Investigation of Gear Rattle Phenomena

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The acceptance by discerning customers of passenger cars is dependent upon both the actual noise level and the subjective noise character.

The subjective noise character itself can contain, among other features, undesirable noise phenomena which become apparent at certain points in the vehicle operating range. One such critical phenomenon is gear rattle, which is mainly present under low speed, high load conditions. Due to changes in the angular velocity of the crankshaft, gear rattle under driving conditions occurs at the unloaded gears and splines. It is influenced by a combination of features, such as the inertia of the flywheel, clutch, and all gears, the transmission drag, and the resonant characteristics of the driveline. The effect on airborne noise is also influenced by the characteristics of the vibration transfer paths from the gear teeth meshes via shafts and bearings to the transmission housing. To analyze the interactions of these parameters, it is necessary to conduct both vehicle tests, either on the road or with a chassis dynamometer, and to investigate the bare engine-transmission unit in an anechoic test cell. The first part of the investigations reported in this article deals with the definition by conventional methods of the noise phenomena in the passenger compartment and a parallel application of a subjective noise character evaluation system. This is followed by a correlation analysis of the noise phenomenon recorded in the passenger compartment with the noise at the source.

Finally, a comprehensive investigation of the noise phenomenon on the bare power unit and the potential of key parameters for reduction at the source will be described.

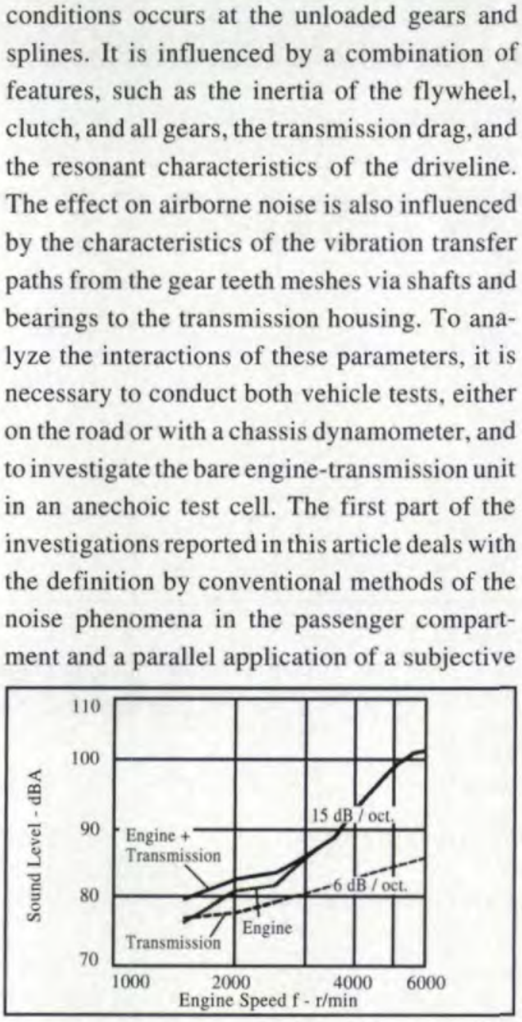

Fig. 1 - Noise of a petrol engine and transmission (full load, fifth gear).

Introduction

Generally, the noise level of today's passenger cars is of an acceptable standard, in terms of both the objective noise level and the subjective noise character. Legislative regulations limit the objective noise level. The subjective noise character, which consists of more than simply the objective noise level,[1] has to be of high quality to meet the demands of discerning customers. Under these circumstances, individual noise phenomena can become annoying, although the objective noise levels are hardly affected. Such a noise phenomenon can arise from the gear rattle of a manual transmission, which occurs either in neutral or under drive conditions.[2] Neutral rattle is generated at idle with the transmission in neutral and the clutch engaged. Drive rattle occurs at the unloaded gear meshing points under driving conditions, usually at higher loads and lower speeds. The annoying character of rattle noise is additionally intensified by the fact that it occurs in the speed range where transmission noise is clearly audible, as it is hardly masked by the engine noise due to the different speed dependencies.

At very low speeds, transmission noise can be similar to or even higher than engine noise, particularly in the case of a petrol engine installation. However, the rate of noise increase with speed is higher for engine noise:[3-6]

$$I_{transm} \text{ a } n^2 \rightarrow 6 \text{ dB/octave} \quad (1)$$

$$I_{engine} \text{ a } n^3 ... n5 \rightarrow 9 ... 15 \text{dB/octave} \quad (2)$$

I = sound intensity

n = speed

As shown in Fig. 1, for a power unit with a 1 ltr. petrol engine, the transmission noise is completely masked by engine noise at speeds above 2500 r/min. The results in Fig. 1 were obtained from noise analysis of the complete power unit and of the engine only (with the transmission wrapped in lead.) The transmission noise was calculated as the difference between these results. However, at speeds higher than 2500 r/min, the difference became too small for an accurate determination of the transmission noise level. For this reason, the curve for the transmission noise was extrapolated (as fine line) from the level at 2500 r/min using a slope of 6 dB/octave. This article will consider only drive rattle phenomena. The analysis of a typical gear rattle problem will be described. This occurred in a front wheel drive passenger car equipped with a 1.1 ltr. petrol engine and five speed manual transmission, and appeared as a disturbing component in interior noise.

Identification

The basic investigations were performed in an acoustic test cell with the bare engine-transmission unit. To ensure that the actual noise problem was treated on the bare power unit, the rattle noise was identified in the vehicle during preliminary road tests. To achieve this, a microphone was located close to the gearbox surface in the engine compartment. An accelerometer was fixed to the gearbox housing and an artificial head measuring system[7] positioned at the co-driver's seat. A digital audio tape recorder was used as a high quality signal storage device. By comparative listening, the actual interior noise, the airborne sound from the gearbox, and the structural vibration of the gearbox housing via headphones - the gear rattle phenomena as well as their typical range of engine operating conditions - could be identified. Fig. 2 shows the frequency spectra of airborne noise measured first near the gearbox, and at the co-driver's seat position during road tests; then in the test cell, again close to the gearbox, and at 1 m from the power unit. The 25% load condition indicates the beginning of gear rattle. Obviously, the gear rattle is a major contributor to high-frequency noise, which can be seen in the good correlation between each diagram; thus, the engine operating range in which gear rattle

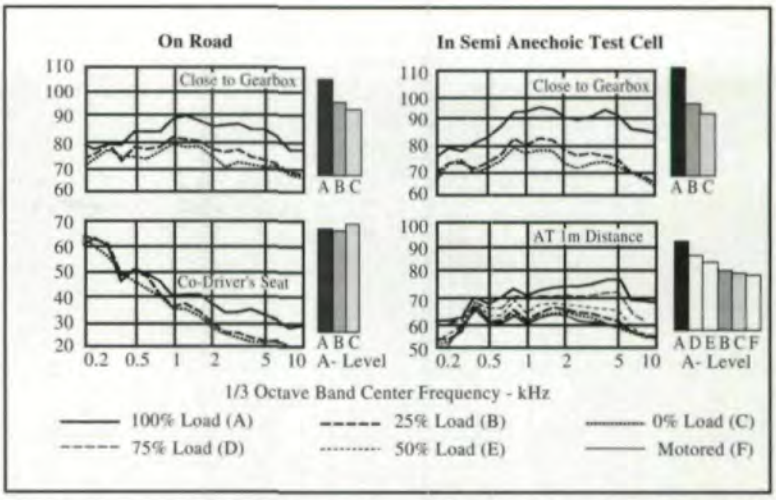

Fig. 2 - Comparison between road test and test cell results (1500 r/min, third gear).

occurred could be limited to the speed range from low idle to 2000 r/min and to the load range from about 25% to full load.

Origin Of Drive Rattle

As described in Ref. 2, drive rattle originates from teeth impacts at the meshes of unloaded gears after passing the backlash point. The impacts occur whenever the angular acceleration is high enough to make the inertia torque at the driven wheel greater than its drag torque:

$$J \cdot \ddot{\phi} > T_{drag} \qquad (3)$$

J = moment of inertia of driven wheel

$\ddot{\phi}$ = angular acceleration of driven wheel

T_{drag} = drag torque acting on driven wheel

Using this relationship, the threshold of gear rattle can be defined in terms of the critical angular acceleration, f crit, at which drive rattle begins to occur:

$$\ddot{\phi}\ crit = T_{drag}/J \qquad (4)$$

Therefore, rattle noise is generated when the angular acceleration reaches high values. This fact is demonstrated in Fig. 3a, representing the time trace of vibration acceleration recorded simultaneously with the speed fluctuation measured at the primary shaft, secondary shaft, and flywheel. Since the angular acceleration is the first derivative of the speed fluctuation (= angular velocity), and since the shape of the speed fluctuation is similar to a sine wave, the maximum angular acceleration occurs during the zero-crossing of the speed fluctuation. Within these time periods, high peaks of structural vibration, which are caused by the rattle impacts, can be observed. The propagation time of typically 80 ms. for the vibration transfer from the gearbox housing is negligible. At an engine speed of 1500 r/min, a time of 80 ms. corresponds to a

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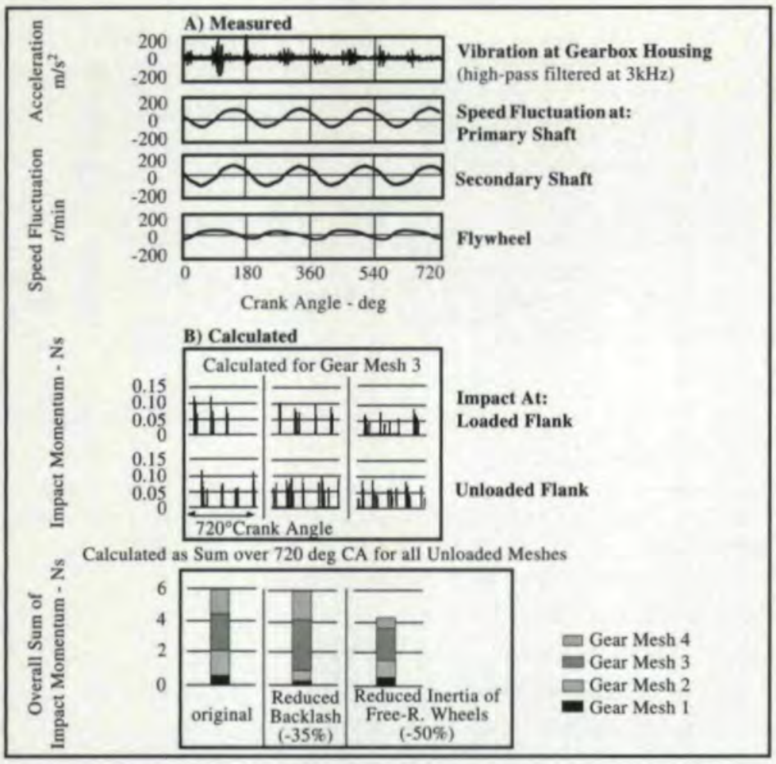

Fig. 3 - Measured and calculated time history of drive rattle impacts (1500 r/min, full load, fifth gear).

crank angle interval of less than one degree.

Some further interesting aspects can be seen in Fig. 3a. At first, the speed fluctuation is dominated by the second order frequency of the engine speed due to the rotary force characteristics of the four-cylinder internal combustion engine. For this reason, impacts occur eight times per engine operating cycle on the fully rattling transmission. Secondly, the speed fluctuations of the primary and secondary shaft are greater than those of the flywheel, and they are additionally shifted in phase. As will be illustrated later, this fact is caused by a torsional resonance in the power train with the clutch as the dominant torsional spring. Thirdly, at one gear meshing point, more than one impact can occur at a given maximum angular acceleration. This is evident from Fig. 3a, where the results were obtained from the transmission equipped - apart from the gear engaged - with only one free-running wheel. In this case all the impacts could only originate from the one unloaded gear mesh. A computer simulation of gear rattle in the fully equipped transmission system, carried out in parallel to the experimental investigation, confirmed this fact and indicated typically several impacts per reversal of tooth loading, and characterized the impacts as elastic collisions. Fig. 3b shows the result of some of these calculations (including the effect of some parameter variations).

Key Parameters

The "primary" parameters which have an influence directly on the generation of rattle noise, can be found in Eq. 4. These are the moments of inertia of the free-running wheel, the angular acceleration, and drag torque acting on the free-running wheel. A fourth parameter not appearing in Eq. 4, but representing a fundamental condition for the generation of rattle noise, is the tooth backlash, since zero backlash precludes any gear rattle. Further possibilities for treating gear rattle are provided by "secondary" parameters acting on the propagation of rattle noise to the outside of the transmission. Here the question arises whether the direct airborne noise contribution or the structural vibration component (coming from the meshes via transfer paths) is more significant in the radiation of airborne noise from the surface of the transmission housing. To find an answer, the insertion loss of the gearbox housing was determined experimentally using a loudspeaker excitation inside the gearbox. A very high insertion loss was measured so that the contribution of the direct airborne sound can be neglected. For example, at 1500 r/min engine speed, full load, and in fourth gear, more than 99% of the total sound power being radiated from the transmission surface originated from the structural vibration component. For this reason, it is useful to consider only those secondary parameters which have an influence either on the vibration transfer from the gear meshes to the outer surface of the transmission or on the airborne sound radiation from the transmission surface.

Backlash. In accordance with experience,[2] the computational analysis indicated only a small effect from backlash on gear rattle (Fig. 3b). Within the limits of current tolerances, the effect of backlash is insignificant. However, zero backlash prevents gear rattle, since the tooth flanks can never loose contact and impact other flanks. Unfortunately, zero backlash is unrealistic for other reasons, except where it can be provided by an additional device such as an anti-rattle plate.[2]

Drag Torque. For lower rattle noise, the drag torque of the free-running wheel has to be increased as shown in Eq. 4. Two major components contribute to the drag torque; the friction due to the lubricant (viscosity, oil depth) and the drag torque in the bearing of the free-running wheel. Within the experimental work, only the bearing drag torque was investigated. For this purpose, the radial clearance between free-running wheel and shaft (via the intermediate needle bearing) was reduced by 40% on all gears. No effect on rattle noise could be found. From the computational analysis it was concluded that to be effective, the drag torque must be increased to a certain extent so that condition (3) is no longer satisfied. If the drag torque increase remains below this limit, then the impact energy at the one tooth flank decreases, but increases at the opposite flank. Clearly the 40% reduction of the radial clearances did not increase the drag torque beyond the limit.

Angular Acceleration. Benefits can be obtained from reducing the angular acceleration acting on the free-running wheels. There are some possibilities for lowering the angular acceleration. First the output speed fluctuation of the engine depends on, among other things, the moment of inertia of the engine flywheel. The properties of the clutch disk also strongly influence the input of speed fluctuation into the transmission due to torsional resonances.[8] Finally, the transmission arrangement; i.e., its geometry and gear ratios, determines the "local" angular acceleration of each free-running wheel. The torsional resonance of the driveline system was found to be the key feature affecting drive rattle. Fig. 4a shows the driveline resonance at full load conditions. The speed fluctuation at the rattle threshold is plotted to illustrate the range of drive rattle (shaded area). Here the speed fluctuation is defined as a percentage ratio of the difference between the maximum and minimum instantaneous speed to the mean speed. Fig. 4a clearly illustrates the procedure for the elimination of drive rattle. The task is either to raise the rattle threshold above the maximum speed fluctuations or to lower the maximum speed fluctuations below the rattle threshold or to move both limits simultaneously. In Fig. 4a the rattle threshold is near the speed fluctuation of the flywheel, and the rattle condition disappears at higher speed due to the "vibration isolation" effect between flywheel and primary shaft. This occurs theoretically - for a simple mass-spring system - at frequencies greater than $\sqrt{2}$x resonant frequency. In Fig. 4b the effect of a lower speed fluctuation output from the engine is demonstrated by a 30% increase of the flywheel inertia. The result is a clear reduction of the speed fluctuation over the whole speed range, but particularly at the resonant speed. The possibility of moving the resonant speed out of the lower operating speed range was investigated by means of a soft clutch (with 27% less stiffness) as well as by a progressively stiff and highly damped clutch. Both clutch disks were tested in combination with the heavy flywheel giving the results in Fig. 4b. It transpired that it is impossible, within practical limits, to shift the resonant speed below the lowest operating range; i.e., to obtain an overcritical condition which would have been the best solution. On the other hand, the stiff clutch drastically reduces the speed fluctuation input to the transmission approaching the rattle threshold. However, the rattle condition now continues up to higher speeds, and the overall noise character becomes rougher in the higher speed range due to the high damping of the clutch, which weakens the vibration isolation between crankshaft and primary shaft in the mid-frequency range. The effect of these changes on the airborne noise radiated from the whole power of 1500 r/min. Since the frequency range above 1 kHz is most significant for gear rattle noise, the range of drive rattle is shaded only above 1kHz in this figure. It indicates the great benefit to noise reduction which is obtained when the torsional resonance is suppressed. By chance, at 1500 r/min engine speed, the speed fluctuation is the same for both the soft and the stiff clutch versions, so both noise spectra nearly

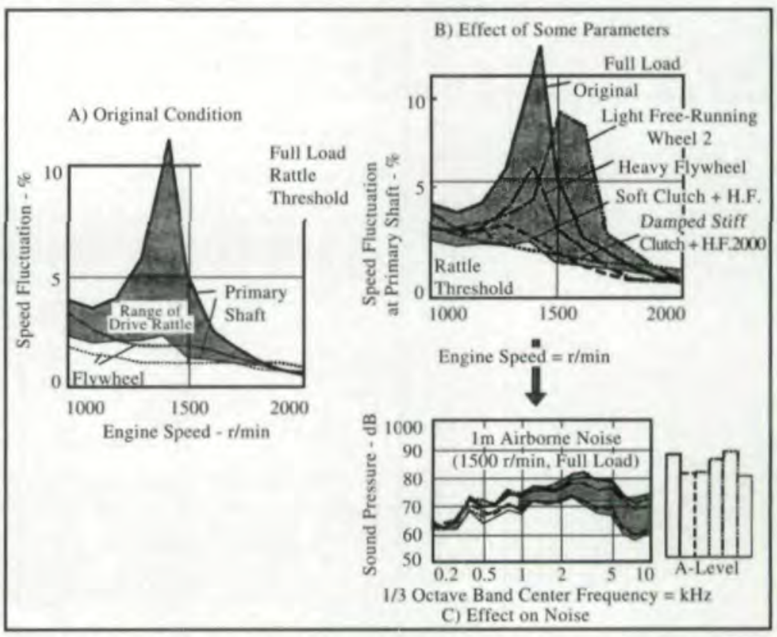

Fig. 4 - Characteristics of driveline torsional resonance (fourth gear).

coincide in Fig. 4c. The gear arrangement in the transmission was found to be disadvantageous with respect to gear rattle. As shown in Fig. 5, the original design (with the free-running wheels of third and fourth gear placed on the primary shaft) results in a very non-uniform distribution of the local angular acceleration. It favors gear rattle particularly at the free-running wheels Nos. 3 and 4, because of the high angular acceleration occurring at these wheels. The situation does not change even when the moments of inertia of the free-running wheels are taken into account (Fig. 5, right side). A re-design of the gear arrangement - placement of the free-running wheels Nos. 3 and 4 on the secondary shaft - would give a far better starting position for low gear rattle. A re-designed transmission was not available as hardware, and it could not be tested, but the benefits for low gear rattle noise in terms of a high rattle threshold are evident in Fig. 5 due to the more even distribution and much lower maximum values of the local angular acceleration.

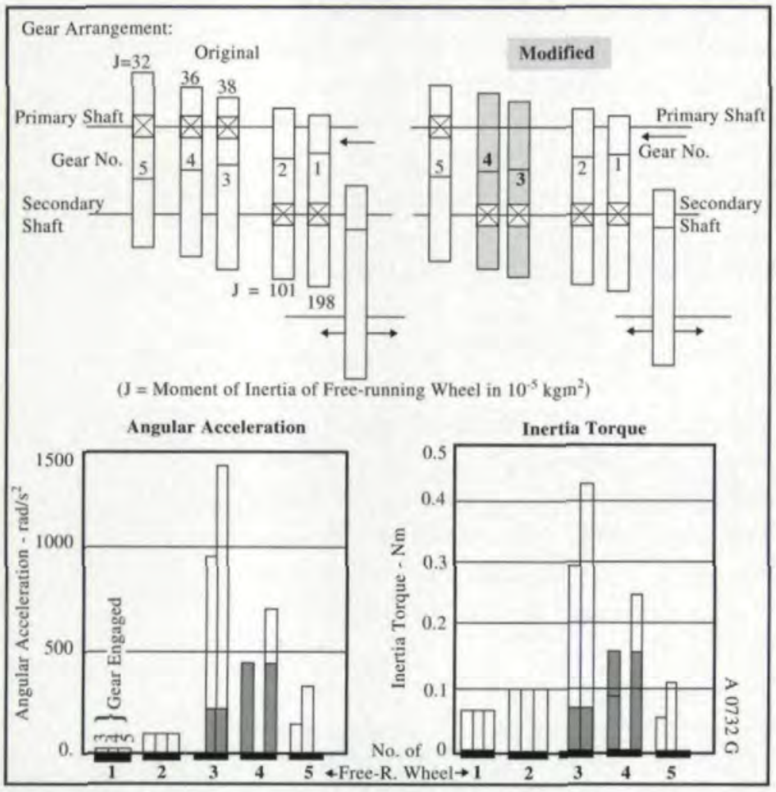

Fig 5 - Effect of gear arrangement on local rattle condition (assuming a speed fluctuation input of 0.5% at 1500 r/min).

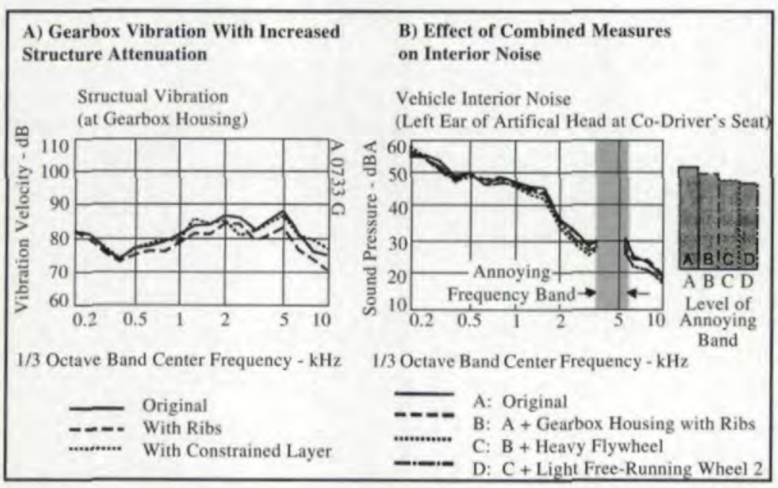

Fig. 6 - Effect of various improvements (road test, 1500 r/min, full load, fourth gear).

Inertia Of Free-Running Gear Wheel. There are three effects on gear rattle arising from the moment of inertia of the free-running wheels. The first and most important one is its influence on the rattle threshold. The rattle threshold is high if the inertia torque and, consequently, the inertia of each free-running wheel is low according to Eq. 4. The second effect is its influence on the characteristics of the tooth impact, and the third one is its effect on the driveline torsional resonance. To test these effects the free-running wheel of the second gear was selected, although the free-running wheels Nos. 3 and 4 were found to be the most prone to gear rattle (cf. Fig. 5). However, the wheels 3 and 4 could not be prepared as low inertia wheels because of their small size. Therefore, the larger wheel No. 2 was used for this test. Its original inertia was about three times greater than that of wheel Nos. 3 or 4 (cf. Fig. 5). Two samples of a low-inertia wheel No. 2 were prepared for experimental purposes. The first was extremely light with an inertia reduction of 38%. The second version, used for the final vehicle tests, was reduced in inertia by 15%. The effect of the extremely low-inertia wheel on the torsional driveline resonance can be seen in Fig. 4b, which shows a clear up-shift of the resonant speed. This speed shift, however, is not only caused by extremely light free-running wheel No. 2, but also to an equal extent by the removal of the sliding sleeve, which could not be installed together with the extremely light wheels for technical reasons. The combined effect (i.e., in combination with the heavy flywheel and damped stiff clutch) on noise is illustrated in Fig. 4c as the lowest spectrum defining the threshold of rattle noise. The computational simulation of a 50% reduction of the inertia of the free-running wheels confirmed the noise reducing effect as shown in Fig. 3b. Moreover it can be seen that the impacts decrease in magnitude and increase in number with a gain in its overall effect.

Vibration Transfer. Significant elements in the vibration transfer path from the mesh to the outer gearbox surface are the shaft bearings and the structure attenuation of the gearbox housing. In Ref. 3 the effect of the bearing design is described with a gain in low noise of up to 4 dB. Within the present work, the influence of the structure attenuation was investigated. From vibration measurements on the running transmission using a laser vibrometer, it was found that at frequencies above 2kHz the shaft vibration was scarcely higher than the vibration at the gearbox housing indicating a very low intermediate structure attenuation.

To determine the potential for noise reduction in terms of the structure attenuation of the gearbox housing two tests were carried out. First the walls of the gearbox housing were damped by means of a constrained layer. Secondly the walls were stiffened by ribs. Fig. 6a shows that the ribbing of the gearbox housing proved to be very effective. The reduction in vibration velocity reached up to 5 dB. The application of the constrained layer also improved the situation, but to a smaller extent. Finally the combined effect of several measures (by successive addition of single measures) was tested in the vehicle during road tests. As can be seen from the noise levels (in the frequency band significant for the rattle audibility in the interior nose) in Fig. 6b, the combination of improved gearbox housing (by ribs), increased flywheel inertia, and reduced inertia of free-running wheel No. 2 (second version) gave the best results, so that the gear rattle in the vehicle cabin was no longer audible.

Conclusions

In automotive transmissions drive rattle is excited by the angular acceleration arising from low frequency speed fluctuation caused by the fluctuating torque output of the combustion engine due to its gas and inertia forces. Work to eliminate gear rattle threshold - expressed by the critical speed fluctuation - above the actual maximum speed fluctuation. The rattle threshold depends upon the drag torque and the inertia of the free-running gear wheels, as well as on the local angular acceleration. Therefore, the following potential for improvement exists. The gear arrangement provide the opportunity to raise the rattle threshold. The proper lay-out of the gear arrangement at the design stage is a powerful tool to minimize gear rattle or at least to provide a good condition for later improvements. In this case, the major item to be taken into account is the wheelshaft arrangement. The great influence of the driveline resonance requires a careful treatment of the clutch and the distribution of inertias. Since the resonance can hardly be avoided, the solution will always be a compromise. Therefore much effort should be concentrated on accompanying measures, such as a low speed fluctuation output from the engine and a high rattle threshold of the transmission. Since the vibration transfer from the gear meshes to the outer surface has a great effect on the rattle noise emitted by a transmission, its structure attenuation has to be of a high level. There can be some potential for improvement by stiffening or damping the housing walls and thereby increasing the "audible" rattle threshold. ■

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