Automotive transmission design using full potential of powder metal

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Abstract
The design of Powder Metal (PM) gears can not be copied from the steel gear that it may be replacing in an automotive transmission. This paper deals with the implications of copying the micro geometry of the steel gear when rebuilding a 6 speed manual transmission for an Opel Insignia 4 cylinder turbocharged 2 litre engine delivering 220hp/320Nm. It will be showed that a different micro geometry of the PM gear teeth coupled with the lower Young's modulus can theoretically enhance performance compared to the solid steel design. A thorough reverse engineering and redesign has been required in order to understand and map the performance of the Solid steel versus the PM gears and the findings from that work will be discussed in this paper. Weight reductions and inertia reductions will be demonstrated with maintained performance levels to put light on some of the advantages with PM technology.

Keywords; Gear, Design, System design, Powder Metal

Introduction
Höganäs AB has demonstrated through its demo cars and design work that PM gear technology has the ability to replace gears in automotive transmissions without sacrificing performance. PM gear technology has also the inherent capability to reduce the weight and inertia of the gear wheel thus reducing mass and energy losses. Another important aspect of lowering the inertia of the gears is to simplify energy dissipation in the synchronization mechanism in manual gear boxes as well as in AMT and DCT type transmissions.

When designing PM gears special attention has to be paid to the use of the correct material properties, meaning Young's modulus and Poisson's ratio. The designers can also improve weight and dynamics by understanding the possibilities that PM offers through its unique production methods. The PM process route also offers direct reduction of the number of manufacturing steps leading to improved cost performance.

Young's modulus and Poisson's ratio can be empirically calculated as a function of density, see Eq. 1 and 2. from [1].

$$E = E_0 \cdot \left(\frac{\rho}{\rho_0} \right)^{3.4} \qquad (1)$$

$$\upsilon = \left(\frac{\rho}{\rho_0} \right)^{0.16} \cdot (1 + \upsilon_0) - 1 \qquad (2)$$

Methodology; system analysis
In order to investigate how much different the micro gear design has to be in relation to solid steel design and also what possibilities for weight reduction that exists a redesign of a GM gearbox was performed. The chosen transmission was a six speed manual transmission rated for 320Nm named "M32". This transmission is used for certain models of Opel Insignia as well as other GM cars. Another aim of this work was to understand how much load that the PM gears have to sustain and from that judge the manufacturing process necessary to fulfill the stress criteria.

The above mentioned transmission was purchased and disassembled while recording pull off forces of gears and bearings as well as measuring axial play in the system. The housing was scanned and imported into FE software see fig. 1. Shafts and gears where measured, modeled and assembled into the housing. An essential part of a system analysis is the bearing stiffness. The bearing representation in this system model is reduced to define the stiffness between two nodes, representing the inner ring and the outer ring, due to the fact that this bearing stiffness is strongly nonlinear and depends on both bearing design and load direction/magnitude.

Simplified modelling techniques were used for bolts, roller bearing contact between gears and shaft as well as gear to gear contacts used in the ***system*** **analysis** where the focus is on deformation of housing, shafts and bearings. This was done in order to save calculation time. The information from the system analysis is then carried over to the **gear analysis**.

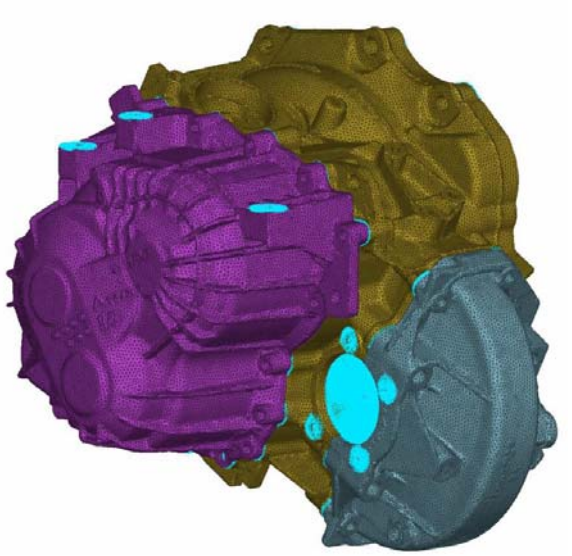


Fig. 1. Scanned and digitized housing

The output from the system analysis is gear misalignment and transmission deflections, which are used as an input to the gear analysis where the micro geometry is tweaked to give the best working behaviour of the gears considering the misalignment and bending from shafts and bearings.

Methodology; gear analysis
The 6-speed transmission was completely dismantled. All parts where then measured and reverse engineered to find current production data for all gears, shafts and housing. Macro geometry of the gears was created with focus on surface stress levels and peak-to-peak transmission error. For 1st, reverse and 2nd the driver member could not be exchanged since they were cut directly on shaft so for these parts, only modification of idler and driven members was performed. The final drive is a straight carry-over.

Modifying the micro geometry of the gears is iterative procedure using the material data, loads and misalignments with the focus on lowering transmission error and contact stresses. This is done by changing the gear design parameters in the iterations, such as crowning, reliefs angular deviations etc.

A duty cycle “typical European consumer usage” based on author’s experience, has been used to evaluate gear life.

Misalignment data is taken from the system analysis above and has been accounted for in the micro geometry of the tooth flanks. Abuse load is 6500Nm on differential cage, based on author’s experience and vehicle data.

The working behaviour of the gears in the system has been modelled for 50%, 100%, 150% and 200% load and different temperatures in order to assure the functionality under different conditions.

All parts are modelled using linear-elastic material properties. Material properties are based on input from Höganäs AB, see table 1.

Table 1. Material data for PM

Material	**Elastic modulus (GPa)**	**Poisson’s ratio**	**Thermal expansion ($^{\circ}C^{-1}$)**	**Fatigue limit, surface (MPa)**	**Fatigue limit, root (MPa)**
Powder metal	160	0.28	$12.5 \cdot 10^{-6}$	1100@$5 \cdot 10^{7}$ Cycles	650@10^{7} Cycles

Several different softwares have been iteratively used to carry out the analysis of the different components and system.

Results
In this paper a few general results are presented since a complete walk through of all the results are beyond the scope of this paper.

A parameter that describes the quality of the mesh cycle of two flanks is the peak to peak transmission error (TE). Transmission error is also to some extent related to the noise of the gears and is generally kept as low as possible. When working with a material with a lower Young’s modulus compared to steel, the TE tends to increase if the geometry is copied from steel [2]. This can be designed away to some extent and improve the working behavior of the PM design. Figure 2. shows the maximum TE for 3 different gear designs during a torque sweep. It is the first gear pair in the transmission and used for taking of from standstill

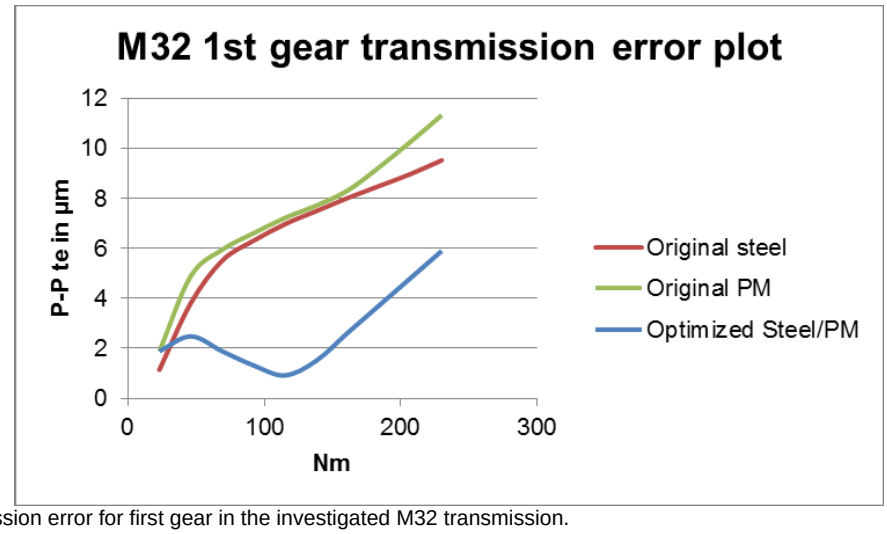


Figure 2. Transmission error for first gear in the investigated M32 transmission.

The first observation is that the TE is quite high. Since this is the first gear, it is only used for taking off and a slightly higher TE can be allowed. More important are the curves; green curve is the PM gear with the steel flank design. The green curve is higher for all torques indicating that the TE will be higher for the copied PM gear which is undesirable. The result from design iterations in order to improve the TE for the PM gear is shown in the blue curve where TE is lower for every torque level and is likely to perform significantly better than the PM gear with the copied design (green curve).

This pattern with an underperforming copied PM gear can be seen for all gears in the transmission. It will not always be better than the steel gear, like in figure 1, but a gear designed for PM will always be an improved design compared to PM gear with the copied steel design.

In table 3 is the contact and bending stress listed for 6:th gear pair in original steel and redesigned PM.

Table 3. Stress comparison

		6th steel		6th PM		Diff	
Bending stress	MPa	564	624	616	677	8,4%	7,8%
Contact stress	MPa	1504		1285		-17,0%	

The 6:th gear was chosen to be presented since the result displays a typical improvement number, -17%, in contact stress and it is a good example of a gear suitable for PM from a performance point of view. Worth noticing is that the bending stress is intentionally increased for the PM gears to be able to design a lower contact stress for the same. Gear design is an iterative tradeoff process, for the 6:th gear pair it was judged to be more beneficial with a lower contact stress and the sacrifice was increased root stress. The root stress can be further reduced with PM technology using the optimization procedure in [3].

The durability of 6:th gear pair is presented in figure 3 below where the duty cycle is taken into account. The red, blue and black lines are S-n curves for sintered and casehardened Astaloy85Mo PM gears with a density of 7.25g/cc and a tolerance class of ISO 7 or better. What can be read from the diagram is that the tooth root bending fatigue is within the acceptable boundaries but he contact stress is still a bit too high. So these gears would need a little higher performance to qualify. The remedy in this case could be increased density to 7.4g/cc by double pressing and double sintering or by shifting the material to a higher performing one. Shot peening to induce higher compressive stresses and/or superfinishing could be other cost efficient methods to increase the fatigue limit another 7% that is necessary to qualify. But without redesign a 25% performance increase (1200MPa to 1500MPa) would have been necessary and that would call for significantly more expensive processes jeopardizing the cost efficiency of PM.

For this particular transmission reverse, 3:rd and 4:th gear pair can be made with the shortest possible manufacturing chain that gives 7.25 density. For the 5:th and the 6:th gear pair some of the above mentioned processes would be necessary to increase performance. First and second gear pair needs densification or more radical redesign with asymmetric gear teeth or non involute gear shape.

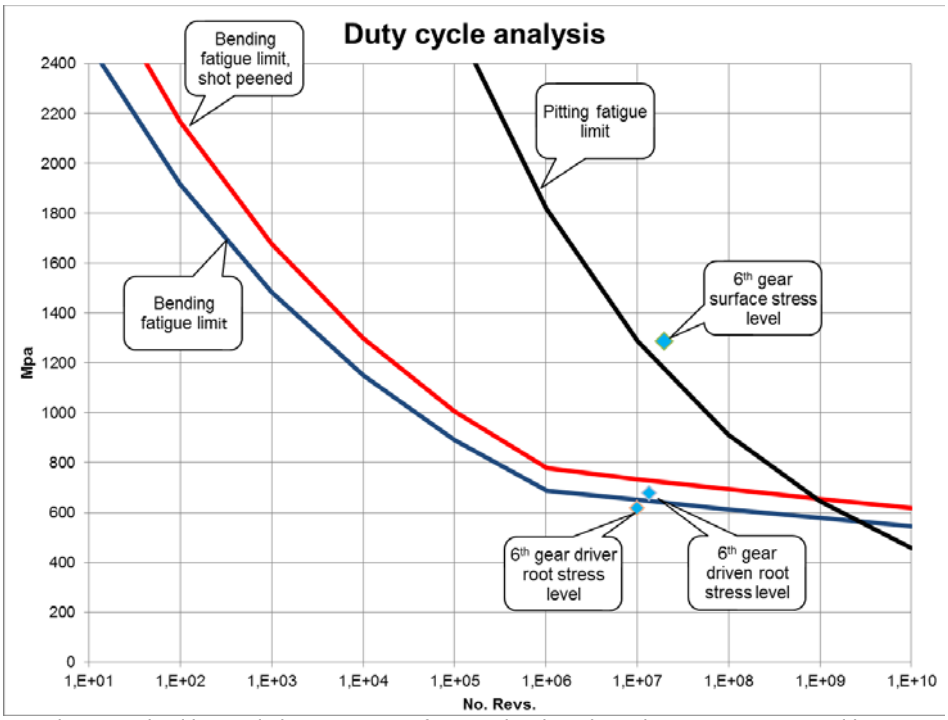


Figure 3. Loads on 6:th gear pair with correlating S-n curves for case hardened Astaloy85Mo PM gears with ISO 7 or better tolerances.

The redesign does not only take micro geometry into account but also macro geometry in order to reduce weight and inertia. Weight and inertia reductions are very important since that reduces material cost and it lowers the weight of the transmission. Inertia reduction is also of great importance since that reduces the losses from the accelerating the gear mass every time the RPM is shifted. A reduced inertia also decreases the amount of heat that has to be dissipated in the synchronization of the gears. Less heat build up will give a more robust synchronization system and longer service life. The reduced energy that has to be turned into friction may also be used to design a simpler and smaller synchronization package reducing overall dimensions or the transmission. Table 4 below summarizes the weight and inertia reduction for the driven gears.

Table 4. Weight and inertia reduction for the redesigned transmission.

	Inertia M32 Steel vs Sinter						
	Inertia Steel	Inertia Sinter			Mass (kg)		
	M32	Copied PM	Optimized PM	Diff	Steel M32	Sinter	Diff
1	2154	1769	1670	22%	1,097	0,896	18%
2	1285	1114	1090	15%	0,953	0,819	14%
3	1991	1605	1532	23%	1,159	0,93	20%
4	983	860	848	14%	0,831	0,73	12%
5	244	224	224	8%	0,323	0,297	8%
6	213	196	196	8%	0,387	0,355	8%
R	1336	1140	1109	17%	0,946	0,791	16%

The redesign will in total for this particular transmission remove 1.1 kg of mass.

Future work

The next step will be to redesign the first and second gear pair using more advanced design philosophies such as non involute gearing and asymmetric gear teeth to be able to prototype the gear box without using any performance enhancing technologies such as hot isostatic pressing (HIP) or other densification technologies. There are a few unknown factors when moving away from the traditional involute curve shape. It is not impossible at all to reduce the contact and bending stress, the difficulty lies when the transmission error has to be kept low for both drive and coast side in order to not have noise issues. This will be modeled to reach a good mesh properties before manufacture.

A few transmissions will then be built according to the optimized design but with different PM technologies built in and then put into a car for everyday driving as a proof of concept. The transmissions will also be put in test rigs to test durability, noise and efficiency according to specified drive cycles to more scientifically prove the possibilities with PM in automotive transmissions.

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