urban environments but they would also minimize, as well as reducing it entirely, the mechanical noise sources of the vehicle and therefore eliminating its noise contribution in many of its operating conditions.

This situation, ideal and desired for a long time, has become a series of inconveniences derived from such an absence of sound. On the one hand, the number of accidents caused by hybrid vehicles, operating in electric mode, regarding the conventional combustion vehicle, has increased [2], on the other hand, the absence of mechanical noises (associated with the combustion engine and its transmission system), as well as the presence of background noise in different urban environments, could mask the low noise produced by this type of vehicle and constitute a risk to pedestrians [3]. Some studies [4] determine that a hybrid vehicle runs twice the risk of causing an accident when it is driven using only the electric drive system than a conventional vehicle powered by a combustion engine. Consequently, Acoustic Vehicle Alerting Systems (AVAS) was proposed to increase their detectability [5–9]. These systems emit warning sounds to alert cyclists and pedestrians about the path of a silent vehicle, an example of these systems is Nissan's VSP technology [10]. However, the random use of this type of sounds does not solve the problem, since they can be inappropriate, excessive and annoying [3]. In January 2017 the regulation CEPE No. 138 [11] was published after the appearance of different kinds of warning systems. This regulation has the purpose of limiting the sound emissions: establishing minimum and maximum sound levels and confining the frequencies allowed.

AVAS and warning sounds have been widely studied in recent years from different aspects such as minimizing noise pollution through directivity [12], studying the effectiveness of the sound signal according to the urban environment [13] or analysing their annoyance [14]. The studies about the perception of these sounds in especially vulnerable population are also remarkable [7,15,16].

Some results from the EVADER project [15] show that the detection distance of a combustion engine vehicle by pedestrians is 36 m, whereas in the case of an electric vehicle equipped with a boarded warning system the detection distance is reduced to 18 m. In that study, the driving condition considered was an accelerating vehicle approaching to the pedestrian from 50 m. Accelerating pass-by of 5 s were used to reproduce this situation. The results of this study revealed that the synthesized warning sounds based on the engine speed obtained a reaction time similar to combustion vehicles despite being 7 dB quieter.

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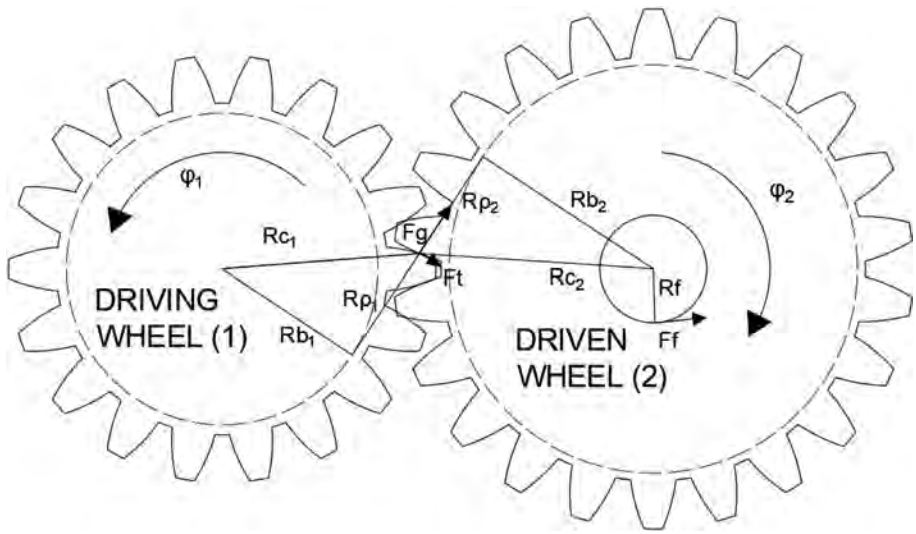

Fig. 1. Gear pair dynamic model.

The use of alternative devices to speaker arrays has also been studied: In [17], the speaker array is replaced by a single speaker attached at the end of a perforated pipe. This pipe allows emitting the warning sound in a directional field. Another instance is proposed in [18], where an array of inertial actuators is used instead of loudspeakers.

Mechanical noise generation models allow to analyse the sound source that can be used for alternative warning systems development: In [19] the sound prediction model of a drive chain is exposed. The noise produced by the interaction between the chain rollers and the sprocket teeth is modelled as the sound produced by the vibration of the chain roller considering it as cylinders. The model has a finite element part which is used to study the dynamics of the drive chain and a numerical part for the sound production.

A model of a gear pair sound emissions was published in [20], where the sound emissions are produced by the irregularities on the surface of the gears. The results are mainly focused on the global sound pressure level.

Several studies about vibrations in gearboxes [21–24] have been published from the Wolfson School of Mechanical and Manufacturing Engineering. In [25], the sound pressure produced by lubricated cylindrical spur gears is determined as a function of time. This model equals the sound produced by the clash of a pair of teeth to the clash of two cylinders.

The prediction of the sound produced by the clash of two cylinders is based on the work presented in [26]. In it, the equations of the sound pressure produced in a clash of two cylinders depending on their impact velocity in a determinate coordinate are presented. The results of both [25] and [26] are validated satisfactorily in a test bench.

This paper describes the earliest phases of developing a new kind of boarded device to improve the perception of silent vehicles by using gears as a sound source, a Mechanical Acoustic Vehicle Alerting System (MAVAS). The main concept behind this new system is its capability of emitting mechanical sounds closer to conventional vehicles than those produced by electronic warning systems.

In the manuscript, it is also established a sound prediction model for spur dry gears, which allows the study of the sound emitted by different combinations of gears as a MAVAS. An initial schematic of the noise prediction analytical model was presented in 2017 by the authors [27]. Later, in 2019, a possible gear configuration was presented to be used as MAVAS [28]. The sound prediction model permits the selection of the gear combination according to the detectability regulations.

2. Model

The implementation of the sound prediction model is divided into two consecutive parts clearly differentiated. On the one hand, a dynamic model that studies the motion of the driven wheel to determine its rattle movement has been developed. On the other hand, the acoustic generation model, which successively produces the noise of the gear teeth that are impacting during the rattle movement has been presented.

2.1. Dynamic model

The scheme of the free gear system is shown in Fig. 1. The driven wheel is mounted on a ball bearing. The motion of the driving wheel φ_1 is imposed so the system has only one rotational freedom degree φ_2 in the driven wheel.

The equation of motion for the driven wheel is shown in Eq. (1):

$$I_2\ddot{\varphi}_2 = F_gRb_2 - F_tR\rho_2 - F_fR_f \tag{1}$$

where I_2 is the inertia of the driven wheel,$\ddot{\varphi}_2$ is the angular acceleration of the wheel. F_g is the gear meshing force while its lever arm is

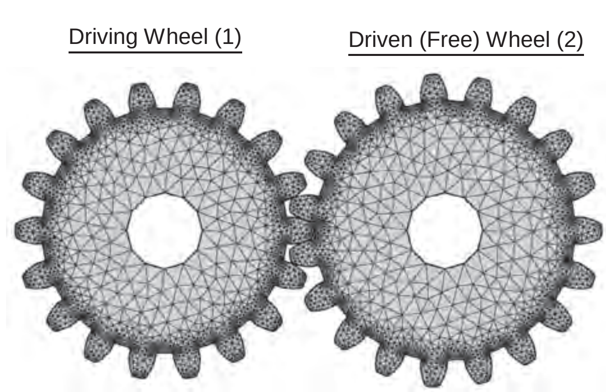

Fig. 2. Gear pair meshed in FEM software.

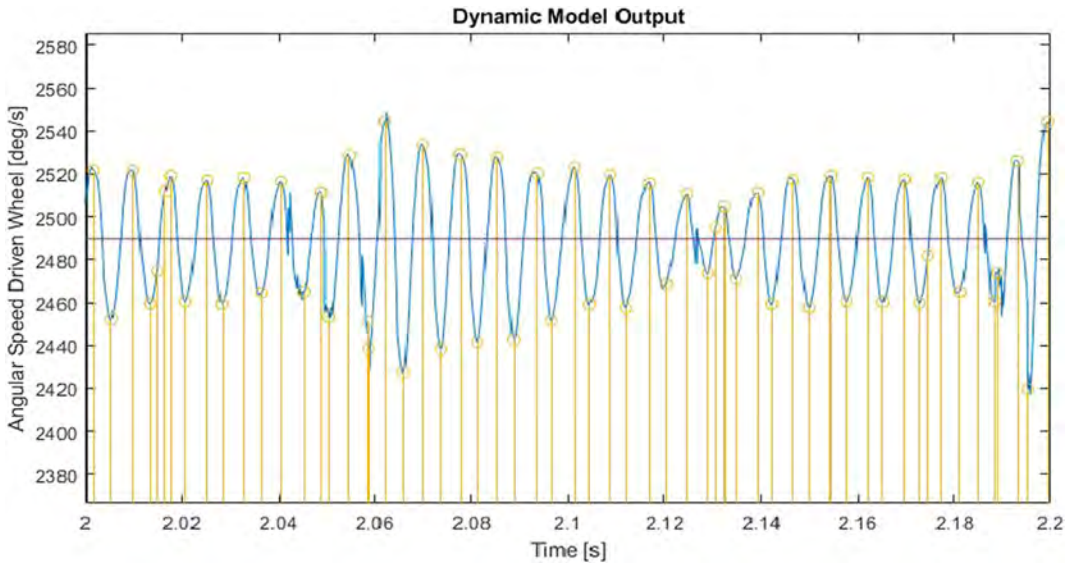

Fig. 3. Output of the Dynamic Model.

the base radius Rb_2. Ff is the frictional force in the bearing which is the only charge in the loose gear pair and Rf is the application radius of the force. Ft is the friction due to the sliding and $R\rho_2$ is the curvature radius of the involute in that point.

The dynamics of the gear pair is solved using a finite element software: The driving wheel has an imposed rotational movement of speed $\dot{\varphi}_1$. The driven wheel has friction on its axle and it is modelled as an elastic solid. The driving wheel is a rigid solid since the model is used to study the rattle movement and not the stresses produced during the gear process. As well, these simulation conditions are suitable for the free load situation of the gear system proposed in this document. Considering one of the solids as rigid and the other as elastic makes calculation time more efficient. The profiles of the teeth of both wheels are modelled as a contact with friction. As shown in Fig. 2, meshing is more detailed on the tooth profiles than on the inside of the wheel to improve contact calculation.

Fig. 3 shows an example of the output of the dynamic model, where the angular output speed of the driven wheel $\dot{\varphi}_2$ is seen oscillating in a rattle movement around the corresponding value given by its transmission ratio. The points marked on the curve show the instant where an impact occurs between the gear teeth. The data of these points is used as an input of the analytical sound model.

2.2. Acoustic model

The fundamental of the model is based on previous researches [24], where those approaches determine the sound produced by a lubricated free gear pair. In the present study, the following shows how the noise produced by the impact of a pair of dry teeth is obtained.

To simulate the clash of the teeth, the gears are simplified as a system of two impacting cylinders [25] which vary in time depending on the gear mesh position. Each radius of these cylinders is given using Pythagoras theorem in Eq. (2).

$$Rc_i = \sqrt{{Rb_i}^2 + {R\rho_i}^2} \tag{2}$$

where Rb_i is the base radius of the pinion or the wheel, and $R\rho_i$ is the involute curvature radius as it is shown in Fig. 1.

To reduce computational cost and simplify the programming, the sound produced by a pair of impacting cylinders can be equated as an equivalent cylinder impacting against a semi-infinite elastic half-space [25]. The equivalent radius is given by Eq. (3) where Rc_1 and Rc_2 are the radii of each cylinder in any instant.

$$Rc_{eq} = \frac{Rc_1 Rc_2}{Rc_1 + Rc_2} \tag{3}$$

The mass of the equivalent cylinder depends on the flank width l and the material density ρ as is shown in Eq. (4).

$$m_{eq} = \pi {Rc_{eq}}^2 l\rho \tag{4}$$

k_1 and k_2 parameters are used to simplify later notation as it is described in [26], where the clash of two cylinders is presented. In the case of the present model, the equivalent cylinders to the clash of the gear teeth change at every moment. Thus, using the initial parameters of the teeth contact is considered.

k_1 and k_2 are shown in Eq. (5).

$$k_1 = \frac{1}{m_{10}} + \frac{1}{m_{20}} \tag{5.a}$$

$$k_2 = \frac{2}{3}\left(2\frac{1-\mu^2}{E}\right)^{-1}\left(\frac{1}{rc_{10}} + \frac{1}{rc_{20}}\right)^{-1/2} \tag{5.b}$$

where m_{i0} is the mass of each cylinder whose radius is the contact radius rc_{i0} (subindex i belongs to the driving wheel when it equals to 1 and to the driven wheel when it equals to 2), both evaluated at

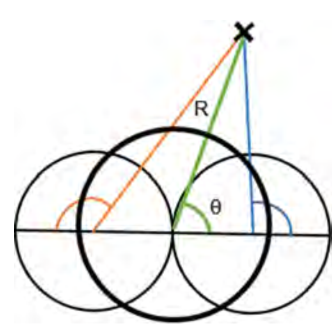

Fig. 4. Coordinate system of the equivalent cylinder.

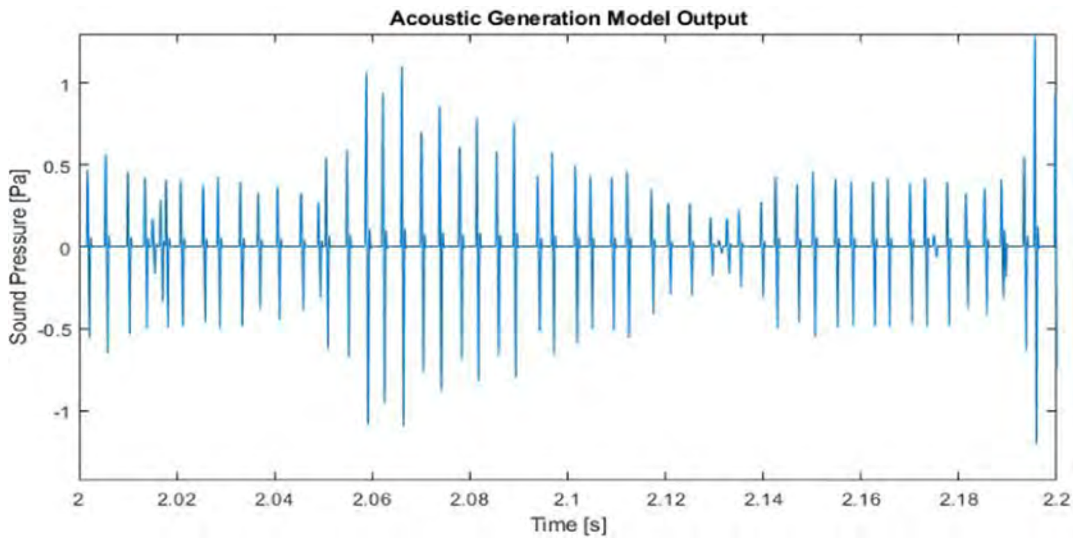

Fig. 5. Output of the Acoustic Model.

the initial geometric data. E is Young's modulus and μ is Poisson's ratio.

v_0 is the velocity in that equivalent cylinder impact. In this model, v_0 is considered as the normal velocity at the teeth surface when the impact occurs. The result is shown in Eq. (6), where $\dot{\varphi}_1$ and $\dot{\varphi}_2$, provided by the dynamic model, are the angular speed when an impact occurs.

$$v_0 = \dot{\varphi}_1 Rb_1 - \dot{\varphi}_2 Rb_2 \tag{6}$$

d_m, is the maximum deformation reached during impact [26] and it is calculated in Eq. (7).

$$d_m = \left(\frac{5}{4k_1k_2}\right)^{2/5} {v_0}^{4/5} \tag{7}$$

While F_m, is the maximum impact force [26] and it is calculated in Eq. (8).

$$F_m = k_2\left(\frac{5}{4k_1k_2}\right)^{3/5} {v_0}^{6/5} \tag{8}$$

Contact time is defined as twice as the time needed by the equivalent cylinder to reach the maximum deformation. In the case of dry gears, it should be calculated according to Hertzian theory detailed in [26]. Contact time t_c is calculated in Eq. (9)

$$t_c = \int_0^{d_m} \frac{\mathrm{d}d_m}{\sqrt{{v_0}^2 - \frac{4}{5}k_1k_2{d_m}^{3/2}}} = 2.94\frac{d_m}{v_0} \tag{9}$$

Contact frequency ω_c is calculated as $\omega_c = \pi/t_c$.

Eq. (10) shows the sound pressure p as a function of time t in a position of known coordinates R y θ [26]. Fig. 4 shows the coordinate system of the equivalent cylinder with radius Rc_{eq} superposed to the two cylinders with radii Rc_1 *and* Rc_2. These cylinders, which are impacting each other, are centred over the gear pair.

The estimation of A, B, D, E, F, G, H, X, Y, l_1 and l_2 parameters could be consulted in [26], where acceleration a is given by Newton as: $a = F_m/m_{eq}$. The sound radiation model of [26] considers the presence of two solids as equivalent cylinders, but therefore does not take into account the diffraction in geometric details such as gear teeth.

Fig. 5. shows an example of an output of the acoustic model where the data of Fig. 3. is used as an input. For each of the impacts marked in Fig. 3, the sound emitted by the collision of a pair of

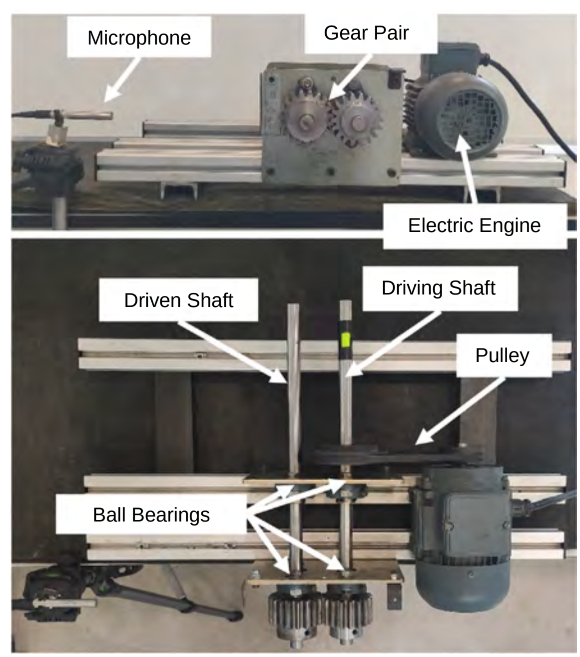

Fig. 6. Experimental set-up, front and top images.

$$p(R,\theta,t) = A\left(B\cos(\omega_c t) + D\sin(\omega_c t) + E\cos(l_1 t)e^{-l_2 t} + F\sin(l_1 t)e^{-l_2 t}\right) \quad \text{For} \quad 0 \le t \le t_c$$
$$p(R,\theta,t) = A\left\{-G\cos((\omega_c + l_1)t_c - l_1 t) - H\sin((\omega_c + l_1)t_c - l_1 t) + X\cos((\omega_c - l_1)t_c + l_1 t) + Y\sin((\omega_c - l_1)t_c + l_1 t)\right] * e^{-l_2(t-t_c)} + \left[E\cos(l_1 t) + F\sin(l_1 t)\right]e^{-l_2 t}\right\} \quad \text{For} \quad t > t_c \tag{10}$$

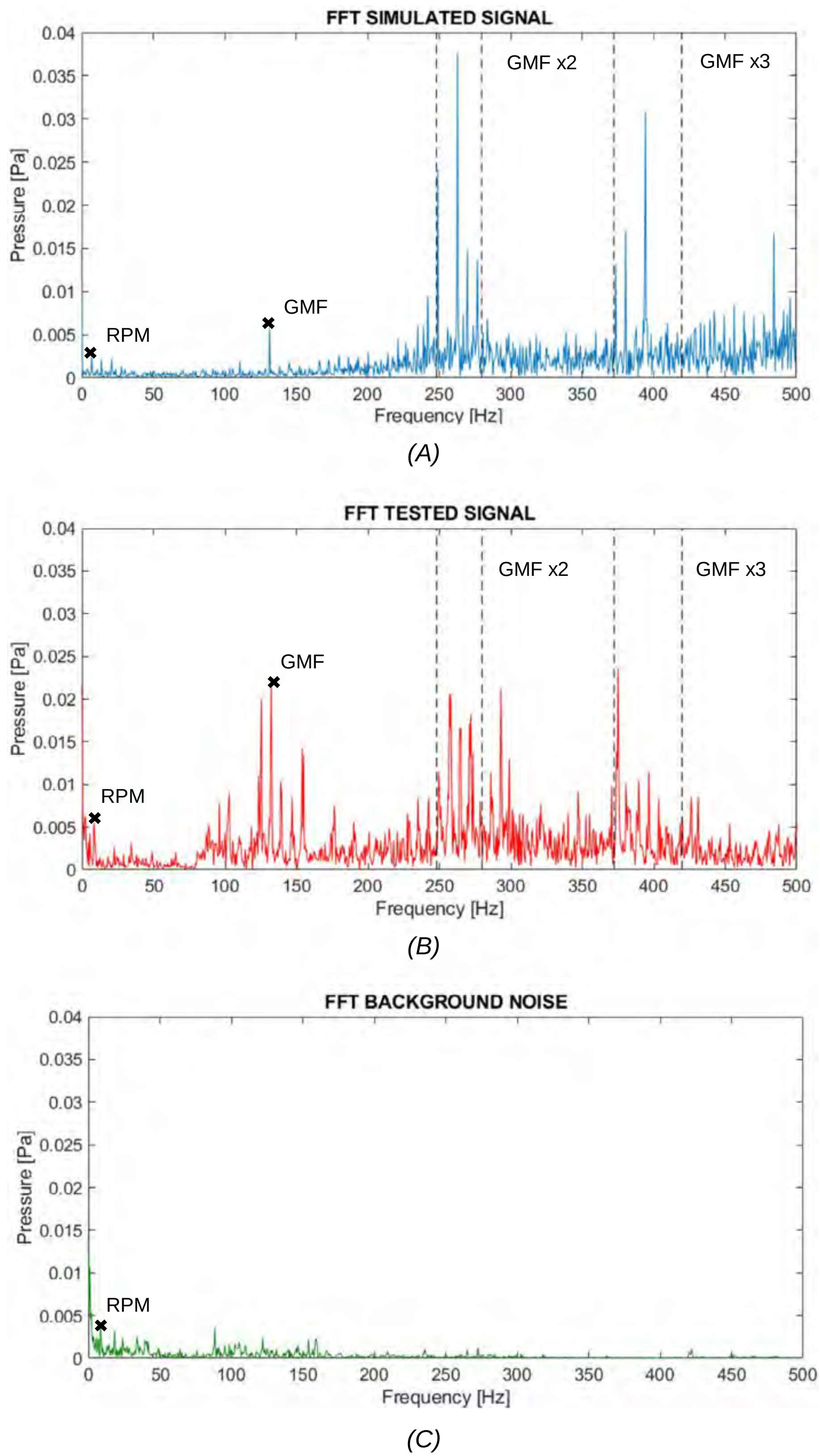

Fig. 7. FFT of the simulated sound (A), FFT of the tested sound (B) and FFT of the background noise (C) with 18 teeth driving wheel at 45.87 rad per second and 19 teeth driven wheel gear pair at R = 0.6 m and θ = 0 rad.

Table 1
Gear pair testing facilities.

Driving wheel	Module 4, 18 teeth, steel, 4 cm thickness
Driven wheel	Module 4, 19 teeth, steel, 4 cm thickness
Microphone	GRAS 40AE with GRAS 26CA preamp
DAQ system	NI cRIO-9233 with NI USB-9162 Chasis
Acoustic calibrator	B&K 4231 type
Tachometer	Manual PCE-151

teeth is generated by the acoustic model, thus obtaining the acoustic pressure signal in function of time.

3. Experimental SET-UP

The assembly used to validate the sound prediction is shown in Fig. 6. It consists of a gear pair driven by an electric engine with a transmission pulley of 2 to 1. The gear pair is mounted on shafts fastened on ball bearings, so the gears could roll free of load. The system is powered by a variable frequency drive.

A microphone in line with the gear pair (θ coordinate of the model equal to zero) through a DAQ system measures the sound signal. The set-up has the gear pair cantilevered to reduce sound reflection. In [26] the sound produced by the collision of two cylinders was validated, both in the aligned and vertical position. The aligned positioning of the microphone shown in Fig. 6. is determined for practical assembly reasons, additionally, the aligned location of the microphone allows to replicate the layout of the following approaching proposal of the MAVAS.

Data collection and microphone calibration are performed using a LabVIEW script. The speed of the driving shaft is measured by a manual tachometer.

Additional information about the gear pair and the measurement equipment can be consulted in Table 1.

4. Validation

The sound produced by the pair of gears of Table 1 was measured in the experimental set-up when the driving wheel is rotating at 7.3 Hz. The background noise was also measured under the

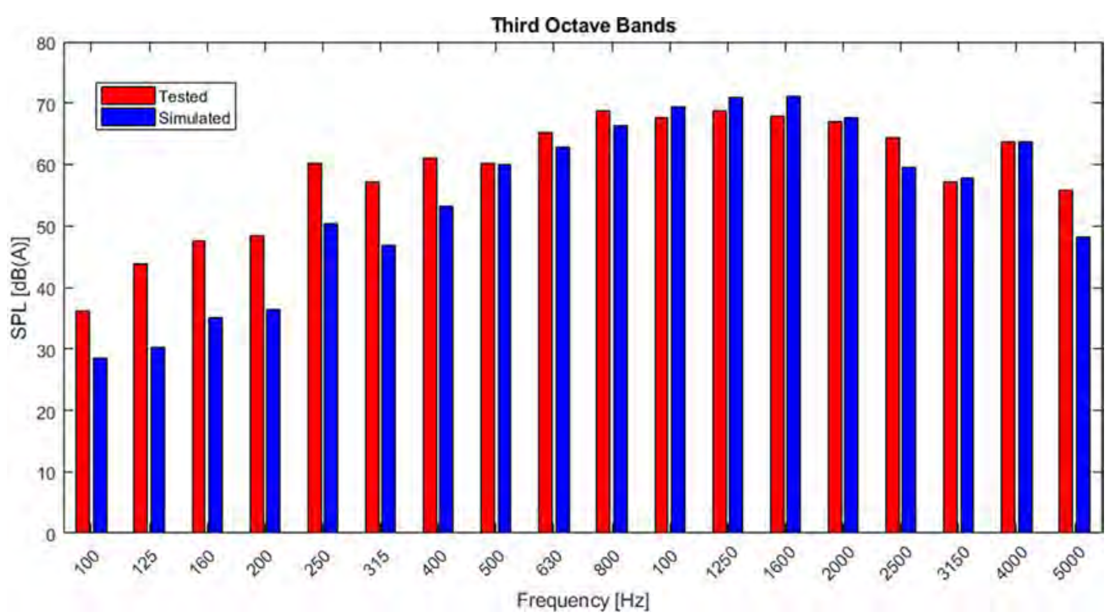

Fig. 8. Comparison between simulated and tested signals in third octave bands.

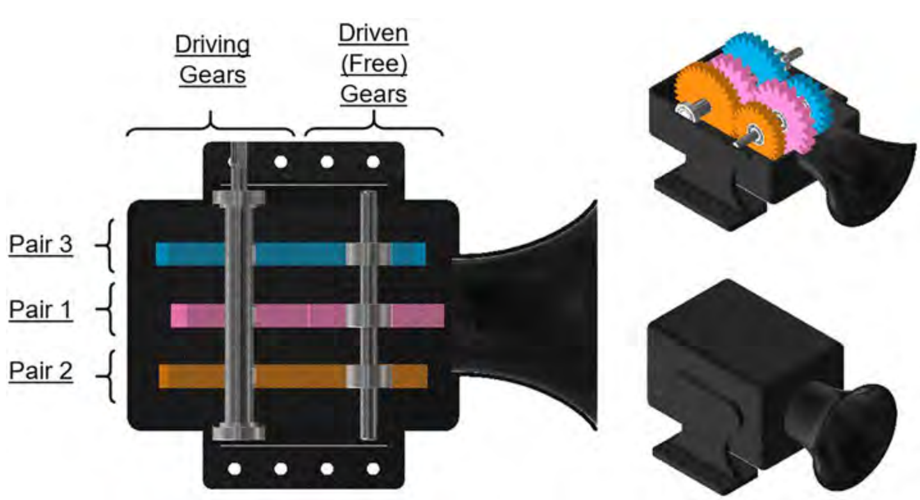

Fig. 9. Sketch of the possible design of the proposal.

Table 2
Gear Pairs module and number of teeth.

Gear Pair	Module (mm)	Driving Wheel Teeth	Driven Wheel Teeth
1	5	17	19
2	3	35	25
3	4	27	18

same conditions: letting the driving shaft roll but removing the gears from the set-up.

The microphone is located aligned to the gears and centred in their thickness. The distance between the microphone and the pitch point of the gears is 60 cm, so the local coordinates of the model are $R = 0.6$ m and $\theta = 0$ rad.

A pair of gears with the same geometric characteristics was simulated using the developed model. The dynamic model has been run under the same constant speed conditions for the driving wheel. The movement data of the driven wheel was entered into the acoustic model to obtain the pressure signal over time at the same coordinates as the microphone position.

The FFT shown in Fig. 7. is performed to make the comparison between simulated and tested signals at low frequency, where specific information of the mechanical gear operation can be appreciated. It is also included the frequency spectrum of the background noise that was measured with the electric engine running without gears. The first pointed peak around 7 Hz corresponds to the rotation speed of the driven and driving wheel (called *RPM* peak). The *RPM* peak of the driving shaft can be distinguished in

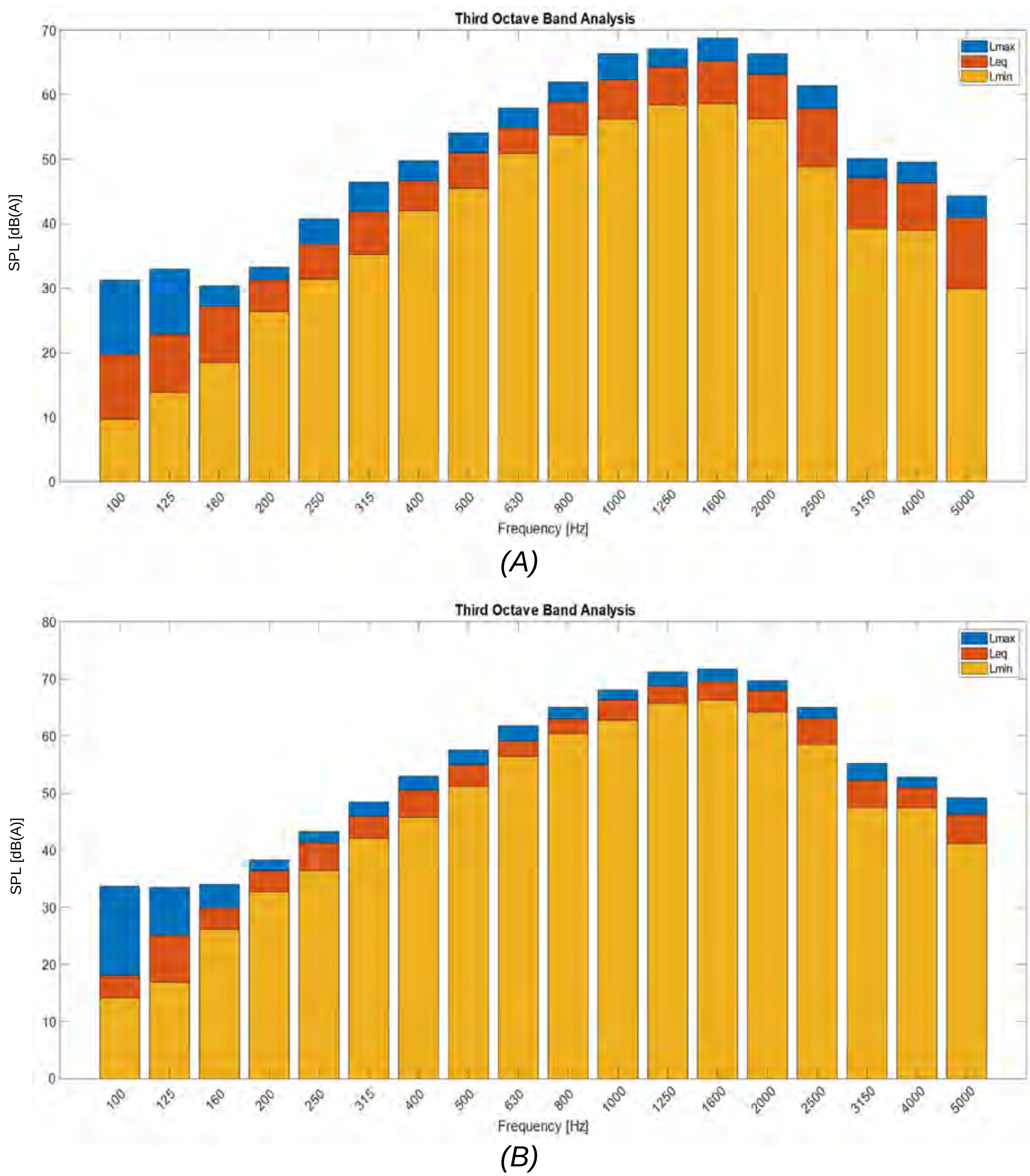

Fig. 10. Third-octave band analysis of the simulated gear configuration at 60 (A) and 120 (B) revolutions per minute.

the background noise spectrum. The peak at 132 Hz is due to the gear meshing frequency (*GMF*). The second and third harmonics are displayed in bands with a frequency of 132x2 ± 6% and 132x3 ± 6% Hz.

Frequencies of *RPM* and *GMF* peaks are coincident in simulated and tested signals. Amplitudes of *RPM* have similar values in both signals, however, the GMF amplitude is significantly higher in the tested signal. The peaks corresponding to the *GMF* harmonics are centred in the highlighted frequency band for the simulated signal (plotted in blue), while for the tested signal (plotted in red) the peaks are displaced in this frequency band range.

Fig. 8. shows the third octave band spectrum of tested and simulated signals after been corrected by the background noise. The bands from 100 Hz to 5000 Hz are displayed as this is the range required by the regulations [11]. It can be appreciated that both signals are coincident between 500 Hz and 4000 Hz bands, although below these frequency bands the differences are noticeable. The total sound pressure level is 76.6 dB(A) for the tested signal and 77.1 dB(A) for the simulated signal.

5. Approaching proposal

The prediction model allows studying the sound signal generated by pairs of spur gears. In this section, the prediction model is used to study the signal produced by a specific example of a proposed MAVAS. This proposal is designed to have a mechanical assembly as simple as possible. Three pairs of spur gears without lubrication are used to produce a more complex sound, all of them with the same centre separation to facilitate the arrangement of the system. As can be seen in Fig. 9 the gear driving wheels are mounted on the same driving shaft, while the gear driven wheels are mounted on bearings over a fixed axis.

The gears are made of steel and have a thickness of 16 mm. The module and number of teeth of each pair can be consulted in Table 2. Bearings allow each driven wheel to rotate at its speed without load. The proposed system can be installed in a box, which has the purpose of increasing the directivity of the sound emission in addition to allowing the positioning of the axes as shown in Fig. 9. However, the acoustic behaviour of the box has not been studied yet.

According to regulation [11], the speed range for the AVAS operation is the range of greater than 0 up to and inclusive to 20 km/h. For this reason, it is considered that the MAVAS drive system provides a rotation speed to the driving shaft from 0 to 120 rpm proportionally to the vehicle speed when it circulates from 0 to 20 km/h, as it could be assumed the same angular speed of a wheel shaft of a commercial vehicle.

In addition, the regulation includes third-octave band requirements for test speeds of 10 and 20 km/h, therefore the sound produced by the three gear pairs at 60 and 120 rpm has been simulated. The total sound pressure is obtained as the sum of the three contributions of each pair of gears. A third-octave band filter has then been applied to the total pressure signal over time.

To perform the simulation, the sound prediction model for each of the three gear pairs has been run at coordinates of $R = 2$ m and $\theta = 0$ rad. With these coordinates, the MAVAS is considered to be located in the longitudinal half of the vehicle, while the sound produced is recorded in the front plane of the vehicle.

Fig. 10 (A) shows the third band analysis at 60 rpm and Fig. 10 (B) at 120 rpm. It could be seen the large amount of energy around the 1600 Hz band in both cases. The overall sound level reached is 71 dB(A) during the 60 rpm simulation and 75 dB(A) in the 120 rpm.

The regulation [11] requires a series of minimum levels below the 1600 Hz band to ensure the detectability of the warning sound. Those minimum levels are already reached by the simulated signal. It also sets the sound produced to a maximum of 75 dB (A), so it has been sought to set this level as the top of the simulations to produce a sound as intense as possible. More extensive analysis of the regulations can be found in [28].

6. Conclusions

A mathematical model of a gear pair sound prediction has been developed with the aim of creating a Mechanical Acoustic Alert System (MAVAS). This system is intended to allow pedestrians to detect the presence of an EV or HEV more effectively than the existing systems by being similar to the sound of the mechanical components of a traditional combustion vehicle.

The sound prediction model generates the pressure signal over time for a pair of spur gears that rotates without lubrication nor transmitting load. The model has been successfully validated using an experimental set-up for those working conditions.

Finally, an approaching example assembly of the MAVAS is proposed, in which the simplicity of the design prevails. The sound prediction model has been used to generate the acoustic signal of the approaching MAVAS allowing to study its adequacy to the regulation.

The MAVAS implementation inside the vehicles, like the location or the design of the drive system, presents a new contribution path, providing a low price and maintenance solution to increase detectability for low-cost quiet vehicles like cargo electric vans. The limitations of this device must also be taken into account, such as mechanical loses or the space necessary to anchor it to the vehicle. Acoustic restrictions arise from the impossibility of adapting the sound generated to the circulation environment, such as reducing the amplitude if it were necessary.

In addition, it is interesting to execute the physical construction of an example of a MAVAS to be able to perform detectability tests against common warning sounds to prove its effectiveness.

CRediT authorship contribution statement

Miguel Fabra-Rodriguez: Methodology, Software, Writing – original draft. **Ramon Peral-Orts:** Conceptualization, Writing - review & editing, Supervision. **Hector Campello-Vicente:** Validation, Resources. **Nuria Campillo-Davo:** Formal analysis, Writing - review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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