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IMPROVEMENT OF POWERTRAIN EFFICIENCY VIA CARF-TRANSMISSIONS

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Abstract

Clatter and rattle noise is an increasing challenge for the development of low-noise transmissions. Efforts to reduce fuel consumption and exhaust emissions of combustion engines combined with a decrease of the idle speed cause increasing cyclic irregularities within modern drivelines which result in an increasing clatter and rattle noise and can even lead to serious damage of driveline components.

Beside these aspects, the improvement of transmission efficiency as part of the powertrain efficiency and therefore fuel consumption of the vehicle is a major task during the development of a new transmission.

In order to design future transmissions that implicate improvements on both tasks simultaneously, new grounds were broken by the Institute of Machine Components (IMA) of the University of Stuttgart, Germany in the field of clatter and rattle noise free automotive transmissions. This is achieved in principle by decoupling all gears which are not in the power flow. On the basis of a patented solution named CARF-Transmission, the ***C***latter ***A***nd ***R***attle noise ***F***ree transmission is presented as well as derived developments. Both targets, i.e. reduction of the emitted clatter and rattle noise as well as the efficiency enhancement by reducing the drag torque can be achieved with this transmission.

Keywords: CARF-Transmission, efficiency enhancement, fuel economy, rattle and clatter noise

1. Introduction

The development of fuel efficient vehicles is one of the main topics of the automotive industry. Beside the development of lightweight motorcars and the reduction of aerodynamic drag, the improvement of powertrain efficiency is one of the most promising targets in automotive design to meet this objective.

Minimising noise has also become an increasingly important factor in automotive development. The importance of this development target is increasing with rising customer expectations in driving comfort and increasingly stringent legal restrictions on noise emissions [1, 2, 3].

Automotive transmission noise can be broken down into several groups according to their causes [1, 2], **Figure 1**. The most dominant type of transmission noise is rolling contact noise from gear pairs under load known as whining and squealing but also as grinding and singing. This type of noise is caused by meshing impacts, parametrically excited vibration and rolling contact noise due to variations in pitch spacing.

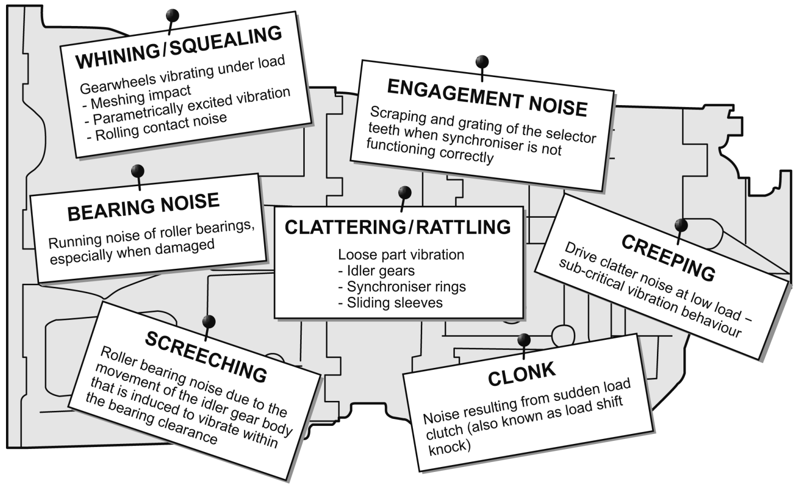

Figure 1. Classification of automotive transmission noises

Clattering and rattling is caused by torsional vibration of loose parts, i.e. parts, such as idler gears, synchroniser rings and sliding sleeves, which are not under load and therefore can move within their functional clearances. The intensity of torsional vibration in the powertrain has been increasing dramatically in recent years due to nearly all means of combustion engine design in order to reduce fuel consumption, such as direct fuel injection, lightweight design, minimisation of the moment of inertia of the crankshaft or usage of supercharged engines. Also the reduction of the idle speed as well as the implementation of an automatic engine cut-off and starting system lead to increasing torsional vibrations. This noise is known as rattling when the transmission is in neutral, and as clattering when any gear is engaged under power or in overrun (pull/push operation).

Clatter and rattle noise is perceived as unpleasant not only because of its high airborne sound pressure level, but also because of its intrusive characteristic. Therefore it constitutes a comfort problem (noise immission) in case of passenger cars, and both comfort and environmental pollution problem (noise emission) in commercial vehicles. Clattering that arises under power with low load, for example at low road speeds and engine speeds, is called creeping.

Gear shifting noise can also arise in automotive transmissions due to scraping and grating of the selector teeth as a result of defective synchroniser functioning, as well as from transient load cycle excitation, referred to as clonk or load shift knock [4]. This noise also can occur by fast engaging or disengaging of the clutch. Bearing noise can appear especially in case of damaged roller bearings, and screeching noise by vibration of the idler body within the bearing clearance.

One of the major transmission losses which affect the efficiency is drag torque. Drag torque T_2 consists of the following four components compression torque T_{Qu}, synchronisation drag torque T_{Sy}, bearing friction torque T_L and churning torque T_{Pl}, **Equation 1**.

$$T_2 = T_{Qu} + T_{Sy} + T_L + T_{Pl} \tag{1}$$

In this paper the design and development of a transmission which meets both required targets, the minimisation of transmission losses and therefore higher powertrain efficiency as well as the elimination of unwanted rattle and reduction of clatter noise emission is presented. This transmission, named CARF-Transmission, was developed at the Institute of Machine Components (IMA) of the University of Stuttgart, Germany.

2. Statement of problem: causes of loose part noise

Clatter and rattle noise in automotive transmissions is mainly caused by torsional vibration transmitted from the combustion engine to the transmission input shaft. This is due to discontinuous combustion processes, resulting in periodically fluctuating drive torque at the crankshaft. Unbalanced engine masses also have a sustainable effect on the rotary movement of the engine crankshaft. This rotary oscillation is superimposed on the rotary movement of the crankshaft, resulting in an irregular rotational speed of the combustion engine, which is in principle sinusoidal. The relevant portion of the torsional vibration in a four-stroke combustion engine is the bisector of the number of cylinders.

The cyclic irregularity of combustion engines is increasing due to efforts to improve fuel consumption and emission levels, such as the increasing use of supercharging, multi-valve technology and direct fuel injection, combined with reducing idling speeds. The engine speed pattern is affected by the operation of additional consumers such as air conditioning and rear-window demister. These aspects further increase the irregularity of rotational speed, resulting in larger angular acceleration amplitudes and changes in their characteristic.

In spite of means of torsional vibration damping, e.g. in the clutch, the torsional vibration transmitted from the internal combustion engine to the transmission excites idle components such as idler gears, synchroniser rings and sliding sleeves to vibrate within their functional clearances. The fixed components encounter idle components at the clearance limits, resulting in impacts perceived as rattle or clatter noise. The intensity and frequency of the impacts are directly related to the airborne sound pressure emitted from the gearbox housing [5, 6].

Clattering and rattling arises chiefly in manual and automated transmissions but also automatic transmissions with engaged lockup clutch can tend to clatter noise emissions.

3. Responsible parameters of clatter and rattle behaviour

The parameters responsible for clattering and rattling can be divided into operating parameters and geometrical parameters. The operating parameters include the excitation frequency as the product of rotational speed and engine order, which is responsible for the number of impulses per rotation of the shaft, and the angular acceleration amplitude which is the main factor determining contact between the fixed gear and idler gear flanks. The diameter of the fixed gear and the geometrical parameters of the idler gear such as diameter, moment of inertia, mass, helix angle and the associated reduction ratio are the main parameters that can be influenced at the gear development stage in terms of their rattle and clatter noise behaviour, **Figure 2**.

However, there is some goal conflict between designing the gears in respect to power transmission and to their service life and low rattling proneness. Mainly the torsional backlash and the axial clearance offer scope for reducing this noise. Reducing the torsional backlash and a defined increase or reduction of the axial clearance results in a reduction of the airborne sound pressure level emitted [5, 7].

Figure 3 illustrates the example of an idler gear whose circumferential and axial movement behaviour relates to the rotational irregularity and its impact performance. The sinusoidal rotational speed profile of the transmission input shaft (for example that of a four-cylinder four-stroke internal combustion engine) is characterised by acceleration and deceleration phases of a fixed gear that transmits its movement pattern to the idler gear, Figure 3a. When the rotational speed profile is decelerating (falling slope of the curve), the idler gear grips the

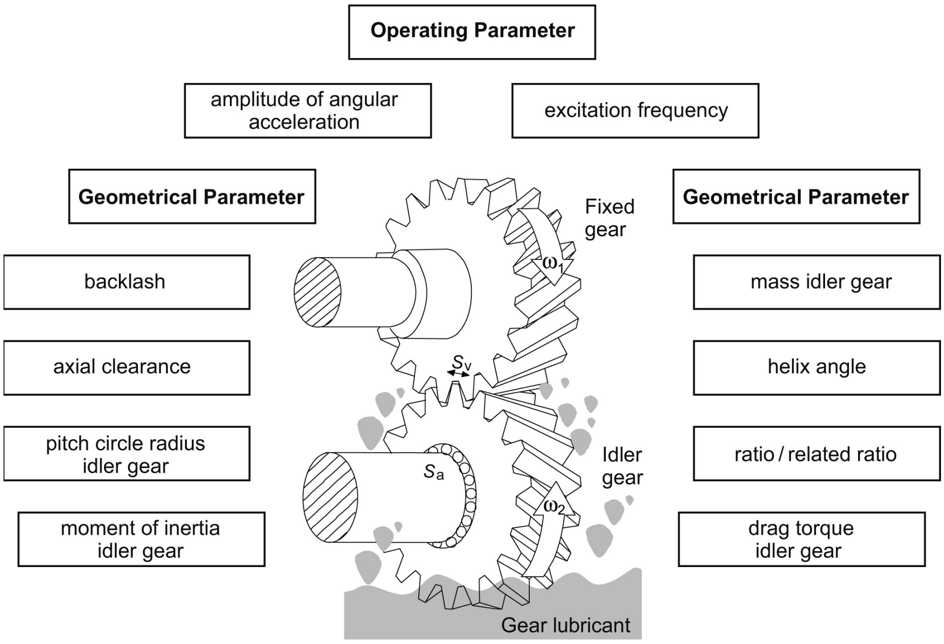

Figure 2. Influential parameters of rattle and clatter noise propensity

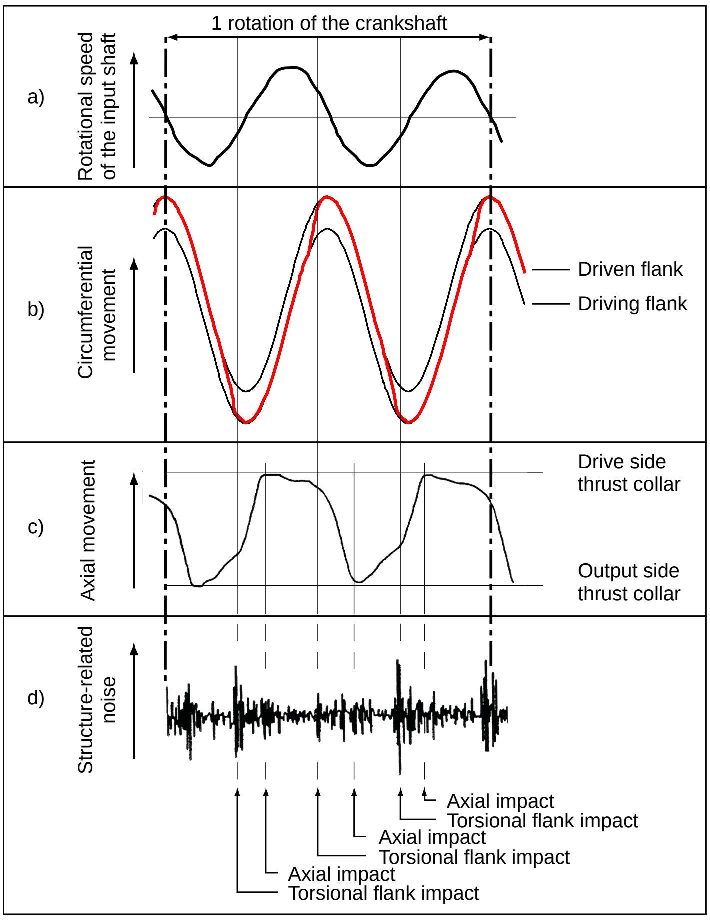

Figure 3. Movement of an idler gear as a result of an irregular rotational speed profile in the case of a fixed/idler gear pair [6, 7]

driven flank of the fixed gear, Figure 3b. As soon as an acceleration phase is entered (rising rotational speed profile), the idler gear comes away from the driven flank of the fixed gear with following flying phase and then impacting against the driving flank of the fixed gear, which is perceived as structure-related noise, Figure 3d. After the torsional flank impact, the helical cut idler gear impacts against the drive side thrust collar, Figure 3c, which can be clearly recognised in the structure-related noise graph, Figure 3d. Each torsional flank impact is followed by an axial impact whose intensity is less than that of the torsional flank impact.

Transmission fluid as an engineering design parameter also has a considerable influence on clatter and rattle noises. The important factors include the type of oil, the additives used, the viscosity (which is directly related to temperature) and the level of oil in the transmission which together act on a gear pair as drag torque, resulting in a reduction in clatter and rattle noise, especially at low speeds and when cold.

4. Construction of a clatter and rattle noise free transmission

Almost all internal means known until today work according to the principle of applying an additional braking torque to the idler gear. In order to counteract the negative accompaniment of reduction of efficiency, new ground was broken at IMA based on the decoupling of all the transmission's gear wheels that currently are not taking part in the power flow. Consequently it is assured that the transmission shaft's torsional irregularities principally are not transmitted to the gear wheels.

This basic thought was protected in the context of a European Patent, [8]. The shelter includes both the idea of decoupling all gearwheels of a transmission that are currently not taking part in the power flow as well as multiple solution suggestions for novel means of coupling gearwheels with their shafts.

The fact that such measurements offer the disadvantage of additional parts and a more complex shifting mechanism, the necessity of actuating two sleeves and therefore also two synchronisers to engage a gear can also be understood as a possibility to reduce shifting time through more synchroniser capacity or to design synchronisers smaller.

Furthermore the drag torque of those gear ratios that are currently not taking part in the power flow is minimised which leads to a measurable enhancement in efficiency. This becomes apparent by examining the easiest case of a gearbox diagram for a manual transmission for front-wheel drive with a longitudinal or transversal engine, **Figure 4**.

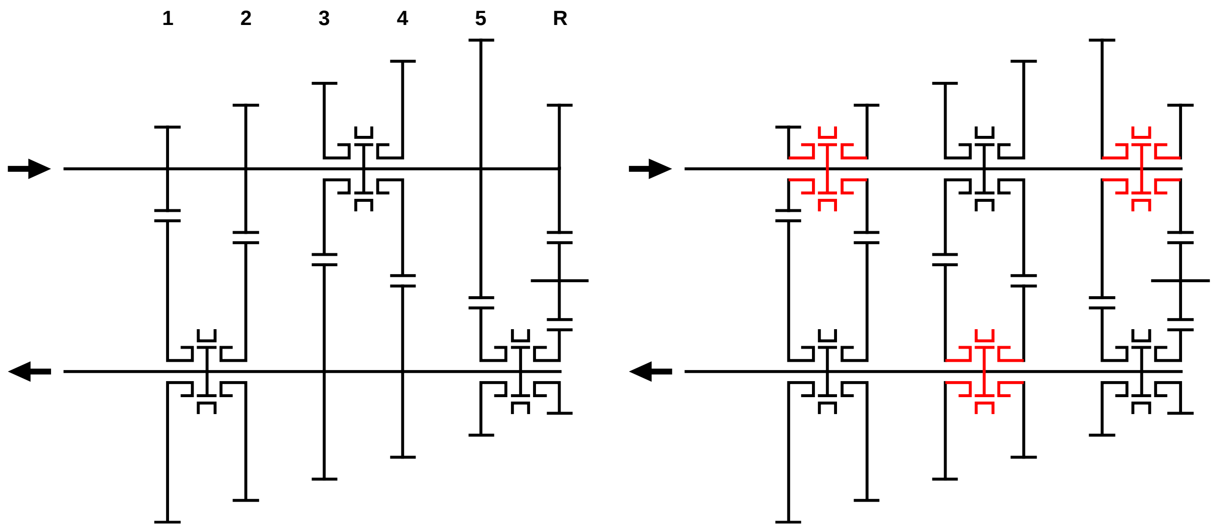

Figure 4. FWD manual transmission redesigned as CARF-Transmission

4.1 Design of a solution

Taking a look at the comparatively high extra costs and the additional weight of a complete CARF-Transmission, the question rises, whether it is possible to achieve a comparable success with only a small extra effort compared to a regular transmission.

A special constellation appears by the appliance of the principle of decoupling on a gearbox in coaxial design. By the insertion of only a single additional synchroniser pack in order to decouple the constant pinion both a complete rattle noise freedom and a remarkable reduction of clatter noise at engaged direct gear can be achieved. **Figure 5** shows such a modified gearbox diagram.

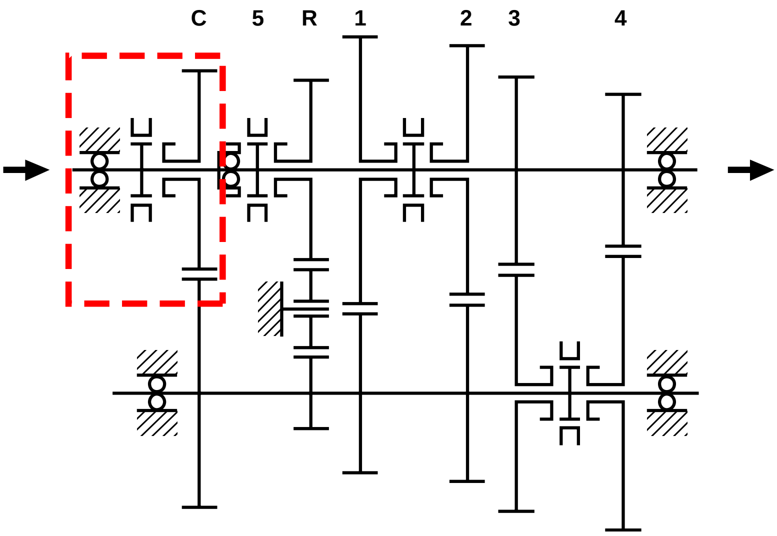

Figure 5. Gearbox diagram of the modified manual transmission in coaxial design, C = constant gear step

During design of the prototype the additional synchroniser for switching the constant pinion and the countershaft on and off was put into an additional housing inside the clutch bell housing due to a lack in space within the actual gearbox housing. The constant pinion is linked to a hollow shaft that also carries its synchroniser hub. **Figure 6** shows a cross-section of the design.

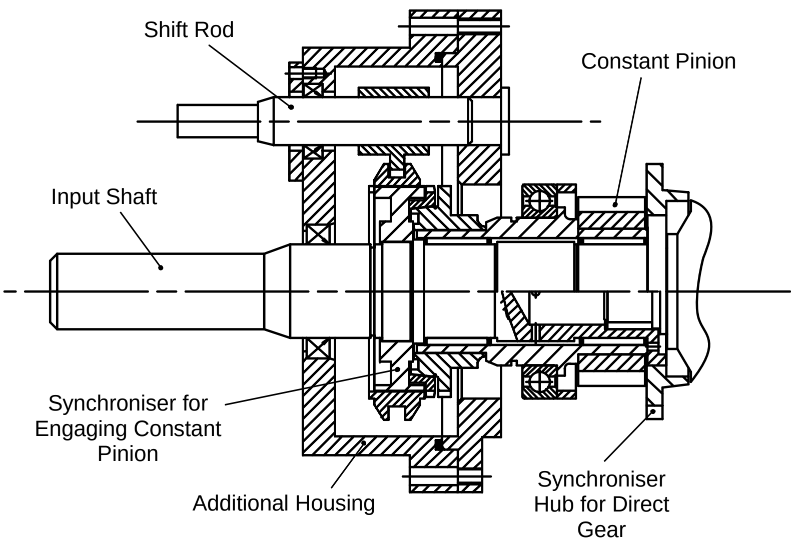

Figure 6. Cross-section of the new module “Shiftable Constant Pinion“

The new input shaft carries the synchroniser body and its sleeve to (de-) couple the constant pinion and is welded to the synchroniser hub for the direct gear at its other end. The sleeve is actuated by an additional shift rod and is connected to a shifting linkage which is temporarily lead outside of the gearbox housing for hand-shifting.

With this prototype it is possible to decouple the constant pinion and thus also the complete countershaft from the constraint rotational movement of the input shaft or to switch it back on during operation. This allows measurements of both the noise behaviour as well as the drag torque at various shifting conditions.

5. IMA rattle and clatter noise test rig

At IMA, a test rig for rattle and clatter noise investigations for automotive transmissions was developed. The drive unit of the test rig, consisting of a highly dynamic brushless three-phase synchronous motor with permanent-magnet excitation, is flanged to a massive steel rack coaxial opposite to the tested transmission. The connection between the drive unit and the transmission is a torsionally stiff metal flex boot coupling, which allows an adherent power transmission without clearance. To eliminate structure-borne sound transfer points, the test configuration is mounted to two wooden beams and is decoupled from the machine base by air springs. The machine base is further decoupled from the foundation by spring elements. **Figure 7** shows the principle arrangement of the test rig.

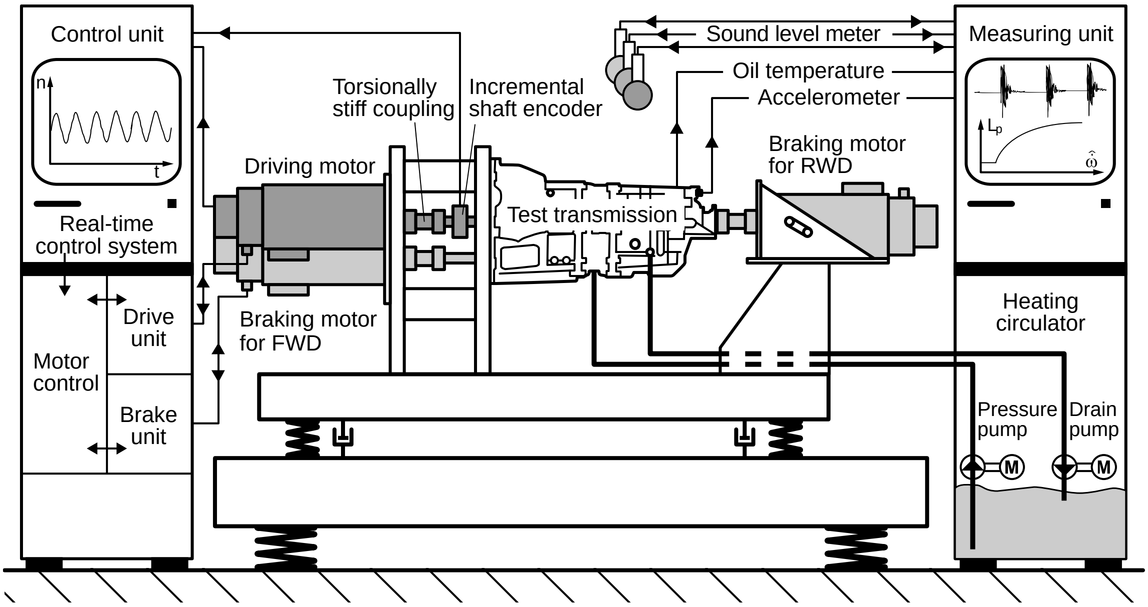

Figure 7. Arrangement of the IMA rattle and clatter noise test rig

To simulate traction and thrust operations, an additional braking motor, also a servomotor, is flanged to the output shaft by means of a metal flex boot coupling. Both servomotors are identical and have a rated power of 12 kW at a rated torque of 30 Nm. The peak torque is 120 Nm and the maximum driving speed at homogeneous rotational motion is 4000 rpm. Due to the small inertia of the shaft of the driving unit, which is $5 \cdot 10^{-3}$ kgm^2, high angular acceleration amplitudes up to 6000 rad/s^2 (without test transmission) can be reached.

The nominal values of torsional vibrations of the servomotors are generated by a computer-controlled functional generator. These nominal values can be transmitted to the motor control by discrete data points or analytically. This allows the simulation of measured speed patterns of combustion engines, e.g. idle mode with or without consuming devices.

The incremental shaft encoder mounted between drive unit and transmission is used to sense the actual condition of speed and angular acceleration amplitude truly transmitted to the test transmission. The pattern of the angular acceleration at the transmission input shaft is recorded with a high resolution of 50 kHz by a digital-analogue converted signal of the incremental shaft encoder. An acceleration pickup which is mounted to the transmission housing surface is used to qualitatively identify the clattering and rattling impacts which are temporally dependent upon the cyclic irregularities. Both the impact impulses and the cyclic irregularities are recorded simultaneously.

The airborne sound pressure level is usually recorded by three sound level meters in the far field of the transmission at a distance of 1000 mm from the transmission housing. Besides sound pressure levels, the temporal pattern of the airborne sound pressure can be recorded as a WAV-file at a high sampling rate to be able to perform frequency analyses later if necessary. It is also possible to measure the sound power of the transmission itself. With this method the sound influence of all other sound sources such as the servomotors can be eliminated.

To simulate realistic operating conditions, the temperature of the tested transmission can be adjusted by means of an oil heating and cooling unit. Also the oil level can be adjusted by tilting the test rig around two axes in order to achieve realistic mounting positions.

6. Noise measurements

At first the rattle noise behaviour of the introduced coaxial gearbox was investigated in idling mode with an immovable output shaft. In case of the constant pinion being coupled to the input shaft the countershaft already appears as a loose part that can be excited to rattle noise. Furthermore the idler gears of the 1st, 2nd and reverse gear as well as the synchroniser rings of all gears become excited. The resulting emitted sound pressure level is plotted versus the angular acceleration amplitude in **Figure 8**.

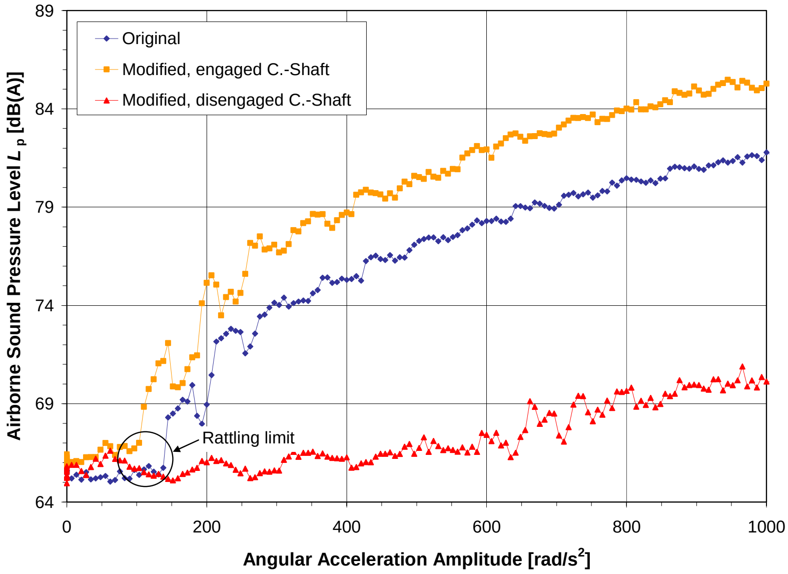

Figure 8. Distribution of sound pressure levels of the modified transmission in idling mode @ 800 rpm

Both the transmission left in original condition and the prototype with engaged countershaft show a distribution typical for rattle noise. The quantitative difference of these two curves can be explained by the modifications of the prototype gearbox whereby characteristic parameters have changed that have direct influence on the rattle noise level. For instance the original bearings were changed as well as the emission behaviour of the gearbox housing by modifications. Secondly a supplementary rattling spot was created by separation of constant pinion and synchroniser hub for the direct gear in design. Next, the synchroniser hub appears as an axial thrust collar for the constant pinion as a loose part in modified condition. Incidentally, it should be mentioned that it required almost six years of production between the prototype transmission in original condition and the original transmission itself used for comparison measurements. Within these years slight design changes positively influencing the rattle noise behaviour were assumed.

When decoupling the constant pinion the only rotating part of the transmission is the new input shaft. Experiments showed that the synchroniser rings of the constant pinion synchroniser pack and the direct gear are excited to vibrations and created rattle noise. Indeed the noise is very small due to the very small moment of inertia of the synchroniser rings so that only a relatively marginal increase of the rattle noise level is recognisable, Figure 8. In this case a synchroniser ring with exceptionally large clearance in axial and tangential direction was used for the constant pinion synchroniser pack in order to show this effect more clearly. If such a modified transmission was to be mounted into a real driveline, it is admissible to discuss complete rattle and clatter noise freedom in combination with common means for torsional damping between combustion engine and transmission.

Furthermore the modified transmission's noise behaviour at engaged direct gear is interesting because now the countershaft is "switched off" by decoupling the constant pinion. Independent from the countershaft's condition the idler gears of the 3rd and 4th gear ratio develop clatter noise at engaged direct gear because their exciting fixed gears are mounted to the main shaft, cp. gearbox diagram, Figure 5.

In order to point out this effect on the noise at a disengaged countershaft, these idler gears including their synchroniser rings and sleeve were demounted. Merely the synchroniser body was left in the gearbox to guarantee the exact axial position of the countershaft fixed bearing. **Figure 9** shows the measured airborne sound pressure levels.

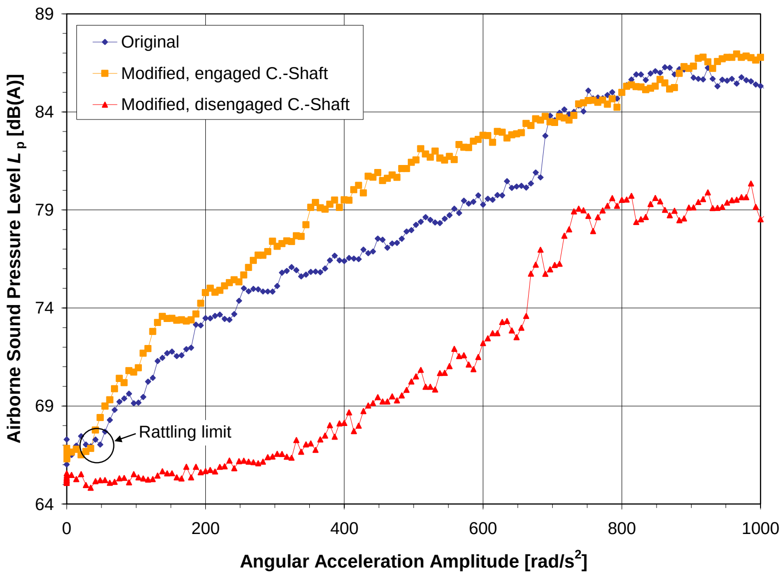

Figure 9. Distribution of airborne sound pressure levels of the modified transmission at engaged direct gear @ 800 rpm

Again, the original transmission and the prototype with coupled constant pinion show typical distributions. However, the distribution of the clatter noise level of the prototype with switched off countershaft turns out to be unusual (bottom curve). Neither a sharp rattling limit like at the upper two curves nor the characteristic distribution at higher angular acceleration amplitudes can be identified. The reason for this is mainly the undefined state of movement of the constant pinion and the countershaft together with all its meshing idler gears.

The constant pinion's decoupling does not mean that it and thus also the whole countershaft stands still. On the contrary a torque balance of accelerating and braking forces is established at the freely rotating parts and especially at the countershaft. Hereby, the oil's drag forces inside the needle bearings that are located between the idler gears on the input and the output shafts and the bearings themselves as well as the drag forces between the friction surfaces of the output shaft's synchroniser packs act in an accelerating manner. In contrast, the splashing of the countershaft in the oil sump and the friction within the countershaft's bearings act in a braking manner. Consequently, it can be explained why the countershaft doesn't stand still at all, as long as either the input or the output shaft rotates.

Up to an angular acceleration amplitude of about 500 rad/s^2 a distribution similar to the one in idling mode with disengaged countershaft can be observed, cp. Figure 8. Afterwards sporadical clatter noise can be heard which becomes stronger for rotational accelerations greater than 700 rad/s^2. At this point at the latest, the countershaft begins to adopt an irregular rotational movement even though it is damped, i.e. with reduced effective angular acceleration amplitude. Nevertheless, the average noise level in decoupled state remains around 6 dB below the noise level of the coupled version, which represents one fourth of the sound energy of the latter. An increase of 10 dB is felt double as loud by the human ear, [9].

With this, the success of the measure taken has been demonstrated in an impressive manner. Also, perpetual clatter noise as just described can be further minimised by systematically influencing the countershaft's speed. Especially the rattling limit from which distinct clatter noise can be heard can be pushed to higher values by influencing the accelerating and braking forces. In combination with external measures, a complete clatter noise freedom at engaged direct gear can thus be achieved. Further investigations at IMA are currently running.

7. Torque measurements

The drag torque of the modified transmission was tested by a torque measuring shaft at the IMA test rig. All tests were performed at two different oil temperatures to simulate cold (ambient) and hot operation. **Figure 10** pictures the measurement results for cold condition.

In **Figure 11**, the results of the drag torque measurement for hot oil is shown. The reduction of the drag torque can be explained by the reduction of the oil viscosity which leads to a lower loss torque. Both figures show that the highest measured drag torque occurs at engaged 5th gear caused by churning losses due to splashing gears in the oil sump. This gear step however has the highest share time e.g. of more than 85 % in service life for commercial vehicles. Therefore the fuel efficiency can be enhanced by the reduction of the drag torque at this gear step.

As described before the countershaft does not stand still at a defined speed of the input shaft in decoupled condition, but rather takes on a steady speed. However, the latter is lower than the constraint speed in coupled condition of the constant pinion.

Consequently a reduction of the drag torque can be expected which was also measured at the IMA test rig. In **Figure 12** the transmission's drag torque in idling mode is plotted versus the speed of the input shaft at various oil temperatures with both switched on and off countershaft.

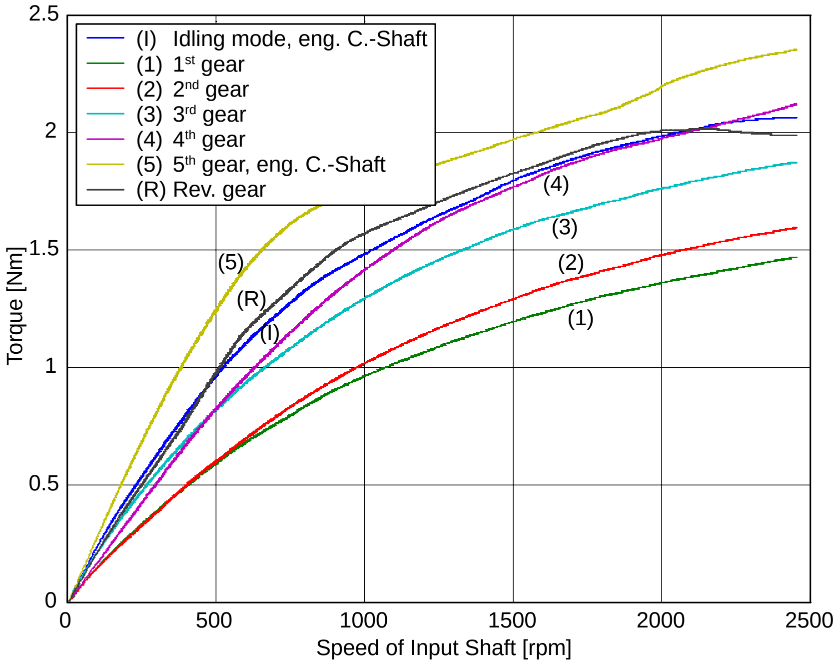

Figure 10. Drag torque overview of the modified transmission at ambient oil temperature with engaged countershaft at idling mode and shifted 5th gear

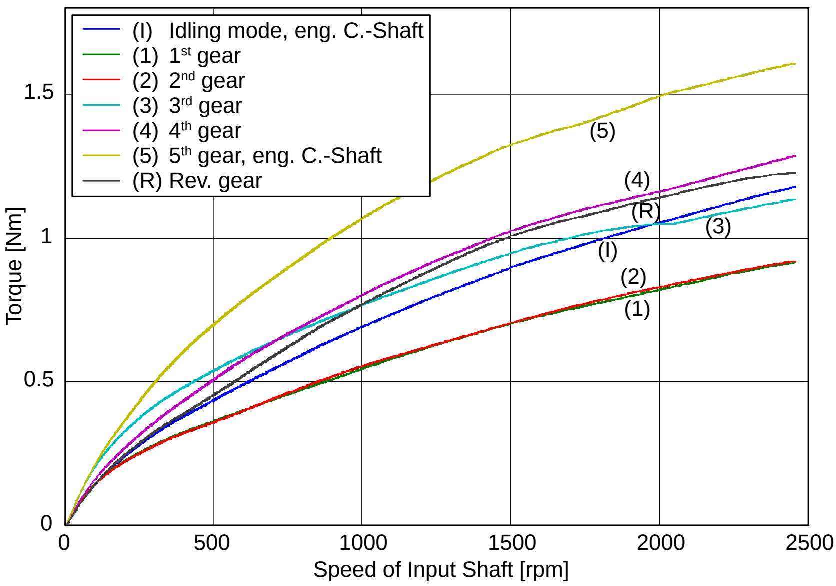

Figure 11. Drag torque overview of the modified transmission at hot oil temperature with engaged countershaft at idling mode and shifted 5th gear

In both cases a remarkable reduction of the gearbox drag torque can be identified in decoupled condition. In practice the observation of the direct gear's behaviour is of much higher meaning because of its major share time, especially for commercial vehicles in long-distance traffic.

In order to come closer to the typical design of commercial vehicle transmissions with all idler gears mounted on the output shaft, the idler gears of the 3rd and 4th gear ratio were removed from the prototype transmission. Originally they are applied to the countershaft and therefore are driven by the output shaft at engaged direct gear and splash loss afflicted in the oil sump. Even though the reduction of the gearbox drag torque is smaller compared to the idling mode here, it is remarkable as **Figure 13** shows and can even dramatically increase depending on the type of transmission.

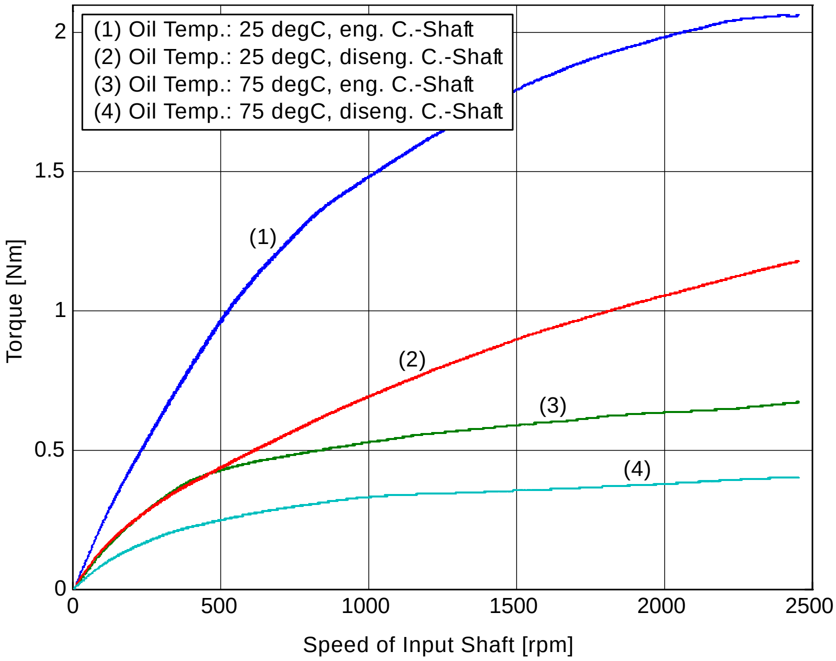

Figure 12. Drag torque of the modified transmission in idling mode with different oil temperature, each with engaged and disengaged countershaft

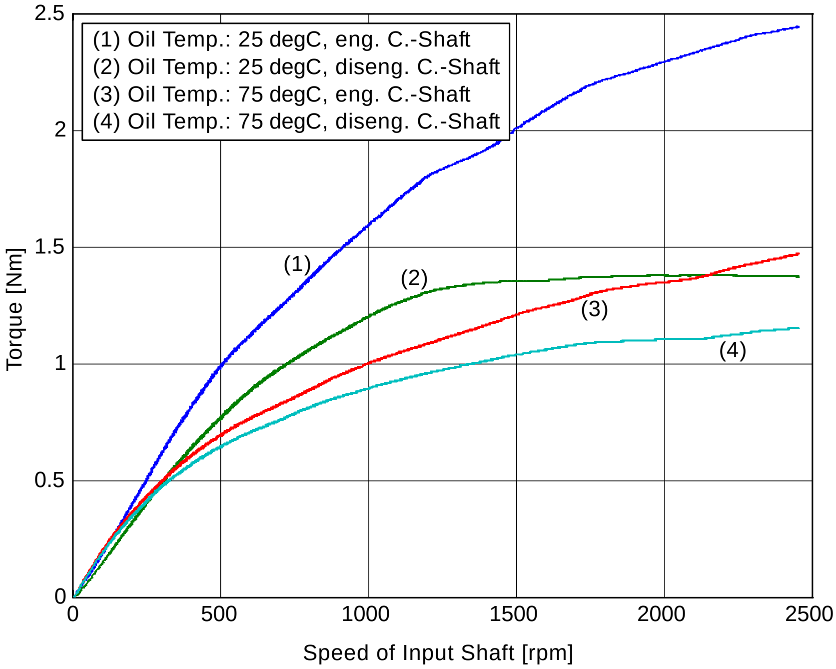

Figure 13. Drag torque of the modified transmission with removed 3rd and 4th idler gear at engaged direct gear with different oil temperature, each with engaged and disengaged countershaft

With this, the transmissions efficiency increases, which directly leads to fuel savings. Furthermore, numerous parts are withdrawn from the influence of rotational irregularities which has a positive effect on the lifetime of the whole transmission. This is to be proved by further durability testing on the IMA power test rig in the future.

As already mentioned above, the application of a supplementary synchroniser pack is accompanied not only by disadvantages in space, weight and costs but also by a more complex shift linkage.

The CARF-Transmission being designed in an automated layout results in programming the activation of both synchroniser packs that have to be engaged when changing gears and leaving the direct gear arise. The necessary frictional work required for changing the speeds of all participating elements can now be spread on two friction surfaces. For this reason, this constellation also offers the possibility of shift time reduction especially when there is a big gear step to the next chosen gear ratio. Even the approach of overlap shifting is considered as two synchroniser packs have to be engaged at the same time when the two synchroniser packs thitherto engaged have to be disengaged.

Unfortunately, the method of decoupling also meets its limits regarding the design. During feasibility investigations regarding CARF-Transmissions, a great number of gearbox layouts on the market and in the patent literature were studied. In doing so, it turned out that an execution as a CARF-Transmission is possible in most cases. Nevertheless, it is very difficult or even impossible especially when planetary gear sets or shift wheels are used in transmissions. The latter are often found in motorcycle gearbox layouts where idler gears work as sleeves for other gears. Such a constellation can also be found in various reverse gear designs, primarily in transmissions of low capacity. Regarding the planetary gear sets, their position in the driveline is the most influential factor. Investigations have shown that for instance range groups in commercial vehicle transmissions only conditionally tend to rattle noise, [10]. On the contrary, planetary gear sets that work as reverse sets, for example used in modern CVT-Transmissions, can be excited to rattle and clatter noise very easily.

8. Summary

Clatter and rattle noise is an increasing challenge for the development of low-noise transmissions. Efforts to reduce fuel consumption and exhaust emissions of combustion engines combined with a decrease in the idle speed cause increasing cyclic irregularities within modern drivelines which result in increasing clatter and rattle noise. This noise emission is subject to excited idle components, for example idler gears, which oscillate within their functional clearances due to inhomogeneous speed patterns and therefore impacting against the fixed gears.

Another important task for the development of automotive powertrains is the efficiency enhancement of all driveline components to gain a more fuel efficient vehicle. Regarding the transmission, an improvement of efficiency can be achieved by reducing transmission losses such as drag toque.

At IMA, a patented solution was developed which meets both the reduction or elimination of unwanted clatter and rattle noise as well as efficiency enhancement. The latter is achieved by a reduction of the churning losses due to the shiftable decoupling of all gears which are not in the power flow. As a complete clatter and rattle noise free transmission requires double the number of synchroniser packs, needle bearings and a more complicate actuation, a gearbox with coaxial layout was modified to achieve a complete rattle noise freedom and a reduced clatter noise emission. This was possible by decoupling the constant pinion and adding only one synchroniser pack. Thus the countershaft can be switched off both in idle mode and direct gear. Thereby, a complete rattle and clatter noise freedom in idle mode and direct gear can be demonstrably achieved, depending on the idler gear arrangement of the transmission.

Another benefit of economical interest appears in particular for commercial vehicles at engaged direct gear that has the highest share time of all gears. Measurements show that the transmission loss torque decreases significantly by decoupled countershaft which leads to an improvement of the transmission efficiency. The so modified transmission also possesses a high potential for a shift-time reduction and a rise in the service life of the countershaft.

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