Automotive Transmission Design Using Full Potential of Powder Metal

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For metal replacement with powder metal (PM) of an automotive transmission, PM gear design differs from its wrought counterpart. Indeed, complete reverse-engineering and re-design is required so to better understand and document the performance parameters of solid-steel vs. PM gears. Presented here is a re-design (re-building a 6-speed manual transmission for an Opel Insignia 4-cylinder, turbocharged 2-liter engine delivering 220 hp/320 N-m) showing that substituting a different microgeometry of the PM gear teeth—coupled with lower Young's modulus—theoretically enhances performance when compared to the solid-steel design.

Introduction

Höganäs AB has established—through its demonstration cars and design work—that PM gear technology is capable of replacing gears in automotive transmissions without sacrificing performance. What's more, PM gear technology has the inherent capability to reduce the weight and inertia of the gear wheel, thus reducing mass and energy losses. Another important benefit of lowering the inertia of the gears is the simplification of energy dissipation in the synchronization mechanism with both manual gearboxes and AMT- or DCT-type transmissions.

When designing PM gears, special attention must be paid to using the correct material properties, as verified through Young's modulus and Poisson's ratio. Designers can also improve weight and dynamics by the awareness and understanding of the possibilities that PM offers through its unique production methods. For example—the PM gear manufacturing process enables a reduction in manufacturing steps—thus providing improved cost performance.

Young's modulus and Poisson's ratio can be empirically calculated as a function of density (Eqs. 1 and 2; Ref. 1).

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

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

Methodology

System analysis. In order to determine the extent of difference between the microgear and solid-steel design, as well as the possibilities existing for weight reduction, a re-design of a GM (General Motors) gearbox was performed. The chosen transmission was a 6-speed manual transmission rated for 320 N-m, named "M32." This transmission is used in certain Opel Insignia models as well as other GM cars.

Another aim of this work was to understand how much load PM gears must sustain and, from that, to identify the best manufacturing process necessary to meet the stress criteria.

The abovementioned transmission was purchased and disassembled while recording the pull-off forces of the gears and bearings, as well as measuring axial play in the system. The housing was scanned and imported into finite element software (Fig. 1). Shafts and gears were measured, modeled and assembled into the housing. An essential part of the system analysis is bearing stiffness. The bearing representation in this system model is reduced to define the stiffness between two nodes—i.e., inner and outer ring—because this bearing stiffness is strongly non-linear and dependent upon both bearing design and load direction/magnitude.

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

The output from the system analysis is gear misalignment and transmission deflections. This data is used as an input for the gear analysis, where the microgeometry is tweaked to realize the best working behavior of the gears, and for addressing the misalignment and bending from shafts and bearings.

Gear analysis. The 6-speed transmission was completely dismantled; all parts were then measured and reverse-engineered to acquire current production data for all gears, shafts and housing. Macrogeometry of the gears was created with a focus on surface stress levels and peak-to-peak transmission error (TE). For first, second, and reverse gear, the driver member could not be exchanged since the gears were cut directly on-shaft;

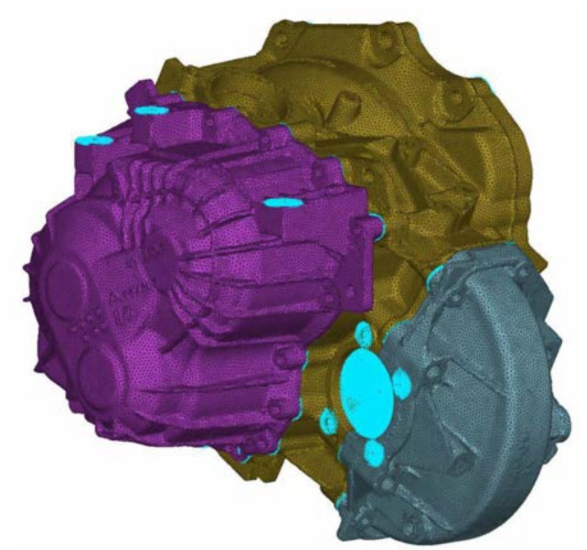

Figure 1 Scanned and digitized housing.

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Table 1 Material data for PM

Material	Elastic modulus (GPa)	Poisson's ratio	Thermal expansion (°C^{-1})	Fatigue limit, surface (MPa)	Fatigue limit, root (MPa)
Powder metal	160	0.28	$12.5\cdot10^{-6}$	$1100@5\cdot10^{7}$ Cycles	$650@10^{7}$ Cycles

thus, for these parts only modification of the idler and driven members was performed. The final drive is a straight carry-over.

Modifying the microgeometry of the gears is an iterative procedure using the material data, loads and misalignments, with the primary intent of lowering both TE and contact stress. This is accomplished by changing the gear design parameters in the iterations, such as crowning, reliefs, angular deviations, etc.

A duty cycle based upon "typical European consumer usage" and the authors' experience was used to evaluate gear life.

The misalignment data gleaned from the system analysis has been accounted for in the microgeometry of the tooth flanks. The abuse load is 6,500 N-m on differential cage—also based on author experience and vehicle data.

The working behavior of the gears in the system has been modeled for 50-percent-, 100-percent-, 150-percent- and 200-percent-load, and at different temperatures in order to assure functionality under various conditions.

All parts were modeled using linear-elastic material properties; material properties are based on input from Höganäs AB (Table 1). Several different software programs were iteratively used to conduct the analysis of the different components and system.

Results

Following are some most pertinent results, as a complete accounting of all the testing is beyond the scope of this paper.

A parameter that describes the quality of the mesh cycle of two flanks is the peak-to-peak 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—as compared to steel—TE tends to increase if the geometry is copied from the steel design (Ref. 2). This can be "designed away" to some extent in the PM design. Figure 2 shows the maximum TE for three different gear designs during a torque sweep; it is the first gear pair in the transmission and is used for switching from an idling standstill.

The first observation is that the TE is quite high. Since this is the first gear, it is only used for initial acceleration and so a slightly higher TE is acceptable. More important are the displayed "curves"; i.e.—the green curve is the PM gear with the steel-flank design, and is higher for all torques, indicating that the TE will be higher for the copied PM gear—an unacceptable development. The result of design iterations for improving the TE for the PM gear is shown in the blue curve, where the TE is lower for every torque level and is likely to perform significantly better than the PM gear with the steel-gear-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 (Fig. 1), but a gear *designed* for PM will always be an improved design compared to a PM gear with the copied steel design.

Table 2 shows the contact and bending stress listed for the sixth gear pair in both original steel and re-designed PM.

The sixth gear was deemed representative in that the result displays a typical improvement number— –17 percent in contact stress—and so is a good example of a gear suitable for PM from a performance point of view. Worth noting is that the bending stress is intentionally increased for the PM gears; this enables designing a lower contact stress for the same gears. Gear design is an iterative trade-off process. As such, the sixth gear pair was judged to be at its best with a lower contact stress—the trade-off being increased root stress. Root stress can also be further reduced with PM technology using the existing optimization procedure (Ref. 3).

The durability of the sixth gear pair is illustrated in Figure 3, where the duty-cycle is taken into account. The red, blue and black lines are S-n curves for sintered, case-hardened, Astaloy85Mo PM gears, with a density of 7.25g/cc and tolerance class of ISO 7 or better. What is learned from the diagram is that, while the tooth root bending fatigue is within acceptable boundaries, the contact stress is still a bit too high, meaning that these gears would require a slightly higher performance level to qualify. The remedy in

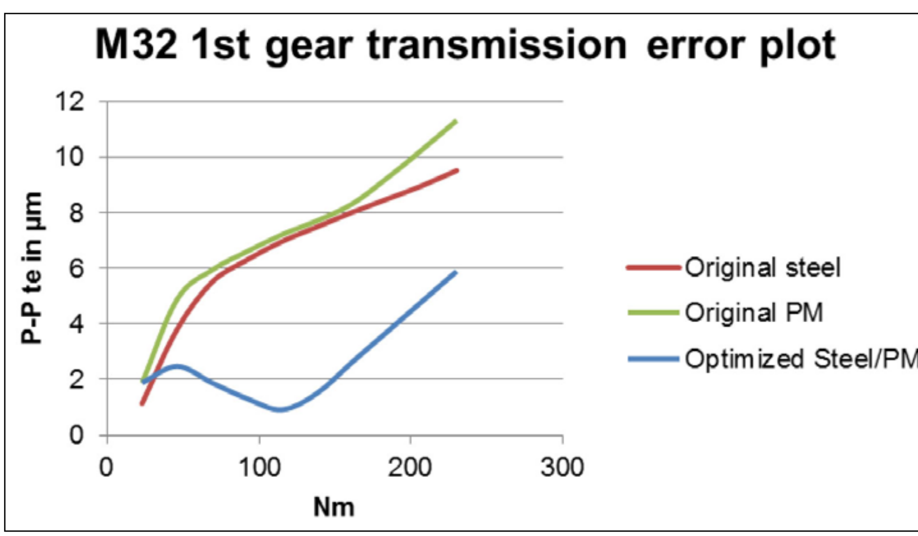

Figure 2 Transmission error for first gear in the investigated M32 transmission.

Table 2 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%	

this case could be increasing the density to 7.4g/cc by double-pressing and double-sintering, or by switching to a higher-performing material. Shot peening to induce higher compressive stresses and/or superfinishing could be other cost-efficient methods to increase the fatigue limit to the additional seven percent necessary to qualify. But without re-design, a 25 percent performance increase (1,200 MPa to 1,500 MPa) would have been necessary, necessitating significantly more expensive processes that would negate the cost-efficiency of PM.

For this particular transmission re-design the third and fourth gear pair can be made with the shortest possible manufacturing time while providing a 7.25 density. For the fifth and sixth gear pair, some of the abovementioned processes would be necessary in order to boost performance. The first and second gear pair requires either densification or a more radical re-design with asymmetric gear teeth or non-involute gear shape.

The re-design not only takes microgeometry into account, but also macrogeometry for attaining the desired weight and inertia reduction. Inertia reduction also off-sets losses from the accelerating gear mass every time the RPM is shifted. What is more, reduced inertia reduces heat dissipated in the synchronization of the gears; less heat build-up provides a more robust synchronization system and longer service life. The energy savings may also be helpful in designing a simpler and smaller synchronization package, thus reducing either overall dimensions or the transmission (Table 3).

Future Work

The next step is to re-design the first and second gear pair using more advanced design methods. These would include non-involute gearing and asymmetric gear teeth for prototyping the gearbox, but without using any performance-enhancing technologies such as hot isostatic pressing (HIP) or other densification technologies. There are a few unknown factors when departing from the traditional, involute curve shape. For example, while it is very possible to reduce contact and bending stress, the difficulty lies when TE must be kept low for both the drive- and coast-sides in order to prevent noise issues. Indeed, modeling to achieve good mesh properties is required *before* manufacture.

Test transmissions will be built according to the optimized design, using the latest available PM technologies, and will be tested in a car for everyday driving as proof of concept. Test rigs will be employed to monitor these transmissions for durability, noise and efficiency—per specified drive-cycles—in order to demonstrably prove the possibilities of PM in automotive transmissions.

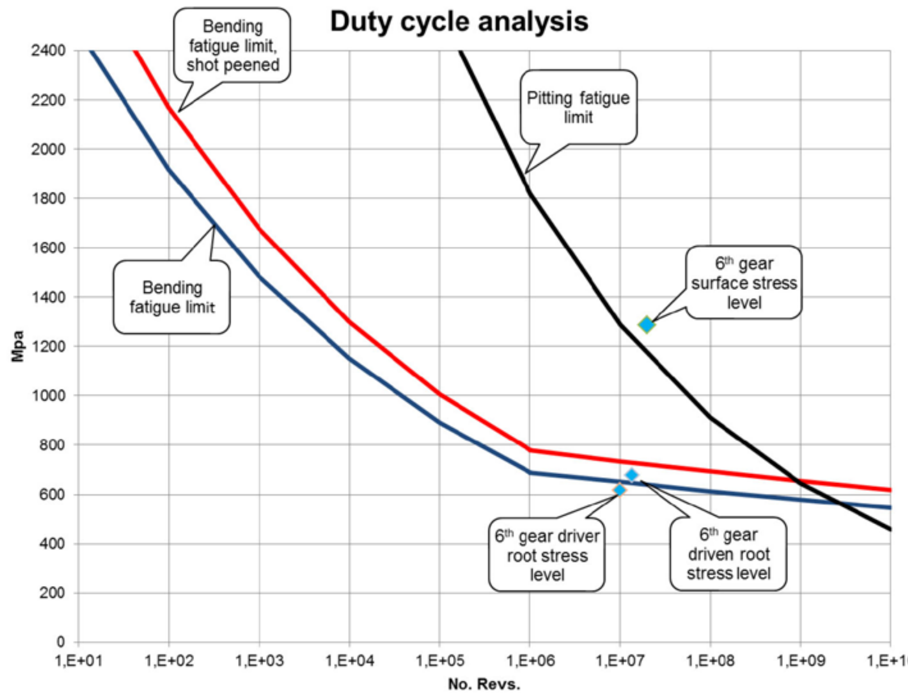

Figure 3 Loads on sixth gear pair with correlating S-n curves for case-hardened Astaloy85Mo PM gears with ISO 7 or better tolerances.

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Table 3 Weight and inertia reduction for 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.