inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 1.1

NOISE OPTIMIZATION OF A REAR AXLE GEAR SET

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Keywords:
GEAR, NOISE, REAR AXLE, MODELISATION

ABSTRACT
Increasing health threats to human beings as a results of environmental impacts have led to a number of directives and regulations which compel automotive manufacturers to minimise emissions from their products. In this paper, we present a calculation method to evaluate at the first stage the ability of an hypoid gear set to quietness. After showing some experimental results, thus validating our approach, we demonstrate on an example the possibility to reduce the emitted noise of a rear axle up to 6 dB on a given application, by slightly modifying the local contact conditions.

1 - INTRODUCTION
Last stage of the power transmission line, trucks, buses and coaches rear axles are built around a bevel or an hypoid gear set. The designer is faced with trade-offs between weight, reliability, and noise level generated by the gear set. Optimising the above parameters lead us to develop different computer programs to evaluate bending stress, contact stress, and ability of the gear set to quietness.
It is well known that gear mechanisms generate characteristic frequencies corresponding to the periodic variation of the mesh stiffness. The amplitude of this variation, known as the transmission error, has become an important parameter when defining a new gear set [1].
Recent advances in computer performances make it practical to conduct a full 3 dimensional analysis, using Finite Element Method (FEM), in order to analyse bevel and hypoid gear sets behaviour under various loading conditions [2,3].
After the description of the calculation method, we will show the clear correlation between measured noise inside a vehicle and the calculated level of transmission error.
Finally, we describe how to optimise the tooth surface (micro-geometry) in order to decrease the transmission error, thus reducing the noise level inside a vehicle.

2 - 3D SOLID MODELLER AND MESH GENERATOR
One of the most difficult task associated with 3D modelling of bevel and hypoid gear set is an accurate description of the gear surface geometry and the root fillet. Starting from the basic parameters of a given gear set (pitch diameter, spiral angle, tooth proportions...), we then generate some theoretical machine tool setting appropriate to cut both the pinion and the crown. We simulate the kinematic motion of the manufacturing process to calculate a set of point coordinates on each flank of both pinion and crown gear [4]. When one member is generated, the surface is the envelope of a family of surfaces of the head cutter: points on the tooth surface are points of tangency to the cutter surface during manufacturing operation. The conditions necessary for envelope existence are given kinematically by the general law of gearing, i.e. the normal of the generated surface must be perpendicular to the relative velocity between the cutter and the gear tooth surface.
The calculated surfaces for both pinion and gear are then restricted to an exact description of the gear blank, to have an accurate evaluation of the contact ratio, i.e. the theoretical number of teeth potentially in contact at a given position in the mesh.
A pre-processor program is used to generate two separate F.E. models for pinion and gear (see figure 1). These models are solved separately to obtain the global stiffness of both members, in a term of gear

teeth deflection under a unitary loaded node. A grid of 6 × 11 nodes on both flanks (drive side and coast side) are successively loaded, to obtain a discrete compliance matrix. On this discrete set of points, we fit an influence function, combination of exponential components in both u and v, normalised coordinates on the tooth flank. This operation gives us the compliance due to bending and shear deformation, excluding the Hertzian contact deformation. This procedure, applied successively to both flank of pinion and gear, is the most costly in terms of computing time. But it has to be done only once whatever is the micro-geometry (ease-off topography) or the loading conditions.
The user interface developed to generate 3D meshes, adequate boundary conditions, and curve fitting of FE results is directly accessible for a non-specialist of finite element modelling technics.

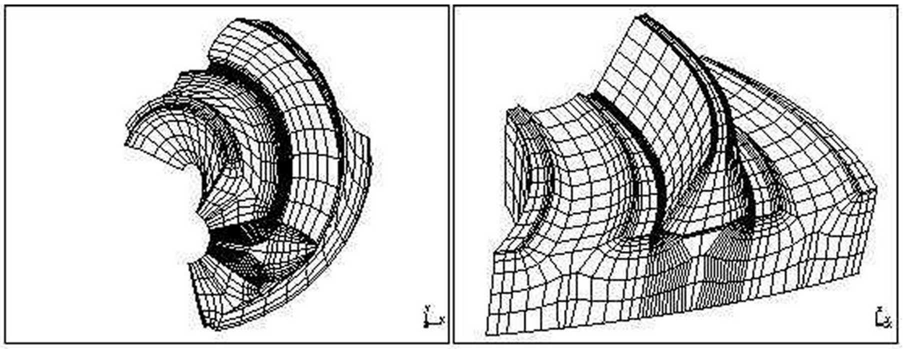

Figure 1: Pinion and crown finite element models.

3 - SOLVING THE CONTACT PROBLEM

Using the general law of gearing, it is possible to calculate the potential lines in contact for a given angular position of the pinion (driving member). Conjugating these lines in term of crown coordinate system, we can determine the theoretical area of contact on both member, for a perfectly conjugated gear set. One can see such lines of contact for the drive side (figure 2) for a 20 × 41 hypoid gear set, cut with OERLIKON Spirac method (outside diameter 315 mm / spiral angles 47° - 37° / offset 28 mm).
For a successive set of engagement pinion position, the potential lines of contact are discretised. For each point, an initial gap is estimated, which is zero if the assembly is working in ideal prefect conditions, and for theoretical tooth flanks. Introducing surface topography given by 3-dimensional measurements on a grid leads to a non-zero gap. Local contact deformation which are calculated using an Hertzian approach (thus introducing an inherent non-linearity in the equilibrium and compatibility equations) is added to the global deformation calculated from previous FE results. An iterative process then shares the total applied load firstly between the potential teeth in contact, and secondly along each line of contact of a given tooth.

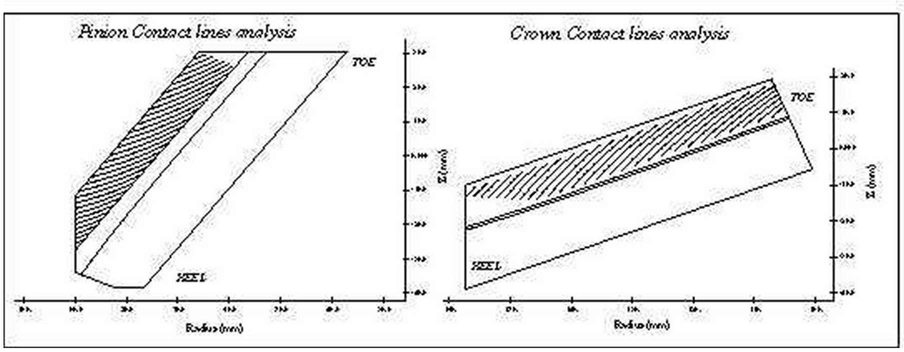

Figure 2: Potential lines of contact for the drive side, gear set 20 × 41.

4 - TRANSMISSION ERROR CALCULATIONS

Following the growing ecological consciousness, legislators have issued more and more severe regulations on the allowable noise levels for trucks. Moreover, the customers do not accept to buy noisy vehicles, felt clearly as a lack of quality. Thus is it a necessary challenge for vehicles manufacturers to reduce, at acceptable cost, the global noise levels. It is now well-admitted that minimising the quasi-static transmission error amplitude (TE) gives better vibration and noise behaviour. However, depending on the application, one could choose to optimise the TE for a given pinion torque, or for a range of torques. Looking to the fact that the evolution of the TE versus torque is very much dependent of the surface topography, it could be interesting to select the correct micro-geometry regarding to the application, keeping in mind the dimensioning criteria related to the transmitted power (bending stress and contact stress). As a result of our computer program, the quasi-static transmission error is calculated for a given macro-geometry, and an ease-off topography resulting from the cutting process. As an example, we can see on figure 3 the resulting evolution of the TE amplitude versus pinion applied torque for a given surface topography, for the drive side and the coast side.

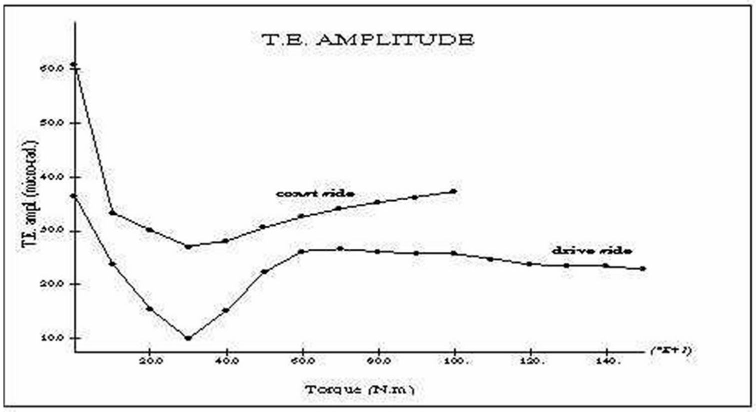

Figure 3: Transmission error versus torque drive side and coast side, 20 × 41 gear set.

5 - EXPERIMENTAL VALIDATION

In order to validate the assumption that the calculated level of transmission error is an accurate indicator the ability of the gear set to quietness, we have perform the following experiment. A given gear set 20 × 41 as been mounted in a rear axle, and the inside noise levels in a vehicle have been measured via a microphone, for different running conditions. The corresponding levels of TE have been calculated, based on the exact ease-off topography of the gear set. One can see the results in table 1 below.

Running condition	Inside noise level (dB)	Calculated transmission error (micro-radians)
80 km/h - grade 0 %	71	11
100 km/h - grade 0 %	72	13
80 km/h - grade +7 %	74	24
100 km/h - grade +7 %	76	23
80 km/h - grade -7 %	76	35
100 km/h - grade -7 %	78	34

Table 1: Experimental datas versus calculated transmission error.

We could see, figure 4, that the measured noise level is very much dependent on the calculated transmission error, thus confirming the original assumption, i.e. by minimising the transmission error, we could reach an acceptable noise inside the vehicle. One should have in mind that the noise level is also depending on of the vibration propagation from the gear set to the microphone. The natural frequencies of the housing, the mounting on the vehicle as well as its own arrangement have a significant contribution to the absolute noise level, and should be optimised in consequence to obtain a low level of noise inside the vehicle.

6 - OPTIMIZATION OF A REAR AXLE GEAR SET

The transmission error of a gear set is directly the resultant of the local conditions of contact between the mating gears. By changing this micro-geometry, we have the possibility, by calculation, to improve the level of transmission error in a given torque range, thus reducing the excitation as the very first

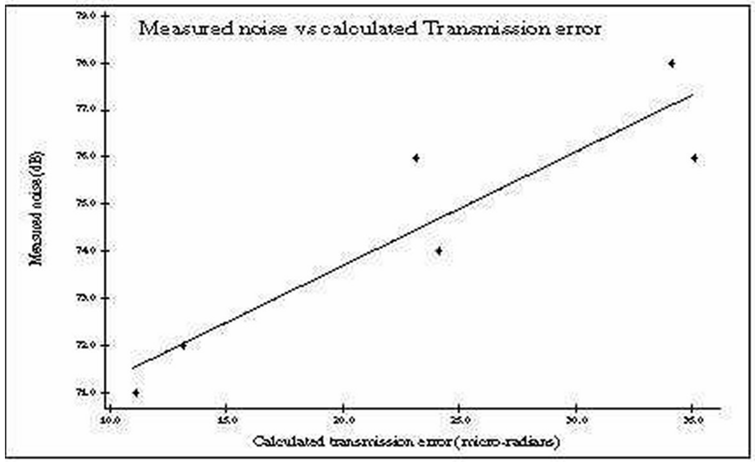

Figure 4: Correlation between measured inside noise and calculated transmission error.

stage, and by that the noise level. Practically, the modification of the micro-geometry can be done by slightly changing the cutting machine settings, and/or re-lapping the gear set if the amplitude of the modifications are rather small. For our application, the second solution has been chosen to demonstrate easily the feasibility of such a method. One can see figure 5 the different ease-off topography before and after re-lapping.
The gear set for this application has the following characteristics: 10 $\times$ 37 / Oerlikon cutting method / Cutter radius 160 mm / offset 39 mm. The result, in terms of calculated transmission error could be seen in table 2 below.

Running condition	Calculated transmission error before re-lapping (micro-radians)	Calculated transmission error after re-lapping (micro-radians)
80 km/h - grade 0 %	50	19
100 km/h - grade 0 %	41	13
80 km/h - grade +7 %	32	25
100 km/h - grade +7 %	38	26
80 km/h - grade -7 %	35	14
100 km/h - grade -7 %	40	14

Table 2: Effect of re-lapping on the calculated transmission error.

The re-lapped gear set has been mounted in a rear axle on the vehicle, and the inside noise measured and compared to a standard gear set (serial production). The improvement in terms of noise reduction was up to 6 dB, thus validating the global approach we have developed.

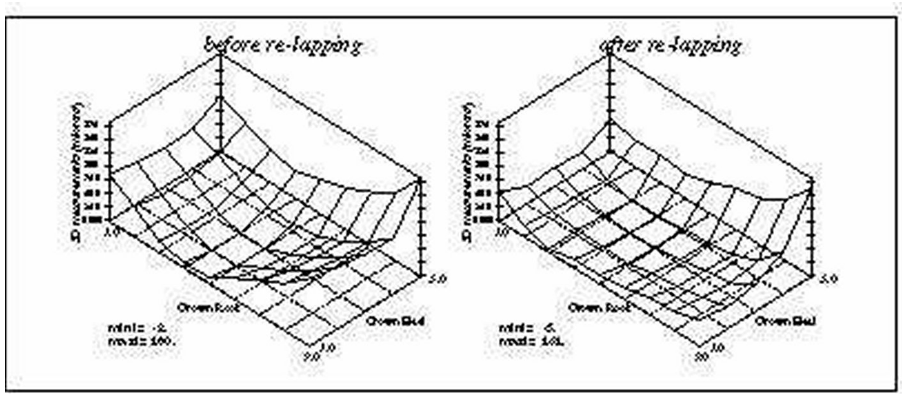

Figure 5: Ease-off topography before and after re-lapping.

7 - CONCLUSION

The package of softwares presented above, developed and validated at Renault VI allow us to accurately calculate the meshing parameters associated to bevel and hypoid gear sets as well as the optimised micro-geometry in terms of life and noise behaviour under running conditions. Successful applications led as to use these methods at the first stage of development of new gear sets.

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