Influential Criteria on the Optimization of a Gearbox, with Application to an Automatic Transmission

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Introduction

In the design of an automatic transmission gearbox, the variation of one parameter can result in different system performances due to the strong interdependencies among all components. For given transmission ratios, component lifetimes and safeties, or space restrictions, improvements in efficiency, noise, and weight can be achieved.

In order to find an optimal solution, it is necessary to perform an analysis of a large amount of gearbox configurations. Using a dedicated design software, an engineer can easily create several variants of a transmission to evaluate. To match the real behavior of the reducer as closely as possible, it is important to take into account the following factors of influence on the simulation results.

In the example of an automatic transmission, when performing a load spectrum calculation, we have to consider the carrier deformation of the planetary stages for the misalignment of the planet axis, and the housing stiffness for the bearing positions. These results have an effect on the shaft deflections and the gear load distributions, and thus indirectly on the reliability of the system. Modifying the carrier shafts and housing design can then be a source of improvement. Thanks to the transmission error and Eigen frequency analysis, it is also possible to estimate the vibration behavior of the reducer. Modifying the shafts dimensions, macro- and microgeometries of the gears, and eventually the positions of the bearings can be necessary in this case. Concerning the power losses calculation, a modification of the macro- and microgeometries of the gears, or the bearings types, can have a considerable impact on the final results.

This paper investigates the influence of the aforementioned parameters on the optimization of a reducer. To validate our analysis, a 6AT gearbox concept is studied and developed in cooperation with the German Ruhr-University Bochum and the Chinese transmission manufacturer Shengrui Ltd.

Presentation of the Model

The model is a 6AT gearbox concept with power variation on the input. Different load spectra are defined in the *KISSsys* interface that is used to perform all the calculations, but only the maximum load condition, representing the most critical case, is used for the optimization below. This spectrum is defined (Table 1) with a requested lifetime of 2 hours for each shifting gear.

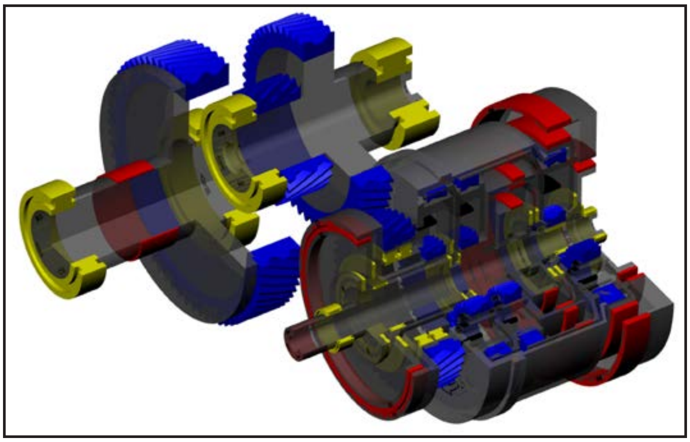

Figure 1 Model overview.

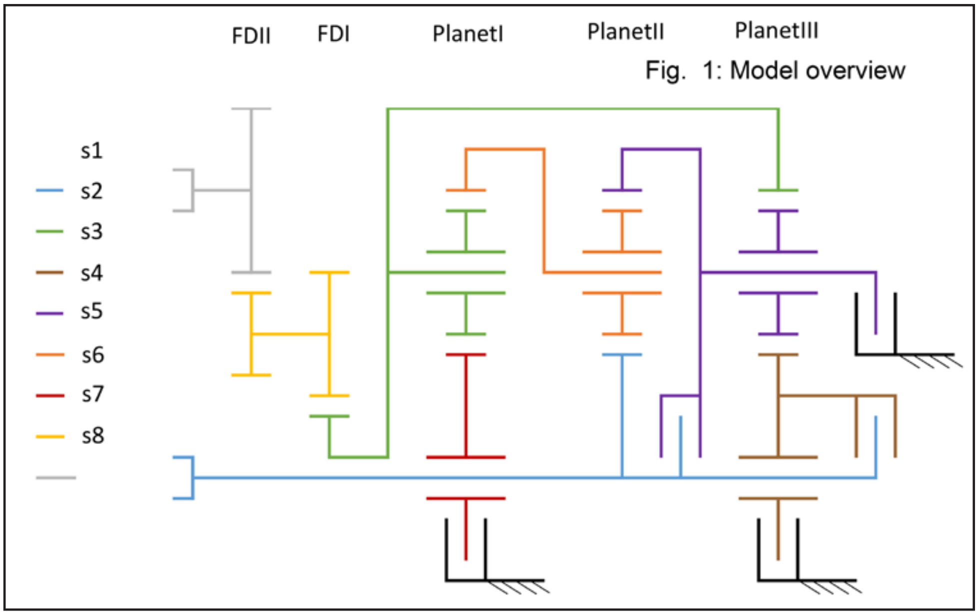

Figure 2 Shafts definition.

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When calculating the strength for each shifting gear independently, we can see that the most critical one is the second — mostly because in this configuration the highest torque is applied on the output shaft. Concerning the reliability of the system for each gear shift, we then get the results in Table 2, which is consistent with our previous observation.

In the current state of the gearbox, the bearings are the most critical elements decreasing system reliability. Because of the axial loads, bearing types, and model size, we don't have a wide range of optimization for these specific elements.

Concerning the casing consideration, we can neglect its effect on the bearing positions — especially in the automotive industry, where these elements are normally very stiff.

With these inputs defined, we can identify the following most influential criteria to manipulate for gearbox optimization:

- For weight: shafts geometry, gears width
- For noise: shafts geometry, bearings and loads positions, gears tooth profiles
- For efficiency: gears tooth profiles

From this list we can easily estimate a good optimization process:

- First, modify the shafts geometry and gears width to reduce the weight of the gearbox while running the strength analysis to keep safeties and lifetimes above required values.
- Then, while reducing the mass, maintain a stiff design for the dynamics analysis, pushing the Eigen frequencies of the system as much as possible above the meshing frequencies.
- Finally, optimize the gears tooth profiles for a reduced transmission error and improved efficiency while considering the shafts deflections and misalignments.

It is also important to mention that the materials of the different elements, which could also be optimized, were not modified in this study.

Table 1 Gear shifts definition

Gear	Ratio	Speed on sl (rpm)	Torque on sl (Nm)	Torque on s8 (Nm)
1	14.555	5000	242.74	−3393.3
2	9.0935	5000	406.51	−3539.8
3	5.6958	5000	450	−2473.4
4	4.2188	5000	450	−1849.5
5	2.9464	4881	450	−1299.5
6	2.229	3905	450	−978.28
R	−9.155	3000	229.75	2020.5

Table 2 Reliability calculation

Gear	Lifetime (h) for 99.9% reliability	Lifetime (h) for 99% reliability	Lifetime (h) for 90% reliability	Reiability (%) for 2h lifetime
1	2.9308	4.2191	5.603	100
2	1.4903	2.074	5.9691	99.121
3	4.0445	5.6288	13.909	100
4	2.5326	3.824	11.236	100
5	7.4115	11.191	32.882	100
6	22.25	33.596	98.716	100
R	3.4608	3.5567	3.924	100

Shafts and Gears Mass Reduction

To avoid dimensioning the coaxial group of shafts, we calculate its maximum transmittable torque in its current state. Thanks to the basic formula of the torsional stress, we can then get a corresponding mean diameter that we can compare to the current one. This method allows us to estimate the potential mass reduction of the system by applying the opposite logic and considering the current torque as the maximum transmittable one. This approach is of course just a rough estimation that consists mostly in scaling down the complete shafts with gears and bearings.

We then calculate the transmittable torque of the coaxial group (s1-s6), shafts and gears, for the second gear shift, and get a maximum of around 500 Nm, instead of the current 406.51 Nm, without decreasing its reliability below 99%. At this point we just have to size some bearings because of low lifetimes for this torque. For this torque difference of 23%, the torsional stress formula gives us an equivalent mean diameter difference of around 7% for the coaxial group of shafts.

At first, and to compare our results afterwards, we also apply this method to the idler and output shafts to estimate the total mass reduction of the system. We calculate their own transmittable torque, so without verifying the connecting gears, and get the following results:

- For s7: 2,400 Nm instead of 1,450.96 Nm — thus a variation of around 18% in diameter.
- For s8: 6,000 Nm instead of 3,539.8 Nm — thus a variation of around 19% in diameter.

For the real optimization and the rest of the study, we in fact only consider the virtual input torque of 500 Nm to resize the two parallel shafts and connecting gears, while keeping their reliability above 99%. Here we modify the shafts geometry and gear widths only to avoid creating some interference by modifying the center distances. As the gear safeties are above required values for the coaxial group, we also slightly optimize their width. We can then apply the scaling down of 7% to the whole system on top of these results to achieve the potential mass reduction for the input torque of 406.51 Nm (Table 3).

We then get around 15% off total mass reduction; and when comparing the optimized and theoretic masses for s7 and s8, we can see that we are quite close to the initial estimation.

Table 3 Mass optimization

Mass (Kg)	Initial	Theoretic	Optimized	Optimized + theoretic (7%)
s1-s6	19.26	17.88	19.14	17.78
s7	5.8	4.74	4.86	4.52
s8	10.74	8.67	8.51	7.9
Total	35.8	31.29	32.51	30.19

Dynamic Analysis of the System

A first evaluation of vibrations in the system requires performing a modal analysis. This allows us to identify the Eigen frequencies of the shafts and their mode shapes. We can then compare these values with the potential excitations coming from the meshing frequencies. For this analysis is quasi stat-

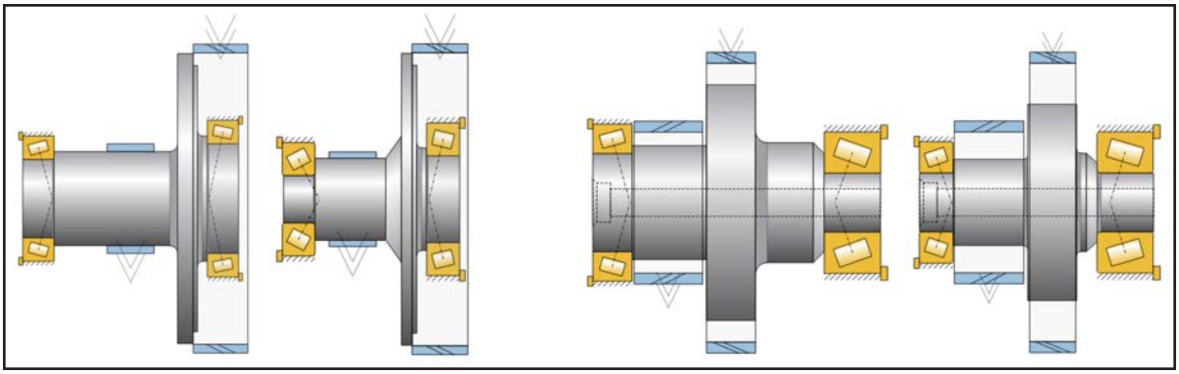

Figure 3 Mass reduction comparison for s8 (left) and s7 (right).

ic, the results should theoretically be the same for all shifting gear. But as we consider the gyroscopic effect to get closer to reality, the Eigen frequencies are different for each operating speed. The same applies to the operating torques since the bearings stiffness calculated from their inner geometry is nonlinear. The study is then made for all shifting gear, but as similar results can be observed, we can summarize the analysis with the second gear again.

We compare the modal analysis of the shafts before and after mass reduction, and can interpret the results quite well on shaft s7. First, we calculate the meshing frequencies of the system (Table 4). Here we don't calculate the harmonics and modulated frequencies with the shafts speeds, as we can already identify some critical frequencies (around 1,300 Hz, and around 1,700 Hz) in the analysis of Table 5.

We can see that the meshing frequency of around 1,700 Hz is no longer critical to the optimized shaft; but the one around 1,300 Hz — more important due to its coming from the excitation of the gear directly mounted on the shaft — is still present and now close to two different mode shapes, i.e. — axial and bending. We can also see that, in general, the Eigen frequencies of the optimized shaft are lower than the initial one for torsion and bending mode shapes. In general, it then seems that the stiffness of the shaft is reduced for these, but slightly increased for the axial deformation.

We can confirm this interpretation by decreasing only the length of the initial design — without changing the diameters — and comparing its modal analysis with the one from the initial shaft (Table 6).

First, we can see that when reducing the shaft length without changing the diameters we manage to push the first torsion mode shape in higher frequencies, thus making the shaft stiffer for this deformation. Concerning the bending mode shapes, we cannot see much difference below 4,000 Hz, simply because the nodes of these modes are mostly located at the position of the right bearing where no change was made for the diameter between all designs.

To get another estimation of the different mode shapes evolution, we compare the distance between loads and bearings when the shaft length and diameters are kept constant, like the virtual shaft displayed in Figure 4. We can clearly see that the shaft gets stiffer against torsion when the loads are close to each other in Table 7, as we can see a very steep part in the torsion mode shape between the two spaced loads as a difference from the close ones (Fig. 4).

Concerning the axial and bending mode shapes, if we look at these

Table 4 Meshing frequencies

	FDI (s7-s2)	FDII (s8-s7)	PlanetI (s6-s2-s5)	PlanetII (s1-s5-s4)	PlanetIII (s3-s4-s2)
Frequency (Hz)	1242.1	632.33	1026.1	1786.9	571.95

Table 5 Eigen frequencies comparison with meshing frequencies

Eigenmode	Initial		Lenght + Diameter reduction	
	Eigenfrequencies	Mode shape	Eigenfrequencies	Mode shape
1	1316.38 Hz	Axial	1219.51 Hz	Bending XY
2	1651.90 Hz	Bending XY	1394.17 Hz	Axial
3	2842.85 Hz	Bending YZ	2381.32 Hz	Bending YZ, Bending XY
4	3785.94 Hz	Bending YZ	3108.07 Hz	Bending YZ, Bending XY
5	4064.81 Hz	Bending YZ	3132.68 Hz	Bending XY
6	16169.46 Hz	Bending XY	15329.62 Hz	Torsion
7	16649.82 Hz	Bending YZ	16137.30 Hz	Bending XY, Bending YZ
8	17928.96 Hz	Torsion	16444.57 Hz	Bending XY
9	18220.94 Hz	Bending XY	19378.30 Hz	Bending XY

Table 6 Eigen frequencies comparison with length reduction

Eigenmode	Initial		Length reduction only	
	Eigenfrequencies	Mode shape	Eigenfrequencies	Mode shape
1	1316.38 Hz	Axial	1211.70 Hz	Bending XY
2	1651.90 Hz	Bending XY	1385.86 Hz	Axial
3	2842.85 Hz	Bending YZ	2375.54 Hz	Bending YZ, Bending XY
4	3785.94 Hz	Bending YZ	3090.85 Hz	Bending YZ
5	4064.81 Hz	Bending YZ	3115.11 Hz	Bending XY
6	16169.46 Hz	Bending XY	18508.19 Hz	Bending XY, Bending YZ
7	16649.82 Hz	Bending YZ	18762.10 Hz	Torsion
8	17928.96 Hz	Torsion	18762.16 Hz	Torsion
9	18220.94 Hz	Bending XY	18775.73 Hz	Bending XY

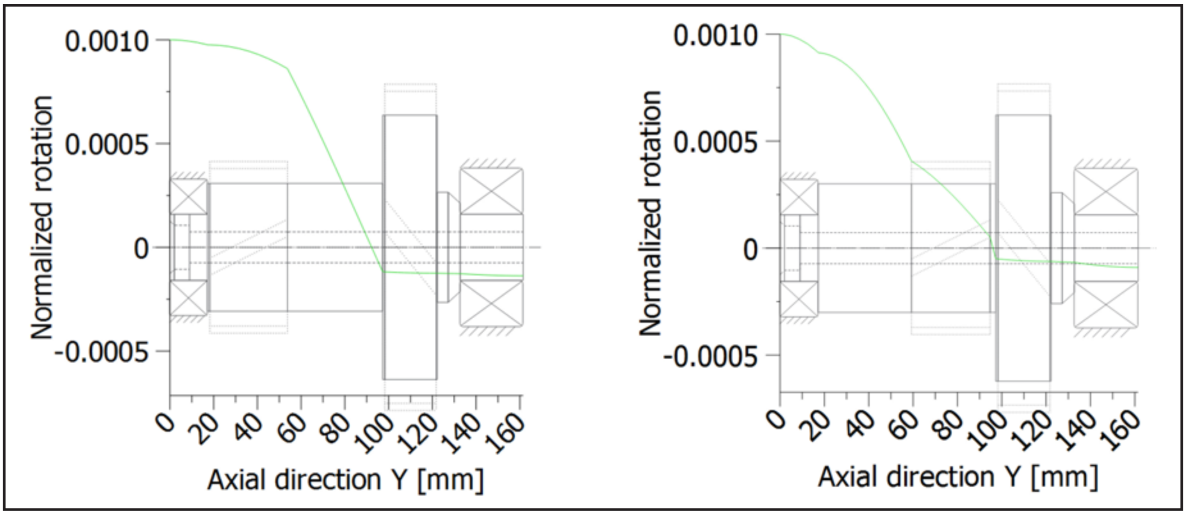

Figure 4 Influence of loads positions, example of torsion mode shape.

results, as well as the ones from the initial shaft where the loads were close and one load was also spaced from a bearing, we can see that spaced loads and bearings seem to be favorable to stiffness against bending, whereas the opposite appears to be favorable to stiffness against axial deformation.

In a general way, we can then say that a more compact design is better to avoid noise generation. But when reducing the mass, the engineer must maintain a stiff design by keeping correct diameters, avoiding mass concentration, and correcting loads positions depending on the mode shapes he wants to attenuate. In this case, the potential excitation emanates from the meshing frequencies.

Table 7 Eigen frequencies comparison for loads positions

Eigenmode	Spaced loads		Close loads	
	Eigenfrequencies	Mode shape	Eigenfrequencies	Mode shape
1	1264.80 Hz	Bending XY	1264.82 Hz	Axial
2	1313.09 Hz	Axial	1324.87 Hz	Bending XY, Bending YZ
3	2269.41 Hz	Bending YZ, Bending XY	2266.30 Hz	Bending YZ, Bending XY
4	3056.80 Hz	Bending XY, Bending YZ	3014.84 Hz	Bending XY, Bending YZ
5	3259.84 Hz	Bending YZ, Bending XY	3296.86 Hz	Bending YZ, Bending XY
6	6805.40 Hz	Torsion	10500.37 Hz	Bending XY, Bending YZ
7	6938.92 Hz	Bending YZ, Bending XY	11025.08 Hz	Bending YZ, Bending XY
8	7269.97 Hz	Bending XY, Bending YZ	13017.89 Hz	Torsion
9	15257.43 Hz	Bending XY, Axial	19157.14 Hz	Bending XY

Gear Sizing for Noise and Efficiency

For the final step of optimization we perform a contact analysis of the different meshes to evaluate and optimize their peak-to-peak transmission error (PPTE), other influential criteria for noise, and power losses. Once again we perform this analysis on the second gear shift with the maximum load provided.

For each gear meshing, we first recalculate the root and flank safeties, considering the shaft deflections, and the tilting of the planet axis from the finite element tool included in the simulation software *KISSsoft*. We can then observe that with the face load factors (KHb) consideration (Table 8), these safeties are much lower than the theoretical ones calculated in the first step of this optimization.

We try then to optimize the gears macro- and microgeometries in terms of transmission error (sizing of the profile modifications as well for a smooth meshing during the same operation) and power losses, considering the face load factors calculations due to the shafts deformations, and for a reliability of the system still above 99%. The center distances and gear widths are kept constant, as well as the gear ratios (with a minimized deviation).

During the sizing functionality, for better wear reduction we then also choose to consider only the solutions that provide a specific sliding below an absolute value of 3, and profile shifts coefficients optimized for a balanced specific sliding along the path of contact between the pinion and the wheel.

As can be seen in Figure 6, the software covers more than 200 solutions for each meshing, from which we can pick the optimum one between transmission error, efficiency and mass. In this example the solution 200 in the top left corner seems to be the best choice in terms of TE and efficiency, and is in the lower range in terms of mass, which doesn't vary

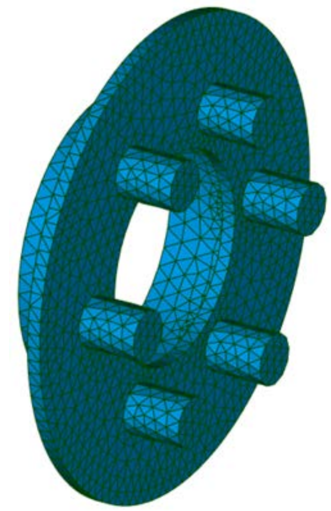

Figure 5 Carrier mesh s2.

so much anyway from the lightest to the heaviest solution. We perform this operation for the 2 gear pairs and the 3 planetary gear sets; and because an improvement of both efficiency and transmission error is not possible in most cases, we therefore tend to accentuate efficiency when the transmission error is quite low; but that is again depending on the direction the engineer wants to take.

We can generally observe in the optimum solutions that the geometries all tend to provide a transverse and overlap contact ratio getting close to 1.5 each. For example, if the overlap ratio is higher, then the helix angle will decrease, and vice versa. For the rest of the parameters, mostly, modules and teeth numbers vary in opposite directions to maintain constant center distances while getting a transverse contact ratio closer to 1.5. And, the pressure angles tend to increase when the bending safeties are much lower than the required ones.

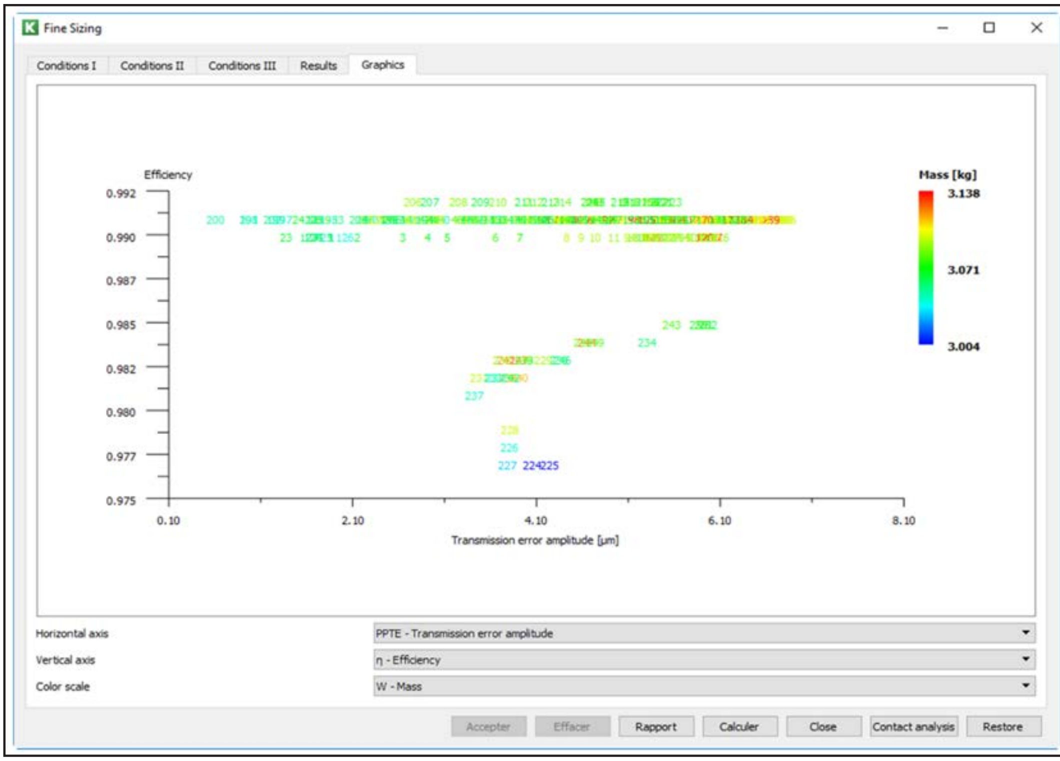

Figure 6 Fine sizing of the gears.

Table 8 Gear sizing results

	FDI (s7-s2)	FDII (s8-s7)	PlanetI (s6-s2/s2-s5)	PlanetII (s1-s5/s5-s4)	PlanetIII (s3-s4/s4-s2)
KHb Initial	1.4381	1.1972	1.3241/1.2374	1.0220/1.0736	1.0910/1.2567
KHb final	1.2775	1.1076	1.2849/1.1926	1.0185/1.0641	1.0891/1.1975
Difference	0.1606	0.0896	0.0392/0.0448	0.0035/0.0095	0.0019/0.0592
PPTE ini. (μm)	2.7733	1.3466	0.210	0.036	0.040
PPTE fin. (μm)	2.5181	0.5793	0.342	0.089	0.135
Difference (μm)	0.2552	0.7673	–0.132	–0.053	–0.095
Efficiency ini. (%)	99.27	97.98	98.90	97.68	99.02
Efficiency fin. (%)	99.39	99.11	99.19	98.67	99.45
Difference (pp)	0.12	1.13	0.3	1	0.45

We can then calculate a total efficiency improvement from around 93% on the initial system, to 96% on the optimized one, when considering only the gear meshing losses. Concerning the transmission error, we can see that the PPTE value is considerably improved for the 2 gear pairs, but slightly bigger for the three planetary stages where the initial value was already very low. Overall, we can say that the optimization for noise reduction on the gears is also successful.

Conclusion

When trying to optimize a gearbox, a considerable amount of solutions exist. If, for a certain system reliability, the perfect solution could be found for noise reduction, it would necessarily be to the detriment of mass and efficiency, and vice versa. Once the objectives of the project have been clarified, an engineer can then prioritize the elements to optimize and find the right balance between the modification of the shafts geometry, bearings and loads positions, and gear tooth profiles. With the help of a designated simulation software like *KISSsoft*, the engineer can then evaluate the dynamic behavior of several geometric variants of a reducer in a very short time, size the corresponding bearings to match the required lifetime, but also evaluate the relation between transmission error, efficiency, mass or other, of hundreds of propositions of gear geometries with optimized profile modifications that match his design limitations. **PTE**

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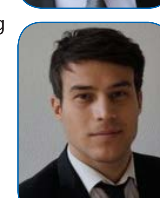

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