Low-tone planetary gears – feasibility study for gear noise reduction

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Introduction

Low-tone gears were formerly presented as *inequidistant gears* [1, 2]. However, with *low-tone gears* a more descriptive name was given to this new technology. Low-tone gears minimize tonal gear noise (*gear whine*) by rendering the teeth's meshing irregular by means of uneven toothing geometries, such as uneven tooth thicknesses and uneven tooth positions, for example. Furthermore, they are also capable of reducing the overall noise level caused by the meshing of the gear wheels. The principle idea is inspired by uneven blade positions in fan blowers, where uneven excitations are used to reduce tonal noise for decades. In recent years, the approach of low-tone gears, developed at the research group SAM at TU Darmstadt, was successfully applied to spur gears and helical gears [1, 2]. However, in many drive train applications, such as in automotive and robotic industry, planetary gears are preferred due to a high efficiency, a compact design, and a high power density. Therefore, this paper evaluates the noise reduction potential of low-tone planetary gears. The requirements for the design of low-tone planetary gears are discussed and exemplary sets of a low-tone planetary gear as well as an equivalent conventional planetary gear are designed. The excitation forces due to the variable gear mesh stiffness are calculated by means of numerical simulations. Results for the calculated airborne sound radiated from a simplified gearbox housing are presented. Finally, chances and risks of low-tone planetary gears regarding their practical application are discussed.

Design requirements for low-tone gears

In literature a variety of planetary gear types are known [3, 4]. In this paper only the basic setup of a planetary gear, consisting of a sun gear, a ring gear, and a carrier with a number of planet gears, is considered, see Fig. 1.

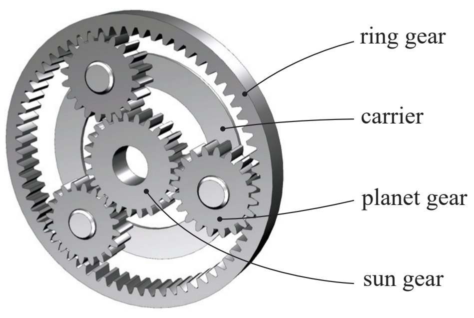

Figure 1: basic design of a conventional planetary gear

For basic planetary gears the tooth numbers of ring gear, sun gear, and planet gears must be in a defined ratio, so that the gear wheels fit together [3, 4]:

$$z_{\mathrm{r}} = z_{\mathrm{s}} + 2\, z_{\mathrm{p}}, \tag{1}$$

where z_{r}, z_{s}, and z_{p} are the tooth numbers of the ring gear, the sun gear, and the planet gears, respectively. For low-tone gears an additional constraint applies. Since their geometry is rendered uneven, the gear wheels need to be matched. In practice, low-tone gears are matched by sections called *uneven patterns*, as illustrated in Fig. 2.

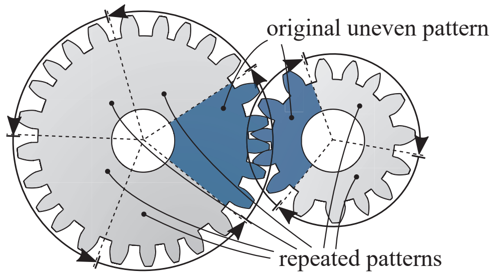

Figure 2: principle of uneven patterns for low-tone gears: original pattern (blue) and repeated patterns (gray)

An uneven pattern consists of a number of teeth Φ that are individually rendered uneven. The pattern repeats several times on a gear wheel. Hence, the design of low-tone planetary gears requires:

The tooth numbers of all gears of a low-tone planetary gear must have Φ as a common integer denominator.

Of course, the uneven patterns of two meshing gear wheels are perfectly matched in order to avoid any transmission error.

The possible number of equally spaced planet gears depend on the ratio of tooth numbers. For basic planetary gears literature states the condition [4]:

$$\frac{|z_{\mathrm{p}}| + |z_{\mathrm{s}}|}{n_{\mathrm{p}}} = int, \tag{2}$$

where n_{p} is the number of equally spaced planet gears and *int* is an integer number. The requirement for a common integer denominator Φ of all tooth numbers in a low-tone planetary gear applies as well. Of course, also unevenly spaced planet gears are possible, which leads to a complex acoustic excitation behavior that might have certain advantages. However, this will be subject to future research.

Due to the requirement of a common integer denominator in the tooth numbers of a low-tone planetary gear, the variety in possible gear ratios is somehow limited. However, since the uneven pattern length Φ and the specific uneven design of the teeth might be chosen individually for every application, a suitable trade-off between a great tonal noise reduction (large numbers of Φ) and the option for many gear ratios (small numbers of Φ) is possible.

Design of a low-tone planetary gear set

To evaluate the gear noise reduction potential of low-tone planetary gears, two sets of planetary spur gears are designed: one set with a state-of-the-art conventional design and one set with a low-tone design. All geometric parameters, except for the uneven tooth thicknesses and the uneven tooth positions of the low-tone planetary gear set, are held constant to allow for a direct comparison of acoustic performances. Table 1 shows the parameters of the planetary gear sets. The tooth numbers z are

Table 1: geometric parameters of the planetary gear sets

name	sun	planets	ring
profile shift coefficient x	–0.3	0.1	–0.3
tooth numbers z	25	20	–65
uneven pattern length Φ		5	
profile type		DIN 867	
modulus m_n		3 mm	
pressure angle α_n		20°	
axial distance a_w		67.5 mm	
face width b		20 mm	
root tip clearance c_P		0.3 mm	
helix angle β		0°	
number of planet gears n_p		3	

chosen to have a common integer denominator of $\Phi = 5$ teeth, thus five teeth on each gear wheel are rendered uneven. The uneven tooth thicknesses and uneven tooth positions of the sun wheel are chosen randomly. The uneven patterns of all other gears are derived from this to perfectly match without any transmission error, as described in previous works [1, 2]. The gear ratios for the chosen design are:

- $i_{s,c} = 3.6$ from sun to carrier for a fixed ring,
- $i_{c,r} = 0.7\overline{2}$ from carrier tor ring for a fixed sun, and
- $i_{s,r} = -2.6$ from sun to ring for a fixed carrier.

The transmission ratio $i_{s,r}$ with a fixed carrier is called *stationary transmission ratio*. The research presented in this paper is going to focus on this *stationary setup*. Fig. 3 shows the conventional planetary gear set and the low-tone planetary gear set designed for the investigations presented in this paper.

Simulation of excitation forces

It is well known from literature that the dominating excitation mechanisms for gear noise are the time-varying gear mesh stiffness and geometric deviations [3, 4]. In this paper geometric deviations (e.g., from manufacturing deviations and local defects) are neglected since they

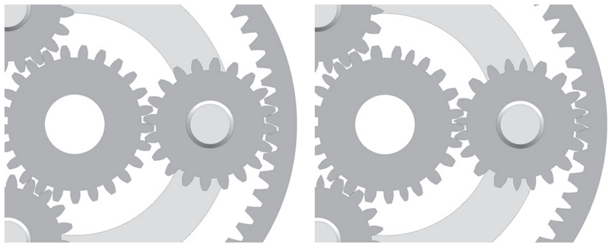

Figure 3: conventional planetary gear set (left) and low-tone planetary gear set with uneven tooth positions and uneven tooth thicknesses (right)

are very difficult to obtain from simulations. Hence, the time-varying gear mesh stiffness is the only excitation mechanism considered. Numerical simulations are performed to obtain the gear mesh stiffnesses between all meshing gears. Hence, three gear mesh stiffnesses are calculated for the mesh of the sun wheel with the three planet wheels at a torque of 100 N m at the sun wheel that is distributed equally to the planets ($33.\overline{3}$ N m each). Another three simulations are performed to obtain the gear mesh stiffnesses between the three planet wheels and the ring wheel with an equivalent torque at the planet wheels of $-26.\overline{6}$ N m each. Quasi-static numerical simulations are performed with the driving wheel being loaded with the defined torque and the driven wheel rotating with a defined rotational speed. The absolute difference between the ideal, undeformed rotational angle and the displaced rotational angle of the driving wheel is the relative angular displacement $\Delta\varphi$. HOOKE's law is used to calculate the gear mesh stiffness c_z as follows:

$$c_z = \frac{T_{load}}{\Delta\varphi \left(\frac{d_b}{2}\right)^2}, \tag{3}$$

where T_{load} is the torque at the driving wheel and d_b is the base diameter of the driving wheel. Fig. 4 shows the gear mesh stiffnesses between the sun wheel and the three planet wheels.

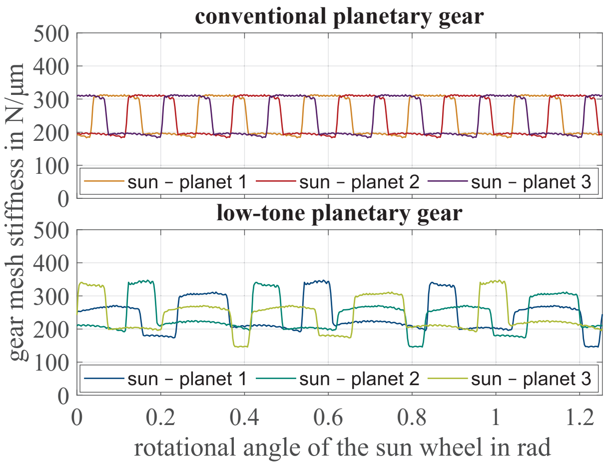

Figure 4: gear mesh stiffnesses between the sun wheel and the three planet wheels for the conventional planetary gear set (top) and the low-tone planetary gear set (bottom), obtained from numerical simulations, $T_{load} = 100$ N m at the sun wheel

The gear mesh stiffnesses obtained from the numerical simulation show a slightly noisy behavior due to the finite element size of the numerical mesh and the very small elastic displacements. However, it is assumed that this will have a negligible effect on the calculation of the airborne sound in the further steps. The gear mesh stiffnesses of the conventional planetary gear set is very regular in both amplitude and temporal behavior, whereas those of the low-tone planetary gear set shows a strongly uneven behavior. This already indicates a less periodic excitation and, therefore, a less tonal noise. Furthermore a phase shift in the shape of the gear mesh stiffnesses of the different gear pairs is visible. Fig. 5 shows the gear mesh stiffnesses between the three planet wheels and the ring wheel. Qualitatively the same observations apply as

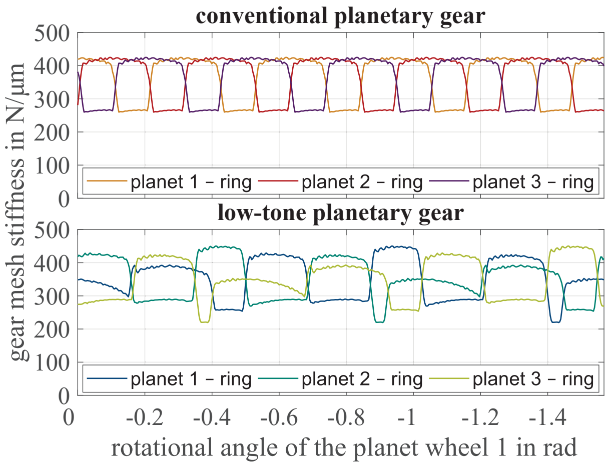

Figure 5: gear mesh stiffnesses between the three planet wheels and the ring wheel for the conventional planetary gear set (top) and the low-tone planetary gear set (bottom), obtained from numerical simulations, $T_{\text{load}} = -26.\overline{6}\,\text{N m}$ at each planet wheel

for the gear mesh stiffnesses in Fig. 4. However, quantitatively the average value is higher for the gear mesh stiffnesses between the planet wheels and the ring wheel due to thicker and, therefore, the stiffer teeth at the ring wheel.

The excitation forces at the gear wheels are calculated from the *force excitation approach* for gear wheels, as known from literature [3, 5]. This approach assumes that in dynamic operation, the gear wheels rotate at such a high speed that no relative motion between the gear wheels occurs. Therefore, the dynamic excitation force F_{z} may be calculated directly from the static torque at the gear wheel T_{load} and the static, average displacement $\overline{x}_{\text{static}}$ that, again, can be calculated from the static load T_{load} and the averaged gear mesh stiffness $\overline{c}_{\text{z}}$:

$$F_{\text{z}} = \frac{c_{\text{z}}}{\overline{c}_{\text{z}}} \frac{2\,T_{\text{load}}}{d_{\text{b}}}. \tag{4}$$

The forces that work on the individual gear wheels are shown in the mechanical model of the planetary gear wheel in Fig. 6. The model is based on the dynamic model of a planetary gear by KAHRAMAN [6]. However, compared to KAHRAMAN's model, it is assumed that all

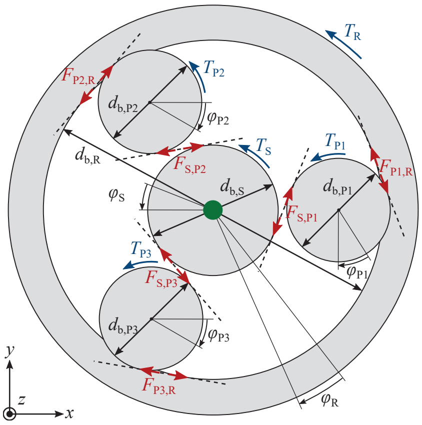

Figure 6: model for the calculation of excitation forces (red) with torque directions (blue), rotational directions (black), lines of action (dashed lines) and the center point with the bearings (green dot)

bearings and the carrier are ideally stiff. Since planetary spur gears are investigated in this paper, only forces in the x-y-plane are considered. The forces at the gear wheels calculated from Eq. (4) are individually transformed to the planetary gear's center point (see green dot in Fig. 6). Since the carrier is assumed to be rigid, the planet wheels' forces are also transformed to the planetary gear's center. Hence, three resulting excitation forces work at the planetary gear's bearings: that from the sun wheel $F_{\text{z,S}}$, that from the ring wheel $F_{\text{z,R}}$, and that from the carrier $F_{\text{z,C}}$ (resulting from the transformation of all planet wheels' excitation forces $F_{\text{z,P1}}$, $F_{\text{z,P2}}$, and $F_{\text{z,P3}}$ to the center point).

Modeling of sound radiation

A simplified gearbox housing for the planetary gear sets is designed, see Fig. 7. The gearbox housing's outer di-

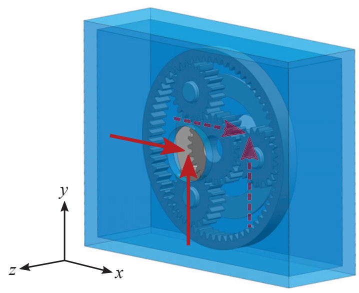

Figure 7: simplified gearbox housing (blue) with force excitations at bearing seats in x and y direction (red arrows)

mensions are 320 mm in length, 250 mm in height, and 100 mm in depth with a wall thickness of 10 mm. Steel is chosen as a material. The three excitation forces from the planetary gear set $F_{\text{z,S}}$, $F_{\text{z,R}}$, and $F_{\text{z,C}}$ work on the bearing seats of the gearbox housing (red arrows in Fig. 7). The fundamental equation of machine acoustics is used to calculate the radiated airborne sound, as described by KOLLMANN [7], for example. The structure-borne sound efficiency for the gearbox housing is calculated by means of numerical simulations. For the radiation efficiency, the model of a monopole is used. The resulting sound power is converted to the time signal of the sound pressure at 1 m distance from the gearbox, assuming free-field conditions.

Simulation results

Figure 8 shows the Campbell diagrams for the sound pressure level excited by the conventional and the low-tone planetary gear sets for rotational speeds from 0 rpm to 2000 rpm and for a load of 100 N m at the sun wheel.

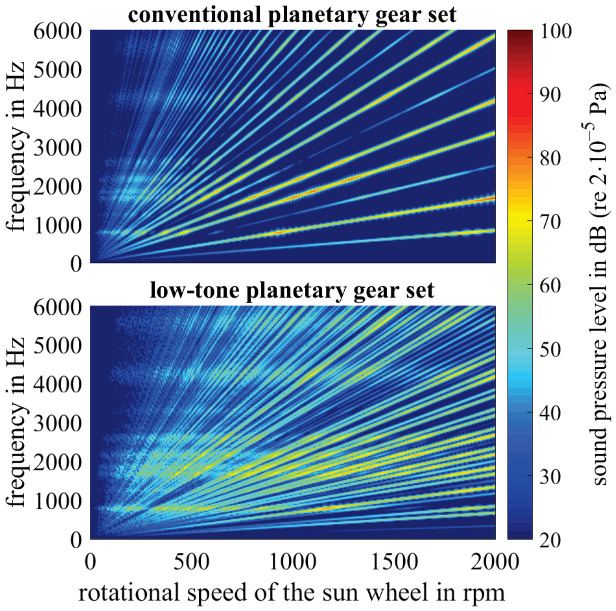

Figure 8: Campbell diagrams for the conventional and the low-tone planetary gear sets; 0 – 2000 rpm with T_{load} = 100 N m at the sun wheel

For the conventional planetary gear set only few orders are excited but with a high level each. For the low-tone planetary gear set five times more orders are excited due to the uneven pattern length of $\Phi = 5$. Each order excites the sound pressure with a much lesser amplitude. Hence, it can be confirmed that low-tone planetary gears decrease the amplitude of tonal noise. Fig. 9 shows the total sound pressure level for both planetary gear sets for rotational speeds from 0 rpm to 2000 rpm and for a load of 100 N m at the sun wheel. For many rotational speeds

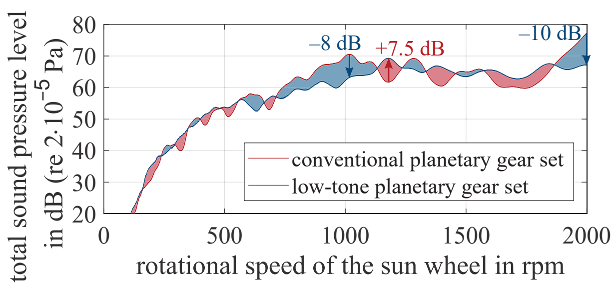

Figure 9: SPL spectra for the conventional and the low-tone planetary gear set; 0 – 2000 rpm with T_{load} = 100 N m at the sun wheel

the total sound pressure level for both planetary gear sets is approximately equal. However, even though for some rotational speeds the total sound pressure level increases for the low-tone planetary gear set (red areas; e.g., 1200 rpm), Fig. 9 shows a decreased total sound pressure for many rotational speeds (blue areas). For 1000 rpm and 2000 rpm the total sound pressure level is decreased by −8 dB and −10 dB, respectively. This shows that low-tone planetary gears are capable of not only reducing tonal noise but also of reducing the total sound pressure level.

Conclusions

Previous works showed that low-tone spur gears and helical gears can be used to considerably reduce gear noise regarding the tonal character (reduction of *gear whine*) and the total sound pressure level. Results from this paper show that these findings are valid for low-tone planetary gears as well. However, manufacturing conventional planetary gears is already challenging. Manufacturing low-tone planetary gears might be slightly more challenging because the correct relative positioning of the gears needs to be ensured due to the uneven toothing geometry. Whether and to what extend the wear increases and the load capacity decreases for low-tone planetary gears due to individual thinner teeth is not investigated yet. These questions will be addressed in ongoing works at the research group SAM at TU Darmstadt. Furthermore, for low-tone planetary gears also other setups than the *stationary setup* will be studied. Optimization methods for the uneven pattern design of low-tone planetary gears will be evaluated to improve the acoustic performance compared to the randomized design approach presented in this paper. Finally, the noise reduction potential of unevenly spaced planets in low-tone planetary gears will be investigated.

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