



**KTH Industrial Engineering  
and Management**

## **GEARBOX NOISE**

**CORRELATION WITH TRANSMISSION ERROR  
AND INFLUENCE OF BEARING PRELOAD**

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**Gearbox Noise – Correlation with Transmission Error and Influence of Bearing Preload**

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## ABSTRACT

The five appended papers all deal with gearbox noise and vibration. The first paper presents a review of previously published literature on gearbox noise and vibration.

The second paper describes a test rig that was specially designed and built for noise testing of gears. Finite element analysis was used to predict the dynamic properties of the test rig, and experimental modal analysis of the gearbox housing was used to verify the theoretical predictions of natural frequencies.

In the third paper, the influence of gear finishing method and gear deviations on gearbox noise is investigated in what is primarily an experimental study. Eleven test gear pairs were manufactured using three different finishing methods. Transmission error, which is considered to be an important excitation mechanism for gear noise, was measured as well as predicted. The test rig was used to measure gearbox noise and vibration for the different test gear pairs. The measured noise and vibration levels were compared with the predicted and measured transmission error. Most of the experimental results can be interpreted in terms of measured and predicted transmission error. However, it does not seem possible to identify one single parameter, such as measured peak-to-peak transmission error, that can be directly related to measured noise and vibration. The measurements also show that disassembly and reassembly of the gearbox with the same gear pair can change the levels of measured noise and vibration considerably. This finding indicates that other factors besides the gears affect gear noise.

In the fourth paper, the influence of bearing endplay or preload on gearbox noise and vibration is investigated. Vibration measurements were carried out at torque levels of 140 Nm and 400 Nm, with 0.15 mm and 0 mm bearing endplay, and with 0.15 mm bearing preload. The results show that the bearing endplay and preload influence the gearbox vibrations. With preloaded bearings, the vibrations increase at speeds over 2000 rpm and decrease at speeds below 2000 rpm, compared with bearings with endplay. Finite element simulations show the same tendencies as the measurements.

The fifth paper describes how gearbox noise is reduced by optimizing the gear geometry for decreased transmission error. Robustness with respect to gear deviations and varying torque is considered in order to find a gear geometry giving low noise in an appropriate torque range despite deviations from the nominal geometry due to manufacturing tolerances. Static and dynamic transmission error, noise, and housing vibrations were measured. The correlation between dynamic transmission error, housing vibrations and noise was investigated in speed sweeps from 500 to 2500 rpm at constant torque. No correlation was found between dynamic transmission error and noise. Static loaded transmission error seems to be correlated with the ability of the gear pair to excite vibration in the gearbox dynamic system.

**Keywords:** gear, gearbox, noise, vibration, transmission error, bearing preload.



## **ACKNOWLEDGEMENTS**

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The financial support of the Swedish Foundation for Strategic Research and the Swedish Agency for Innovation Systems – VINNOVA – is gratefully acknowledged. Volvo Construction Equipment is acknowledged for giving me the opportunity to devote time to this work.

Professor Sören Andersson is gratefully acknowledged for excellent guidance and encouragement. I also wish to express my appreciation to my colleagues at the Department of Machine Design, and especially to Dr. Ulf Sellgren for performing simulations and contributing to the writing of Paper D, and Dr. Stefan Björklund for performing surface finish measurements. The contributions to Paper C by Dr. Mikael Pärssinen are highly appreciated. All contributions to this work by colleagues at Volvo are gratefully appreciated.

Eskilstuna, December 2008.

Mats Åkerblom



## **LIST OF APPENDED PAPERS AND DIVISION OF WORK**

### **Paper A**

Åkerblom M. "Gear Noise and Vibration – A Literature Survey", TRITA-MMK 2001:11 / ISSN 1400-1179, ISRN/KTH/MMK/R-01/11-SE, Stockholm, 2001.

### **Paper B**

Åkerblom M. "Gear Test Rig for Noise and Vibration Testing of Cylindrical Gears", Proceedings OST-99 Symposium on Machine Design, pp 183–189, ISSN 1400-1179, Stockholm, 1999.

### **Paper C**

Åkerblom M. and Pärssinen M. "A Study of Gear Noise and Vibration", TRITA-MMK 2002:8, ISSN 1400-1179 / ISRN/KTH/MMK/R-02/8-SE, Stockholm, 2002.

Åkerblom initiated and planned the work, specified the test gears, performed the transmission error computations, drew the conclusions, and wrote the major part of the paper.

Pärssinen performed the noise and vibration measurements, contributed to the planning of the work, wrote section 6, and contributed to the writing of some of the other sections.

### **Paper D**

Åkerblom M. and Sellgren U. "Gearbox Noise and Vibration – Influence of Bearing Preload", TRITA-MMK 2008:17 / ISSN 1400-1179, ISRN/KTH/ MMK/R-08/17-SE, Stockholm, 2008.

Åkerblom planned the experiments and wrote the major part of the paper.

Sellgren performed the finite element analysis and contributed to the writing.

### **Paper E**

Åkerblom M. "Gear Geometry for Reduced and Robust Transmission Error and Gearbox Noise", TRITA-MMK 2008:18, ISSN 1400-1179 / ISRN/KTH/ MMK/R-08/18-SE, Stockholm, 2008.

## **ADDITIONAL PUBLICATION**

Sellgren U. and Åkerblom M. "A Model-Based Design Study of Gearbox Induced Noise", International Design Conference – Design 2004, Dubrovnik, May 18–21, 2004.



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### **APPENDED PAPERS**

- Paper A
- Paper B
- Paper C
- Paper D
- Paper E



# 1 INTRODUCTION

## 1.1 Background

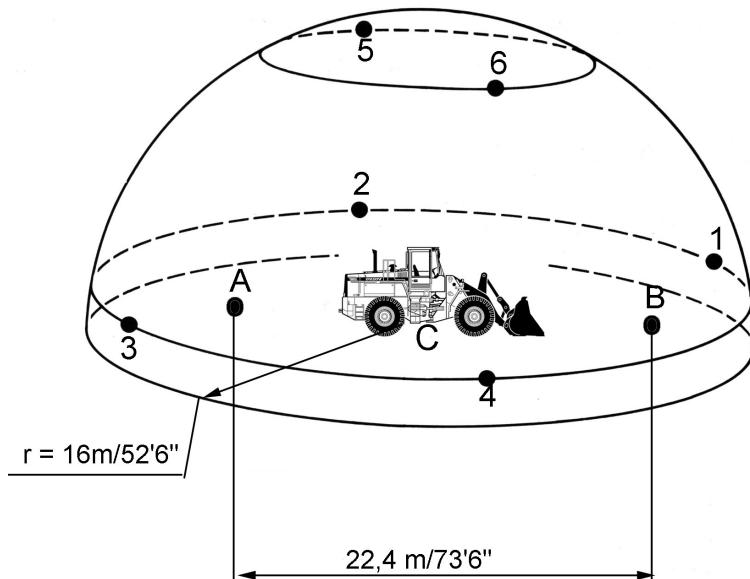
Noise is increasingly considered an environmental issue. This belief is reflected in demands for lower noise levels in many areas of society, including the working environment. Employees spend a lot of time in this environment and noise can lead not only to hearing impairment but also to decreased ability to concentrate, resulting in decreased productivity and an increased risk of accidents. Quality, too, has become increasingly important. The quality of a product can be defined as its ability to fulfill customers' demands. These demands often change over time, and the best competitors in the market will set the standard.

Noise concerns are also expressed in relation to construction machinery such as wheel loaders and articulated haulers. The gearbox is sometimes the dominant source of noise in these machines. Even if the gear noise is not the loudest source, its pure high frequency tone is easily distinguished from other noise sources and is often perceived as unpleasant. The noise creates an impression of poor quality. In order not to be heard, gear noise must be at least 15 dB lower than other noise sources, such as engine noise.

The legal requirements for the exterior noise of construction machinery are shown in Table 1.1.1. These requirements must be met in order to sell construction machinery in the markets shown. The ISO 6395 method of noise testing is described in Figure 1.1.1. Test methods BAV 741.41 and STVZO §49 are pass-by tests.

Legal exterior sound requirements for construction machinery			
Market / Regulation	Test method	Engine power P [kW]	Demand
EU and EES / 2000/14/EU Stage I, as from 2002	ISO 6395	P<66	$L_{WA} \leq 105 \text{ dB(A)}$
		66<P<500	$L_{WA} \leq 85+11*\log P \text{ dB(A)}$
EU and EES / 2000/14/EU Stage II, as from 2006	ISO 6395	P<66	$L_{WA} \leq 102 \text{ dB(A)}$
		66<P<500	$L_{WA} \leq 82+11*\log P \text{ dB(A)}$
Switzerland / BAV 741.41	BAV 741.41	Road cert. Machines	$L_{pA} \leq 88 \text{ dB(A)}$
Germany / STVZO §49	STVZO §49	Road cert. Machines	$L_{pA} \leq 89 \text{ dB(A)}$

**Table 1.1.1 Legal exterior sound requirements for construction machinery**



**Figure 1.1.1 Sound measurement according to ISO 6395. Measurement is carried out as the machine moves from position A to position B and while simulating a work-cycle. Microphone positions are denoted 1 to 6**

There are also special customer and market demands which, while not essential, may in some cases offer economic advantages to the customer who buys a machine with low sound emission. Such equipment may also be required on certain work sites.

<b>Special customer and market exterior sound requirements for construction machinery</b>			
<b>Market / Regulation</b>	<b>Test method</b>	<b>Engine power P [kW]</b>	<b>Demand</b>
Germany / RAL-UZ53 "Blauer Engel"	ISO 6395	P<90	$L_{WA} \leq 79+11*\log P \text{ dB(A)}$
		P>90	$L_{WA} \leq 101 \text{ dB(A)}$
Holland / VAMIL	ISO 6395	P>25	$L_{WA} \leq 81+11*\log P \text{ dB(A)}$
Japan / "Low Noise Construction Machinery"	Japan MOC ISO 6395	P<55	$L_{WA} \leq 102 \text{ dB(A)}$
		55<P<103	$L_{WA} \leq 104 \text{ dB(A)}$
		103<P<154	$L_{WA} \leq 107 \text{ dB(A)}$

**Table 1.1.2 Special customer and market exterior sound requirements for construction machinery**

Sound levels in the cab may not exceed 85 dB(A), measured in accordance with ISO 6396. The sound level is usually considerably lower than this. Even more important, however, as previously indicated, is the fact that gear noise must be much lower than the overall noise level in order not to be perceived as irritating or unpleasant. Interior noise is thus even more difficult to handle than exterior noise, for different people can have different opinions about what is acceptable.

## 1.2 Gear noise

This dissertation deals with the kind of gearbox noise that is generated by gears under load. This noise is often referred to as “gear whine” and consists mainly of pure tones at high frequencies corresponding to the gear mesh frequency and multiples thereof, which are known as harmonics. A tone with the same frequency as the gear mesh frequency is designated the gear mesh harmonic, a tone with a frequency twice the gear mesh frequency is designated the second harmonic, and so on. The term “gear mesh harmonics” refers to all multiples of the gear mesh frequency.

Transmission error (TE) is considered an important excitation mechanism for gear whine. Welbourn [1] defines transmission error as “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 may be expressed as angular displacement or as linear displacement at the pitch point. Transmission error is caused by deflections, geometric errors, and geometric modifications.

In addition to gear whine, other possible noise-generating mechanisms in gearboxes include gear rattle from gears running against each other without load, and noise generated by bearings. In the case of automatic gearboxes, noise can also be generated by internal oil pumps and by clutches. None of these mechanisms are dealt with in this work, and from now on “gear noise” or “gearbox noise” refers to “gear whine”.

MackAldener [2] describes the noise generation process from a gearbox as consisting of three parts: excitation, transmission, and radiation. The origin of the noise is the gear mesh, in which vibrations are created (excitation), mainly due to transmission error. The vibrations are transmitted via the gears, shafts, and bearings to the housing (transmission). The housing vibrates, creating pressure variations in the surrounding air that are perceived as noise (radiation). Gear noise can be affected by changing any one of these three mechanisms. This dissertation deals mainly with excitation, but transmission is also discussed in the section of the literature survey concerning dynamic models, and in the modal analysis of the test gearbox in Paper B. Transmission of vibrations is also investigated in Paper D, which deals with the influence of bearing endplay or preload on gearbox noise. Differences in bearing preload influence a bearing’s dynamic properties like stiffness and damping. These properties also affect the vibration of the gearbox housing.

## 1.3 Objective

The objective of this dissertation is to contribute to knowledge about gearbox noise. The following specific areas will be the focus of this study:

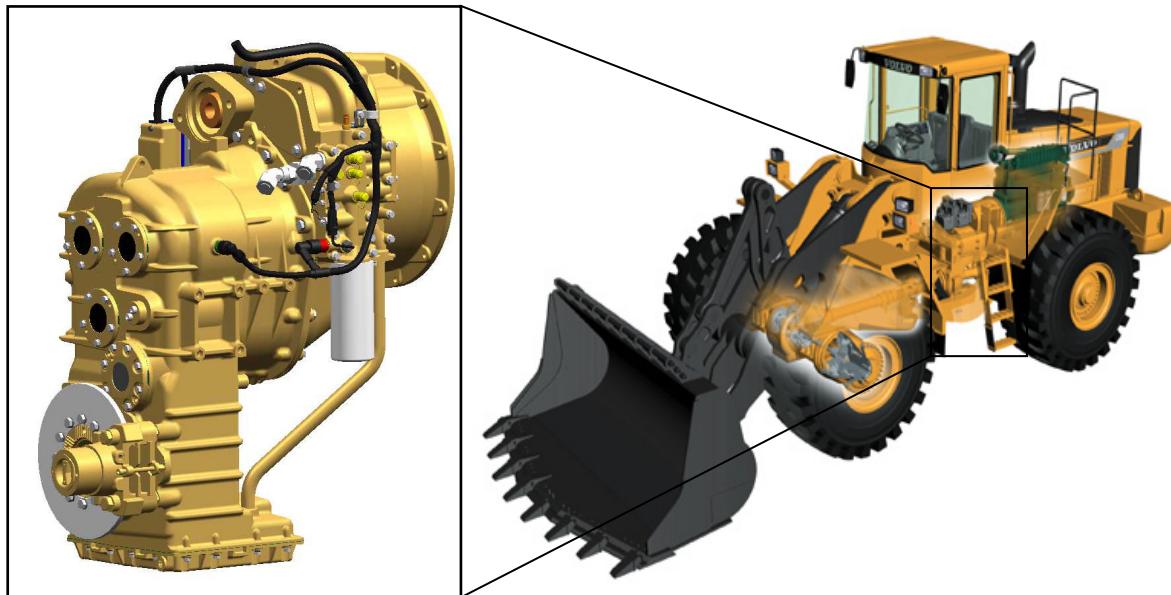
1. The influence of gear finishing method and gear modifications and errors on noise and vibration from a gearbox.
2. The correlation between gear deviations, predicted transmission error, measured transmission error, and gearbox noise.
3. The influence of bearing preload on gearbox noise.
4. Optimization of gear geometry for low transmission error, taking into consideration robustness with respect to torque and manufacturing tolerances.

## 2 AN INDUSTRIAL APPLICATION – TRANSMISSION NOISE REDUCTION

### 2.1 Introduction

This section briefly describes the activities involved in reducing gear noise from a wheel loader transmission. The aim is to show how the optimization of the gear geometry described in Paper E is used in an industrial application. The author was project manager for the “noise work team” and performed the gear optimization.

One of the requirements when developing a new automatic power transmission for a wheel loader was improving the transmission gear noise. The existing power transmission was known to be noisy. When driving at high speed in fourth gear, a high frequency gear-whine could be heard. Thus there were now demands for improved sound quality. The transmission is a typical wheel loader power transmission, consisting of a torque converter, a gearbox with four forward speeds and four reverse speeds, and a dropbox partly integrated with the gearbox. The dropbox is a chain of four gears transferring the power to the output shaft. The gears are engaged by wet multi-disc clutches actuated by the transmission hydraulic and control system. The oil for this hydraulic system is supplied by an internal oil pump driven by the input shaft. The transmission and its position in the wheel loader are shown in Figure 2.1.1.

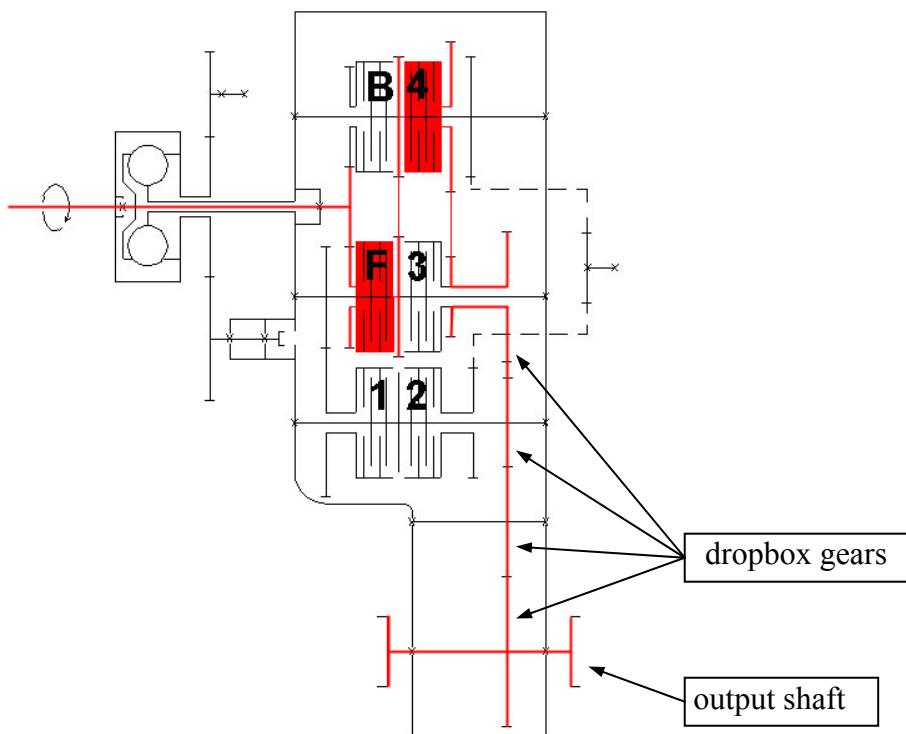


**Figure 2.1.1** Transmission and wheel loader

## 2.2 Gear noise target for the new transmission

Experience has shown that the high frequency gear noise should be at least 15 dB below other noise sources such as the engine in order not to be perceived as disturbing or unpleasant. Measurements showed that if the gear noise could be decreased by 10 dB, this criterion should be satisfied with some margin. Frequency analysis of the noise measured in the driver's cab showed that the dominant noise from the transmission originated from the dropbox gears. The goal for transmission noise was thus formulated as follows: "The gear noise (sound pressure level) from the dropdown gears in the transmission should be decreased by 10 dB compared to the existing transmission in order not to be perceived as unpleasant."

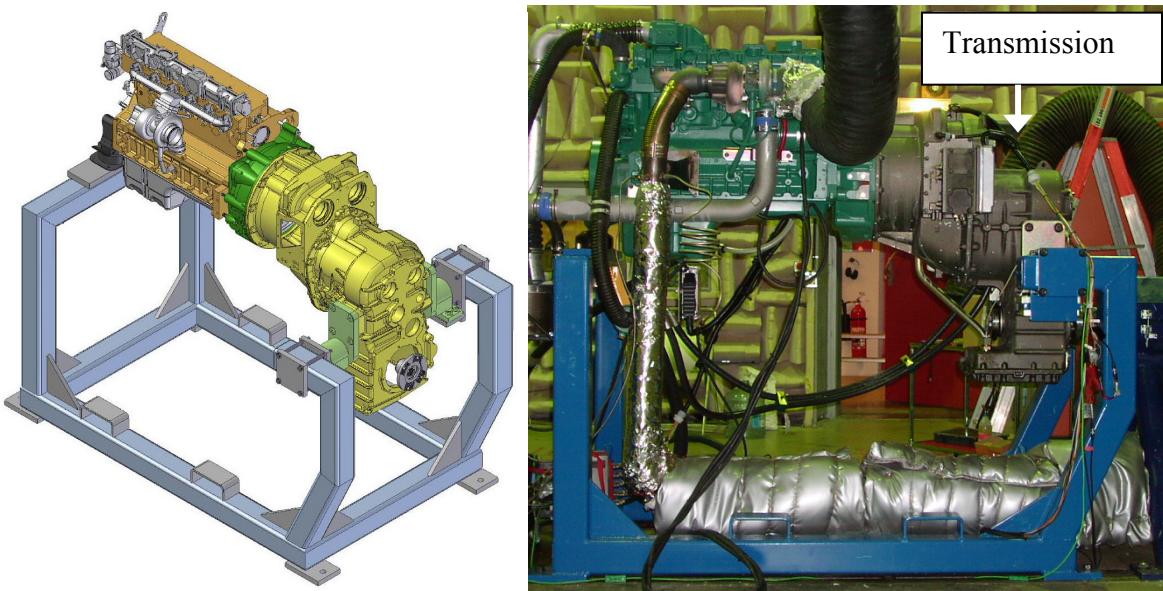
The position of the dropdown gears is shown in Figure 2.2.1. It was assumed that it would be necessary to make changes to both the gears and the transmission housing in order to decrease the gear noise sound pressure level by 10 dB.



**Figure 2.2.1** Explanatory sketch of a wheel loader transmission showing the position of the noisy dropdown gears

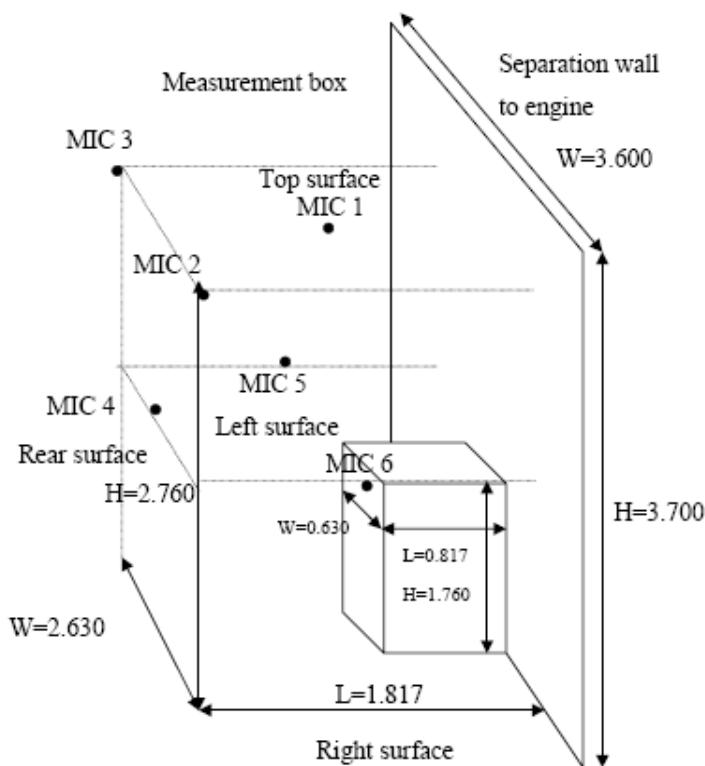
## 2.3 Noise and vibration measurements

In order to establish a reference for the new transmission, noise and vibration were measured for the existing transmission. The test set up is shown in Figure 2.3.1. The transmission is driven by the same type of diesel engine used in a wheel loader. The engine and transmission are attached to the stand using the same rubber mounts that are used in a wheel loader in order to make the installation as similar as possible to the installation in a wheel loader. The output shaft is braked using an electrical brake.



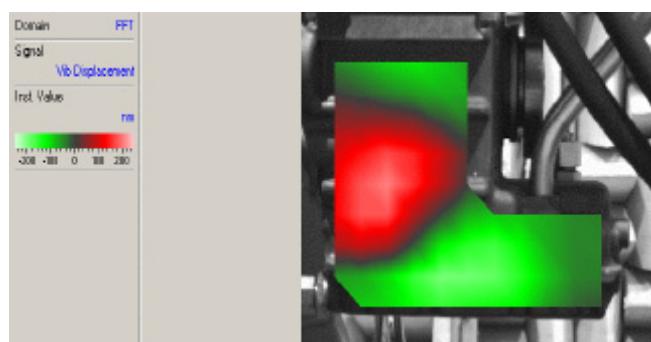
**Figure 2.3.1** Test set-up for noise and vibration measurement. CAD model to the left and photo from noise testing to the right (Krister Fredriksson, Volvo Technology)

Test method EN ISO 3744:1995 was used to measure the total sound power radiating from the transmission, with an imaginary rectangular box enclosing the test object. The engine was shielded using a separation wall because only transmission noise, and not engine noise, was to be measured. This is the special case in EN ISO 3744 for a free field over one reflecting plane with one reflecting wall. The microphones were mounted at a distance  $d = 1\text{ m}$  from, and perpendicular to, the box surfaces. Figure 2.3.2 shows the microphone positions and the size of the reference box.



**Figure 2.3.2** Microphone positions and size of reference box for sound power measurement  
(Krister Fredriksson, Volvo Technology)

The vibration of the transmission housing was measured using a Polytec scanning laser vibrometer. The goal was to find out which surfaces of the gearbox vibrated with the highest surface velocity. The results of the measurements were used to calibrate the FE model used to optimize the housing, as will be described in section 2.5. Figure 2.3.3 shows a result for measurement of the vibration displacement at the side of the transmission dropdown housing. The noise and vibration measurements were performed by Krister Fredriksson at Volvo Technology and by Kaj Bodlund at the Noise and Vibration Laboratory at Volvo Trucks.



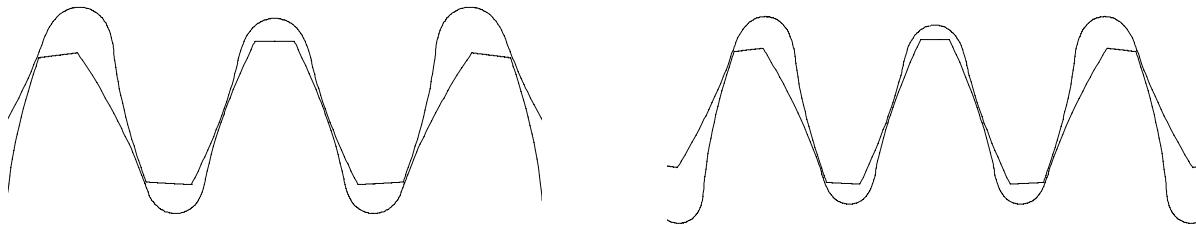
**Figure 2.3.3** Typical result of the scanning laser vibrometer measurement  
(Krister Fredriksson, Volvo Technology)

## 2.4 Optimization of gears

Noise-optimized dropbox gears were designed by choosing macro- and microgeometries giving lower transmission error than the original (reference) gears. The gear geometry was chosen to yield a low transmission error for the relevant torque range, while also taking into consideration variations in the microgeometry due to manufacturing tolerances. The optimization of one gear pair is described in more detail in Paper E.

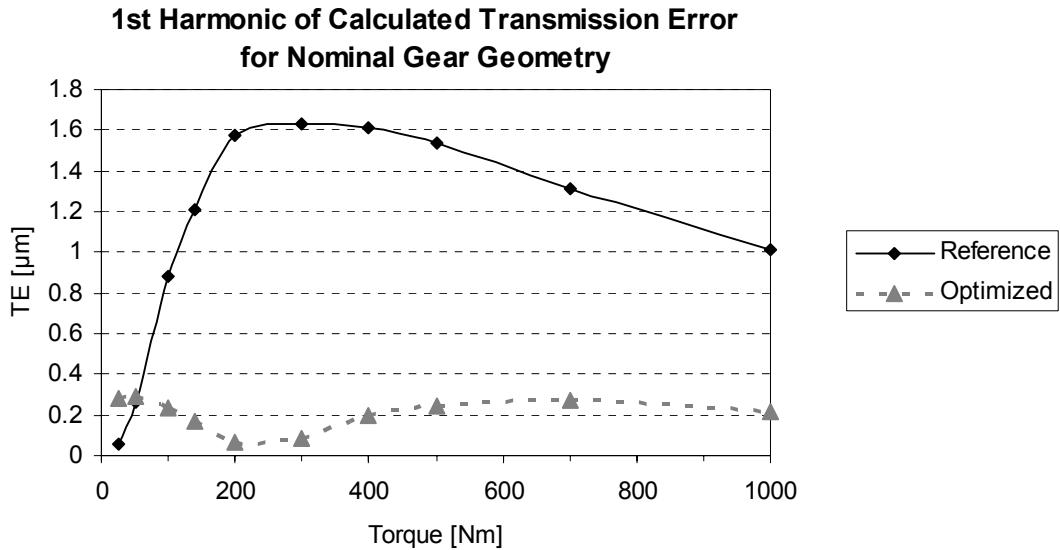
Transmission error is considered an important excitation mechanism for gear whine. Welbourn [1] defines it as “the difference between the actual position of the output gear and the position it would occupy if the gear drive were perfectly conjugate.” In this project the aim was to reduce the maximum predicted transmission error amplitude at gear mesh frequency (first harmonic of gear mesh frequency) to less than 50% of the value for the reference gear pair. The first harmonic of transmission error is the amplitude of the part of the total transmission error that varies with a frequency equal to the gear mesh frequency. A torque range of 100 to 500 Nm was chosen because this is the torque interval in which the gear pair generates noise in its design application. According to Welbourn [1], a 50% reduction in transmission error can be expected to reduce gearbox noise by 6 dB (sound pressure level, SPL). Transmission error was calculated using the LDP software (Load Distribution Program) developed at the Gear Laboratory at Ohio State University [3].

The “optimization” was not strictly mathematical. The design was optimized by calculating the transmission error for different geometries, and then choosing a geometry that seemed to be a good compromise, considering not only the transmission error, but also factors such as strength, losses, weight, cost, axial forces on bearings, and manufacturing. Figure 2.4.1 shows the macrogeometry for the reference and optimized gears. The face width for the optimized gears was increased by 30% in order to achieve a higher load-carrying capacity and also to some extent to compensate for the decreased load-carrying capacity due to the decreased module.



**Figure 2.4.1** Gear macrogeometry for the reference (left) and optimized gears (right)

Figure 2.4.2 shows the predicted amplitude of the first harmonic of the gear mesh frequency transmission error for the nominal gear geometry at different torque levels. Transmission error is given as displacement at the pitch diameter. Nominal gear geometry means that no errors are present and all tolerances are centered in the tolerance zone.



**Figure 2.4.2** First harmonic of predicted transmission error for the reference and optimized gears with nominal gear geometry

When choosing microgeometric modifications and tolerances, it is important to take manufacturing options and cost into consideration. The goal was to use the same finishing method for the optimized gears as for the reference gears, namely grinding using a KAPP VAS 531 and CBN-coated grinding wheels.

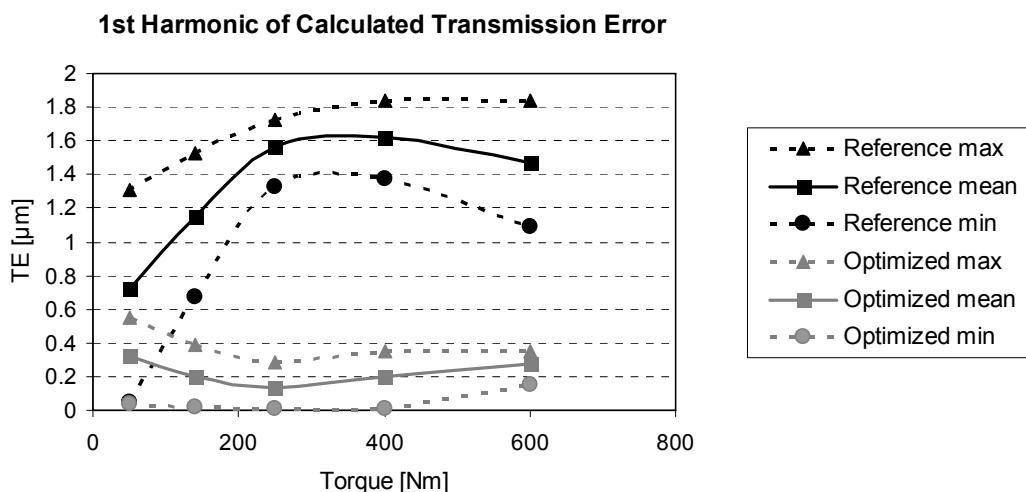
For a specific torque and gear macrogeometry, it is possible to define a gear microgeometry that minimizes transmission error. For example, at no load, if there are no pitch errors and no other geometrical deviations, the shape of the gear teeth should be true involute, without modifications like tip relief or involute crowning. For a specific torque, the geometry of the gear should be designed in such a way that it compensates for the differences in deflection related to stiffness variations in the gear mesh. However, even if it is possible to define the optimal gear microgeometry, it may not be possible to manufacture it, given the limitations of gear machining. Consideration must also be given to how to specify the gear geometry in drawings and how to measure the gear in an inspection machine. In many applications there is also a torque range over which the transmission error should be minimized. Given that manufacturing tolerances are inevitable, and that a demand for smaller tolerances leads to higher manufacturing costs, it is important that gears be robust. In other words, the important characteristics, in this case transmission error, must not vary much when the torque is varied or when the microgeometry of the gear teeth varies due to manufacturing tolerances.

LDP [3] was used to calculate the transmission error for the reference and optimized gear pair at different torque levels. The robustness function in LDP was used to analyze the sensitivity to deviations due to manufacturing tolerances. The “min, max, level” method involves assigning three levels to each parameter. Thus the parameters in Table 2.4.1 were assigned deviations from the nominal values, according to Table 2.4.1. Because there are five parameters at three levels each, there are  $3^5 = 243$  different combinations. For each torque level, transmission error was calculated for all 243 combinations.

Figure 2.4.3 shows the calculated first harmonic of the transmission error. The maximum, mean and minimum values are plotted for the original and optimized gear pair, after substituting in the variations from Table 2.4.1. Figure 2.4.3 shows that the mean and maximum transmission error for the optimized gears decreased to less than 50% of the value for the reference gears in the torque range of 50 to 600 Nm.

	min	mean	max
<b>Lead crowning dev. [μm]</b>	-2	0	+2
<b>Involute alignment dev. [μm]</b>	-10	0	10
<b>Involute crowning dev. [μm]</b>	-2	0	2
<b>Lead alignment dev. [μm]</b>	-10	0	10
<b>Bias dev. [μm]</b>	-5	0	5

**Table 2.4.1** Gear microgeometry deviations used for robustness analysis



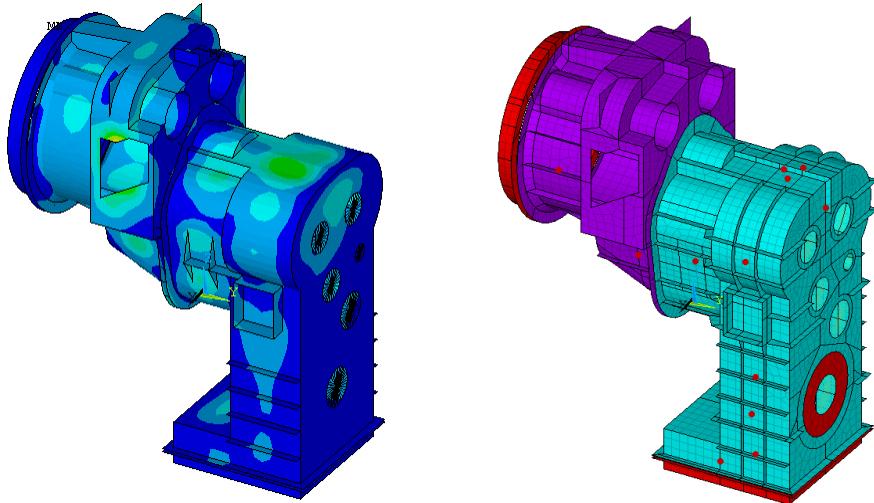
**Figure 2.4.3** Predicted first harmonic of transmission error for reference and optimized gear pair when taking into consideration the deviations given in Table 2.4.1

## 2.5 Optimization of transmission housing

Finite element analysis was used to optimize the transmission housing. The optimization was not performed in a strictly mathematical way, but was done by calculating the vibration of the housing for different geometries and then choosing a geometry that seemed to be a good compromise. Vibration was not the sole consideration, also weight, cost, available space, and casting were considered. A simplified shell element model was used for the optimization to decrease computational time. This model was checked against a more detailed solid element model of the housing to ensure that the simplification had not changed the dynamic properties too much. Experimental modal analysis was also used to find the natural frequencies of the real transmission housing and to ensure that the model did not deviate too much from the real housing.

Gears shafts and bearings were modeled as point masses and beams. The model was excited at the bearing positions by applying forces in the frequency range from 1000 to 3000 Hz. The force amplitude was chosen as 10% of the static load from the gears. This choice could be justified because only relative differences are of interest, not absolute values. The finite element analysis was performed by Torbjörn Johansen at Volvo Technology. The author's contribution was the evaluation of the results of different housing geometries.

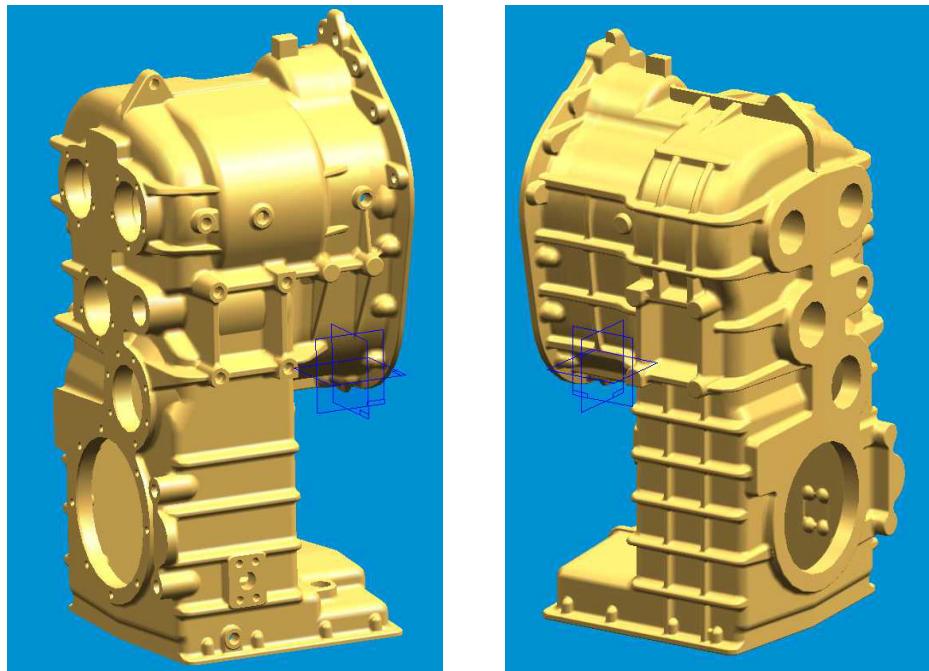
A number of measuring points were chosen in areas with high vibration velocities (Figure 2.5.1). At each measuring point the vibration response due to the excitation was evaluated as a power spectral density (PSD) graph. The goal of the housing redesign was to decrease the vibrations at all measuring points in the frequency range 1000 to 3000 Hz.



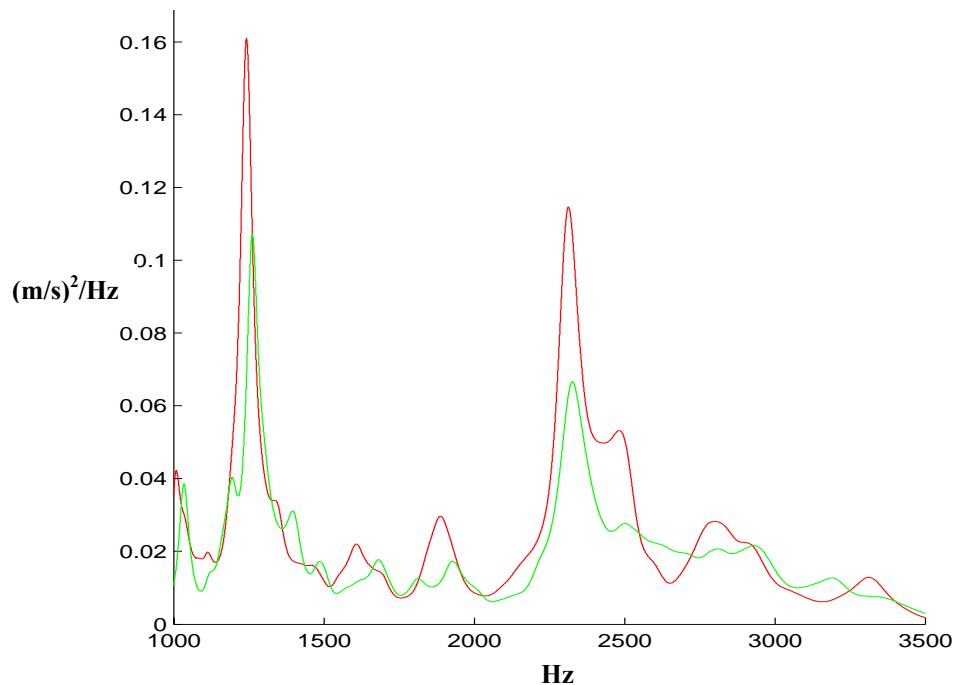
**Figure 2.5.1** Simplified model of the transmission housing. Vibration velocities (left) and measuring points (red dots) for evaluation of modifications (right), (Torbjörn Johansen, Volvo Technology)

The optimization of the housing showed that increased stiffness decreases the vibrations, and that ribs increase the stiffness more than increased wall thickness does. It was also found that stiffening the housing at one location in order to decrease vibration sometimes increased the vibration at another location.

The casting and machining options, available space, weight, and cost were also considered when making design changes to the housing. The housing before and after optimization is shown in Figure 2.5.2. Figure 2.5.3 shows the PSD plots before and after the housing optimization. The plot is the sum of all the measuring points.



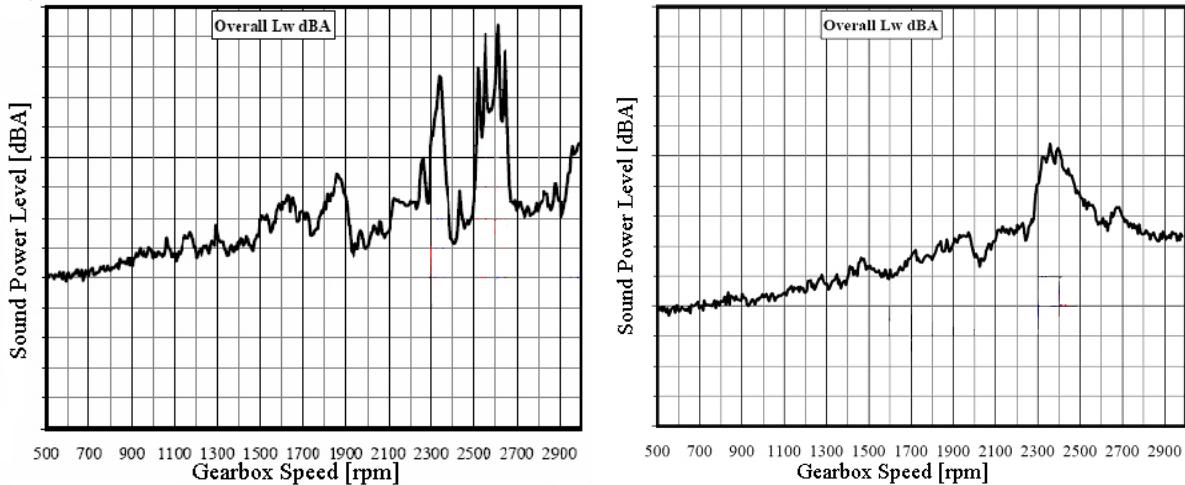
**Figure 2.5.2** Model of the transmission housing before (left) and after (right) optimization



**Figure 2.5.3** Power spectral density before (red) and after (green) optimization (Torbjörn Johansen, Volvo Technology)

## 2.6 Results of the noise measurements

The noise and vibration measurements described in section 2.3 were performed after optimizing the gears and transmission housing. Figure 2.6.1 shows the result of the noise measurements. The total sound power level decreased by 4 dB.



**Figure 2.6.1** Results of the sound power measurements before (left) and after (right) optimization of gears and transmission housing (Krister Fredriksson, Volvo Technology)

Noise measurements in the driver's cab of a number of wheel loaders verified that the gear noise level after the optimization is more than 15 dB below the total sound pressure level, which was the goal of the noise reduction.

## 2.7 Discussion and conclusions

It seems to be possible to decrease the gear noise from a transmission by decreasing the static loaded transmission error and/or optimizing the housing. In the present study, it is impossible to say how much of the decrease is due to the gear optimization and how much to the housing optimization. Answering this question would have required at least one more noise measurement, but time and cost issues precluded this. It would also have been interesting to perform the noise measurements on a number of transmissions, both before and after optimizing the gears and housing, in order to determine the scatter of the noise of the transmissions. Even though the goal of decreasing the gear noise by 10 dB was not reached, the goal of reducing the gear noise in the wheel loader cab to 15 dB below the overall noise was achieved. Thus the noise optimization was successful.

## **3 SUMMARY OF APPENDED PAPERS**

### **3.1 Paper A: Gear Noise and Vibration – A Literature Survey**

This paper presents an overview of the literature on gear noise and vibration. It is divided into three sections dealing with transmission error, dynamic models, and noise and vibration measurement.

Transmission error is an important excitation mechanism for gear noise and vibration. It is defined as “the difference between the actual position of the output gear and the position it would occupy if the gear drive were perfectly conjugate” [1]. The literature survey revealed that while most authors agree that transmission error is an important excitation mechanism for gear noise and vibration, it is not the only one. Other possible time-varying noise excitation mechanisms include friction and bending moment. Noise produced by these mechanisms may be of the same order of magnitude as that produced by transmission error, at least in the case of gears with low transmission error [4].

The second section of the paper deals with dynamic modeling of gearboxes. Dynamic models are often used to predict gear-induced vibrations and investigate the effect of changes to the gears, shafts, bearings, and housing. The literature survey revealed that dynamic models of a system consisting of gears, shafts, bearings, and gearbox casing can be useful in understanding and predicting the dynamic behavior of a gearbox. For relatively simple gear systems, lumped parameter dynamic models with springs, masses, and viscous damping can be used. For more complex models that include such elements as the gearbox housing, finite element modeling is often used.

The third section of the paper deals with noise and vibration measurement and signal analysis, which are used when experimentally investigating gear noise. The survey shows that these are useful tools in experimental investigation of gear noise because gears create noise at specific frequencies related to the number of teeth and the rotational speed of the gear.

### **3.2 Paper B: Gear Test Rig for Noise and Vibration Testing of Cylindrical Gears**

Paper B describes a test rig for noise testing of gears. The rig is of the recirculating power type and consists of two identical gearboxes, connected to each other with two universal joint shafts. Torque is applied by tilting one of the gearboxes around one of its axles. This tilting is made possible by bearings between the gearbox and the supporting brackets. A hydraulic cylinder creates the tilting force.

Finite element analysis was used to predict the natural frequencies and mode shapes for individual components and for the complete gearbox. Experimental modal analysis was carried out on the gearbox housing, and the results showed that the FE predictions agree with the measured frequencies (error less than 10%).

The FE model of the complete gearbox was also used in a harmonic response analysis. A sinusoidal force was applied in the gear mesh and the corresponding vibration amplitude at a point on the gearbox housing was predicted.

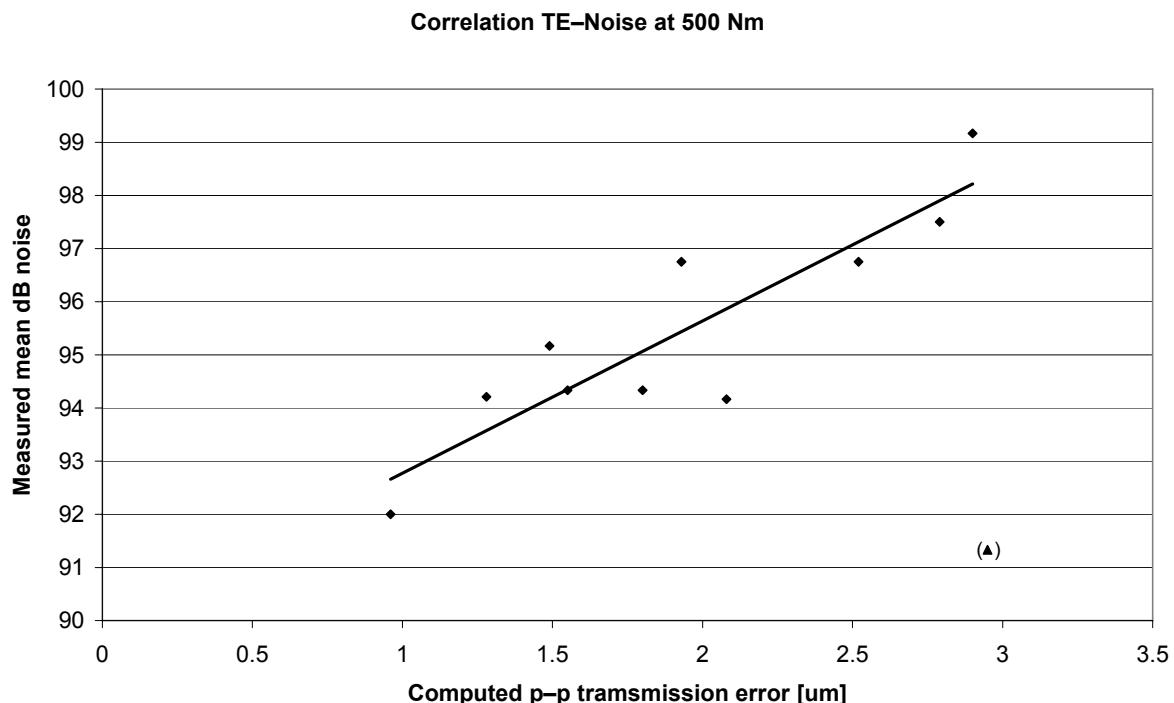
### 3.3 Paper C: A Study of Gear Noise and Vibration

Paper C reports on an experimental investigation of the influence of gear finishing methods and gear deviations on gearbox noise and vibration. Test gears were manufactured using three different finishing methods and with different gear tooth modifications and deviations. Table 3.3.1 gives an overview of the test gear pairs. The surface finishes and geometries of the gear tooth flanks were measured. Transmission error was measured using a single flank gear tester. LDP software from Ohio State University was used for transmission error computations. The test rig described in Paper B was used to measure gearbox noise and vibration for the different test gear pairs. The measurements showed that disassembly and reassembly of the gearbox with the same gear pair might change the levels of measured noise and vibration. The rebuild variation was sometimes of the same order of magnitude as the differences between different tested gear pairs, indicating that other factors besides the gears affect gear noise. In a study of the influence of gear design on noise, Oswald et al. [5] reported rebuild variations of the same order of magnitude.

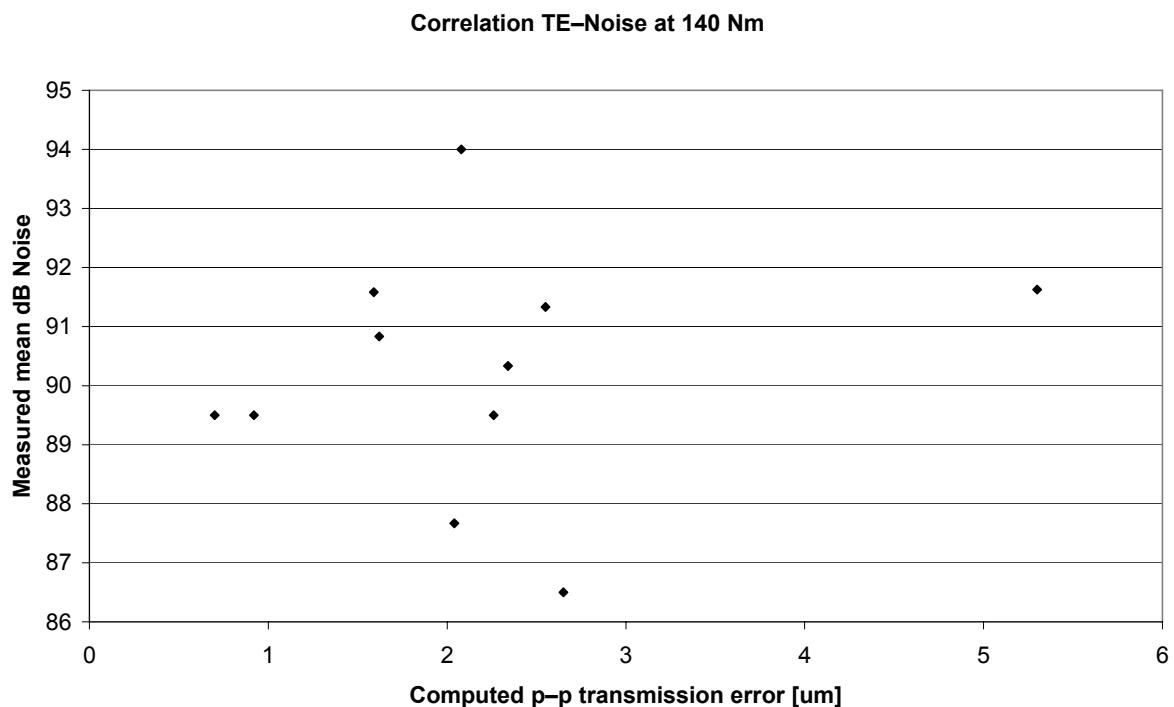
Gear pair	Description	Finishing method
A	Reference gears	Profile grinding (KAPP)
B	Shaved	Shaving
C	Gleason ground	Threaded wheel grinding (Gleason)
D	Rougher surface	Profile grinding (KAPP) "B126"
E	Increased face-width	Profile grinding (KAPP)
F	Pitch errors	Profile grinding (KAPP)
G	Increased lead crowning	Profile grinding (KAPP)
H	Decreased lead crowning	Profile grinding (KAPP)
I	Involute alignment error	Profile grinding (KAPP)
J	Helix angle error	Profile grinding (KAPP)
K	Decreased lead twist (bias)	Profile grinding (KAPP), single flank

**Table 3.3.1** Overview of the different test gear pairs

Different gear finishing methods produce different surface finishes and structures, as well as different geometries and deviations of the gear tooth flanks, all of which influence the transmission error and thus the noise level from a gearbox. Most of the experimental results can be explained in terms of measured and computed transmission error. The relationship between predicted peak-to-peak transmission error and measured noise at a torque level of 500 Nm is shown in Figure 3.3.1. There appears to be a strong correlation between computed transmission error and noise for all cases except gear pair K. However, this correlation breaks down in Figure 3.3.2, which shows the relationship between predicted peak to peak transmission error and measured noise at a torque level of 140 Nm. The final conclusion is that it may not be possible to identify a single parameter, such as peak-to-peak transmission error, that can be directly related to measured noise and vibration.



**Figure 3.3.1** Relationship between measured mean noise (dB) and computed peak-to-peak transmission error for the different test gear pairs at 500 Nm. Line fitted by the method of least squares. Gear pair K excluded



**Figure 3.3.2** Relationship between measured mean noise (dB) and computed peak-to-peak transmission error for the different test gear pairs at 140 Nm

The rebuild variation makes it hazardous to draw conclusions from the noise and vibration measurements. However, there is some support for the following statements:

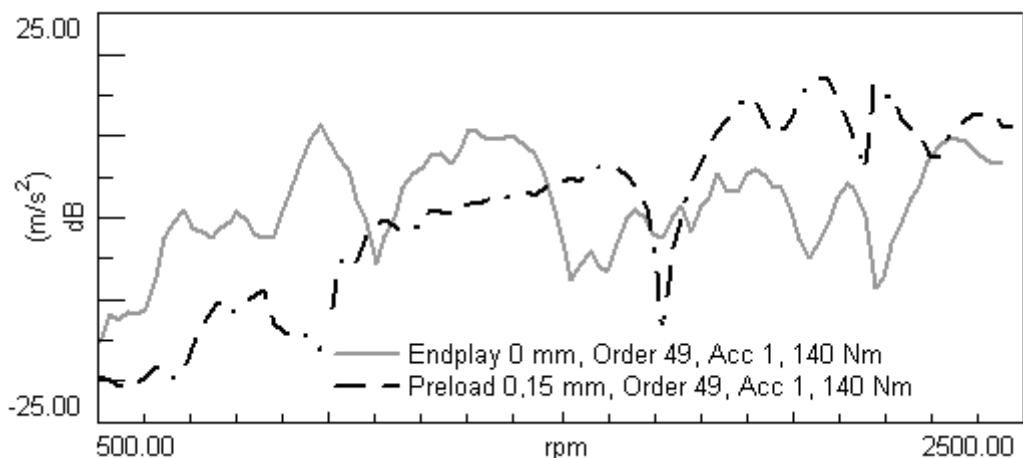
- Shaved gears do not seem to be noisier than ground gears even if their gear deviations are larger.
- Factors that seem to reduce noise, compared to the profile ground reference gears, are threaded wheel grinding, increased face-width, decreased lead crowning, increased pitch errors, and decreased lead twist.
- A rougher surface finish, increased lead crowning, and helix angle error seem to increase noise.

### 3.4 Paper D: Gearbox Noise and Vibration – Influence of Bearing Preload

The influence of bearing endplay or preload on gearbox noise and vibrations is investigated in Paper D. Measurements were carried out on a test gearbox consisting of a helical gear pair, shafts, tapered roller bearings, and a housing. Vibration measurements were carried out at torque levels of 140 Nm and 400 Nm with 0.15 mm and 0 mm bearing endplay and with 0.15 mm bearing preload. The results show that the bearing endplay or preload influence gearbox vibrations. Compared with bearings with endplay, preloaded bearings show an increase in vibrations at speeds over 2000 rpm and a decrease at speeds below 2000 rpm.

Figure 3.4.1 is a typical result showing the influence of bearing preload on gearbox housing vibration. After the first measurement, the gearbox was not disassembled or removed from the test rig. Only the bearing preload/endplay was changed from 0 mm endplay/preload to 0.15 mm preload. Therefore the differences between the two measurements are solely due to different bearing preload. FE simulations performed by Sellgren and Åkerblom [6] show the same trend as the measurements here.

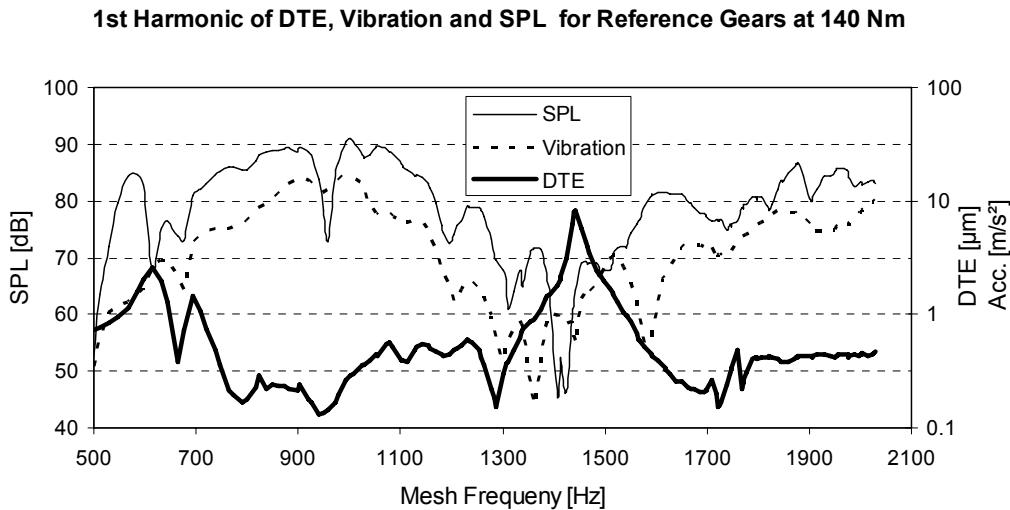
For the test gearbox, it seems that bearing preload, compared with endplay, decreased the vibrations at speeds below 2000 rpm and increased vibrations at speeds over 2000 rpm, at least at a torque level of 140 Nm.



**Figure 3.4.1** Measured gearbox housing vibration for 0 and 0.15 mm bearing preload at a torque of 140 Nm

### 3.5 Paper E: Gear Geometry for Reduced and Robust Transmission Error and Gearbox Noise

In Paper E, gearbox noise is reduced by optimization of gear geometry for decreased transmission error. The optimization was not performed strictly mathematically. It was done by calculating the transmission error for different geometries and then choosing a geometry that seemed to be a good compromise considering not only the transmission error, but also other important characteristics. Robustness with respect to gear deviations and varying torque was considered in order to find gear geometry with low transmission error in the appropriate torque range despite deviations from the nominal geometry due to manufacturing tolerances. Static and dynamic transmission error as well as noise and housing vibrations were measured. The correlation between dynamic transmission error, housing vibrations, and noise was investigated in a speed sweep from 500 to 2500 rpm at constant torque. No correlation was found between dynamic transmission error and noise (Figure 3.5.1).



**Figure 3.5.1.** First harmonic of dynamic transmission error (DTE), gearbox housing vibration and noise (SPL) measured at 140 Nm pinion torque

The static loaded transmission error seems to be a good measure of the ability of a gear pair to excite dynamic systems. A gearbox can be considered a complex dynamic system with a high number of degrees of freedom (dof). Some dof's are excited by transmission error. A torsional dof might lead to high values of dynamic transmission error when the gear mesh frequency corresponds to the natural frequency of that specific dof. This does not necessarily lead to high levels of housing vibration and noise because the dof of the housing that is most likely to radiate noise might not be excited at a sufficiently high amplitude. On the other hand, high levels of housing vibrations and noise might occur when the gear mesh frequency corresponds to natural frequencies of the housing. These high vibration and noise levels do not necessarily correspond to high levels of dynamic transmission error.

## 4 DISCUSSION AND CONCLUSIONS

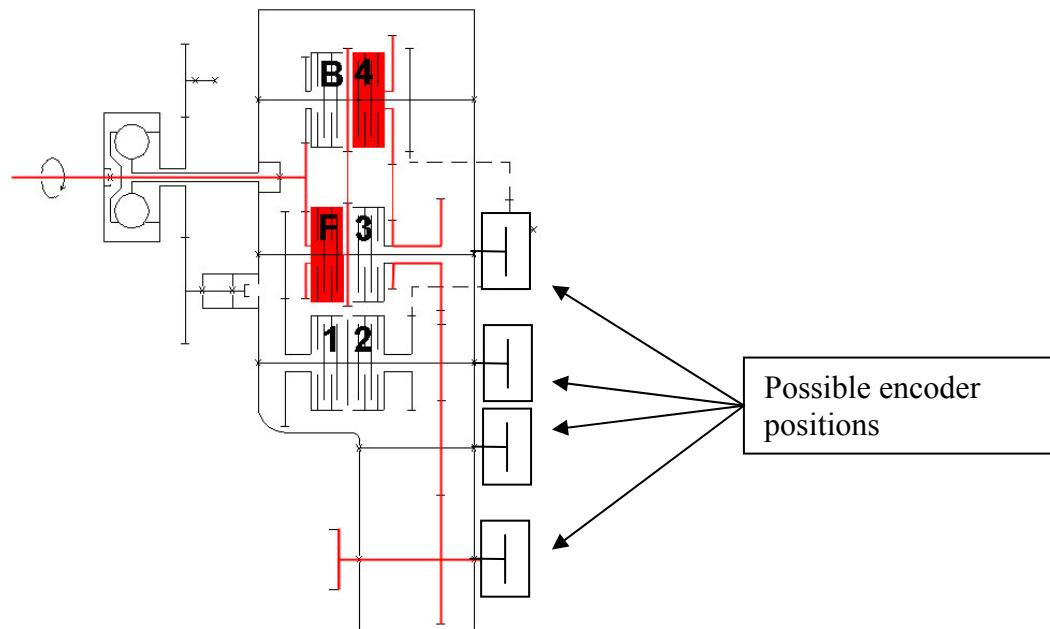
Static loaded transmission error seems to be strongly correlated to gearbox noise. Dynamic transmission error does not seem to be correlated to gearbox noise in speed sweeps in these investigations. Henriksson [7] found a correlation between dynamic transmission error and gearbox noise when testing a truck gearbox at constant speed and different torque levels. The different test conditions, speed sweep versus constant speed, and the different complexity (a simple test gearbox versus a complete truck gearbox) may explain the different results regarding correlation between dynamic transmission error and gearbox noise.

Bearing preload influences gearbox noise, but it is not possible to make any general statement as to whether preload is better than endplay. The answer depends on the frequency and other components in the complex dynamic system of gears, shafts, bearings, and housing.

To minimize noise, the gearbox housing should be as rigid as possible. This was proposed by Rook [8], and his views are supported by the results relating to the optimization of a transmission housing described in section 2.5. Finite element analysis is a useful tool for optimizing gearbox housings.

## 5 FUTURE RESEARCH

It would be interesting to investigate the correlation between dynamic transmission error and gearbox noise for a complete wheel loader transmission. One challenge would be to measure transmission error as close as possible to the gears and to avoid resonances in the connection between gear and encoder. The dropbox gears in a typical wheel loader transmission are probably the gears that are most easily accessible for measurement using optical encoders. See Figure 5.1.1 for possible encoder positions.



**Figure 5.1.1** Explanatory sketch of a wheel loader transmission showing possible positions for optical encoders making it possible to measure static and dynamic transmission error for the dropdown gears

Modeling the transmission in more detail could be another challenge for future work. One approach could be to use a model of gears, shafts, and bearings using the transmission error as the excitation. This could be a finite element model or a multibody system model. The output from this model would be the forces at the bearing positions. The forces could be used to excite a finite element model of the housing. The housing model could be used to predict noise radiation, and/or vibration at the attachment points for the gearbox. This approach would give absolute values, not just relative levels.

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