Designing Powder Metal Gears

In this article Hoganas demonstrates some design aspects that can—and need to—be considered when working with powder metal gears.

By Dr. Anders Flodin, Prof. Dr.-Ing. Christian Brecher, Dipl.-Ing. Christof Gorgels, Dipl.-Ing. Dipl.-Wirt. Ing Tobias Röthlingshöfer, and Dipl.-Ing Jannik Henser
INTRODUCTION
Powder metal gears have been available for quite some time, initially only in light duty applications such as toys and power tools. As material quality and processes have improved, so has the confidence in PM. Today PM gears are present in many engine platforms, and certain car makers use nothing but PM gears for the cam shaft synchronization. Inroads are made to diesel engine gear trains, both for commercial vehicles such as heavy trucks and high end cars, but also for lighter duty vehicles such as ATVs and motorcycles. The reason for this is lower total cost and improved performance.

The reluctance from the automotive industry to fully embrace PM gears comes from the fact that PM gears in the sintered state fail before the wrought steel gears during endurance testing in rigs. Part of this is due to design sub-optimization, and part is due to the porosity of the material. With the correct design, more applications that now are regarded as too heavily loaded could benefit from the PM technology.

NVH is another factor that a gear designer has to tackle during design. For PM gears, the problem is twofold. First the material, density, grain size, and structure plays a role, and second is that the gear design has to be altered. The crowning in radial and axial direction, $C_a$ and $C_s$, will be different as well as the angle deviations $fha$ and $fhß$. There are also possibilities with PM to design lightweight gears by thinning the web and introducing design features that removes material where it is not needed.

This paper deals with these issues and maps out some of the differences the gear designer needs to investigate when designing gears for powder metal, as well as giving some practical examples.

GEAR MODELING: MICRO GEOMETRY
The software Zako3D from WZL in Aachen was used to investigate the working behavior during meshing for different gear designs [1]. The input to Zako3D is the geometric data of the flank and displacement influence coefficients, which can be derived from an FE-Analysis of a gear sec-
For the consideration of PM properties, including surface densification, the Young's modulus and Poisson's ratio can be adjusted in the FE model.

The tooth flank is modeled as a 21 by 21 grid. During the tooth contact analysis the contact distances between the flanks are calculated, and by considering the influence coefficients the forces in each node are calculated. Varying density from material boosting processes such as surface densification can also be modeled. This method has been checked against ISO 6336 and correlated well with the results from that calculation. The grid density has also been increased, but with negligible difference of the results. To avoid excessive calculation times, the grid density has not been increased in this work.

By varying angular deviations $f_{ha}$ and $f_{ha}$ and the crowning parameters $C_a$ and $C_b$, and studying the effect they have on contact stress and transmission error, some interesting differences could be noted.

Fig. 2: Design parameters that may be changed at no cost to reduce weight.

Fig. 3: Schematics of experimental setup for measuring the internal damping of the noise. The two high-pass filters are there to improve quality of the signal.
between PM and wrought steel gears that will be presented further on in this paper.

**GEAR MODELING: MACRO GEOMETRY**

With powder metal manufacturing technology it is possible during design to remove material on a gear wheel where it is not needed at no additional manufacturing cost. The high stresses dimensioning the gear are found in and under the contact zone on the flank and in the root area. The web, rim, and hub can be modified for low weight and low inertia saving fuel.

To investigate the possibilities, the gear train from a Smart Fortwo was modeled in Solid Works and loaded to max torque for each gear pair to evaluate the stresses and deformations. Then material was removed in the web to reduce weight while maintaining deformations and stresses (fig. 1a).

The design changes that are possible are a reduction of rim (R), web (W), and hub (H) as well as holes (O) in the web, chamfers (C), and arcs. In this investigation we studied changes in web thickness (W) and introduction of holes (O), as seen in fig. 2. Other features are possible, such as design for assembly features, but that is outside the scope of this paper. The point is that it is possible to do near net shape manufacturing, thus saving cost.

**NVH AND MATERIAL DAMPING**

Sandner et al. [2] have reported that PM gears in an engine application have been less noisy than the original design.
There are only speculations as to why. So an experiment was conducted at the Marcus Wallenberg Laboratory in Stockholm, where test bars of different density were excited with an impact hammer and the sound pressure measured (fig. 3). The reverberation time was measured and the damping was calculated according to equation 1 and 2.

\[ T_{\text{rev},60} = 60 \text{dB} \times \frac{\Delta t}{\Delta \text{dB}} \quad (1) \]

Definition of Reverberation time \( T_{\text{rev},60} \): Time until the sound pressure level is reduced by 60dB

The damping is then calculated as:

\[ \zeta = 6 \cdot \ln10 \cdot 100 \left( \frac{\omega}{\omega T_{\text{rev},60}} \right) \quad (2) \]

Where \( \omega \) is the eigen angular frequency of the individual test specimen.

**MACRO GEOMETRY RESULTS**

The hypothesis investigated was that a reduced weight of the gear train would reduce losses from spinning up the total gear mass for every acceleration of the vehicle. For this particular gearbox some of the gears were difficult to convert to PM since they were cut directly on the shaft, and thus left without redesign. The third, fourth, and fifth gear pair could be converted as well as the reverse idler and the first gear. Figure 4 shows an example of the differences between stock geometry and the PM geometry. The weight difference is 17 percent, or 150 gram for this particular gear, and the cost reduction resulting from less usage of PM material would be around 0.20-0.25 Euro per gear.

This type of design was employed for the suitable gears mentioned above. Only two gears were suitable for lightening holes, and the rest has a thinner web. In table 1 only the thin web design is accounted for, since that was judged as a safer design.
From table 1 it can be seen that inertia and weight can be reduced by 10-20 percent, even without introducing holes in the web. Maximum displacement is notably higher, up to 27 percent, and this is partly due to the lower Young’s modulus rather than the redesign and is found at the gear tip. That is why it is important to employ the right micro-geometric changes discussed earlier. However, attention should not be paid to the exact figures of the deformation since it is also partly an artifact from applying a load on nodes in the FEM program.

If a fictive acceleration is made with the gearbox (as depicted in fig. 1a), according to table 2 the corresponding energy needed to accelerate the gear trains in steel and PM will be 1954 J and 1836 J, respectively. The PM gear train in this particular example from the Smart car, where all gears have not been converted, still reduces inertia losses with 6 percent. Figure 5 shows the energy difference plot between steel gears and PM gears for the Smart Fortwo-08.

**NVH/MATERIAL DAMPING RESULTS**

Three different densities on the test bars were investigated together with a reference test bar made of solid steel. The chemical composition was not identical between the PM test bars and the solid steel bar. However, the manufacturing and geometry was. The test bars were excited and the sound pressure measured according to fig. 3, and the damping coefficients were obtained through equations 1 and 2.

Figure 6 shows the result with a clear influence of the density on the damping. However, the solid steel reference was found to dampen the excitation like the 7.0

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**Fig. 8:**
Pressure map through mesh cycle with variation in fha. Tolerance box with best minimum pressure found in map and corresponding max pressure within tolerance box.
density PM test bar. Obviously, there are more factors that come into play. Due to intergranular friction in the grain boundaries, grain size—and thus the length of grain boundaries—will play an important role and the metallographic analysis showed that the solid steel test bar had significantly finer grains compared to the PM test bars. In [2] the gears are case hardened leading to a similar grain size in both materials, which partly helps to explain the lower noise (fig. 7). There are several physical effects that affect hysteresis in steel, however, and different manufacturing methods and geometric gear quality will also play a significant role on the noise generated and emitted in the testing in [2].
There are several design parameters that will influence the working behavior of a gear, but in this particular study only $C_a$ and $C_{\alpha}$ have been used to optimize and minimize the influence of manufacturing errors $f_{ha}$ and $f_{ha,\gamma}$. The PM gears will also be compared to a similarly optimized solid steel gear. The gear pair studied is the fourth gear pair in the Smart gearbox (gear data is given in appendix 1).

In fig. 8, the lowest surface pressure is depicted with an X. This would be the optimal point and would give a desired $f_{ha,1}$ of 30 µm and a $f_{ha,2}$ of 30 µm as well. However, it is impossible to hit those values in production. There will be a variation in tolerances when manufacturing, represented by the white dashed box, and from which can be seen that the maximum pressure the gear flanks will be subjected to is 2461 MPa, which is too high. This is not a robust design, so another area of the map has to be chosen where pressures that may arise from manufacturing errors are within allowable limits for the material.

Figure 9 shows the surface pressures when a more robust design point is chosen. Maximum pressure is significantly reduced, and the minimum pressure only marginally increased compared to fig. 8. This type of iterative search for a good tradeoff between low pressure and robust design was performed for variations of $f_{ha,\gamma}$ as well, and then crowning parameters $C_a$ and $C_{\alpha}$ were introduced in order to reduce stress levels further. The result is summarized in fig. 10.

Optimization in gear design is a tradeoff between different parameters; there are more than one set of parameters that will

<table>
<thead>
<tr>
<th>Gear</th>
<th>Stress (Pa)</th>
<th>% Diff</th>
<th>Displ (mm)</th>
<th>% Diff</th>
<th>Mass (g)</th>
<th>% Diff</th>
<th>$I (g/mm^2)$</th>
<th>% Diff</th>
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<tr>
<td>4 Pin</td>
<td>3.8E+08</td>
<td>4</td>
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<td>200024</td>
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<td>1.6E-02</td>
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<td>12</td>
<td>399691</td>
<td>11</td>
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<td>17</td>
<td>65833</td>
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</table>
Fig. 11: Mitsubishi EVO 9 rally car, 300 HP, 600Nm torque equipped with 5 speed manual dog-box. 4:th gear and pinion made in PM.

give similar values for transmission error and stresses. The trend in fig. 10 is lower surface stresses, higher transmission error, and lower spread for the PM gears. The transmission errors are defined as the amplitude (peak to peak) and not zero to peak.

Also worth mentioning when discussing the optimization of gear geometry is the root radius. Since PM manufacturing is not dependent on a hob that cuts the gear and the following kinematic relations defining the root radius, a stress-optimized root can be designed and manufactured as well. The methodology has been described, for instance, by Kapelevich [4], and it holds true for PM gears as well.

**PRACTICAL APPLICATIONS**

The car in fig. 1b has been equipped with PM gears. The gears are copies of the original design and gear pairs 3, 4, 5, and the reverse idler gear have been made by PM and case hardened plus ground to final tolerances. At the time of writing the fourth gears have passed 35,000,000 revolutions without sign of any damage or measurable wear. Density is 7.25 g/cm³ and the material is a standard powder alloyed with 0.85 percent Mo and 0.25 percent C.

In order to showcase the durability of PM, a Mitsubishi EVO 9 rally car has been equipped with PM gears and competes successfully nationally and internationally in Group N (fig. 11).

<table>
<thead>
<tr>
<th>MPH</th>
<th>Time(s)</th>
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<tbody>
<tr>
<td>0-27</td>
<td>4,3</td>
</tr>
<tr>
<td>27-46</td>
<td>9,7</td>
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<tr>
<td>46-71</td>
<td>22,4</td>
</tr>
</tbody>
</table>

Tab. 2: Acceleration times and speeds gear 1-3 from [3].
CONCLUSIONS

This paper has demonstrated some design aspects that can and need to be considered when working with PM gears. Advantages of lightweight design and its immediate implications on performance have been modeled. Inertia losses in the gearbox can easily be reduced 6-7 percent, with the possibility to reduce costs at the same time. It has also been shown that the damping is clearly affected by the density of the material and is a feature that can be used in order to reduce noise emission from the gears.

PM gears also appear as fault tolerant and suffer less from variations in manufacturing quality. PM’s lower Young’s modulus gives a slightly higher Peak to Peak Transmission error but also help to reduce surface stress. Further work is underway to compare the optimized PM gears with optimized wrought steel gears with respect to NVH and inertia losses by rig testing at WZL in Aachen.

REFERENCES:

3) www.caranddriver.com/var/ezflow_site/storage/original/application/0866ce87d6af4f956f6b783b9e6846f.pdf

Appendix 1:

Gear data in micro design study

\[ \frac{m_n}{z_2} = 1.7 \text{ mm} \]
\[ \frac{z_1}{z_2} = \frac{35}{33} \]
\[ \frac{b_1}{b_2} = \frac{12.7 \text{ mm}}{12.1 \text{ mm}} \]
\[ \beta = 30^\circ 15' \]
\[ a = 68 \text{ mm} \]
\[ a = 16^\circ \]
Core Density: 7.2 g/cm³