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A MODEL-BASED DESIGN STUDY OF GEARBOX-INDUCED NOISE

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1. Introduction

Volvo Construction Equipment (Volvo CE) is one of the world's leading manufacturers of construction machines, with a product range encompassing wheel loaders, excavators, articulated haulers, motor graders and more. Construction equipment, such as the wheel loader and the articulated hauler shown in figure 1, is heavily loaded during normal operation, which gives strength and fatigue predications a high priority. In recent years there have also been increased legal and customer demands for lower noise levels for construction machinery.

Figure 1. A Volvo wheel loader and an articulated hauler

The gearbox is sometimes the dominant source of noise in wheel loaders and articulated haulers. Noise-generating mechanisms in gearboxes include gear rattle from unloaded gears, gear whine from gears under load, vibrations in bearings, and, in the case of automatic gearboxes, noise generated by internal oil pumps and clutches. Even if gear whine is not the loudest source, its pure tone, which consists of frequencies that correspond to the gear mesh frequency and multiples thereof, is easily distinguished from other noise sources and it is often perceived as highly unpleasant. Furthermore, this type of gear noise creates an impression of poor quality. A general design requirement is to keep the gear whine noise at least 15 dB lower than noise from other sources, such as the engine. Consequently any reduction in the noise from these other sources necessitates a simultaneous reduction in the gear whine noise. Gear whine originates from vibrations excited mainly by transmission error (TE). TE is defined as the nonconjugacy of a gear pair [Welbourn 1979], that is, the motion error defined by the difference between the output gear's actual position and its position if the gear teeth were perfect in shape and infinitely stiff. In equation form, TE is defined as:

$$TE = R_{bp}\left[\theta_p - \left(\frac{N_g}{N_p}\right)\theta_g\right] \tag{1}$$

where θ_p and θ_g are the rotations of the pinion and gear shafts, and N_p and N_g are the number of teeth on the two wheels, respectively. R_{bp} is the base radius of the pinion wheel. The major causes of TE are manufacturing and assembly imperfections and elastic deflections of gear teeth, shafts, bearings, and the housing. Figure 2 shows a recorded TE and its low-frequency run-out and high frequency tooth-to-tooth components

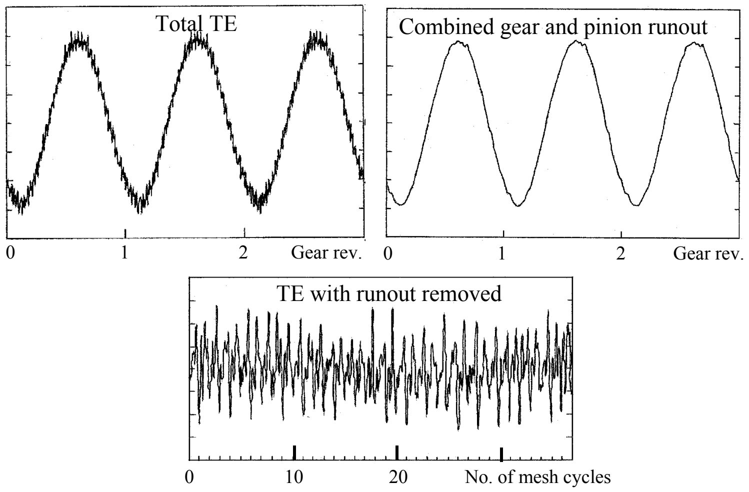


Figure 2. Recorded TE and its runout and tooth-to-tooth components [Houser 2004]

The TE-induced vibrations are transmitted to the housing via the gears, shafts, and bearings. The housing then vibrates and radiates pressure variations in the surrounding air that are perceived as exterior noise. The level and frequency of the gear noise can be influenced by changing any one of the highly complex excitation, transmission, or radiation mechanisms. This paper reports on ongoing model-based research at Volvo CE that is focused on investigating the influence of design parameters, gear-finishing methods, and assembly operations on the level and frequency content of the gearbox-induced vibrations and exterior gear whine. It deals with the approach chosen to explore the excitation mechanisms and some of the results obtained.

2. Method

The excitation problem was approached using a combination of rig tests of physical models (e.g., simplified gearboxes) and numerical modeling (e.g., finite element modeling) and simulation. The test rig, which is also designed to perform gear life testing and efficiency measurements, consists of a simplified test gearbox and a master gearbox connected with two universal joint shafts and driven by an electric motor (see figure 3). The transmission error for each tested gear pair is predicted with LDP software from Ohio State University [Ohio 2000]. It can also be measured with encoders on the free shaft ends in the test gearbox. The two gearboxes are identical, each comprising a single gear pair that represents the average gears in a wheel loader transmission. Technical data for the test gears are listed in table 1.

Table 1. Technical data for the test gears [Åkerblom 2002]

	Pinion	**Gear**
Number of teeth	49	55
Normal module (mm)	3.5	3.5
Pressure angle (degrees)	20	20
Helix angle (degrees)	-20	20
Center distance (mm)	191.9	191.9
Face width (mm)	35	33
Profile shift coefficient	+0.038	-0.529
Tip diameter (mm)	191	209

The noise is recorded with microphones, and the vibration on the test gearbox housing is measured with accelerometers. To reduce the risk of the gears in the master gearbox interfering with the test gears, the master gearbox is equipped with precision ground gears.

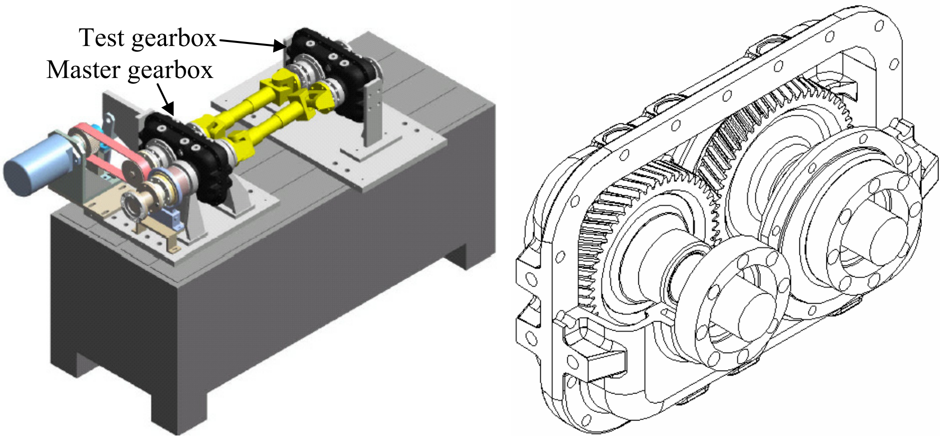


Figure 3. The test rig (left) and gearbox (right) design models

Other researchers (e.g., [Harris et al. 2000]) have shown that a housing that is not sufficiently stiff can have a significant influence on the gear mesh alignment and thus on the TE. Therefore, the test gearbox housing was designed to be similar in character to typical wheel-loader transmission housings. The same is true for the shafts and the bearings in the test gearbox. The master design model of the test gearbox is shown in figure 3 (right). The physical test gearbox and a finite element (FE) model of the gearbox are shown in figure 4.

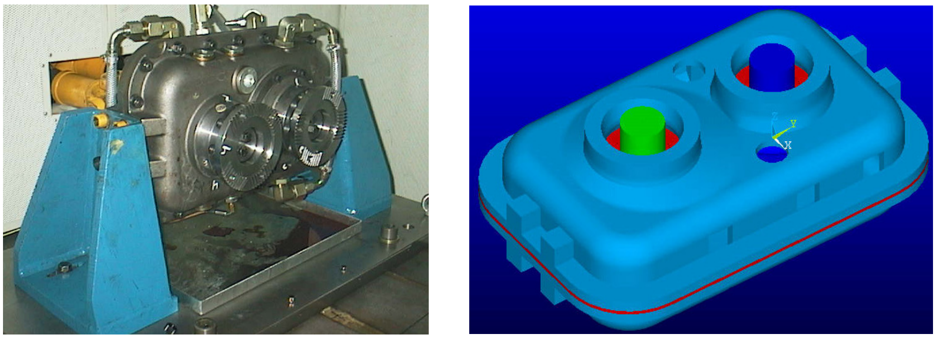
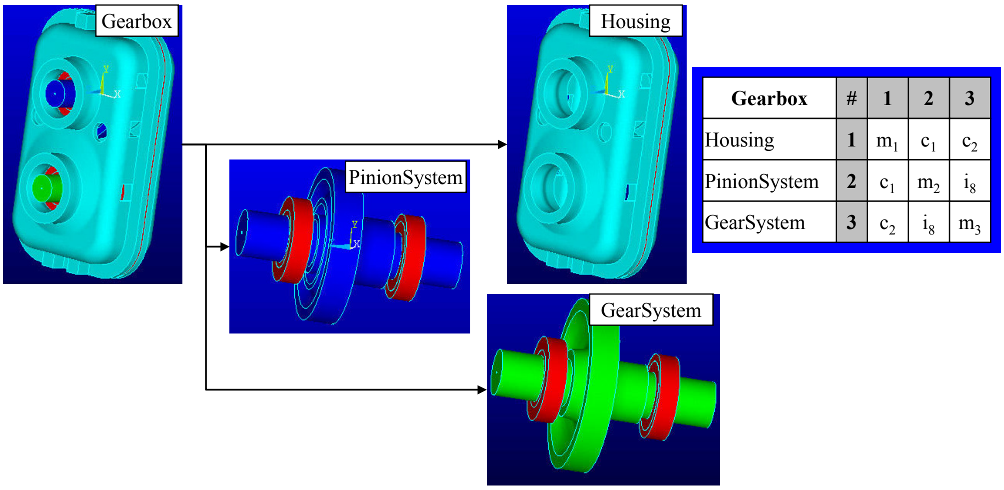


Figure 4. The physical test gearbox (left) and a finite element representation (right)



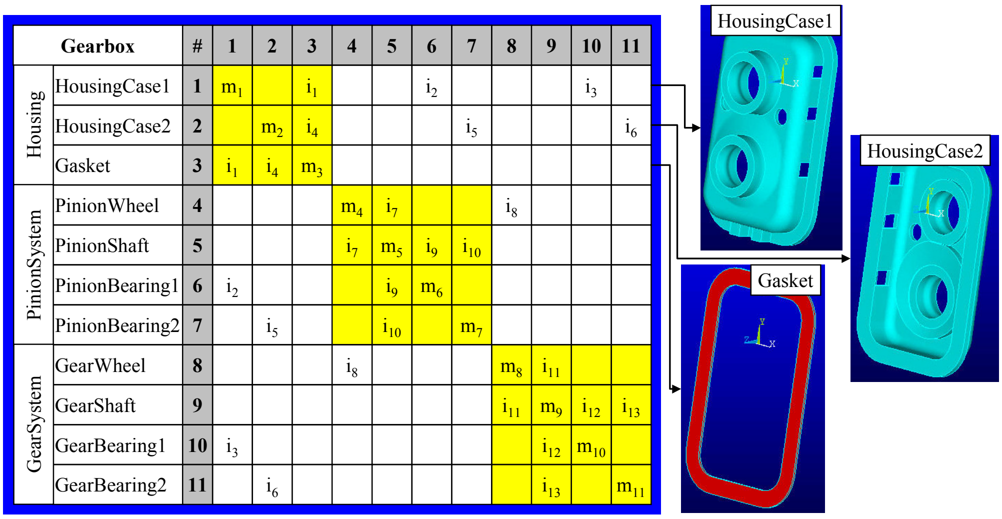
Gearbox	**#**	**1**	**2**	**3**
Housing	**1**	m_1	c_1	c_2
PinionSystem	**2**	c_1	m_2	i_g
GearSystem	**3**	c_2	i_g	m_3

Figure 5. The first decomposition level of a gearbox FE-model

To enable efficient and flexible studies of the important noise excitation and transmission mechanisms and how they are related to each other, a modular model architecture has been developed. The building blocks of a model configuration are classified as component models, interaction models or interface features, and environment models [Sellgren 2003]. The FE model variant of a gearbox shown in figure

5 consists of three submodels: a *Housing*, a *PinionSystem*, and a *GearSystem*. These submodels interact at two composite interfaces, c_1 and c_2, and one elementary interface, i_8, that connects the cylindrical representations of the gear and pinion wheels. Interface i_8 is represented by a single bar element with an axial stiffness equal to the sum of the bending stiffness of the gear pair and the Hertzian contact stiffness. In the simulations, the effect of TE is modeled as a force pair applied at the two ends of the bar and acting in the contact direction.

A second-level decomposition of the FE model of the gearbox is shown in figure 6. Each of the four bearing submodels can be further decomposed into two submodels and an interface feature.



Gearbox		#	1	2	3	4	5	6	7	8	9	10	11
Housing	HousingCase1	1	m_1		i_1			i_2				i_3	
	HousingCase2	2		m_2	i_4				i_5				i_6
	Gasket	3	i_1	i_4	m_3								
PinionSystem	PinionWheel	4				m_4	i_7			i_8			
	PinionShaft	5				i_7	m_5	i_9	i_{10}				
	PinionBearing1	6	i_2				i_9	m_6					
	PinionBearing2	7		i_5			i_{10}		m_7				
GearSystem	GearWheel	8				i_8				m_8	i_{11}		
	GearShaft	9								i_{11}	m_9	i_{12}	i_{13}
	GearBearing1	10	i_3								i_{12}	m_{10}	
	GearBearing2	11		i_6							i_{13}		m_{11}

Figure 6. The second decomposition level of the FE-model of the gearbox

3. Results

The test rig was used to study the effect of different gear-finishing methods on the TE and the gearbox noise level at different torque levels [Åkerblom 2002]. Most, but not all, of the tested gear pairs showed a strong correlation between TE and noise. The measurements also revealed that the disassembly and re-assembly of the gearbox with the same gear pair could significantly change the housing vibration and noise levels. The rebuild variation was sometimes in the same order of magnitude as that obtained when testing gear pairs with different topography and surface structure. Oswald et al. [1998] reported rebuild variations of the same order. Further efforts to control gear whine require a proper explanation of the observed rebuild variations.

The cause of the significant scatter in the recorded noise levels was studied by experimenting with variants of the gearbox FE model shown in figures 5 and 6. Figure 7 presents the computed accelerations of the housing surface midpoint for four different free-free model variants. The curve labeled *rigid* in figure 7 shows the acceleration response to a harmonically varying excitation force in the pinion and gear interface feature. The force amplitude corresponds to a TE amplitude of 1 µm. An overall 1% relative damping was chosen for the model. The results for the *soft gasket* model variant show the limit case response of a gearbox with *HousingCase1* and *HousingCase2* loosely interconnected. The *all soft bearings* curve shows the influence from four bearings with significantly reduced stiffness in all three spatial directions. The response curve in figure 7 that is labeled *2 axially soft bearings* emulates the effect of a frictionless shaft contact for two of the four bearings.

On the basis of these and other simulations, it was hypothesized that the "observed rebuild variations are caused by differences in the axial bearing pre-loading between different re-assembly operations." New rig tests are currently being carried out to validate the findings from these simulations, and the preliminary results seem to validate the hypothesis. If the hypothesis holds, the rebuild variations can be reduced significantly by using a more controlled re-assembly procedure.

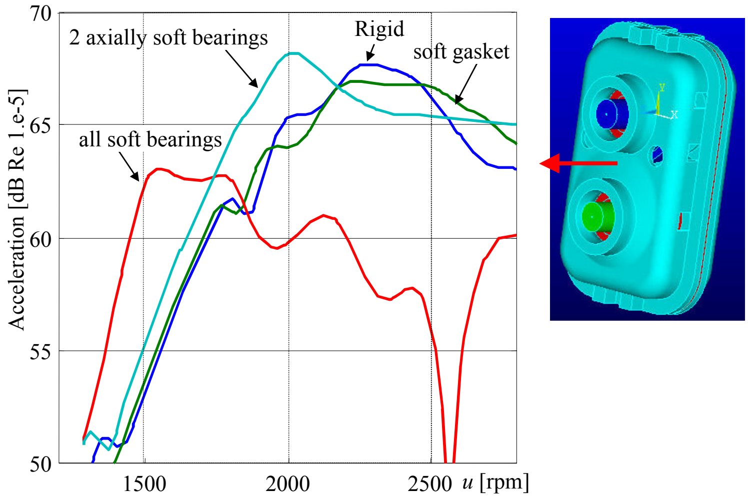


Figure 7. Housing vibrations in dB (relative to 1x10^{-5} m/s^2) for four gearbox model variants

A number of simulations were also performed with three different dual gearbox models. Two instances of the FE gearbox model were configured and connected with three different representations of the two universal joint shafts that connect them. Figure 8 shows the three model variants. From left to right they represent a *rigid rotational constraint only*, a *rigid rotational and an axial constraint*, and a *flexible body constraint* between the shafts of the two gearboxes. Each gearbox is connected to two rigid brackets. The general purpose of these models was to study the interaction between the two gearboxes in the test rig. The models also served the specific purpose of verifying that TE in the master gearbox does not significantly influence the vibration behavior of the test gearbox. Figure 9 (left) shows the accelerations of the housing surface on a single master gearbox and on the housing surfaces of two dual gearbox variants. In all simulations, a harmonically varying TE with an amplitude of 1µm excites the master gearbox. Bearing in mind that the master gearbox is equipped with high-precision ground gears, the results of the simulation leave us confident that the actual TE in the master gearbox has a negligible influence on the level of the high frequency gear whine from the test gearbox.

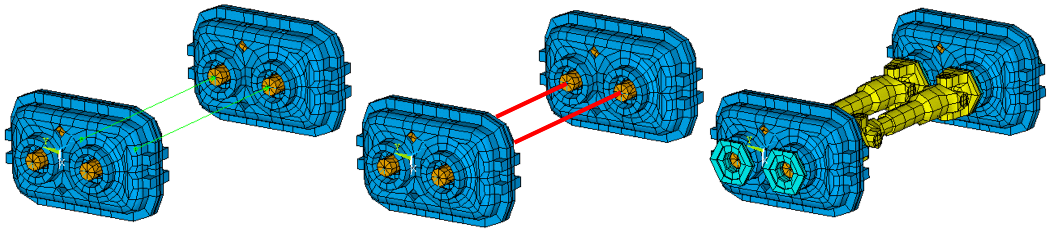


Figure 8. Three model variants of two gearboxes with different representations of the connection

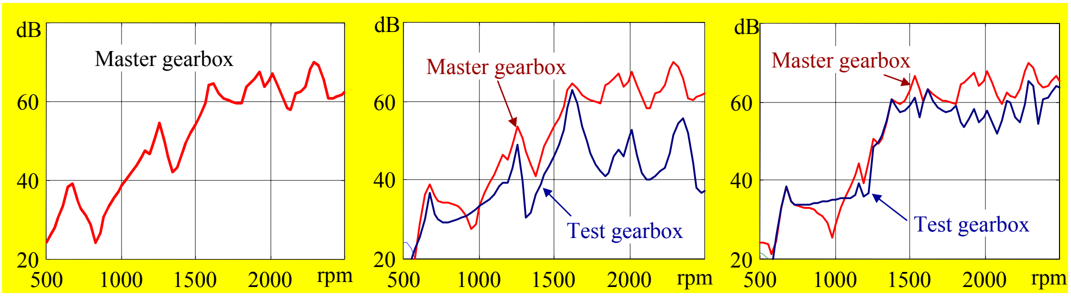


Figure 9. Accelerations (rel. 1x10^{-5} m/s^2) on a single master gearbox (left) and two dual gearbox variants - rigid rotational constraints only (center) and rigid rotational and axial constraints (right)

4. Conclusions

The test results demonstrate that noise generation is a complex mechanism and that there is no single excitation parameter, such as peak-to-peak transmission error, that can be directly related to the measured noise and vibration. The origin of gear whine is gear mesh misalignment, which has design, manufacture, assembly, and operation-related causes. The level of the perceived noise depends on the characteristic dynamic properties of all gearbox components and the interfaces between them.

Achieving the goal of controlling gear-generated exterior noise with design and manufacturing measures involves complex problem solving. Before any solutions can be proposed, the problem itself must be known and defined. There is thus a need for thorough investigation of the effect of variations in characteristic designs, manufacturing, and assembly parameters on the excited, transmitted, and radiated noise. The most efficient approach to exploring the problem involves a balanced combination of reduced physical testing and numerical simulation. While thoroughly planned tests yield valuable knowledge, such tests are tedious and often cannot provide a satisfactory explanation of the underlying causes of the observed behavior. The cognitive activity of experimentation with a numerical model, however, may give valuable insight into basic physical phenomena. This is especially true as regards the phenomena that occur in the interfaces between the interacting components of the artifact, which are difficult to observe and measure physically.

Many of the simulations required for the studies presented here were of an exploratory, and thus non-routine, character. There were thus significant benefits to having a flexible and modular model architecture that enabled models to be modified and reconfigured to address new and changing questions. The presented study has significantly increased our knowledge of how the exterior noise level of construction machinery is related to gearbox-induced vibrations and how these vibrations are related to characteristic gearbox design parameters, gear-finishing methods, assembly sequences, and operations. This knowledge is a stepping-stone to further efforts to control the gearbox noise from construction machines in general and gear whine in particular.

Acknowledgements

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