# Gearbox Noise and Vibration Prediction and Control

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This paper will review practical techniques and procedures employed to quiet gearboxes and transmission units. The author prefers solving the gear noise problem at the very source to introduce an enclosure as a means to reduce radiated noise, which seems to be easy but its effect on the sound pressure level is small. The gearbox noise problem solution is focused on the improvement of gear design; on the verification of its effect on the radiated noise and the determination of the gears' contribution to the truck's or car's overall noise levels and on the analytical and/or numerical computer-based tools needed to perform the signal processing and diagnostics of geared axis systems. All of the analytical methods are based on the time and frequency domain approach. Special care is addressed to the smoothness of the drive resulting from the transmission error variation during a mesh cycle. This paper will review the progress in technique of the gear angular vibration analysis and its effect on gear noise due to the self excited vibration. This presentation will include some examples of the use of such approaches in practical engineering problems.

## 1. INTRODUCTION

The major functions of a gearbox are to transform speed and torque in a given ratio and to change the axis of rotation. The main parts of the gearbox are the gears, shafts, bearings, housing, and the outside of the gearbox clutches and couplings. The gears are machine elements that transmit motion by means of successively engaging teeth. The transformation of speed is without slip and loss of synchronization. Thus, the relationship between the angular velocities of the driving gear to the driven gear, or the velocity ratio, of a pair of mating teeth results from the number of teeth of the driving and driven gear. A gear train consists of two or more gears (a pinion and wheels), which serve to transmit motion from one axis to another. The gear train operational condition depends on the drive systems. Some gearboxes work with the steady-state rotational speed while the rotational speed of the others varies in a certain range. Independently on the rotational speed, the transmitted power can be constant or variable.

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's 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 to prevent noise radiation — with the consequences of low efficiency and difficult 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.

## 2. HISTORICAL NOTES

The author of this paper was a member of the team for gearbox noise reduction at TATRA, a Czech company that produces off-road and on-road trucks. TATRA trucks are used to carry goods and have a maximum authorized total mass exceeding 12 tonnes. In the past, TATRA truck gearboxes were noted for a very robust design. Their only disadvantage was that they were "a little noisy." The introduction of the 80 dB limit for the peak pass-by noise level of on-road trucks in Europe in 1994 created a challenge to start research works on gearbox noise reduction. It should be noted that the pass-by noise measurements shall be made using the time F-weighting and the fre-quency A-weighting on the dB scale at a distance of 7.5 m from the centre of the test track. In the USA, the SPL is measured at the distance of 50 feet (doubled in comparison to the European test method), therefore 6 dB has to be added to the USA limit to make it correspond to the European level. The truck's operational condition during pass-by noise measurements is based on the full accel-eration test specified in the International Standard, ISO R362-1982 (E) "Acoustic-Measurement of Noise Emitted by Accelerating Road Vehicles-Engineering Methods" or SAE J366 Surface Vehicle standard — (R) "Exterior Sound Level for Heavy Trucks and Buses (Issued 1969-07, Reaffirmed 1987-02)." The test requires the vehicle to be driven through a test track at full acceleration.

The noise level of road vehicles was decreased in intervals as it was possible to fulfil the requirements given by the vehicle noise legislation. The peak SPL of the mentioned category of trucks was set to 84 dB before 1994. The 84 dB trucks were easy to produce but the last decrease of the SPL by 4 dB required fundamental improvements. The truck engine is the noisiest unit. Engine noise reduction for TATRA trucks of the T 815-2 type, introduced in production in 1989, was reached by using covers and shields that fit over the engine compartment and, also, by reducing the rate of the rise in cylinder pressure. the reduction of the SPL to reach the last 4 dB was not easy. To solve the gearbox noise problem it was necessary to decide if the special enclosure should be extended over the gear-box or not. As mentioned before, the designers decided to solve the gearbox noise problem at the very source. The experiments show that the effect of the enclosure, having an opening for the Cardan drive shafts, on the SPL is less than the 3 dB reduction at a distance of 1 m. The TATRA truck power train has the gearbox separated from the engine and these units are connected to it by the Cardan shaft, therefore, it is not easy to design the enclosure protecting noise radiation of the gearbox. This paper describes the research work on quieting the truck gearbox. As previously stated, the tuck gearbox is without enclosure and their operating rotational speed and load are not steady but are variables. The experience gained from research work on the truck gearbox noise reduction can be applied, generally, to any other transmission.

## 3. VEHICLE PASS-BY NOISE MEASUREMENTS

Transmission noise should be assessed in terms of external vehicle noise. In order to analyze the contributions of each individual simple-gear train to the overall noise level, it is preferable to measure the time history of sound pressure and engine revolutions per minute (RPM) during the pass-by test. The vehicle transmission units contain multiple shafts that may run coherently through fixed transmissions, or partially related through torque converter slippage, or independently, as for instance a cooling fan in an engine compartment. The emitted tonal noise consists of harmonic components whose frequency is a multiple of the corresponding part's rotational speed (RPM). It is supposed that the instantaneous rotational speed is measured in the form of a pulse train and is then transmitted via a radio transmitter to be recorded together with the noise and threshold signals. The overall pass-by vehicle noise level is almost given by the contributions of the mentioned tonal components. Therefore, it is possible to analyse contributions of each individual simple gear train as well as the engine firing noise and the tire noise to the overall noise level. The noise spectrum analysis requires extending the measurement instrumentation by introducing methods to evaluate the instantaneous engine RPM during pass-by tests. In contrast to the measurement with a stationary vehicle, during a pass-by noise test, the effect of the relative speed of the noise source and the microphones is translated into a Doppler phenomenon and causes positive (approaching) or negative (receding) frequency shifts (typically from 2 to 5%) in the signals received by the microphones. The frequency $f_0$ of sound waves that are emitted by a source moving at velocity $v_R$ relative to the stationary microphone receiving it, shifts the sound waves frequency to the value of $f_1$:

$$f_1 = \frac{c}{c + v_r} f_0; \qquad (1)$$

where c is the sound velocity. The Doppler frequency shift can lead to considerable errors for order tracking using a narrow-band pass filter.

The analysis of the truck pass-by noise sources based on the frequency spectra decomposition was introduced at the TATRA company in 1994 (first published in 1996[2]).[1] An example of the results is shown in Fig. 1 and contains a plot of the relative sound power contributions (root mean square (RMS) squared) versus time elapsed from the moment when the truck crosses the test track entry point. The data corresponds to gearbox improvements discussed later. It is assumed that all the shafts run coherently through fixed transmissions. Each power contribution is the sum of 5 partial contributions associated to corresponding harmonics of a base frequency. The base frequency and five of its harmonics of the sound waves excited by the truck's 8-cylinder, 4-stroke engine divided into the engine firing (E1) and timing gears (E2 through E5), gearbox gears (gear trains designated by N, 3, and SG), axle gears (Axles), and eventually tire profile were evaluated using the engine rotational speed measurement (RPM) after correction according to Eq. (1) and prediction of the truck's instantaneous velocity and position (Dist) on the track, taking into account the Doppler shift and the 1% relative passband width around the centre frequency. Note the cursor values at the peak value (Max) of the sound pressure. The phase modulation effect in the noise signal during pass-by tests was analysed by Kubena.[3]

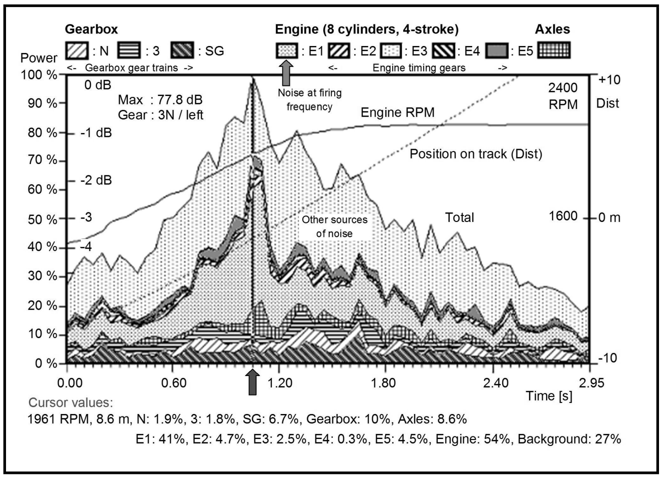


**Figure 1.** Truck noise sources contribution to overall pass-by noise level.

Regardless of these previous successful results, software was later developed for pass-by noise analysis based on the Vold-Kalman order-tracking filtration[4–6], which included a mathematical model of the tire slip as a function of the longitudinal and normal force acting on the tire by the road surface.[1] Assuming that the tire slip and the vehicle position at the end of the test track end is corrected by less than 1 m, then it is negligible compared to the truck dimensions.

## 4. FACTORY LIMIT FOR THE GEARBOX SPL

To fulfil the pass-by noise level limit for the truck tests according to the aforementioned International Standard, the factory internal limit for the gearbox noise level measured on a test stand by microphones located aside at a distance of 1 m was proposed by technicians having experience with noise problems. This limit level was used to check gearbox quality and as a criterion for improving gearbox design. The gearbox noise was measured on the test stand at the gearbox operational condition, simulating the vehicle pass-by noise test. The SPL limit was setup according to the comparison of the result of the many pass-by and test stand measurements. The mentioned SPL limit for gearboxes is a maximum of the overall SPL in the RPM range at the gearbox input shaft corresponding to the engine RPM range during acceleration on the test track (where the pass-by noise tests take place). The reason for this is that the peak vehicle SPL should be less than the required limit for all the tested gears.

There are two possible arrangements of the test stand, an open-loop test stand and a back-to-back test rig. In contrast to the open-loop test stand, the back-to-back test rig configuration saves drive energy. The torque to be transmitted by the gearbox is induced by a planetary gearbox. The gearbox under testing is enclosed in a semi-anechoic room. The quality of the semi-anechoic room is of great importance for the reliability of the results at engineering accuracy. According to ISO 3744, the reverberation time should satisfy a condition required for the ratio of the room absorption to the measurement surface area (greater than 6) in the frequency range from at least 200 Hz to 3 kHz. The input shaft speed is slowly increased from minimal to maximal RPM while the gearbox is under a load corresponding to full vehicle "acceleration" during the pass-by tests.

In the TATRA company, noise is measured by two microphones located by the side of the gearbox under a test at a distance of 1 m. Accelerometers that are attached on the surface of the gearbox housing, near the shaft bearings, can extend information about the noise sources. A tacho probe, that generates a string of pulses, is usually employed to measure the gearbox's primary-shaft (input shaft) rotational speed.

The effect of the gearbox improvement has to be demonstrated statistically. The gearbox design, namely the gear design parameters, and the production quality before the introduction of the noise reduction improvements in 1994 can be described by the empirical distribution function of the tested gearbox noise level. As it was explained before, the gearbox SPL is detected during the run-up tests on the test stand as a maximum of overall SPL in the RPM range corresponding to the engine RPM during the pass-by tests. The characteristic gearbox noise level is considered the result of the measurement of the noisiest gears at the noisiest side of the gearbox.

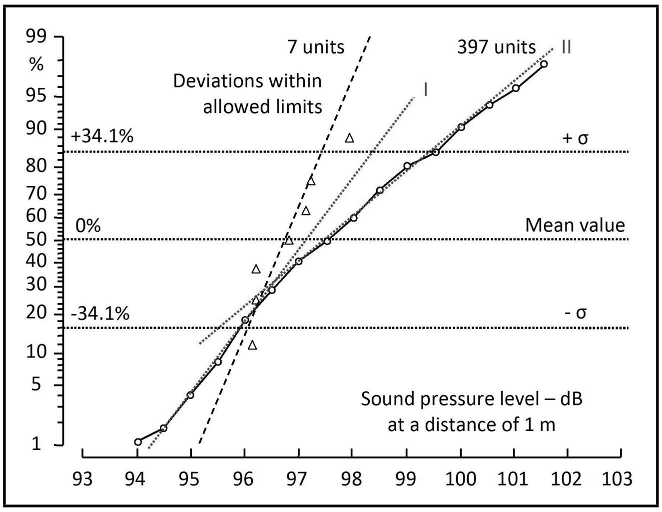


**Figure 2.** Experimental distribution functions of the gearbox SPL equipped with gears produced before 1994.

In 1993, a total number of 397 newly built units were tested and the measurement results are shown in Fig. 2 in the form of the normal distribution probability plot, which is a graphical technique for assessing whether a data set is approximately normally distributed. In fact, the probability plot presents an experimental distribution function (cumulative histogram) in transformed coordinates. The data are plotted against a theoretical normal distribution in such a way that the points should form an approximate straight line. Deviations from this straight line indicate deviations from normality. The normal probability plot is formed by the vertical axis (probability) and horizontal axis with equally probable ordered values. A mean value which is identical with the median for normally distributed data corresponds to the 50% probability. The standard deviation is indicated by $\sigma$ in Fig. 2.

The empirical distribution function for 397 units is a broken-line (I and II sections), which indicates two subsets of the normally distributed data differing in the mean value and the standard deviation. It may be guessed that these subsets probably differ significantly in quality. When discussion about the strategy for quieting gearboxes started, one of the suggestions recommended avoiding the extreme values of the gear deviations. The measurement result of the gearboxes with gear deviations within the allowed limits is shown in Fig. 2, as well. As it is evident, the improvement of production quality is not sufficient for considerable noise reduction. The SPL median of the checked gearboxes is less than the median of the others by approximately 0.8 dB. The only effect of selecting the gearbox is a reduction of the standard deviation of the SPL level values.

## 5. DESCRIPTION OF THE TATRA TRUCK POWERTRAIN SYSTEM

The truck powertrain system consists of the engine, gearbox, differentials, and axles. All these units contain gears. Due to the high rotational speed and transferred torque, gears in a gearbox and axles play the key role in emitting noise. All gears in the TATRA gearbox are of the helical type and the gears in the axles are of the spiral bevel type. The problem of the axle noise will be noted marginally. The main interest is focused on the transmission, which consists of the main and secondary gearbox. The secondary gearbox is sometimes called the drop gearbox due to the fact that this gearbox reduces the rotational speed. In the case of the TATRA trucks, the drop gearbox transfers power to the level of the central tube, which is the backbone of the chassis structure. The main gearbox is two stages and has five basic gears and reverse. As all the basic gears are split (R, N) the number of the basic gears is extended to ten forward and two reverse gears. The gears are designated by a combination of the number character (1 through 5 or 6) and letter (R or N), for example '3N'. According to the EEC regulations valid at the beginning of the 1990s, the basic gears selected for the pass-by tests are No.s 3, 4, and 5. The drop gearbox is either the compound gear train with an idler gear or the two-stage gearbox, extending the number of gears to 12. TATRA does not use a planetary gearbox as the drop gearbox. The kinematic model of gear arrangement as well as the gearbox housing are shown in Fig. 3.

## 6. SOURCES OF GEARBOX NOISE AND VIBRATION

Gearbox noise is tonal. This means that 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

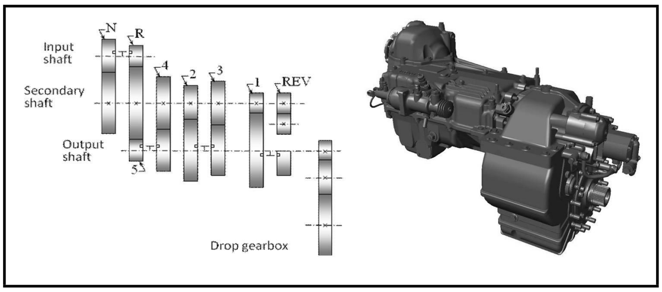


**Figure 3.** TATRA truck (T815-2) gearbox arrangement.

and the number of teeth are referred to as the base toothmeshing frequency or gear meshing frequency $f_{GMF}$. A simple gear train (a pair of meshing gears extended optionally by idler gears) is characterized by only one toothmeshing frequency. All the basic spectrum components are usually broken down into a combination of the following effects[7]:

- 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 toothmeshing frequency and their sidebands due to the modulation effects that are well audible; the noise and vibration of the geared axis systems originated 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 to 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 index-wheel teeth, these ghost components disappear after running-in;
- components originating from faults in rolling-element bearings usually of the low level noise except for fatal bearing faults as the cracking or pitting of the inner or outer race or of the rolling element itself.

Except for the frequency spectrum components originating from the rolling bearings, the frequency of the components, which are associated with the meshing gears, is an integer multiple of the shaft rotational frequency. There are subharmonic components as well. These components are excited at the half of the teeth resonant frequency in high-speed units (thousands RPM) due to the non-linearity of tooth stiffness. Other subharmonics originate from the rate at which the same two gear teeth mesh together (hunting tooth frequency) $f_{HTF}$,

$$g_{HTF} = f_{GMF}\frac{gcd(n_1, n_2)}{(n_1 n_2)}; \tag{2}$$

where $gcd(n_1, n_2)$ is a greatest common divisor of both the numbers $n_1, n_2$ of teeth.[8]

All these spectrum components are designated as orders of the shaft rotational frequency. The described spectrum components can be identified at all the gearbox operational condition, even when the gear pairs are unengaged or unloaded and are oscillating within their backlash range. The unloaded gears produce the idle gear rattle. The dominating components in the frequency spectrum can be identified after averaging either in the time or frequency domain.

## 7. TOOLS FOR GEARBOX NOISE AND VIBRATION ANALYSIS

Analytical methods for gearbox noise and vibration are described in Crocker's handbook.[7] Instrumentation of the test stands is of the Brüel & Kjær origin, namely the signal analyzers of the BK 2034 and 3550 type.

### 7.1. Frequency Domain Analysis

The basic tool for signal processing in diagnostics is the Fourier Transform of signals. The base frequency of all the exciting forces is related to the gearbox shaft rotation frequency. The gearbox is tested during steady-state rotation or run-up/coast-down. Clear information about the origin of extensive vibrations cannot be given by a single frequency spectrum. Extensive vibration is excited when the toothmeshing frequency or its harmonics meet the structural resonance frequency of the gearbox structure. It should be mentioned that any driven unit does not rotate at a purely constant speed but its speed slowly varies around an average value. Spectrum components of the diagnostic signal result from the simultaneous amplitude and phase modulation of the so-called carrying harmonic components that correspond to the excitation at a purely steady-state rotation. An amplitude modulation of harmonic signals arises from the non-uniform periodic load while a phase modulation is due to non-uniform rotational speed. Rotational speed variations at fixed-signal sampling frequency cause the smearing of the dominating components in the frequency spectra.

Analysis of signals from machines running in cyclic fashion is preferred in terms of order spectra rather than frequency spectra. The order spectra are evaluated using the Fourier Transform of time records that are measured in dimensionless revolutions rather than seconds and the corresponding spectrum components are associated with dimensionless orders rather than frequency in Hz. This technique is called order analysis or tracking analysis, as the rotation frequency is being tracked and used for analysis. The resolution of the order spectrum is equal to the reciprocal value of the revolution number per record that is an input data vector of the length, equal to a power of two, for the Fast Fourier Transform (FFT).

The signal/noise ratio is improved by averaging in both the frequency and time domains. Signal resampling to a fixed number of samples per revolution is employed in order to eliminate the phase modulation effect at the tacho pulse frequency.

An example of gearbox noise spectra in dB (Hanning time window and A-weighting) during the run-up test under load is shown in Fig. 4. The input shaft rotational speed was slowly increased from 1000 RPM to 2200 RPM. The spectral map illustrates how the various harmonics fall along radial lines and can, thus, be separated from the constant frequency components due to excessive amplification by a structural resonance. The first five harmonics of the toothmeshing frequency is usually sufficient to set up the frequency range for measurements.

To demonstrate gear noise analysis, the dependence of the overall SPL in dB (Total) and the levels of 5 toothmeshing

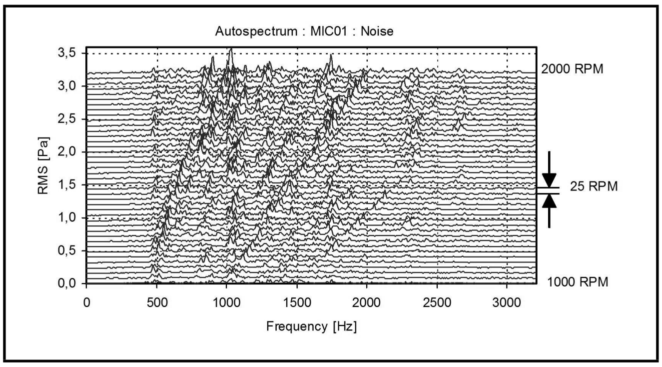


**Figure 4.** Running noise autospectra of the gearbox noise in RMS.

harmonics of all the gear trains under load on the input-shaft rotational speed for the 3N gear are shown in Fig. 5. Except for the 5R gear, only three pairs of the engaged gears, which are designated by N, 3, and SG (Secondary Gearbox), are under load. The panels of the diagram in Fig. 5 titled Gear N, 3, and SG, corresponds to the mentioned gear pairs. The curve in these panels marked by 'Sum' is a sum of the power contributions of 5 harmonic components resulting in the noise level excited only by the appropriate pair of the gears. As the pass-by vehicle noise test is based on the maximum of the overall SPL, the maximum of the gearbox overall SPL (MaxTot) and a maximum of the 5 tonal components SPL (MaxSum) can be chosen as a gear quality criterion. Optionally, the maximum is evaluated for the input shaft rotational speed range either from 1000 to 2200 RPM or for an interval corresponding to the engine rotational speed during the pass-by tests. Due to the low rotational speed of the secondary gearbox gear train, its contribution to the overall (Total) SPL is negligible. The right lower panel in the diagram in Fig. 5 compares the contribution of all the gear train under load to the overall SPL of the gearbox. The minimum of the difference between the overall SPL and the contributions of the N, 3, and SG gears for the mentioned RPM range is designated by MinDiff. As was noted in the introduction section, the main sources of the gearbox noise are gears under load.[7]

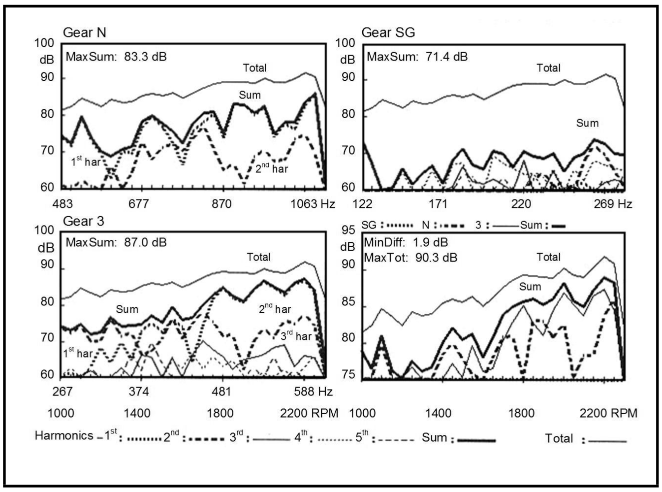


**Figure 5.** Overall (Total) SPL and level of the of 5 toothmeshing harmonics of all the gear trains under load for the 3N gear vs. input-shaft rotational speed.

## 7.2. Time Domain Analysis

### 7.2.1. Averaging synchronised with the rotational frequency

To study the excitation at each gear mesh cycle, the time domain analysis based on synchronous time-domain averaging is more suitable than the frequency domain analysis. This technique is also known as a signal enhancement which has been treated by many authors (e.g. McFadden,[9] Angelo,[10] etc.) and can be regarded as a "magnifying glass" (Angelo), whereby one can "focus" on the shaft to be examined. Noise and vibration signals are sampled using a tracking technique in such a way that the length of the time record is equal to the time interval of a gear revolution. Time records are triggered synchronously with a shaft rotation and all the records contain the same number of samples (equal to a power of two). The time domain averaging of the samples on the same position in all the records results in a comb filtering with centre frequencies coinciding with the integer multiples of the rotational frequency. The frequency response function of synchronous averaging is dis-cussed in Chapter 46 of Crocker's handbook.[11]

Averaging in the time domain results in the reduction of the RMS of an uncorrelated random signal, which is an obvious part of measured signals, that is proportional to the reciprocal of the square root of the average number. For instance, if the averaging number equals 100 then the RMS (Root Mean Square) of random signals are reduced by 20 dB. Synchronous averaging attenuates considerably the tonal components with frequencies which are a fractional multiple of the trigger frequency. The attenuation concerns the components excited by rolling bearing and by the gears rotating at the frequencies which are a fractional multiple of the trigger frequency. Except for engine timing gears, an ideal set of gears for even wear on each tooth does not have tooth numbers with a common factor other than one, which results in the mentioned fractional multiple. The gearing ratio of the timing gear train driving the camshaft is 2:1 for a 4-stroke engine and the common factor is therefore greater than one.

### 7.2.2. Average toothmesh and envelope analysis

The averaged time records corresponding to a gear rotation are corrupted by modulation signals, which are in correlation with gear geometry errors or varying gear load. The modulation signals give rise to sidebands around the carrying components with a frequency that is corresponding to the harmonics of the toothmeshing frequency. If all the spectrum components, except harmonics of the toothmeshing frequency, are removed then the purely periodic toothmesh waveform without the modulation effects is obtained. In view of this, the shape of this function may be illustrated by only one period corresponding to the time interval of one tooth pitch rotation. In this way, filtered signals are called average toothmesh signals.[12] The average toothmesh acceleration measured on the gearbox housing close to the shaft bearing is proportional to the dynamic forces acting between the teeth in mesh. The average toothmesh is a tool to represent the average mesh cycle.

It can be observed that both the average toothmesh signals corresponding to meshing gears have the same shape (see Fig. 6). This fact follows from Newton's third law. To assess a uniformity of toothmeshing during a complete gear rotation, an

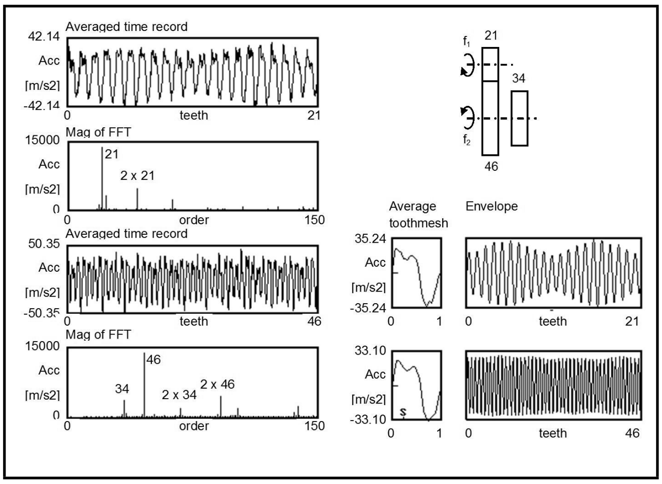


**Figure 6.** Acceleration responses of the 21- and 46-tooth gear, envelops and averaged toothmesh signals.

envelope analysis can be employed. The theory of analytical signals and the Hilbert Transform are tools for envelope detecting. In contrast to the similarity of the averaged toothmesh signals, the envelopes of signals generated by meshing gears may differ from each other.

The average toothmesh based on the acceleration signal is a useful tool for the analysis of the tooth contact during a tooth pitch rotation. The shape of the average toothmesh is almost independent on the rotational speed.[12]

### 7.2.3. Averaging synchronised with the hunting tooth frequency

The time records may be averaged synchronously with the hunting frequency as well.[8] Like averaging synchronised with rotational frequency, it is assumed that the records of the noise or vibration signal, corresponding to a complete gear rotation, is resampled to the new records with the same number of samples. Resampling could be done for a complete revolution of either the pinion or wheel. If the number of the pinion teeth is equal to n1 and the number of the wheel teeth is equal to $n_2$ and simultaneously the greatest common divisor $gcd(n_1, n_2)$ is equal to 1 then the tooth of the pinion will enter into the mesh with the same tooth of the wheel after $n_2$ rotations of the pinion. Let $n_2$ records be connected to a multiple record, whereas each of them is corresponding to a complete rotation of the gear with $n_1$ teeth. If the mentioned multiple record is averaged in the time do-main then the samples represent averaged values corresponding to the mesh cycle of the same tooth pair.

The arrangement of the averaged and squared signal samples into a matrix with rows corresponding to the gear revolution results in a 3D colour map giving information about the geometrical inaccuracy of both the gears. An example for the gear train composed of the 27- and 40-tooth gear is shown in Fig. 7. The colour on the vertical column shows the resulting vibration amplitude of a tooth of the 40-tooth gear with any tooth of the 27-tooth gear. As the gears are newly produced, the picture shows an effect of the residual surface waviness.

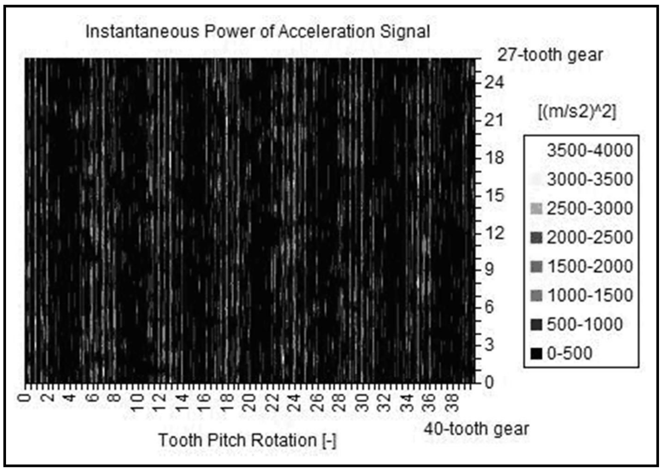


**Figure 7.** Averaging the synchronised with the hunting tooth frequency.

## 8. GEARBOX IMPROVEMENT AIMED AT NOISE REDUCTION

The research work on noise reduction employs different methods to identify the dynamic properties and to discover the dynamically weak parts of gearbox structures. In reality, the effect of these improvements, which are limited by the given gearbox structure, on the overall SPL is of small significance. However, they are important because, after introducing these improvements, the SPL of newly built units varies in correlation with errors of gear geometry. The most efficient improvement can be reached when the noise problem is controlled at its very source, which is the tooth contact of meshing gears.

### 8.1. Gearbox Housing Stiffness

The modal properties of gearbox housing were identified using operational deflection shapes and experimental modal analysis. The effect of the gearbox load on the vibration response of each gear in the time domain using synchronous averaging with the rotational frequency was analysed. The "power" can flow through the gearbox in many ways according to the gear used. The gearbox housing, shafts, and gears are not an ideally rigid structure. The compliant gearbox structure is deformed under load. The tooth mesh of the R gear pair depends on the gear pair under load between the secondary and output shaft (see Fig. 3). The averaged vibration responses of the 21-tooth gear, belonging to the R gear train, for the 3$^{rd}$, 4$^{th}$, and 5$^{th}$ gears under load are shown in Fig. 8. The responses are differing due to the gearbox deformation.

To prevent uncertain tooth contact, the gearbox housing was stiffened by using ribs, inside the housing the ribs ware perpendicular to the shafts while two massive ribs at the housing surface were parallel to the shafts. The additional ribs ensure that the bearing close to the 21-tooth gear is stiffened enough and the vibration responses were not differing any more.

### 8.2. Geometric Design of Gears

The averaged toothmesh vibration signals are contact-ratio sensitive. It is well known that the contact ratio (simply, though not quite accurately stated, contact ratio is the average number of teeth in contact during a mesh cycle) is one of the most important parameters determining gear tooth excitation and,

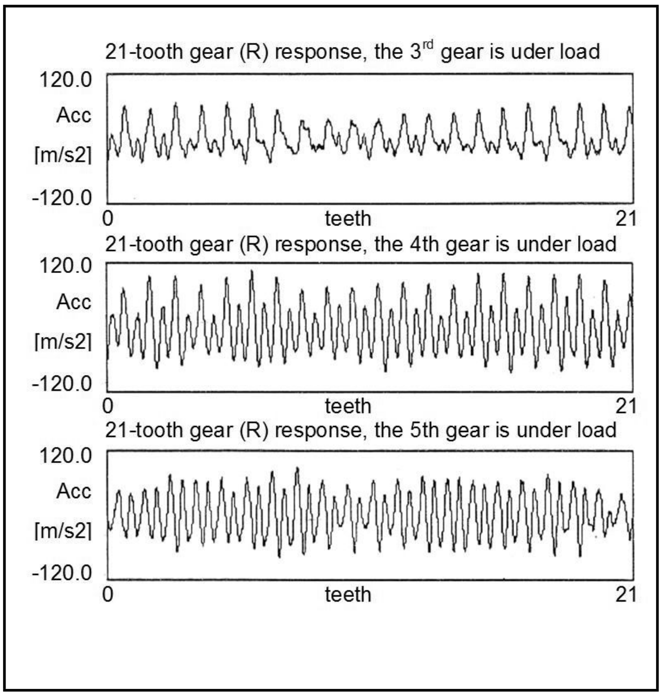


**Figure 8.** Synchronously averaged response of the 21-tooth gear under load.

thus, gearbox noise level.[13–15] Three examples of the average toothmesh acceleration signals versus tooth pitch rotation, shown in Fig. 9, deal with helical gears under load differing in profile contact ratio ($\varepsilon_\alpha$) while the overlap ratio ($\varepsilon_\beta$ face contact ratio) is approximately equal to 1.0. The total contact ratio ($\varepsilon_\gamma$) as a sum of the profile contact ratio and overlap ratio is indicated by the diagram legend.

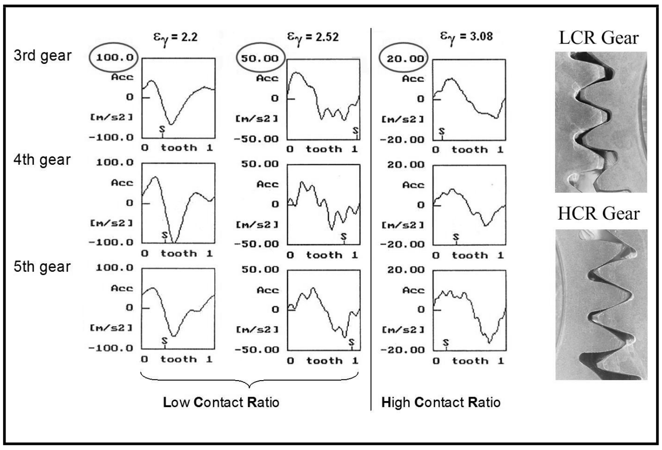


**Figure 9.** Effect of the total contact ratio on the average toothmesh acceleration signal of the N gear train for the 3$^{rd}$, 4$^{th}$, and 5$^{th}$ gear train under load.

To confirm reproducibility of the tooth mesh cycle, the average toothmesh of the N gear train for the 3$^{rd}$, 4$^{th}$, and 5$^{th}$ gear train under load is shown in Fig. 9. The value of the profile contact ratio, which is less than 2.0, is called Low Contact Ratio (LCR) gearing while the gearing with this parameter equalled to 2.0 or more is designated as High Contact Ratio (HCR). The integer value for the profile contact ratio and the overlap ratio result in considerable reduction of gearbox vibration and noise. It can be estimated that introducing the HCR toothing results in reducing the noise level of gearboxes with LCR toothing by approximately 6 dB. For example, experiments[16] with the HCR gears of the contact ratio ranging from 4.0 to 5.0 show that the transmission sound level decreases by approximately 10 dB per 1.0 increase in contact ratio.

Evaluation of the averaged toothmesh signal is an effective method for verifying contact ratio, detecting a regular error in tooth profile geometry, and improving toothing by modification of tooth profile and lead while the envelope of acceleration signal during a complete revolution is important only for quality control.

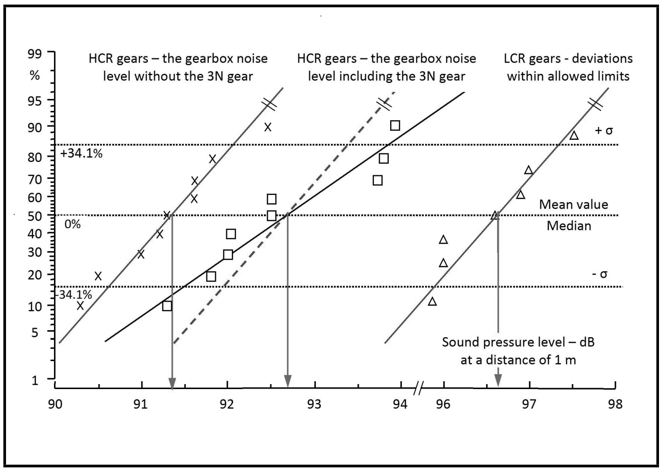


**Figure 10.** SPL experimental distribution functions of the gearbox equipped by LCR and HCR gears.

The effect of the tooth design~~, described in Chapter 5,~~ on the 1
gearbox noise level is shown in Fig. 10, which represents the normal distribution probability plot containing three sets of data. The first one as a reference measurement of the gearbox noise with the LCR gears ($\varepsilon_\gamma = 2.52$) produced before 1994 originates from the tests of the gearboxes with geometric deviations within limits. The remaining sets of data correspond to the gearbox with the HCR gears ($\varepsilon_\gamma = 3.08$), production of which was started by TATRA in 1994, therefore, at the same time as for example in Germany (Ecosplit 3 type, ZF Friedrichshafen AG). The empirical distribution functions of these two measurements are computed including with the measurement when the 3N gear is used and without this measurement. The intersection of the distribution function with the horizontal line 50% results in the mean value (median for normally distributed data) of the gearbox SPL indicated on the horizontal scale.

## 9. EFFECT OF GEAR QUALITY AND OPERATION CONDITIONS ON THE GEARBOX SPL

The gear tooth quality is given by the permissible maximum values of individual variations. Individual variations are those variations from their nominal values, which are exhibited by the various parameters of the gear teeth, such as pitch, profile shape, base diameters, pressure angle, tooth traces and helix angle. It is known that a gear could be absolutely perfect referring to these variations, and yet be very noisy in the conditions of meshing. We can find in the tolerance data a few microns and have many times more deformations due to the loading. In reverse some low noisy gears are imperfect.

An example of the effect of the gear tooth quality on the emitted noise in SPL is shown in Fig. 11. The data was

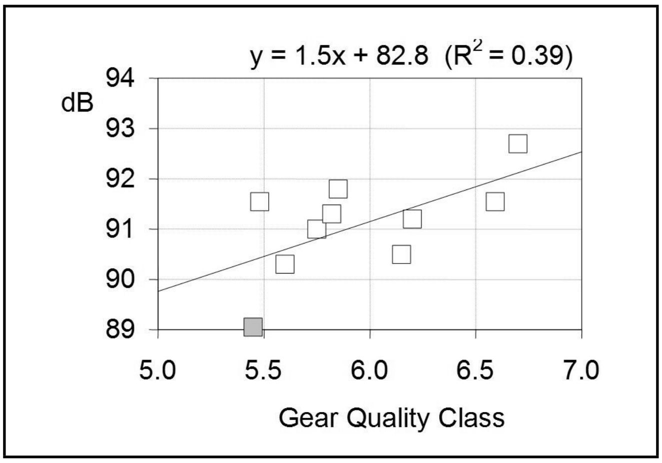


**Figure 11.** Effect of mean gear quality on noise level in dB at 1 m.

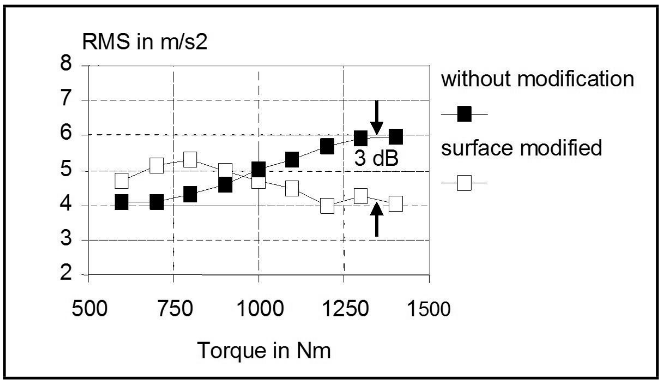


**Figure 12.** Effect of load on RMS of acceleration.

taken from measurements on a truck gearbox with gears finished by grinding. It may be noticed that the SPL minimum (89 dB) corresponds to gears produced by a shape grinding machine (MAAG) while the others correspond to the grinding machine employing the continuous shift grinding method (Reishauer). However, it can be estimated that an improvement by class 1 of DIN 3961 quality results in the SPL reduction by approximately 1.5 dB. Gear Quality Class is a quantity that is influenced mainly by the variations in the tooth profile (tooth trace form or tooth alignment angle) and weakly in the radial run-out. This quantity is computed as a weighted sum of the tooth trace form and tooth alignment angle (the value of the weighting coefficient for both deviations is equal to 0.4) and the radial run-out.[17] The gearbox noise level correlates with the gear geometry deviations if the gearbox structure is stiff enough.

An example of the effect of gearbox load on the RMS value of gearbox acceleration is shown in Fig. 12. The gearbox design input torque is equal to 1200 Nm. To prevent the increase of vibration and, consequently, the noise level due to the teeth and gearbox structure deformation, the modification of the tooth profile and lead by crowning across the face and tapering the lead is introduced.[18] Comparison of the modified toothing with the toothing, which is without modification, shows that the gear pair with the modified surface meshes more smoothly at nominal loading than the gears without modification. This design improvement can result in reducing the noise level by 3 dB during the tests on the test stand.

The positive effect of introducing the HCR gears into the TATRA gearboxes and the tooth surface modification on the noise level of 6 gears (3R through 5N) which are important for the pass-by noise measurement is shown in Fig. 13. The truck noise sources contribution to overall pass-by noise level for the nosiest gear (3N) is shown in Fig. 1. It should be noted that the gear design of the tested truck was improved by introducing the HCR gears. The gearbox noise was not only reduced but the gearbox life time was doubled due to decreasing of dynamic forces acting between teeth. The mentioned gearbox quieting caused some problems for the truck manufacture in decreasing the SPL in the cab in such a way that drivers were able to distinguish the tonal noise emitted by the other unit, for example, by axles, mainly when they used engine braking. Cab environment is important from the point of view of the subjective perception of noise. The axle noise reduction is beyond the topic of this paper.

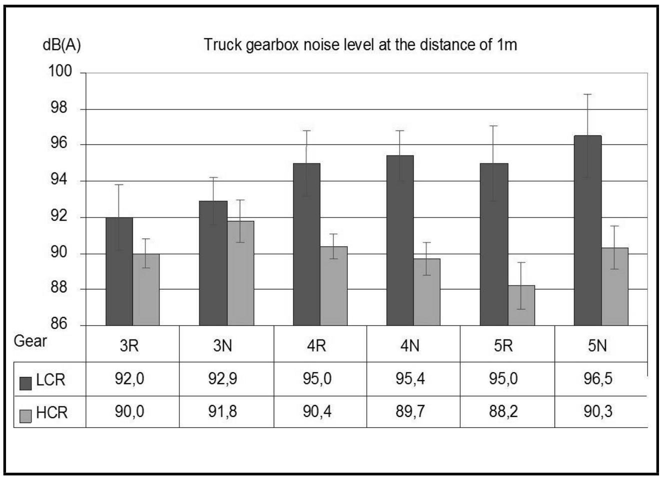


**Figure 13.** The gearbox overall noise level of the HCR gears in comparison with the LCR gears for 6 engaged gears important for the pass-by noise level measurement.

## 10. TRANSMISSION ERROR MEASUREMENTS

Noise and vibration problems in gearing are mainly concerned with the smoothness of the drive. The parameter that is employed to measure smoothness is the Transmission Error[19] (TE). This parameter can be expressed as a linear displacement at a base circle radius defined by the difference of the output gear's position from where it would be if the gear teeth were perfect and infinitely stiff. Many references have attested to the fact that a major goal in reducing gear noise is to reduce the transmission error of a gear set. Experiments[16] show that decreasing TE by 10 dB (approximately 3 times less) results in decreasing transmission sound level by 7 dB. The TE of the TATRA gearbox was measured later by using the method developed at the VSB — Technical University of Ostrava.

The basic equation for TE of a simple gear set is given as

$$TE(m) = \left(\Theta_2 - \frac{n_2}{n_1}\Theta_1\right) r_2; \tag{3}$$

where $n_1, n_2$ are teeth numbers of pinion and wheel respectively, $\Theta_1, \Theta_2$ are angles of rotation of the mentioned gear pair and $r_2$ is a wheel radius.

TE results not only from manufacturing inaccuracies, such as profile errors, tooth pitch errors and run-out, but from bad design. The pure tooth involute deflects under load due to the finite mesh stiffness caused by tooth deflection. A gear case and shaft system deflects due to load, as well. While running under load, one of the important parameters, tooth contact stiffness, varies, which excites the parametric vibration and, consequently, noise.

There are many possible approaches to measuring TE but, as Derek Smith points out in his book,[20] in practice, measurements based on the use of encoders dominate. An example of the TE measurement using the encoders, shown in this paper, deals with the gear train consisting of the 27- and 44-tooth gears under test. The incrementary rotary encoders are attached to the pinion and wheel shaft. Both of the encoders generate a string of pulses. As a consequence of Shannon's sampling theorem, a few pulses have to be recorded during each mesh cycle. This means that the number of pulses produced per encoder revolution must be a multiple of the tooth number. If five harmonics of toothmeshing frequency are required then the number of pulses per gear revolution must be at least ten times greater than the number of teeth. The encoder generating 500 pulses per revolution seems to be an optimum.

There are two approaches to processing the pulse signals. The first is based on the measurement of the time interval between adjacent pulses using the high frequency pulse generator (for instance, up to 1 GHz in the Rotec signal analyzer) and the counter, which is triggered by the encoder output signals. The instantaneous angular velocity is primary information for TE evaluation and needs integration to obtain the rotation angle variations. Henriksson and Pärssinen present an example of this measurement.[19] The second method is based on the phase demodulation of the sampled pulse signals, which is described below. This method gives as primary information the instantaneous rotation angle and is especially suitable for the FFT analyzers.[21]

A perfectly uniform rotation of the gear produces an encoder signal having in its frequency spectrum a single component at the frequency that is a multiple of the gear rotational frequency. The phase of this carrying component is a linear function of time. As the phase is the argument of the cosine function, it can be associated with the gear rotation angle. The non-uniform rotation results in a small variation of the rotation angle around the mentioned linear function of time. In this case, the basic frequency of the pulse signal is modulated, which gives rise to sidebands around the carrying component in the frequency spectrum.

The phase modulation signal is a part of the phase of an analytical signal that is evaluated using the Hilbert Transform technique. The complex analytical signal is compounded from the real part, which is the sampled pulse signal, and the imaginary part, which is the mentioned Hilbert Transform of the signal real part. As the angle of the complex values ranges from $-\pi$ to $+\pi$, the phase of the analytical signal (as the time function) contains jumps by reaching the value of $-\pi$ or $+\pi$ radians. The true phase can be obtained by the so-called phase unwrapping algorithm. The un-wrapping algorithm is based on the fact that the absolute value of the difference between two consecutive samples of the phase is less than $\pi$. The described algorithm is called phase demodulation.

The Hilbert Transform can be evaluated by using either FFT or a digital filter (Hilbert transformer). Before evaluating FFT, it is recommended to resample the measured signal according to the gear rotational frequency in such a way that the length of the resampled time record equals to a power of two, namely to at least 2048 samples per gear revolution for the 500-pulse encoder. The pulse signal order spectrum for the 2048-sample record ranges to the value of 800 orders. There is a space for $\pm 300$ sideband components around the carrying component of the order equal to 500.

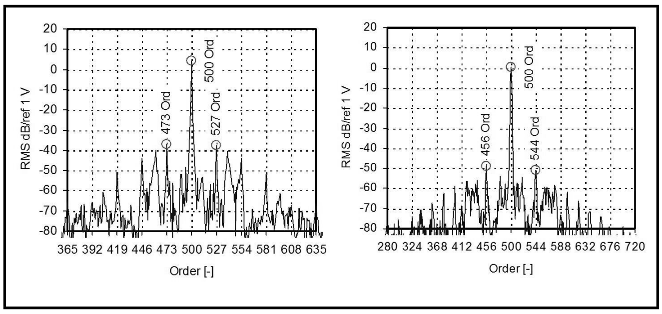


**Figure 14.** Frequency spectrum of phase modulated signal generated by the encoders that are attached to the secondary shaft (27-tooth gear) and the output shaft (44-tooth gear) of the gearbox shown in Fig. 3.

The gear speed variation at the toothmeshing frequency results in the phase modulation of the impulse signal base frequency. As noted above the phase-modulated signal contains sideband com-ponents around the carrying component situated in Fig. 14 at $500 \pm 27k$ orders for the 27-tooth pinion and at $500 \pm 44k$ orders for the 44-tooth wheel, where $k$ = 1, 2,.. is an integer. The spectrum frequency axis in Fig. 14 is in order which is dimensionless quantity corresponding to a multiple of the gear rotational frequency. Take notice of the fact that the dominating components in both the sideband families exceed the background noise level at least by 20 dB or even more. Both the spectra were evaluated from time signals that are a result of the synchronized averaging of 100 revolutions of gears under testing.

Phase demodulation of the averaged pulse signals results in the phase variation during a rotation of both the meshing gears. As in the case of the average toothmesh of the vibration signal, both the phase variation signal can be averaged again to obtain an averaged representative for angular vibration during the gear tooth pitch rotation. Angle variations can be easily transformed into arc length variations.

The arc length difference is not a final step for evaluating TE. As both the encoder signals are recorded separately, the true phase delay between these signals is to be detected. This problem can be solved thanks to the fact that the average toothmesh responses (for example, in acceleration of some point on the gear case) to dynamic forces acting between meshing teeth are theoretically the same. Therefore, both the encoder pulse signals are sampled together with the acceleration signal. A two-stage averaging of the twice-measured acceleration signal gives average toothmesh responses that are delayed against each other. The lag for the maximum correlation gives the value of relative delay.

TE is given as the difference between the angular vibration signals in the arc length produced by the meshing gears. The result for three levels of loading of the $2^{nd}$ gear with the N and R gear under load is shown in Fig. 15. Note that the transmission error is almost independent of the torque, which means that the tooth surface modification is optimal. The design of the TATRA truck gear-box results in TE, which is within the range corresponding to car gearboxes. As Derek Smith claims, the truck gears work with TE in the range of +/-10 microns.

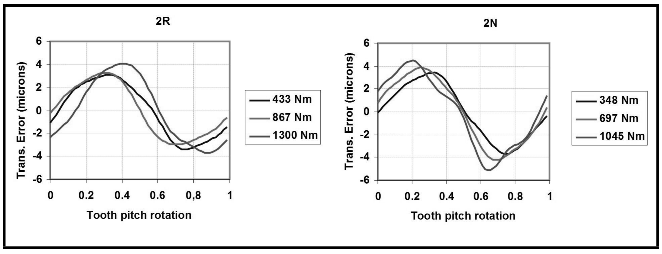


**Figure 15.** Transmission error of the 27- and 44- tooth gears against rotation angle in tooth pitch rotation.

## 11. CONCLUSION

This paper reviews the research work on reducing truck gearbox noise. Gear design improvement is preferred to protection against noise by using a gearbox enclosure. This approach is more effective than the additional cover. The measurement methods and the effect of improving the gear design focused on the problem of reducing gear noise are described. Gearbox noise and vibration analysis is based on the time and frequency methods. Gearbox noise is tonal with a set of dominating frequency components. The sum of the power contributions of the toothmeshing harmonic components results in the noise level of an individual gear pair. Averaging the acceleration signal in the time domain, synchronised by revolutions and tooth-pitch rotations, results in an averaged tooth mesh response serving to compare the effects of improving gear design. Gear design and accuracy may be tested by the transmission error measurement.

The paper reviews the effect of the most efficient improvements reducing noise excited by gears, as well. Concerning the gearbox noise problem, one can conclude that a low noise gearbox requires sufficiently rigid housing, shafts and gears, and the HCR gears and the tooth surface modification for design load. The positive effect of introducing the HCR gears depends on the gear quality class. Gears finished by grinding are needed. All these improvements introduced by the TATRA company result in a decrease of the gearbox noise, which was measured on the test stand at the distance of 1 m by 8 dB at minimum. The TATRA truck gearboxes do not require an enclosure to fulfil the requirements given by the vehicle noise legislation.

## 12. ACKNOWLEDGMENT

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## ~~REMARKS~~

1. ~~Chapter 5 of what book? (Or section 5 of this paper?)~~