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Estimation and Reduction of Noise in Gears

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ABSTRACT

The paper presents the literature for noise in gears. It includes the methods of calculation of gear noise, mathematical modeling for gear noise prediction. Also experimental methods to measure noise parameters. It is mainly divided into three parts “Causes of Noise”, “Mathematical modeling”, “Noise measurement”. Mathematical models consisting of gear noise prediction is useful in order to design gear box with less noise. Noise and vibration measurement and signal analysis are important tools when experimentally investigating gear noise gears create noise at specific frequencies, related to number of teeth and the rotational speed of the gear.

Keywords : Gear, Noise, Gear Ratio, Mathematical Modeling, Noise Measurement.

I. INTRODUCTION

The most frequently used type of gear profile is the involute. It is used for cylindrical spur and helical gears as well as for conical gears like beveloid, hypoid and spiral bevel gears. Some characteristics of involute (cylindrical) gears that have made them so common are:

- Uniform transmission of rotational motion, independent of small error in centre distance.
- The sum of the contact forces is constant and the direction of the total contact force always acts in the same direction.
- An involute gear can work together with mating gears with different number of teeth.

Manufacturing is relatively easy and the same tools can be used to machine gears with different numbers of teeth. (Applies to hobs, shaper cutters, grinding worms, shaving cutters but not to profile tools like milling cutters and profile grinding wheels). If the gears were perfectly rigid and no geometrical errors or modifications were present, the gears would transmit the rotational motion perfectly, which means that a constant speed at the input shaft would result in a constant speed at the output shaft. The assumption of no friction leads to that the gears would transmit the torque perfectly, which means that a constant torque at the input shaft would result in a constant torque at the output shaft. No force variations would exist and hence no vibrations and no sound (noise) could be created. Of course, in reality, there are geometrical errors, deflections and friction present, and accordingly, gears sometimes create noise to such an extent that it becomes a problem. [1]

The gearbox is a source of vibration and consequently noise. Except for bearing fatal defects or extreme structure-resonance amplification, gears are the main sources of high frequency vibration and noise, even in newly built units. The gearbox overall sound pressure level (SPL), compared to the SPL associated with the meshing gears, is only by some 5 dB higher at maximum. There are two possible solutions for keeping a transmission unit quiet. Introducing an enclosure for preventing noise radiation with the consequences of low efficiency and difficulties in maintenance is the easiest one. The more sophisticated and much more efficient solution is based on solving the noise problem at the very source. It means to introduce improvement aimed at the gear design and manufacturing, which results in the greatest reduction of the SPL. [16]

High contact ratio spur gears could be used to exclude or reduce the variation of tooth stiffness. Kasuba[19] established experimentally that the dynamic loads decrease with increasing contact ratio in spur gearing. Sato, Umezawa, and Ishikawa [20] demonstrated experimentally that the minimum dynamic factor corresponds to gears with a contact ratio slightly less than 2.0 (1.95). The same result was found experimentally by Kahraman and Blankenship [21] and theoretically by Lin, Wang, Oswald, and Coy [22]. The increase in contact ratio can be implemented in two ways: 1) by decreasing pressure angle and 2) by increasing tooth height. Obviously, the use of a standard pressure angle and standard tools is preferable. In the author certificate (Nikolayev and Podzharov) [23] a simple method of design of high contact ratio spur gears with standard basic rack of 20° profile angle was presented. This method allows us to design gears with a contact ratio nearly 1.95. Vulgakov [24] proposed a method of design of nonstandard gears in generalized parameters and found that spur gears with a contact ratio of more than 2 and a pressure angle more than 20° worked considerably quieter. Rouverol and Watanabe [25, 26] proposed maximum-conjugacy gearing which has a low pressure angle at pitch point and which increases slowly at the tip and at the root. The measurements also show a considerable reduction in the noise level compared with standard gears.

II. TRNSMISSION ERROR

"The difference between the actual position of the output gear and the position it would occupy if the gear drive were perfectly conjugate". Transmission error (TE) is considered to be an important excitation mechanism for gear noise and vibration.

The causes of transmission error are deflections, geometrical errors and geometrical modifications.
Examples of deflections:

- Contact deformations (hertzian) in the gear mesh
- Gear teeth bending deflections
- Gear blank deflections
- Shaft deflections
- Bearing and gearbox casing flexibility
- Examples of geometrical errors:
- Involute alignment deviations
- Involute form deviations
- Lead deviations
- Lead form deviations
- Gear tooth bias
- Pitch errors
- Run-out
- Error in bearing position in the casing

Examples of some common geometrical modifications:

- Lead crowning
- Helix angle modification
- Profile crowning
- Tip relief and root relief

Transmission error can be measured statically/ dynamically (high and low speed), loaded as well as unloaded.

TABLE I. STATIC /DYNAMIC, LOADED/UNLOADED TE

		Speed	
		Low	**High**
Load	**Low**	Static Unloaded	Dynamic Unloaded
	High	Static Loaded	Dynamic Loaded

The usual cause of gear noise is its harmonics. The TE per revolution is problem because the frequency is relatively low, but the once per revolution transmission error, due to for example run-out, causes side bands to the gear mesh frequency, with the frequency of tooth mesh frequency +/- the shaft rotational frequency. [2]

“Phantom” frequencies may also originate from the dressing wheel, when grinding gears. [3]

Often it is too difficult to measure transmission error, due to inaccessibility of free shafts. [4]

Transmission error is often measured with optical encoders, which gives typically several thousands of pulses per revolution. The transmission error is acquired by comparing the signals from the two encoders on each shaft. [5]

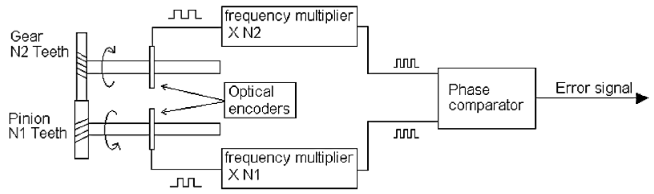

FIGURE I. TE MEASUREMENT USING OPTICAL ENCODERS

Sasaoka[6] suggested that frequency range between the first and second mode can be avoided by installing torsional stiffness adjustment sections in power transmission system to control the natural frequencies.

The knowledge of elemental gear errors leads to transmission error but transmission error does not lead to knowledge about the elemental gear errors.[6] The calculation of transmission error is useful for several purposes, some examples are:

- To choose appropriate gear geometry to minimize the variations in mesh stiffness, i.e. determine module, helix angle and contact ratio.
- Determine gear tooth modifications like crowning and tip relief (magnitude and starting point) to minimize transmission error.
- Investigate how different manufacturing errors influence gear noise and vibration characteristics.
- To obtain input to dynamic models of gear systems. [1]

In addition to TE there are also other possible excitation mechanisms. [8]

There is no physical basis for a ideal correlation between transmission error, vibration and noise in general case. [5]

TABLE II. MEASUREMENT OF PARAMETERS

TE	Optical encoders
Vibrations	Accelerometer
Noise	Microphone

A resonance and mobility check indicates that the primary wheel was the source for dramatic change between dynamics of system at different speeds. [9]

True involute gears cause lower vibration level than bias out gears. [10]

Dynamic incremental load (which is function of velocity and operating load with parameter of tooth surface modifications) can be used as an index of dynamic performance (like noise and vibrations) of gear pair. [11]

Small misalignment like 60 μm can cause considerable transmission error and high dynamic forces. [12]

The transmission error can be improved (decreased) by increasing the real contact ratio as much as possible. This can be realised by modifying and correcting gear misalignment resulting from transmission case production error and other defects, and hence shifting the tooth bearing point to the tooth surface centre, reducing the curvature of tooth surface and obtaining larger bias-in modification [13].

III. MATHEMATICAL MODELING

To understand and control gear noise, it is necessary not only to have knowledge about the gears, but also about the dynamic behaviour of the system consisting of gears, shafts, bearings and gearbox casing. The noise characteristics of a gearbox can be controlled already at the drawing board when designing the gearbox, because all the components have an important effect on the acoustical output [14]. For relatively simple gear–systems it is possible to use lumped parameter dynamic models with springs, masses and viscous damping. For more complex models, which include for example the gearbox casing, finite element modelling is often used. The first dynamic models were used to determine dynamic loads on gear teeth, and they were developed in the 1920s, the first mass–spring models were introduced in the 1950s [15].

A. Lumped parameter dynamic models

Özguven and Houser [30] reviewed the literature on mathematical models used in gear dynamics,from 1915 and up to 1986. The review is very extensive and includes 188 references. They classified the models in five groups:

TABLE III. CLASSIFICATIONS OF LUMPED PARAMETRIC DYNAMIC MODELS

Sr. No	Type	Description
1	Simple dynamic factor models	a) most of the early studies b) gear root stress formulae is determined c) include empirical and semi-empirical approaches
2	Models with tooth compliance	a) include only the tooth stiffness b) flexibility of shafts, bearings, etc. are all neglected c) single degree of freedom spring–mass system
3	Models for gear dynamics	a) Include the flexibility of the other elements as well as the tooth compliance b) Torsional flexibility of shafts and the lateral flexibility of the bearings and shafts along the line of action.
4	Models for geared rotor dynamics	The torsional vibration of the system is usually considered
5	Models for torsional vibrations	a) flexibility of gear teeth is neglected b) viewed as pure torsional vibration problems, rather than gear dynamic problems

B. Noise prediction equations

In order to obtain a more accurate prediction method of gear noise, a new prediction equation was proposed by Masuda et al. The equation was obtained by adding a dynamics term to Kato's equation.

Katos equation:

$$L = \frac{20(1-\tan(\beta/2)) \bullet \sqrt[8]{u}}{f_v \sqrt[4]{\varepsilon_\alpha}} + 20\ \log\ W\ dB(A)$$

Where:

L : overall noise level at 1 meter from a gearbox

β : helix angle

u : gear ratio

ε_α : transverse contact ratio

W : transmitted power in hp

f_v : speed factor (analogous to dynamic factor in JIS – B1702)

The new prediction equation was derived by replacing the speed factor fv by AGMA's recommendation

$f_{v0} = \sqrt{(5.56 /(5.56 + \sqrt{(v)})}$ and adding the effects of dynamics.

New prediction equation:

$$L = \frac{20(1-\tan(\beta/2)) \bullet \sqrt[8]{u}}{\sqrt[4]{\varepsilon_\alpha}} \sqrt{\frac{5.56+\sqrt{v}}{5.56}} + 20\ \log\ W + 20\log \bar{X}\ dB(A)$$

Where:

L : overall noise level at 1 meter from a gearbox

β : helix angle

u : gear ratio

εα : transverse contact ratio

W : transmitted power in kW

v : pitch line speed in m/s

X : Vibration displacement amplitude normalized by static deflection, calculated by vibration analysis using a simple torsional dynamic model.

Predicted noise levels were compared with experimental noise measurements for hobbed gearsand gears ground with two different grinding methods and the correlation was good.

IV. NOISE MESUREMENT

Noise and vibration measurement and signal analysis are important tools when experimentally investigating gear noise. Gears create noise at specific frequencies, related to the rotational speed and number of teeth of the gear. It is also possible to detect different errors like for example run out (eccentricity) due to side-band generation. Closely related is also vibration measurement and signal analysis for the purpose of gear fault detection, used in machine diagnostics in order to detect gear failures before catastrophic failure occurs [1].

V. CASE STUDIES

A. Transmission and Gearbox Noise and Vibration prediction and Control

Jiri Tuma reviews practical techniques and procedures employed to quiet gearboxes and transmission units. The paper describes the research work on quieting the truck gearbox. The tuck gearbox is without enclosure and their operating rotational speed and load are not steady but are variable. The experience gained from research work on the truck gearbox noise reduction can be applied, according to the opinion of this paper's author, generally to any other transmissions.

Two possible test arrangements are presented as open loop test stand and open loop test rig configuration. And noise was measured from two microphones located by side of the gearbox under a side of the gearbox under a test at a distance of 1 m.

Gearbox noise is tonal. It means the noise frequency spectrum consists of sinusoidal components at discrete frequencies with low-level random background noise. The frequency that is the product of the gear rotational speed in Hz and the number of teeth are referred to as the base tooth meshing frequency or gear meshing frequency f_{GMF}.

All the basic spectrum components are usually broken down into a combination of the following effects:

- low harmonics of the shaft speed originating from unbalance, misalignments, a bent shaft, and resulting in low frequency vibration, therefore without influence on the gearbox noise level
- harmonics of the base tooth meshing frequency and their sidebands due to the modulation effects, that are well audible; the noise and vibration of the geared axis systems is originating from parametric self-excitation due to the time variation of tooth-contact stiffness in the mesh cycle, the inaccuracy of gears in mesh and non-uniform load and rotational speed
- Ghost (or strange) components due the errors in the teeth of the index wheel of the gear cutting machine, especially gear grinding machines employing the continuous shift grinding method that results in high frequency noise due to the large number of the index wheel teeth, these ghost components obviously disappear after running-in
- components originating from faults in rolling-element bearings usually of the low level noise except for fatal bearing

Other subharmonics originate from the rate at which the same two gear teeth mesh together (hunting tooth frequency) f_{HTF}

$$f_{HTF} = f_{GMF} \frac{\gcd(n_1, n_2)}{(n_1 n_2)}$$

Where gcd (n_1,n_2) is a greatest common divisor of both the numbers n_1 ,n_2 of teeth.

The dominating components in the frequency spectrum can be identified after averaging either in the time or frequency domain.

The conclusion was low noise gearbox requires sufficiently rigid housing, shafts and gears, and the HCR gears and the tooth surface modification for design load [16].

B. Static and Dynamic Transmission Error in Spur Gears

High precision and heavily loaded spur gears the effect of gear errors is negligible, so the periodic variation of tooth stiffness is the principal cause of noise and vibration. High contact ratio spur gears could be used to exclude or reduce the variation of tooth stiffness. The analysis of static and dynamic transmission error of spur gears cut with standard tools of 20° profile angle is presented in this paper. A simple method to design spur gears with a contact ratio nearly 2.0 is used. It consists of increasing the number of teeth on mating gears and simultaneously introducing negative profile shift in order to provide the same center distance. Computer programs to calculate static and dynamic transmission error of gears under load have been developed. The analysis of gears using these programs showed that gears with high contact ratio have much less static and dynamic transmission error than standard gears.

The conclusions are

- The analysis of static and dynamic transmission errors in high precision heavy loaded standard gears, high contact ratio gears of standard tooth height and high contact ratio gears with slightly increased tooth addendum showed that in the last type of gears the static and dynamic transmission errors can be almost completely excluded.
- Preliminary experiments show that high contact ratio spur gears have noise level considerably less than standard gears [17].

C. Gear noise evaluation through multibody TE-based simulations

This paper presents a methodology for the calculation of gear bearing forces, useful for the acoustic analysis of gearboxes and applicable to spur as well as helical parallel gear systems. The methodology is based on the implementation of a procedure for the computation of the dynamic transmission error (DTE) in a multibody environment. The DTE is obtained from the static transmission error (STE), i.e. the static relative displacement between meshing teeth, which is variable along the mesh cycle. The adopted multibody technique enables to overcome the principal drawbacks of FEM, achieving good computational efficiencies, and of analytical models, avoiding to lump the system in one or few degrees of freedom. These goals are reached by means of a user-defined force element, acting as teeth meshing force, which stems from the integration of the multibody software, LMS Virtual.

The static mesh stiffness is imported in the multibody software, the contact forces are calculated and applied to the gears by a user-defined force element which reads the instantaneous value of the mesh stiffness based on the actual position along the mesh cycle.

Since the contact analysis is captured in the instantaneous static mesh stiffness by GCAS, including three Dimensional teeth microgeometric modifications and manufacturing errors, teeth global and contact stiffness, shaft deflections and assembly errors, the gear system can be modeled as rigid in the multibody environment, achieving a good computational efficiency. The term "static" mesh stiffness indicates that GCAS calculation is based on the assumption that the gears reach the equilibrium under static torque.

Nevertheless the static mesh stiffness is variable along the mesh cycle, for example due to a different number of contacting tooth pairs or due to a profile modification. With more detail, GCAS returns as an output the STE, which is defined along the line of action as the difference between the real and the ideal displacement of the driven gear.

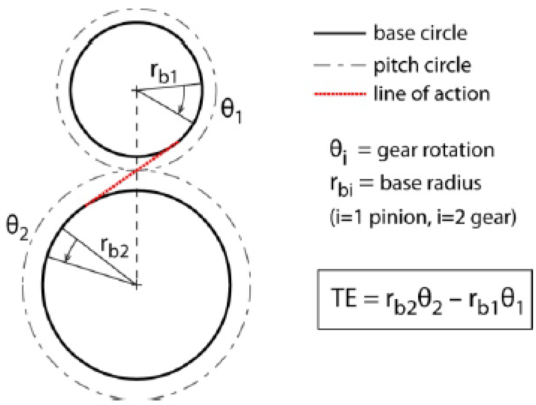

FIGURE II. TRANSMISSION ERROR DEFINITION

The results of the multibody simulation are calculated solving the system of equations of motion, which can be condensed like in Equation 1.

$$M\ddot{x} + C\dot{x} + Kx = F \qquad - (1)$$

Where a dot in accent position indicates the time derivative, x is the vector of the Lagrangian coordinates, M, C and K are respectively the mass, damping and stiffness matrices and F is the vector of the applied loads. Referring to this formulation and recalling the definition of TE, the dynamic formulation of gear meshing can be considered as the scalar Equation 2 which belongs to the vector Equation 1:

$$c\dot{DTE} + kDTE = f_{contact} \qquad - (2)$$

In the Equation 2 the inertial contribution is taken into account implicitly into the DTE, when resolving the system of equations of motion.
Two aspects about this equation are worthwhile to be mentioned in order to explain how the dynamic analysis is performed. The first is that stiffness k is the static mesh stiffness which is variable along the mesh cycle and is imported from GCAS. The second is that, since the Equation 2 is part of the Equation 1, the DTE and the contact force are both influenced by all the multibody model parts in terms of inertia, damping and stiffness.

The bearing forces, which can be user later for an acoustic analysis of the gear train, are part of the solution found for the Equation 1, hence they are available in the results of the multibody simulation.

The static transmission error is higher when only one tooth pair is in contact and lower when two pairs mesh simultaneously, being the contact ratio between 1 and 2[18].

VI. CONCLUSION

Transmission error is main cause for vibrations and ultimately for noise. Transmission error is present due to modifications. “Ghost” or “phantom” parameters also present in system which causes noise such as misalignment in gear pair, grinding wheel inaccuracies, tooth bending. Prediction methods are useful for implementation of noise less gear pair. High precision heavy loaded standard gears, high contact ratio gears of standard tooth height and high contact ratio gears with slightly increased tooth addendum showed that in the last type of gears the static and dynamic transmission errors can be almost completely excluded.

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