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Loose part vibration in vehicle transmissions - Gear rattle

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Abstract

Minimising noise is becoming an increasingly important factor in motor vehicle development. The importance of this development goal is increasing with rising customer expectations and increasingly stringent legal restrictions on noise emissions. The cause of rattling and clattering noise is torsional vibration of transmission components that are not under load, that move backwards and forwards within their functional clearances. This noise is perceived as distinct from other sources of noise, and is intrusive because of its undesirable character. The transmission parameters backlash, axial clearance and main centre distance were varied by experimental analyses in test stand trials, showing the effect on propensity to rattle and clatter. By optimising these parameters, it was possible to minimise the rattling and clattering noise. Measures internal to the transmission to reduce loose part vibration in vehicle transmissions were also considered. The calculated noise level with the EKM-Simulation program correlates with the measured noise level. Parameter studies with the EKM-Simulation program, which contains all relevant parameters for the excitation of rattling noise caused by idle gears, shows the significant parameters for the investigated transmissions.

Key Words: Transmission noise minimising, Gear rattle, Torsional vibration, Backlash, Axial clearance

Şanjıman gevşek parça titreşmesi-Dişli çark tıkırdaması

Özet

Otomotiv sanayisinde, giderek güçlenen gürültü azaltma eğiliminde şanjıman dişli tıkırdamasının önem kazandığı görülüyor. Diğer ses kaynaklarından daha etkin ve karakteristik biçimde olması, şanjıman tıkırdamasının azaltma eğilimini daha da güçlendiriyor. Bu dişli tıkırdamasının temel sebebleri; yüklü olmayan şanjıman kısımlarının serbestlik derecesi içerisinde titreşimidir. Deneysel analizlerde çark parametrelerden diş boşluğunun, mihver boyunun ve ana mihver açıklığının sistematik şekilde değiştirilmesi ile tıkırdama sesinin üzerindeki tesiri araştırılmaktadır. Bu parametrelerin optimum hale getirilerek tıkırdama sesinin azaltılması gösterilmektedir. Ayrıca, şanjıman içerisindeki gevşek parçaların titreşimini önleyen tedbirlerin ortaya çıkartılması ve deneysel analizlerde tesiri araştırılmaktadır. Şanjıman tıkırdamasının simulasyon hesaplama yardımıyla (EKM-Programı) pratikteki deneyimlerin karşılaştırılması gösteriliyor. Hedef, şanjıman sesine karşı uygun yöntemler gerçekleştirebilmek ve ses izolasyonunu uygulamaktır.

Anahtar Sözcükler: Şanjımanda gürültü azaltması, Dişli tıkırdaması, Burulma titremesi, Diş boşluğu, Mihver boyu

Introduction

Noise reduction in power trains is a constant development goal in automotive engineering, emphasised by increasing requirements in terms of consumption and emissions, weight reduction, reliability, performance, efficiency and comfort. One consequence of this trend is the reduction of engine running speeds to achieve good fuel consumption and emissions characteristics, generally entailing substantially greater rotational irregularity. This irregularity creates excitation throughout the power train. Together with the engine, bodywork and chassis, the transmission is a factor determining the overall level of noise. The rattling and clattering noise produced by vehicle transmissions is perceived as particularly offensive. Measures for reducing loose part vibration in vehicle transmissions are being drawn up on the basis of experimental analyses using test stand trials. A simple computer simulation model to describe the vibrational phenomenon of loose parts has been developed at the institute of machine components, university of Stuttgart. In order to determine the model parameters and to verify the simulation results, the rattling noise tester has been designed and built at the institute. Both computer and experimental simulation provide good correlation of their results. The examination of the influence of all important parameters concerning the transmission such as backlash, axial clearance, drag torque, and moments of inertia, has led to design recommendations enabling the prevention or reduction of transmission rattling noises.

1. Classification of transmission noise

The importance of minimising noise in motor vehicle development is increasing with rising customer expectations and increasingly stringent legal restriction on noise emission. Vehicle transmission noise arises from the types of causes identified in Figure 1 [1, 2, 3, 4, 5]. Whining and squealing noises arise from meshing impact; parametrically excited vibration and rolling contact noise arise from gearwheels under load. This can be reduced by refinement of the gearing parameter design. Gear-shifting errors give rise to engagement noise caused by scraping and grating of the selector teeth, which can in principle be reduced by engineering of the synchroniser. Screeching is caused by vibration of idler gears that are not under load when the vehicle is moving off, and is affected by the gearing geometry and lubricant viscosity [3]. Bearing noise takes the form of rolling-bearing running noise, especially where bearings are damaged. Rattling and clattering noises are caused by loose part vibration of components moving backwards and forwards within their clearances when not under load, such as control gears, synchronisers and sliding sleeves. Vehicle transmissions rattle in neutral and clatter under power and in overrun.

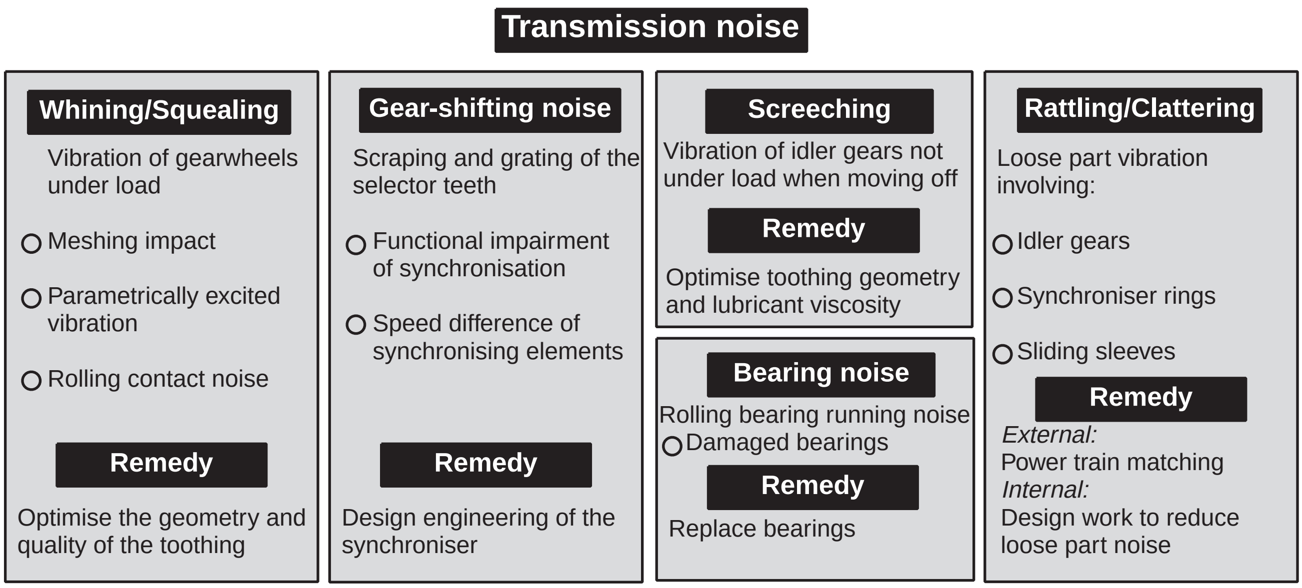
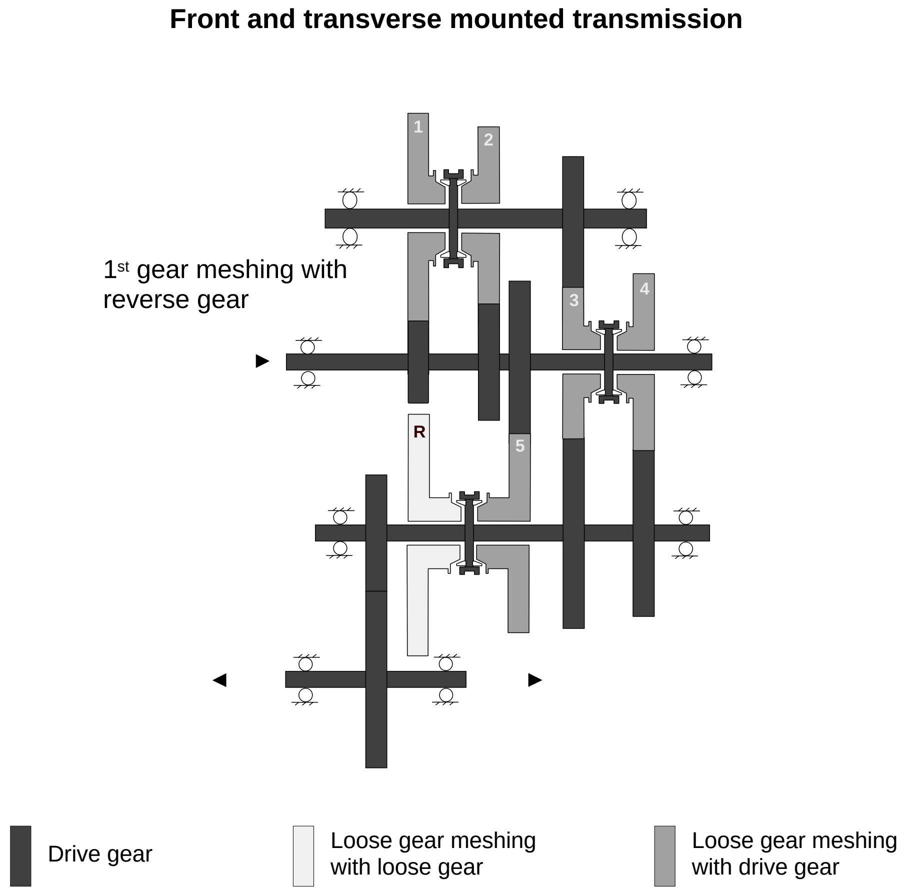


Figure 1. Classification of transmission noise



Gear / Status	1st gear	2nd gear	3rd gear	4th gear	5th gear	Reverse gear	Lay shaft
Neutral	☆	☆			☆	☆	
1st speed		☆		☆	☆	☆	
2nd speed	☆			☆	☆	☆	
3rd speed	☆	☆		☆	☆	☆	☆
4th speed	☆	☆	☆		☆	☆	☆
5th speed	☆	☆	☆	☆		☆	☆
R. speed		☆	☆	☆	☆		☆

Figure 2. Rattling Meshes in a 5-speed manual front and transverse mounted transmission

2. Problem definition

The power train is excited by the driving torque of the internal combustion engine pulsating with the ignition frequency, and by the unbalanced engine mass, giving rise to torsional vibration. This torsional vibration is induced in the transmission through the transmission input shaft, and causes rattling and clattering noise from transmission components that are not under load, and vibrate backwards and forwards within their functional clearances. The noise is usually intrusive not because of its high airborne sound pressure level, but because of its distinctive character that distinguishes it from the other sources of noise in the vehicle.

Depending on the transmission type (standard, front and transverse mounted) and on the operating condition, a different number of loose parts may cause rattling and clattering noises. Figure 2 shows a 5-speed manual transmission. For the front and transverse mounted transmission type-non co-axial input and output shaft-in neutral condition only the idle 1^{st}, 2^{nd}, 5^{th} and reverse gears are rattling. Driving in 1^{st} and 2^{nd} speeds all idle gears-without the engaged 1^{st}, 2^{nd}, 3^{rd} idle gears and the lay shaft-are excited to cause rattling and clattering noises.

For the 3^{rd}, 4^{th} and 5^{th} speeds all idle gears-without the engaged 3^{rd}, 4^{th} and 5^{th} idle gears-are excited to rattle. In reverse gear only the 1^{st} and reverse idle gear are not excited to rattle.

The shaft speed function of a four-cylinder four-stroke internal combustion engine is shown in Figure 3; its first derivative is the critical factor determining torsional vibration. The amplitude of the vibration is related to the degree of rotational irregularity, which increases when additional loads cut in, such as air conditioning, heating and headlights.

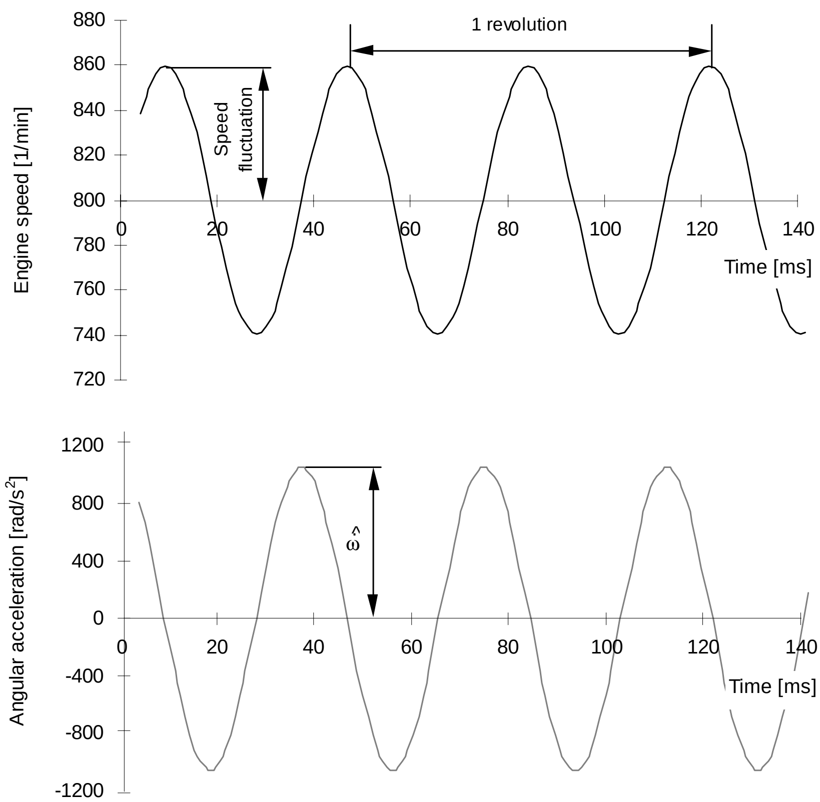


Figure 3. Shaft speed and angular acceleration function of a 4-cylinder 4-stroke internal combustion engine

Numerous publications and patent specifications offer means of suppressing rattling and clattering noise. These measures have so far however been constrained by technical limitations, and are either not economically viable, or viable only in combination with other measures. The possible ways of reducing or avoiding rattling and clattering noise can be subdivided into external and internal measures (Figure 4).

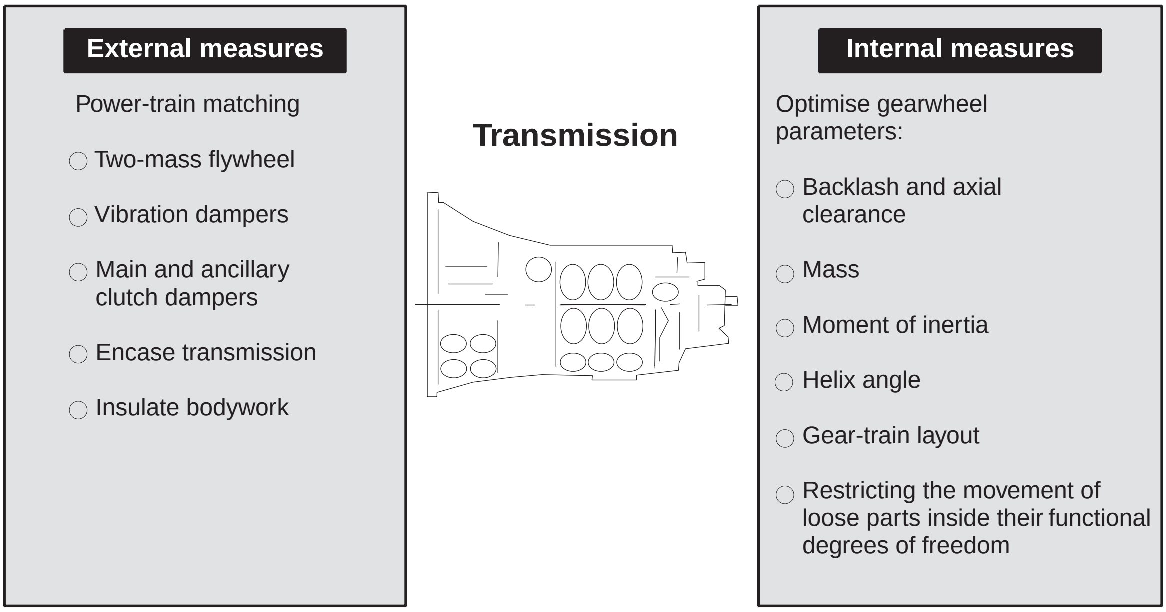


Figure 4. External and internal measures to reduce loose part vibration and noise

External measures include isolating the combustion engine from the transmission by means of clutch dampers or a two-mass flywheel, in order to reduce torsional vibration in the power train. Vibration dampers are suitable for attenuating resonance peaks in the power train; sound transmission and radiation in the passenger compartment can be reduced by encasing the transmission housing. These external measures are, however, subject to physical and economic constraints, and do not always yield the desired results. Solutions therefore have to be sought within the transmission itself.

Internal measures within the transmission can limit the freedom of movement of loose parts within their functional clearances. Effort can be cost-effectively targeted at the main sources of noise, but a different combination of the numerous solutions available has to be individually contrived for each power train unit. There is no prospect of a universal solution for all vehicle transmissions.

3. Ingear rattle test stand

The ingear rattle test stand developed by the IMA at the University of Stuttgart enables realistic operating conditions to be simulated without interference from impinging variables from the vehicle (Figure 5).

With this test stand it is possible to study both standard transmissions and front and transverse mounted transmissions, both during idling and under part load. The test stand consists of a welded structure on which the drive unit is mounted, a highly dynamic, brushless, permanently excited 3-phase synchronous motor. This drive unit enables simulation of various internal combustion engines, with different numbers of cylinders.

A braking motor is also flange-mounted on the driven shaft for investigating rattling noise under power and in overrun. The servomotors are directly linked to the test transmission by means of torsion-proof metal bellows joints to provide a backlash-free and positively engaged connection. The servomotors have a nominal output of 13 kW at a nominal torque of 30 Nm and a natural moment of inertia of the rotor of 0.005 kgm^2. The peak torque arises at 120 Nm, and the maximum input speed with uniform rotational movement is 5000 1/min. On this test stand, car transmissions can be driven in neutral with an angular acceleration amplitude of up to 4000 rads^{-2}, depending on their mass moment of inertia.

With the aid of a PC controlled function generator, both ideal and actual engine speed profiles can be specified to the motor control in the form of discrete plots, or analytically. The incremental sensor on the drive motor serves to record the speed and synchronise the function generator with the angular position of the drive motor. The irregularity thus has a fixed phase relation to the rotational angle of the servomotor. The rattling noise level in the near field of the test transmission is recorded by an integral airborne sound pressure meter. In addition to the speed, the structure-borne noise is synchronously recorded with an acceleration sensor on the surface of the transmission housing to identify the rattling and clattering impacts that are related in time to the rotational irregularities. The test transmission can also be heated up to a maximum of 130°C by oil circulation with an external oil unit, to simulate realistic operating conditions.

4. Interpretation of the rattling noise level curve

Before the results of the trial are described, the rattling noise level curve is interpreted below with the example of a complete transmission. The rattle curve describes the air sound pressure level curve as a function of the angular acceleration amplitude, and is divided into three characteristic sections: the basic noise, the rattling noise limit and the level curve (Figure 6).

The **basic noise** occurs up to the rattling noise limit, and is made up of bearing running noise, churning noise, and gearing noise. No impacts occur in the structure-borne noise curve 1 shown. The **rattling noise limit** 2 identifies the point in the rattling noise level curve at which the angular acceleration amplitude has become so great that the loose parts start to become detached from the driving fixed gears. The rattling noise level starts to rise as angular acceleration amplitudes become greater, and the first rattling impacts are visible in the structure-borne noise signal. The **level curve** (points 3 and 4) shows the noise behaviour at angular acceleration amplitudes above the rattling noise limit. In this case both the torsional and axial impacts are clearly evident.

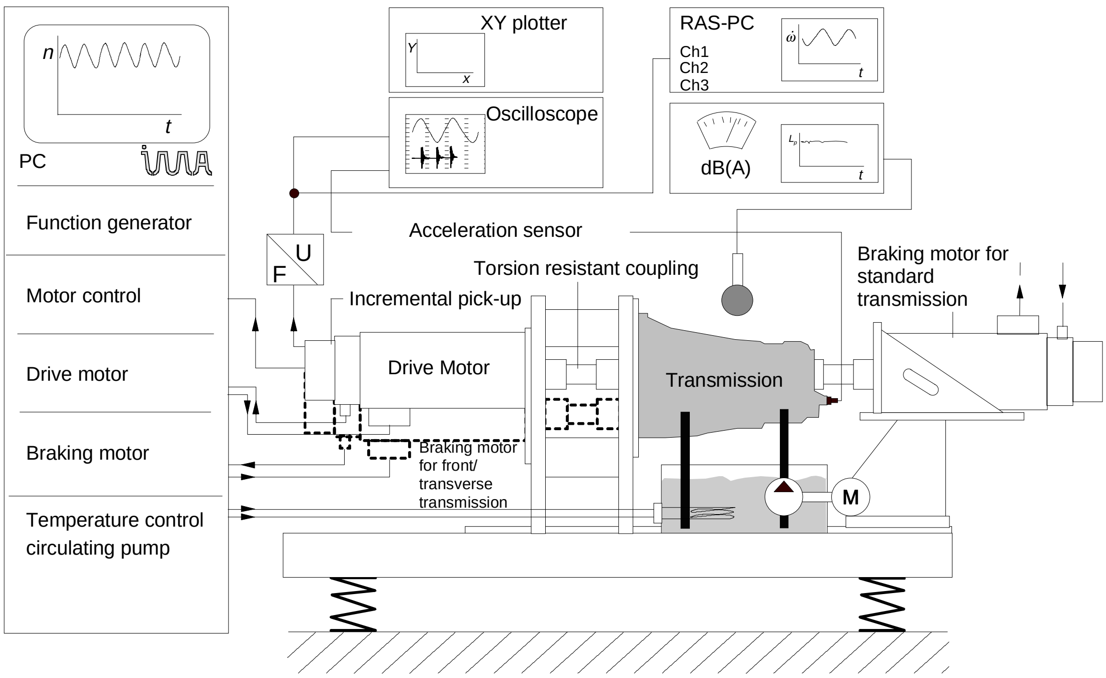


Figure 5. Ingear rattle test stand

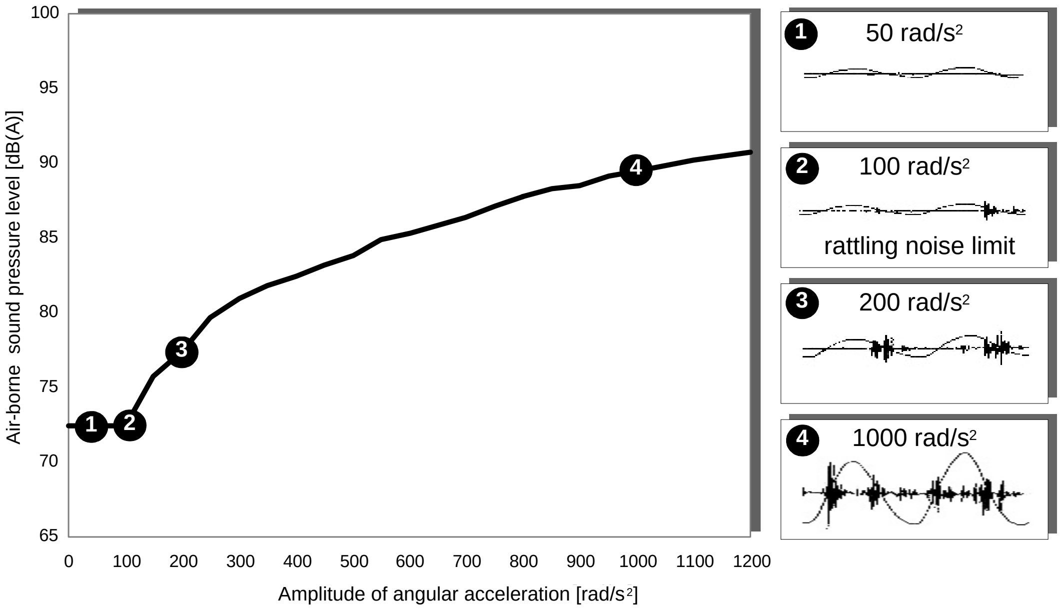


Figure 6. Rattling noise level curve

5. Computer Simulation EKM

EKM ("**E**infachst-**K**lapper-**M**odell") is a simple computer simulation model to describe the vibrational phenomenon of loose parts. The EKM simulation parameters are as follows [2, 3, 4]:

J_2	[kgm^2]	moment of inertia idle gear
k	[-]	correlation factor
T_2	[Nm]	drag torque idle gear
m_2	[kg]	mass idle gear
r_{b1}	[mm]	pitch circle radius fixed gear
r_{b2}	[mm]	pitch circle radius idle gear
s_a	[mm]	axial clearance
s_v	[mm]	backlash
β	[°]	helix angle
μ	[-]	friction coefficient
ω_{an}	[rads^{-1}]	excitation frequency
$\hat{\omega}$	[rads^{-2}]	amplitude of angular acceleration

The simple rattling model has two degrees of freedom. The clashing of the exciting and loose part is modelled with the basic impact law.

Figure 7 shows the modelling of loose gears and synchronizing rings. The rotational irregularities of the drive shaft may be considered as an up and down movement of the exciting part m_1. Loose part m_2 rattles within the backlash if acceleration amplitude of the exciting part exceeds the acceleration of the loose part. The axial clearance is considered as a movement between the left and right functional clearances. The examination of the influence of all important parameters concerning the transmission will lead to design recommendations enabling the prevention or reduction of transmission rattling noises caused by the impacts of unloaded gears and synchronization parts within their functional backlash and axial clearance. The issue is to get the quantitative influence of all relevant design parameters affecting the excitation of rattling noises in automotive transmission. The influence of the noise on the path of the vibration from the gear flank to the ear of the passenger is not being investigated. In all cases, except in neutral and direct gear of the standard type, a rattling idle gear is excited by a single drive gear, and the vibration of an idle gear does not influence the vibration of any other gear. The mass of the exciting part is much higher than the other one, and thus the impacts do not affect the movement of the exciting part. Concerning the drag torque it is assumed that the main speeds of the shafts are constant so that the drag torque also stays constant. Therefore each rattling part can be simulated separately and independently.

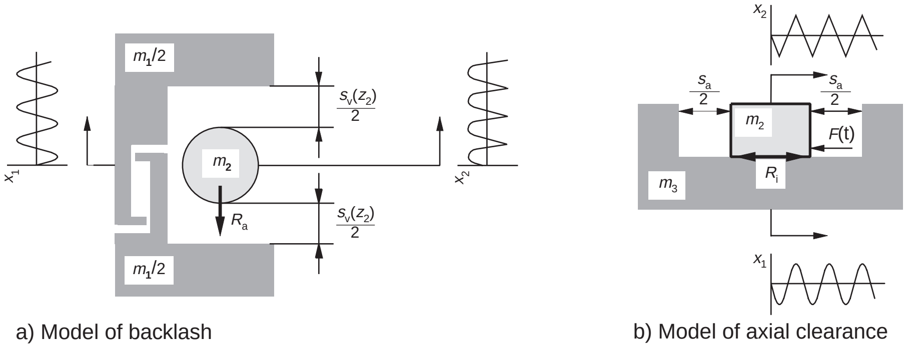


Figure 7. Model for loose gears and synchronizing rings

5.1. Calculation of the Rattling Noise Level

The noise level which is measured in the vicinity of the gearbox cannot be compared directly with the rattling noise level in vehicles because the passengers of a car do not hear only the gear noise, and direct radiation of airborne noise is prevented by the shape of the car. But for the comparison of different gearboxes and vibration conditions within gearboxes, sound level measurement on the rattle test stand is quite useful.

The goal of the simulation is to predict the rattle noise level caused by an idle gear or synchronizing at a determined excitation function. It is assumed that the time-average impact impulses are proportional to the noise level.

The calculation of the average intensity of impacts and the rattle noise level is divided into the following steps [3]:

1/Non-dimensional axial friction force:

$$C_{fa} = \frac{F_R \cdot \tan\beta}{m_2 . r_{b1} \cdot \hat{\dot{\omega}}_1} \tag{1}$$

Non-dimensional circumferential backlash:

$$C_{sv} = \frac{s_v \cdot \omega_{an}^2}{r_{b1} \cdot \hat{\dot{\omega}}_1} \tag{2}$$

Non-dimensional axial clearance:

$$C_{sa} = \frac{s_a \cdot \omega_{an}^2 \cdot \tan\beta}{r_{b1} \cdot \hat{\dot{\omega}}_1} \tag{3}$$

2/Non-dimensional intensity of impacts:

$$C_{lm} = \sqrt{C_{sv}} \cdot \left(1.462 - \frac{0.714 \cdot C_{fa} . C_{sa}}{-0.016 \cdot C_{fa} + 0.12 C_{sv}}\right) \tag{4}$$

3/Average intensity of impacts:

$$I_m = m_2 \cdot \hat{\omega}_1 \cdot r_{b1} \cdot C_{lm} \tag{5}$$

4/Airborne rattle noise level:

$$L_{Klapper} = 10 \cdot log(k \cdot I_m) \tag{6}$$

The following equations are applied, if the boundary is exceeded.

Non-dimensional intensity of impacts:

$$C_{lm} = 0.748 \cdot \sqrt{C_{sv}} \tag{7}$$

Non-dimensional axial clearance:

$$C_{sa} = -0.016 + \frac{0.12 . C_{sv}}{C_{fa}} \tag{8}$$

or

$$C_{sa} = \frac{0.2}{C_{fa}} . (1 - C_{mv}) \tag{9}$$

For the case β=0 (C_{fa}=0) or (C_{sa}=0):

Non-dimensional intensity of impacts:

$$C_{lm} = 1.462 \cdot \sqrt{C_{sv}} \tag{10}$$

Non-dimensional drag torque:

$$C_{mv} = \frac{T_2 \cdot r_{b2}}{J_2 \cdot r_{b1} \cdot \hat{\dot{\omega}}_1} \tag{11}$$

Average intensity of impacts:

$$I_m = 1.462 \cdot m_2 \cdot \sqrt{r_{b1}} \cdot \sqrt{s_v} \cdot \sqrt{\hat{\dot{w}}_n . w_{an}} \tag{12}$$

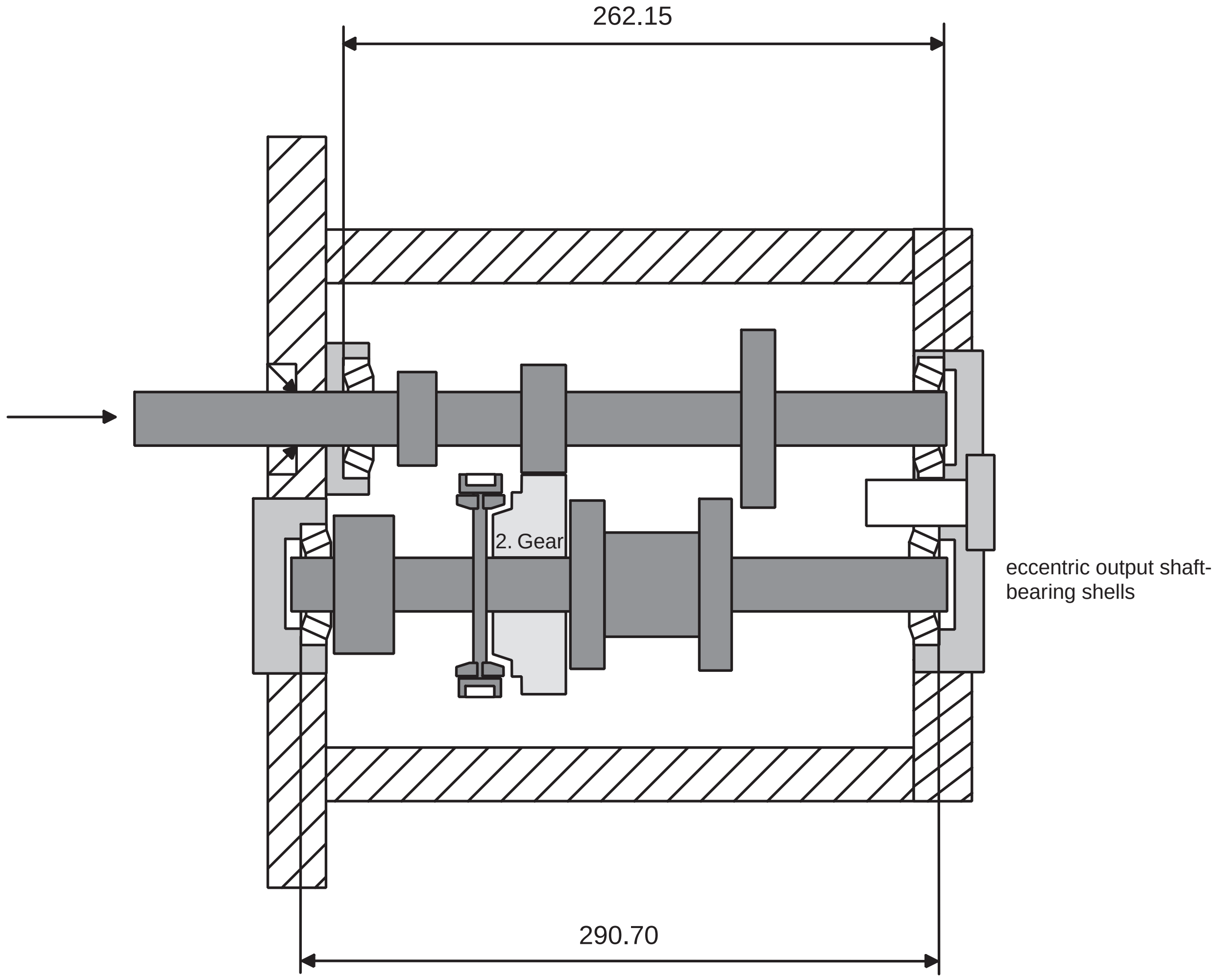


Figure 8. Elemental test transmission

5.2. Range of validity of the rattling noise level

The approximative solution is valid for the following referred parameters of the EKM and real idle gear parameters:

The appropriate parameters, such as synchronizer rings, sliding sleeves or synchronizer bodies, can be used instead of the idle gear parameters to calculate the rattling noise level.

- Non-dimensional drag torque C_{mv} <0.5
- Non-dimensional axial friction force C_{fa} <0.7
- Axial clearance $s_a \geq 0.2$mm
- Amplitude of angular acceleration $\hat{\dot{\omega}}_1 > 300 rad/s^2$
- Excitation sinus 2^{nd} order

6. Test stand trials

With the ingear rattle test stand developed by the IMA at the University of Stuttgart, systematic studies on production, prototype and elemental test transmissions were carried out to reduce rattling and clattering noise, with the practically relevant rotational irregularities of a 4-cylinder 4-stroke internal combustion engine.

The following sections review suitable internal measures for minimising vehicle transmission noise, on the basis of test stand trials.

6.1. Elemental test transmission trials

To vary the transmission parameters relevant to rattling noise, an elemental test transmission was developed at the IMA with eccentric output shaft-bearing shells on which the backlash and axial clearance of loose parts can be infinitely adjusted [3]. The rig enables infinite adjustment of backlash by changing the main shaft distance between the input and output shafts, and of axial clearance by altering the position of the synchroniser body on the output shaft. This elemental test transmission enables the noise behaviour of individual gearwheel stages to be studied, as well as several gear trains.

6.1.1. Influence of backlash

The elemental test transmission was equipped with the second gear stage in order to illustrate the influence of the backlash on this gear stage.

All other loose parts were removed for the series of trials. Figure 9 shows the standard backlash s_v of the second gear stage for an axial clearance s_a of 0.2 mm reduced or enlarged by 0.06 mm, and the results for the angular acceleration amplitude of 600 rad/s^2. Reducing the standard backlash by 0.06 mm has the effect of minimising the air-borne sound pressure for this gear stage by up to 2 dB(A), whereas increasing it by the same amount results in an increase in the value of 4 dB(A).

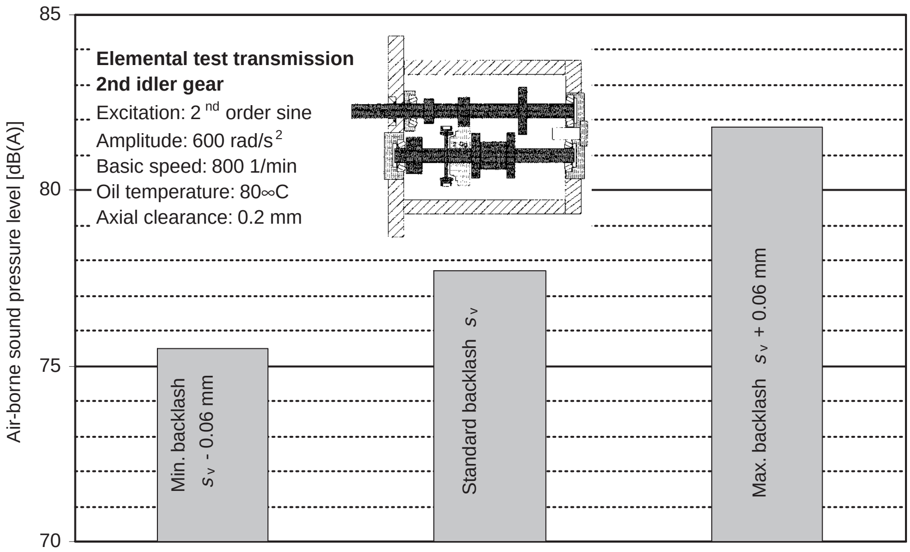


Figure 9. Effect of backlash on the rattling noise level at an angular acceleration amplitude of 600 rad/s^2, based on results for the second gear idler gear

6.1.2. Influence of axial clearance

In order to illustrate the influence of the axial clearance the elemental test transmission was also equipped with the second gear stage.

In Figure 10, the backlash of the second gear idler gear is kept constant at 0.06 mm, and the axial clearance is infinitely varied by the changing position of the synchroniser body in 0.1 mm increments.

Increasing the idler gear axial clearance up to a specific limit of $s_a = 0.4$ mm, at which the idler gear can no longer impact against its axial thrust collars, has the effect of reducing the airborne sound pressure level by up to 3 dB(A) in comparison to the standard axial clearance of 0.2 mm at an angular acceleration amplitude of 600 rad/s^2. Reducing the axial clearance also has the effect of reducing the value, but there is a danger of impairing the gear-shifting action, and of inadequate lubrication of the gear stage under consideration.

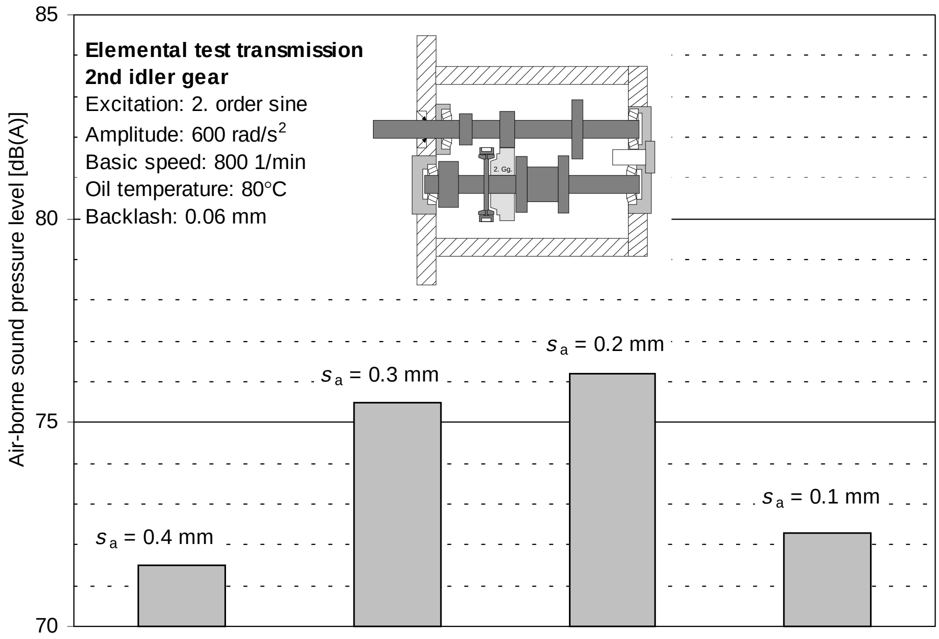


Figure 10. Effect of axial clearance on the rattle noise level at an angular acceleration amplitude of 600 rad/s^2, based on the example of the second gear idler gear

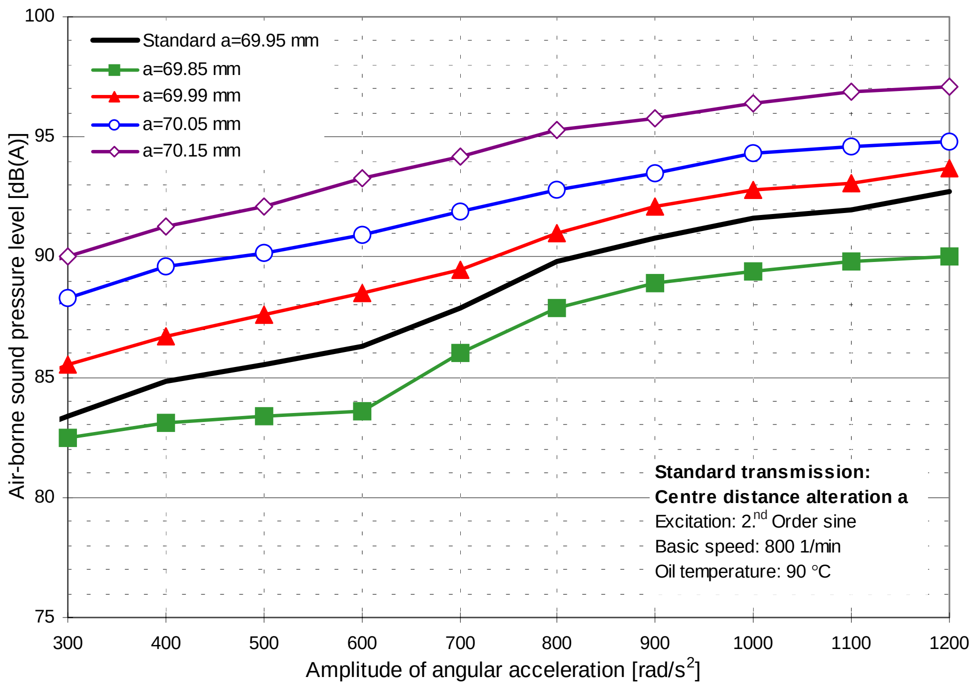


Figure 11. Effect of centre distance on the rattle noise based on the example of a standard transmission

6.1.3. Influence of the main centre distance alteration

The influence of the main centre distance a was investigated on a standard transmission.

In Figure 11, the main centre distance is infinitely varied by the changing position of the countershaft in 0.04 mm, 0.1 mm and 0.2 mm increments.

Reducing the main centre distance up to a specific limit of 69.85 mm has the effect of minimising the air-borne sound pressure for this standard transmission up to 2.5 dB(A), whereas increasing it by 0.2 mm (70.15 mm) results in an increase in the value of 7 dB(A).

6.2. Restricting the movement of loose parts

6.2.1. Minimising the axial impacts of loose parts

In the previous section the transmission parameters backlash, axial clearance and centre distance were infinitely varied within suitable limits. If optimising the transmission parameters does not lead to any satisfactory reduction in the noise level, then the targeted design engineering of the loose part can be undertaken to reduce the freedom of movement and thus minimise rattling and clattering noise.

In the elemental test transmission, a vibration damping device was installed based on the second gear stage, and compared to the standard version. Apart from the second gear stage, all other loose parts were again removed during the measurements. The second gear idler gear was fitted with elastic thrust collars in order to minimise the axial impacts, which can make a significant contribution to rattling and clattering noise. The face of the idler gear was fitted with elastomer inserts to reduce axial impact (Figure 12). The standard axial clearance was increased by the amount of the elastomer insert. This allowed the same axial vibration movement as in the standard version.

Figure 11 shows the trial results for the second gear idler gear with and without axial impact reduction by fitting elastic axial thrust collars. The basic noise is not significantly greater with this arrangement. The rattling noise limit on the other hand is displaced to higher angular acceleration amplitudes. This is attributable to the elastomer insert initially impeding the idler gear in its circumferential direction, thus reducing the intensity of the torsional impacts. The air-borne sound pressure level is also significantly below the standard version with higher angular acceleration amplitudes.

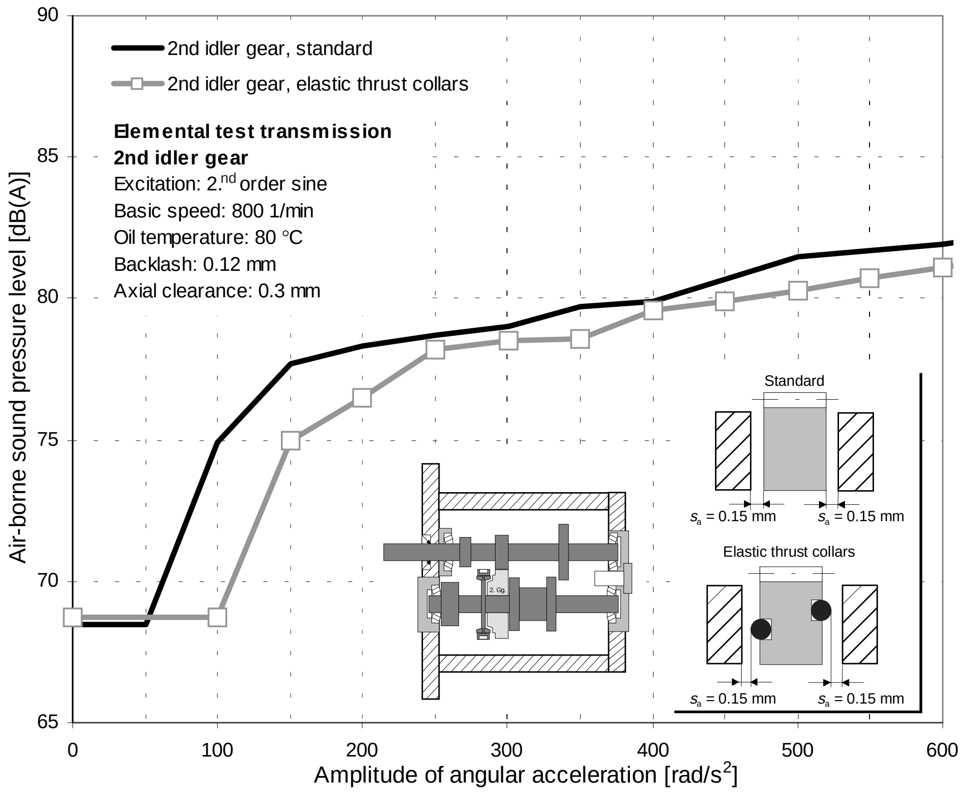


Figure 12. Reducing the axial impact by fitting elastic thrust collars, based on the second gear stage

6.2.2. Minimising the torsional impacts of loose parts

In a second series of experiments the elastomer inserts were installed between the idler gear and the two thrust collars with different pre-compressions of 10 N and 20 N. The idler gear could only move by means of elastic deformation of the elastomer in an axial direction.

In Figure 13, the rattling noise level curves for the elastomers with a pre-compression of 10 N and 20 N are contrasted to the standard version up to an angular acceleration amplitude of 1600 rad/s^2. The rattling noise limit is displaced to significantly higher angular acceleration amplitudes by the increase of the pre-compression of the elastomer. From the rattling noise limit, the air-borne sound pressure level is up to 4 dB(A) below the standard version for 10 N elastomer pre-compression, and up to 7 dB(A) below for 20 N.

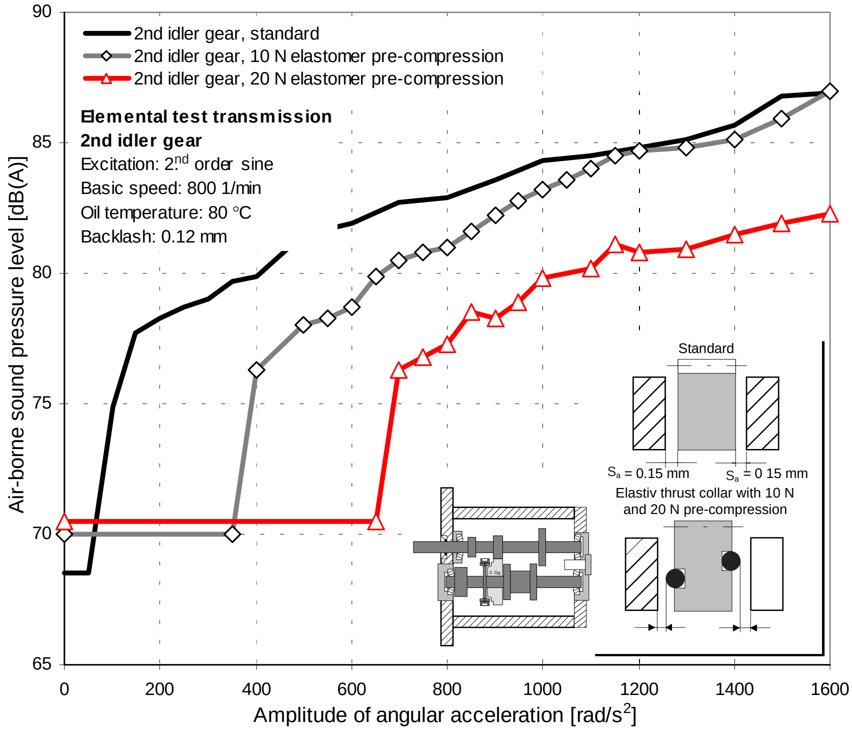


Figure 13. Pre-compressed elastomer inserts for minimising rattling and clattering noise, based on the second gear stage

7. Comparison of measurement and EKM-Simulation

For direct comparison with the measured airborne noise level the calculated impact intensity is multiplied by the correlation factor k, which refers to the air-borne noise level in Decibel A picked up by the microphone at a distance of 10 cm from the transmission housing. This distance has not changed for the other transmissions. The following figures show the parameters and the results of measurement and EKM-Simulation for the 5-speed manual front and transverse mounted transmissions (see Figure 2) during driving in 5^{th} speed, Figure 15.

ω_{an}	[rad/s]	230						
μ	-	0,30						
k	-	25500000						
L_{pBasic}	[dB(A)]	65						
		1st gear	**2nd gear**	**3rd gear**	**4th gear**	**5th gear**	**Lay shaft**	**Reverse gear**
Amplitude 500 rad/s^2	[rad/s^2]	500	500	618	618	-	833	144
β	[°]	26.25	28.25	27.00	32.50	31.00	27.00	26.25
r_{b1}	[mm]	16.16	24.75	42.27	36.75	41.12	31.37	55.94
r_{b2}	[mm]	55.94	48.42	31.37	37.62	33.25	40.91	53.46
J_2	[10^{-4}kgm^2]	21.51	14.50	4.15	5.57	4.51	8.95	17.26
T_2	[Nmm]	24.00	19.90	13.20	14.70	19.50	21.70	21.50
m_2	[kg]	1.20	1.10	0.56	0.60	0.54	1.53	1.07
5-speed manual front and transverse mounted transmission 1								
s_f	[mm]	0.111	0.109	0.099	0.102	0.098	0.196	0.22
s_a	[mm]	0.32	0.2	0.05	0.08	0.32	0.15	0.34
$L_{p,sim}$	[dB(A)]	81.5	82	83.1	82.6	-	88.8	82.5
$L_{pges,sim}$			[dB(A)]			92.1		
5-speed manual front and transverse mounted transmission 2								
s_f	[mm]	0.108	0.116	0.089	0.066	0.132	0.176	0.214
s_a	[mm]	0.29	0.15	0.37	0.21	0.33	0.18	0.36
$L_{p,sim}$	[dB(A)]	81.5	83	80.3	79.7	-	88.4	82.4
$L_{pges,sim}$			[dB(A)]			91.5		
5-speed manual front and transverse mounted transmission 3								
s_f	[mm]	0.116	0.106	0.19	0.099	0.123	0.293	0.222
s_a	[mm]	0.34	0.12	0.37	0.11	0.25	0.18	0.3
$L_{p,sim}$	[dB(A)]	81.6	83.1	83.6	82.2	-	89.7	82.5
$L_{pges,sim}$			[dB(A)]			92.7		

Figure 14. Simulation parameter and transmission data for the 5-speed manual front and transverse mounted transmissions driving in 5th speed

8. Summary

The cause of rattling and clattering noise is torsional vibration of transmission components that are not under load, that move backwards and forwards within their functional clearances. This noise is perceived as distinct from other sources of noise, and is intrusive because of its undesirable character. The transmission parameters backlash, axial clearance and main centre distance were varied by experimental analyses in test stand trials, showing the effect on the propensity to rattle and clatter. By optimising these parameters, it was possible to minimise the rattling and clattering noise. Measures internal to the transmission to reduce loose part vibration in vehicle transmissions were also considered. The effectiveness, in terms of minimising clattering and rattling proneness, of making the axial thrust collars elastic with and without pre-compression of the elastomer was considered. All internal transmission measures discerned as effective in reducing rattling and clattering noise need to be examined in terms of service life and possible side effects in all the operating states arising in a vehicle transmission.

The calculated noise level with the EKM-Simulation program correlates with the measured noise level. Parameter studies with the EKM-Simulation program, which contains all relevant parameters for the excitation of rattling noise caused by idle gears, shows the significant parameters for the investigated transmissions.

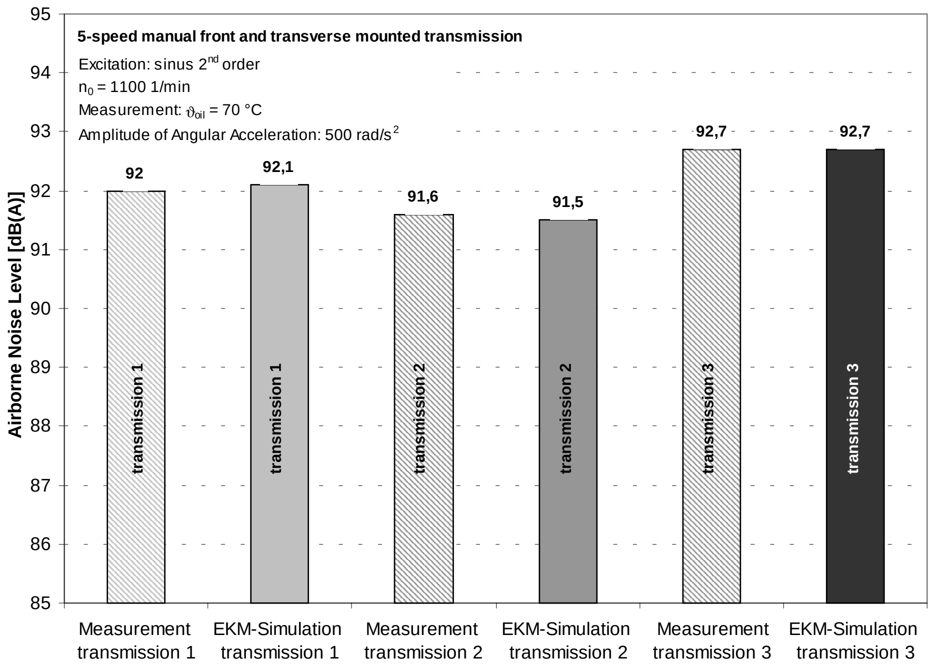


Figure 15. Comparison of measurement and EKM-Simulation

9. Nomenclature

C_{lm}	[-]	non-dimensional intensity of impacts
C_{mv}	[-]	non-dimensional drag torque
C_{fa}	[-]	non-dimensional axial friction force
C_{sa}	[-]	non-dimensional axial clearance
C_{sv}	[-]	non-dimensional circumferential backlash
D	[mm]	pitch circle diameter
F_R	[N]	friction force
I_m	[N]	average intensity of impacts
J	[kgm^2]	moment of inertia
k	[-]	correlation factor
T	[Nm]	drag torque
m	[kg]	mass
$L_{Klapper}$	[dB]	air-borne rattle noise level
$L_{p,Basic}$	[dB]	basic noise level
$L_{p,sim}$	[dB]	simulated air-borne rattle noise level of one gear
$L_{pges,sim}$	[dB]	simulated air-borne rattle noise level of a transmission
r_b	[mm]	pitch circle diameter
s_a	[mm]	axial clearance
s_v	[mm]	backlash
β	[°]	helix angle
μ	[-]	friction coefficient
ω	[rads^{-1}]	angular velocity
ω_0	[rads^{-1}]	average angular velocity
ω_{An}	[rads^{-1}]	excitation frequency
$\hat{\dot{\omega}}$	[rads^{-2}]	amplitude of angular acceleration

10. Indices

1 fixed gear
2 idle gear

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