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Prediction of airborne radiated noise from lightly loaded lubricated meshing gear teeth

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ABSTRACT

This paper introduces a novel analytical method for determination of gear airborne noise under lightly loaded conditions, often promoting gear rattle of loose unengaged gear pairs. The system examined comprises a single gear pair, modelled through integrated contact tribology and inertial transient dynamics. Lubricant film thickness, structural vibration and airborne gear noise are predicted and correlated with experimental measurements undertaken in a semi-anechoic environment. Good agreement is noticed between the numerical predictions and the experimental measurements. The presented model is capable of estimating the airborne radiated gear noise levels and the dynamic behaviour of gear pairs under different operating conditions, with superimposed impulsive input speed harmonics.

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1. Introduction

Automotive transmission gear noise has received increased attention in recent years. The phenomenon is considered as a major noise source in the automotive industry. It is perceived as a vehicle built quality issue and is caused by impacts of the loose (unengaged) meshing gear teeth pairs through their backlash. The problem is also exacerbated by the oscillatory crankshaft vibration signal (engine order vibrations), which has a greater poignancy with the higher torque fluctuations of diesel engines at multiples of engine order vibration [1,2]. Modern downsizing philosophy has led to compact transmissions, thus a greater tendency for the interactions of the loose meshing gear pairs. As the result, the engine torsional oscillations, resident on the transmission input shaft, exacerbate the teeth pair impacts through their lubricated conjunctions [3,4]. The accelerative nature of these impacts causes radiated noise, which is widely termed as gear rattle [5].

A large volume of numerical analyses has been reported, particularly for parallel axis gearing systems. Most analyses consider dry contact of meshing pairs, which is representative of highly loaded cases, where elastohydrodynamic conditions may be reasonably approximated by the classical Hertzian theory [6,7]. Other analyses have included the effect of lubricant in the contact, which is particularly important for lightly loaded contacts, where a hydrodynamic regime of lubrication would be prevalent, such as in the case of idle gear rattle [1,8–10]. The contact stiffness under lightly loaded hydrodynamic conditions is well below that obtained through use of the classical Hertzian theory. In addition, the temperature dependence of lubricant viscosity significantly affects its load carrying capacity, as well as its shear characteristics, thus influencing gear dynamics [3,9,10].

Much attention has been paid to the estimation of radiated noise from meshing gear pairs with the aim of determining a threshold for the onset of unacceptable gear rattle. These have been mostly experimental, often involving determination of coefficients of restitution to describe the effect of lubricant damping through its squeeze film motion [11], as well as any hysteretic elastic deformation of the impacting solid surfaces. Using a torsional vibration model, gear rattle noise was calculated for a 5-speed gearbox, employing the main design parameters and use of various empirical formulae [12]. Following an optimisation study, the gear noise was shown to be reduced by 14%. The influence of different parameters on lubrication conditions and structure-borne noise of gear transmissions was also studied by Fietkau and Bertsche [13]. This enabled direct determination of structure-borne noise for the rattling loose gear pairs, as well as for the loaded gear pairs. The findings were validated experimentally.

Radiated structure-borne noise from a gearbox was calculated using three-dimensional Finite Element Analysis (FEA) of the structure, combined with the Rayleigh integral method [14]. A simplified gearbox, excited internally by the gear teeth meshing stiffness was used, where the vibro-acoustic coupling between the elastic housing, the air-cavity and the free acoustic field was

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[1] Research data for this paper are available on request from Dr. Stephanos Theodossiades (S.Theodossiades@lboro.ac.uk).

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Nomenclature

C	clearance in gear wheel bore-retaining shaft conjunction (m)
C_b	half normal teeth pair backlash (m)
$\frac{\partial h}{\partial t}$	squeeze film velocity (m/s)
l	instantaneous contact length of the meshing teeth (m)
l_1	contact length in the gear wheel–shaft conjunction (m)
u_{ent}	speed of entraining motion of lubricant in the meshing pairs (m/s)
v	speed of entraining motion in the wheel–shaft conjunction (m/s)
a_{eq}	equivalent radius of curvature of the meshing teeth pair (m)
α_n	normal pressure angle (rad)
η_0	lubricant atmospheric dynamic viscosity (Pa s)

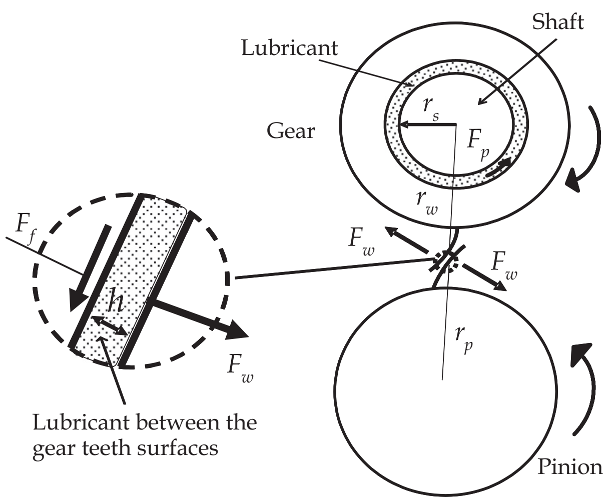

Fig. 1. The gear pair model.

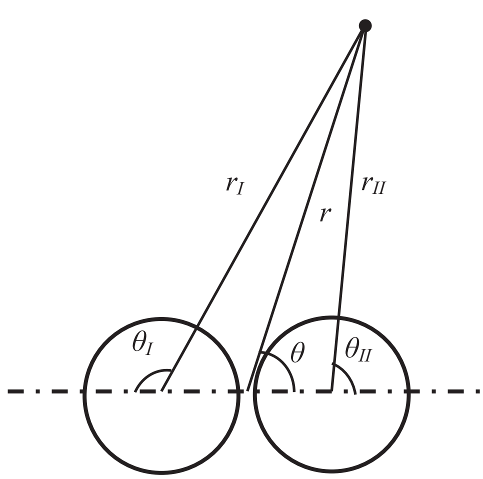

Fig. 2. System of reference for noise radiation for a pair of impacting equivalent cylinders.

considered. Mucchi et al. [15] presented a method for determination of noise and vibration analysis of gear pumps, comprising a combination of numerical analysis and experimental measurements. The numerical method included lumped parameters, integrated with FEA and Boundary Element Method (BEM). The lumped parameter model comprised loaded bearings and gears, whilst the FEA was used for the casing and plates. The use of BEM resulted in the prediction of the emitted noise levels. The experimental measurements included inertial acceleration and acoustic pressure, which were verified through simulation results.

A model relating the acceleration response of chain drive components (sprocket teeth against chain rollers) to the generated sound was developed using finite element techniques and numerical schemes by Zheng et al. [16]. Sound pressure levels at different locations on a virtual cylindrical surface around the chain were evaluated and validated against experimental measurements, showing good agreement. The work was based, to a large extent on that reported by Yufang et al. [17], where the radiated sound from the impact of two rigid cylinders was calculated through use of Hertzian impact theory, and verified experimentally.

In this paper, an analytical method to predict the airborne radiated noise from the meshing gear teeth under light loads is presented. The method is based on rigid body dynamics, coupled with hydrodynamic lubricated contacts, as well as far field sound pressure calculations. This analytical approach has not hitherto been reported in literature. In the following sections, the methodology for sound radiation predictions is presented initially, as well as a flowchart for the numerical calculations. The experimental configuration is then described, followed by analytical results and discussion. The numerical predictions show good agreement with the experimental measurements obtained from a single stage gearbox.

2. Methodology

The gear pair system studied is shown schematically in Fig. 1. The entire physical assembly is depicted in Fig. 3. The input pinion shaft is driven by an electric motor. The gear wheel is mounted onto a shaft and is resisted through generated friction at the supporting bearings. The spur gear pair is modelled by a single degree of freedom rotational inertia (gear wheel) with the pinion's motion known *a priori* (this is a kinematic non-holonomic constraint, see Section 5). The remaining 5 degrees of freedom of the gear wheel are constrained because the associated motions are deemed negligible due to the light loads transmitted. The equation of motion for the gear wheel (Fig. 1) is obtained as:

$$I_g\ddot{\varphi}_g = F_w r_{bg} - F_f r_f - F_p r_s \tag{1}$$

where I_g is the gear wheel inertia with φ_g being the corresponding rotational degree of freedom. F_w is the meshing teeth contact force. r_{bg} is the gear base radius; F_f is the flank friction with r_f being its moment arm. F_p is the bearing generated friction whilst r_s is the outer contacting radius of the output gear wheel retaining shaft. For the lightly loaded meshing of loose gear pairs, flank friction is quite insignificant and may be neglected in the analysis [8]. The tooth hydrodynamic contact force is given by [4,18]:

$$\left.\begin{array}{ll} F_w = \frac{l\eta_0 a_{eq}}{h}\left(2u_{ent} - \frac{3\pi}{\sqrt{\frac{2h}{a_{eq}}}}\frac{\partial h}{\partial t}\right), & \text{if} \quad \frac{\partial h}{\partial t} < 0 \\ F_w = \frac{l\eta_0 a_{eq}}{h}(2u_{ent}), & \text{if} \quad \frac{\partial h}{\partial t} \geq 0 \end{array}\right\} \tag{2}$$

Eq. (2) provides the lubricant reaction under assumed iso-viscous rigid hydrodynamic regime of lubrication, where the term $\frac{\partial h}{\partial t}$ is the squeeze film contribution. When $\frac{\partial h}{\partial t} < 0$, the meshing

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