Design and Optimization of a Hybrid Vehicle Transmission

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Introduction

The hybrid vehicles seem to be the fastest solution for the containment of consumption and of pollution for personal transportation.

The designer of a hybrid transmission has to address additional issues with respect to the classical cases, in particular the high speed of the electric unit and the bidirectional motor/generator operation. In this case, a lot of attention should be paid to how to consider the four combinations of signs for torque and speed in the load spectrum for the gear calculation. The paper presents some topics, like several approaches for the alternating bending factor, the effects of the asymmetric crowning (especially the helix modification, tapered or parallel) and how to consider the housing stiffness in the TCA.

Finally, the paper presents an interesting solution from the kinematic point of view, the compound planetary, relatively well known in the automotive, but much less so in the industrial gearboxes design.

The data presented in the paper are taken from some recent jobs of the author. For reasons of confidentiality (the projects are still under development) there will be not quantification of the parameters, but only the description of the procedures followed, with details indicated in the bibliographical reference, and of the software used. The latter two types of information are probably the most interesting part of the paper, since they enable the reader to repeat the procedures by himself on his own context. The paper is therefore almost a guideline for the designers.

The need to reduce air pollution has led to the introduction of electric engines for the propulsion of road vehicles. Without making a general introduction in this regard, suffice it to say that the design of the transmission for this new type of vehicle requires special attention. A comprehensive discussion on the subject is available in (Ref. 1). This paper examines two cases addressed by the author: the optimization of a single-gear transmission (two helical stages speed reducer) for full electric vehicles and the design and optimization of the transmission for a hybrid vehicle. The topics discussed in the first case provide a basis for the second case, which involves greater kinematic complexity because it is based on a compound epicyclic. In particular, the paper presents the calculation method, critical points, bibliographical references and software tools. More information about this type of automotive transmission are in (Refs. 2–3).

Single-Speed Transmission with Double-Stage Helical Gear Reduction

Two different configurations were examined for different applications. The first one is more classic and simple, and the second one is more compact (Fig. 1). The same approach, described below, has been followed for both cases: some macro-geometric configurations had been proposed (module, number of teeth, pressure angle, helix angle, tooth height) with the need to optimize the micro-geometry, i.e. the contact pattern. Several applications (for different vehicles) were studied in the first case, and only one in the second.

Contact pattern optimization. An optimization procedure starts with the definition of one or more objectives, variables "to play with" and constraints to respect. The contact pattern between gears is not only a "visual" objective and an analytical expression is necessary to be able to proceed with numerical optimization. The contact pattern should be expressed and evaluated as an objective metric. Two parallel procedures are followed.

Crowning. The effect of the crowning can be quantified with the root and flank safeties in accordance with ISO 6336 (Ref. 4), as long as the real distribution of the load along the flank is taken into account. In this regard, the method

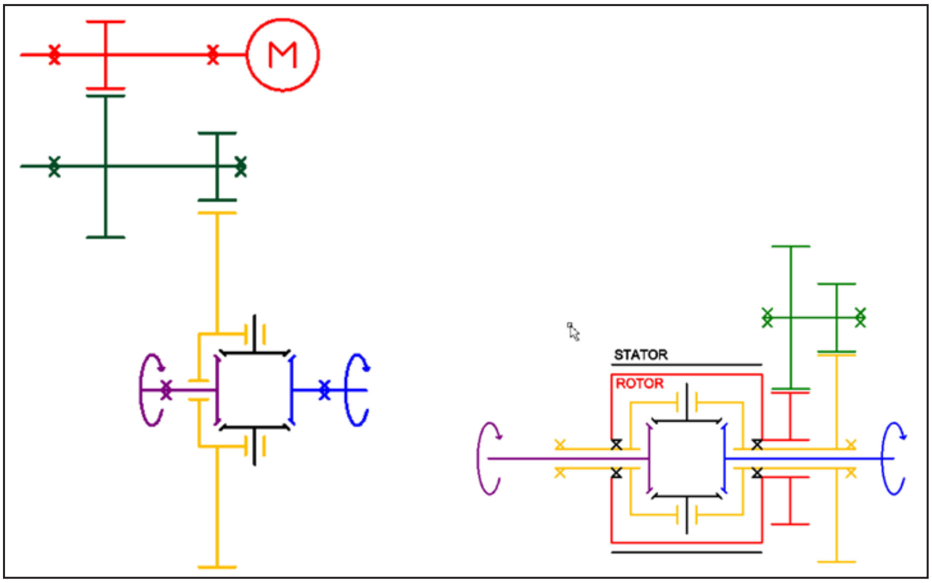


Figure 1 Two different layouts for 2-stage single-speed transmission.

described in ISO 6336-1 Annex E was used, considering the micro-geometry of the gears and the deformation of the shafts. This leads to reliable results in a short time and provides the value of the $K_{H\beta}$ factor necessary for the calculation of the load capacity. The implementation of this method, which utilizes single dimension LTCA, is described in (Ref. 5). SF and SH are thus the objective, i.e.—how these values have been calculated using the load spectrum.

The variables chosen are the two dimensions separately defining the two components of the asymmetric crowning, set as the sum of the symmetric crowning and the helix angle modification. The helix angle modification, conical (tapered) or parallel, has been added as a further variable.

Tip relief. After selection of the optimal (asymmetric) crowning, the tip relief is determined in order to minimize the PPTE (Peak to Peak Transmission Error) and peak contact pressure. Since these values depend on the load, a graph has been made on various torques levels, letting the human eye choose the relief value leading to the graph of PPTE without high value at lower torque.

Alternating bending, drive & coast flanks and spectrum. As stated previously, the first problem faced was to calculate SF and SH with the load spectrum in Table 1. ISO 6336-5 explains how to calculate cumulative damage with the Palmgren-Miner's Rule. This method is widely used but is only valid when the driving gear is the same one and the drive flank does not change. In the present case, however, the spectrum shows all four combinations of signs for the torque and speed, since the electric engine acts as a motor or brake in one direction and the other. It is therefore necessary to take this into account in calculating the

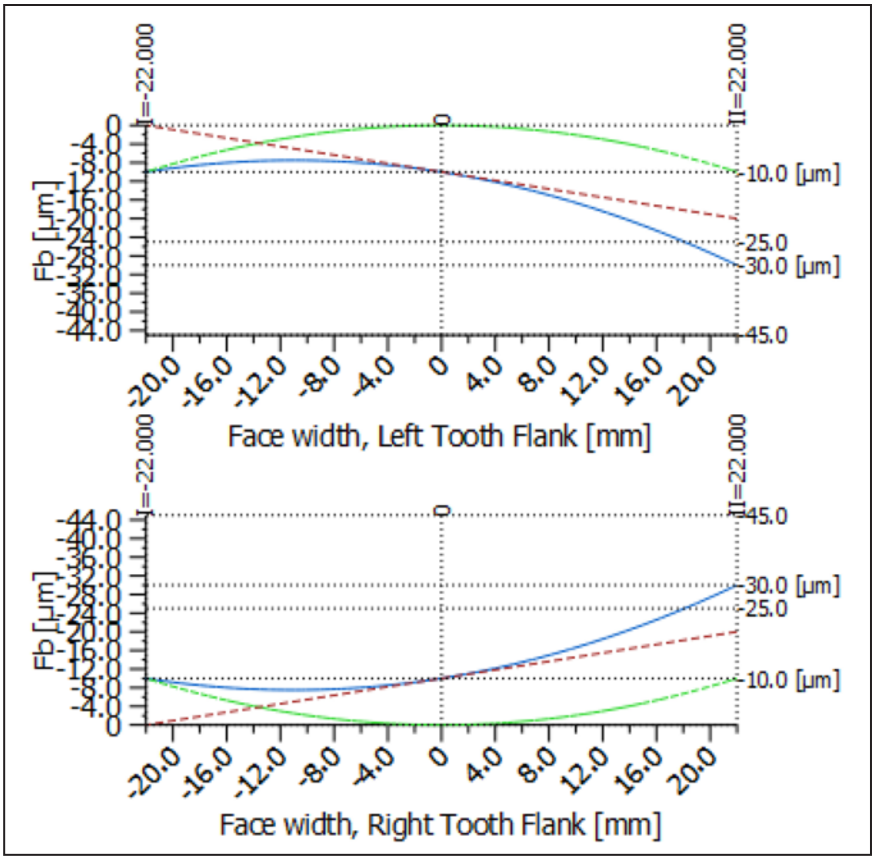


Figure 2 Asymmetric crowning (blue) = symmetric crowning (green) + helix angle modification (red).

Table 1 Load spectrum with mixed sign on torque and speed; shows torque and speed factors with respect to nominal data

% Time (Σ = 100)	% Torque	% Speed
3.673	100.000	100
10.034	100.000	200
1.233	100.000	400
10.116	72.370	600
11.062	61.852	–800
2.558	30.701	1000
10.697	33.333	1200
0.032	29.630	1400
6.125	–81.580	–100
2.221	–81.481	200
10.693	–81.481	333
6.280	–81.481	467
2.060	–71.370	600
3.453	–55.556	720
12.002	–48.148	890
7.762	–40.741	1000

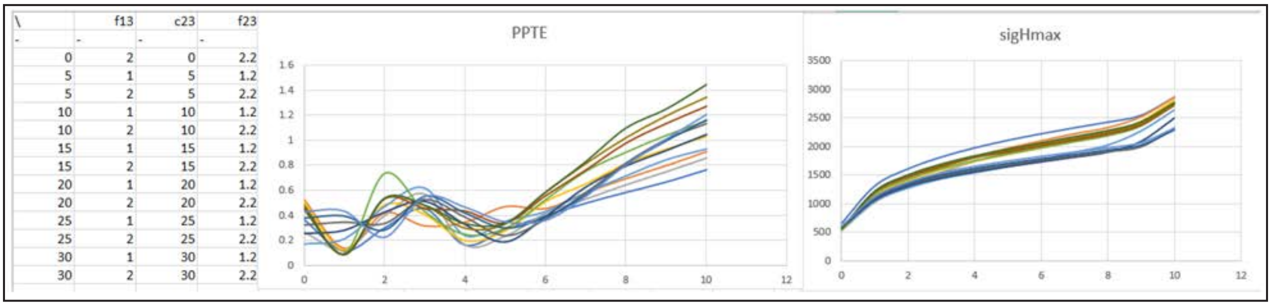


Figure 3 PPTE and max pressure in different load condition from 1 to 100% of the Max Torque, with different microgeometry solutions.

bending (which is neither pulsating nor fully reversing) and pitting, for which the number of hours on each flank is not the same and it is also different for bending calculation. For each line of the spectrum the $K_{H\beta}$ factor (function of the crowning and the deformation of the shafts) must obviously be recalculated.

For the calculation of surface durability, two separate calculations are made, one for each flank, each for the number of hours actually occurring on that flank, as shown from the spectrum. The flank safety is the lowest of those calculated.

The calculation of tooth bending strength is a little more complex.

The rules for the calculation of the load capacity of gears show the fatigue limits of materials with one way bending. In the case of fully reversing bending, AGMA 2001-D04 (Ref. 6) states that this should be reduced to 70%.

ISO 6336-5:2016 also reduces resistance for fully reversing bending to 70%. In the other cases, ISO 6336-3:2006 Annex B states that it should be dimensioned by a Y_M factor, a function of the stress ratio (i.e. the ratio between the loads on the two flanks of the same tooth), of the material and, for case hardened gears, of the tooth root form (Table 2).

$$Y_M = \frac{1}{1 - R\,\dfrac{1-M}{1+M}} \qquad (1)$$

where

$$R = -1.2 \cdot \frac{\text{load per unit facewidth of the lower loaded flank}}{\text{load per unit facewidth of the higher loaded flank}}$$

M considers the mean stress influence on the endurance (or static) strength amplitudes, and is defined as the reduction of the endurance strength amplitude for a certain increase of the mean stress divided by that increase of the mean stress, as defined in 6336-3:2006 Annex B

Table 2 Mean stress ratio, M (Ref. 4)

Material	M
Case hardened	$0.8 + 0.15\,Y_s$
Case hardened and shot peened	0.4

A few years ago, a paper was presented on how to calculate this reduction in fatigue resistance for alternating bending according to the theories of Gerber and Goodman (Ref. 7). Both theories put Y_M at levels lower than 0.7 for pure alternated bending, respectively 0.569 and 0.699.

The same "evolution" can also be seen in the change in Y_M from 0.7 to 0.65 shown in the two subsequent editions of the book on cylindrical gears (Refs. 8–9). This shows the formulation for intermediate cases, referring to the number of inversions, with the stress factor in any case unchanged, i.e. with the same value of force applied to the two flanks (Fig. 4).

For the calculation here, it was chosen the formulation most adapted to the data contained in the spectrum involved. As explained in (Ref. 10), two calculations are performed:

1. with $Y_M = 1$ for the lines with positive torque and $Y_M = 0.7$ for those with negative torque
2. with $Y_M = 0.7$ for the lines with positive torque and $Y_M = 1$ for those with negative torque

The final safety *SF* is the best between the two (not the worst). This is called the "most realistic" one because, since the number of inversions is not known, choosing the worst value (as done however in the calculation for pitting) would be excessively conservative. It should be observed that the spectrum used is already condensed and does not represent the actual load history (in the sense of chronological sequence); the change between positive and negative sign of the spectrum lines does not represent a change in the functioning of the transmission.

Contributions to the contact pattern. As discussed previously, the contact pattern is a function of the shaft microgeometry (to be optimized) and the shaft deformation. The latter is affected by the stiffness of the gear body, the shaft, the casing, the bearings (with the operating clearance) and the accuracy of the bearings arrangements.

A detailed description of the method used is available in (Ref. 11). A model of the entire transmission has been made for making a totally automatic iterative calculation without operator intervention. The calculation is as follows:

1. Kinematic calculation of the gears (forces acting on the tooth)
2. Application to the shafts of the forces calculated in the previous point
3. Calculation of the reactions, considering only the stiffness of the bearings with the relative pretension
4. Calculation of the displacement of bearings housing on the casing (FEM)
5. Calculation of the real deformation of

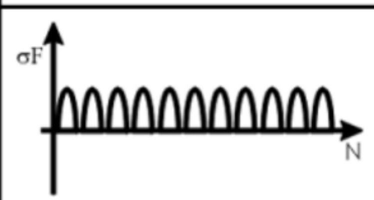
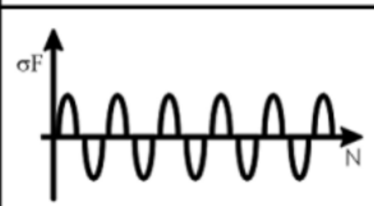
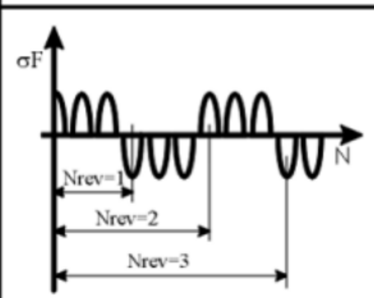
Operating Mode	Alternating Bending Factor (Mean Stress Influence Factor) Y_M	Load Direction
Pulsating	1	
Alternating	0.7 (1) 0.65 (2)	
Oscillating	$0.85 - 0.15 \cdot \frac{\log N_{rev}}{6}$ (1) $0.85 - 0.20 \cdot \frac{\log N_{rev}}{6}$ (2) $(1 \le N_{rev} \le 10^6)$ 0.7 (1) 0.65 (2) $(N_{rev} \ge 10^6)$	

Figure 4 Alternating fatigue according to (Ref. 8) (1) e (Ref. 9) (2).

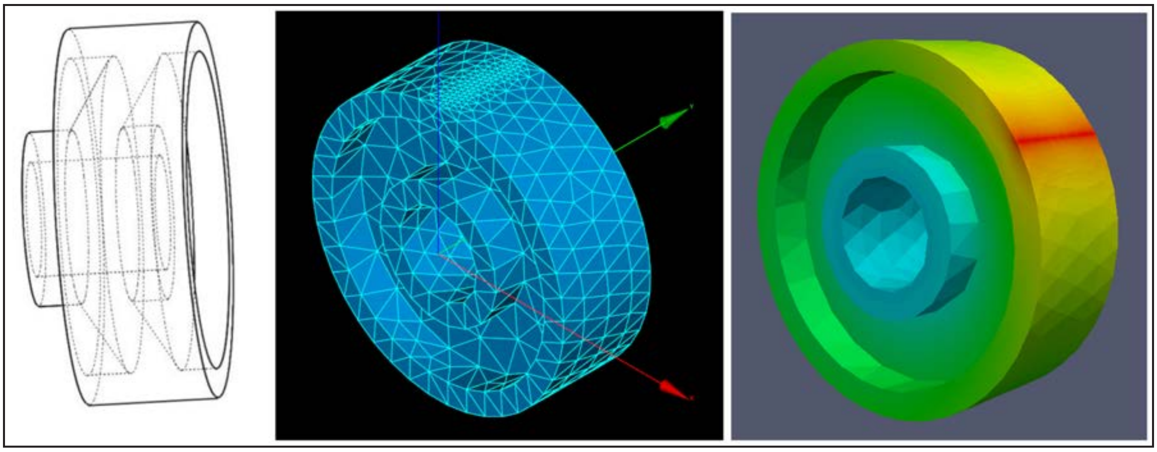


Figure 5 Gear body FEM calculation (Refs. 10 and 15).

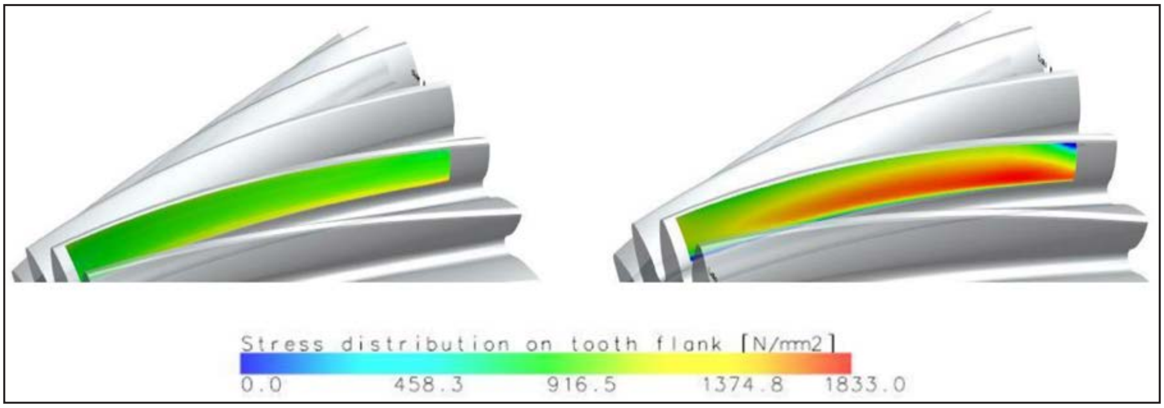


Figure 6 Contact trace at the max torque calculated with the LTCA before optimization (A) and after optimization (B).

the shaft

6. LTCA on gears with the misalignments
7. Updating of the figures in point 1 and repetition of the calculation until convergence.

The shaft calculation was conducted according the Timoshenko Beam theory. The bearings stiffness calculated as explained in ISO/TS 16281 (Ref. 12). Detailed geometric data of the bearings were requested from the producer, that provided also a software (Ref. 13) to check the results. The contribution of casing stiffness was taken into consideration by exporting from FEM the condensed stiffness matrix in the geometrical centers of the bearings (Ref. 14) and importing it to the calculation model. The contribution of the stiffness of the gear body was also considered in the model. The FEM used in this case is open-source (Ref. 15). Without these integration from the various software, allowing for exchange of data (loads and deformation) without operator intervention, the calculation of deformation would have simply been one-shot and less accurate.

Among the results of this calculation, it was also decided to include the various lives of the bearings (Ref. 16), with the accumulated damage method, also taking into account contamination of the oil on each single bearing. The maximum contact pressure was calculated too, because this value is a good indicator of bearing operation conditions.

Optimization algorithm. It has been previously mentioned the two separate calculations for contact optimization on the two directions of the flank and of the tooth profile. Now that all the interactions of the model are known, the algorithms behind them can be examined. As already explained, in this phase a professional optimizer was not used as was done for the study (Ref. 17), but two different research loops are simply defined.

For longitudinal optimization, the 4 variables for crowning and helical modification were evaluated on 10 values, 10 times from 1% to 100% of the nominal torque. To avoid the proliferation of solutions (up to 10^5), the corrections of the two gears are increased synchronously, thus obtaining only 10^3 variants. The range of variants for the crowning was from 0 to double the value necessary to compensate the deformations at the maximum torque. For the helix modification, variation was between the two values that are positive/negative opposites to the one suggested, thus transiting through 0. It was observed that on the basis of the spectrum, for configuration of Figure1a, the optimal micro-geometry consisted of helical corrections that were sometimes parallel and sometimes conical, in a way not predictable *a priori*.

For the profile modification, the progressive tip relief only was used (avoiding the root relief as being hard to implement), with variations of 10 values between 0 and the value necessary to compensate the deformation of the tooth at the maximum torque, between the long one and the short one (Re. 18). As already stated, the best combination is the one maximizing the safeties against bending and pitting with the given spectrum.

Since it is not possible to establish whether the optimal relief value and the optimal crowning value, when combined,

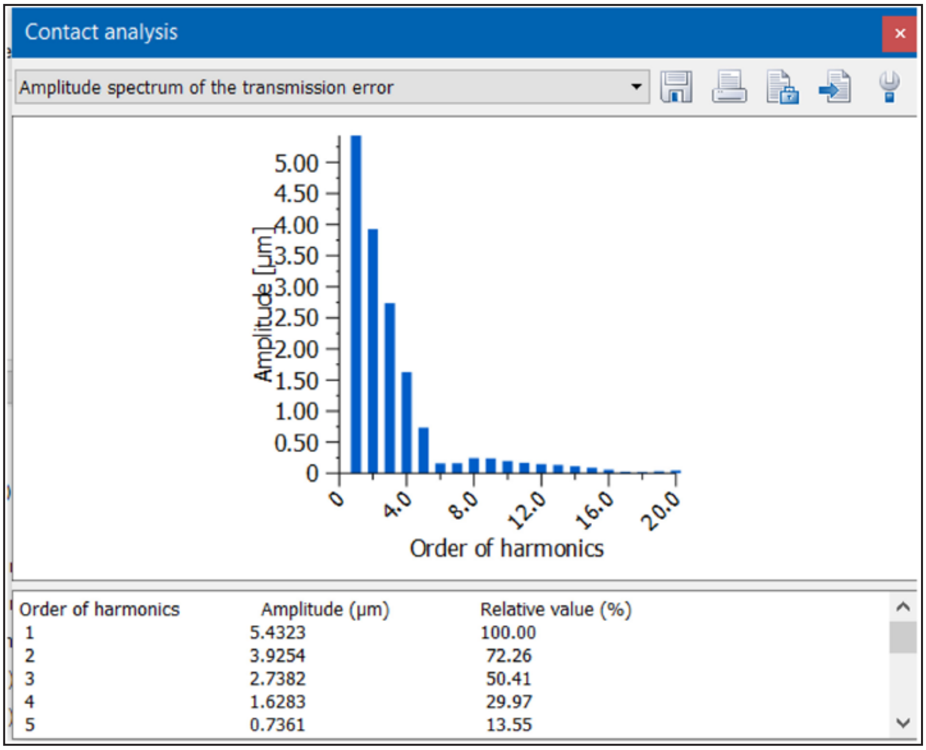


Figure 7 FFT of Transmission Error for a torque level.

Table 3 Empty example of efficiency map

		Input speed										
		0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
Input Torque	0											
	400											
	800											
	1200											
	1600											
	2000											
	2400											
	2800											
	3200											
	3600											
	4000											

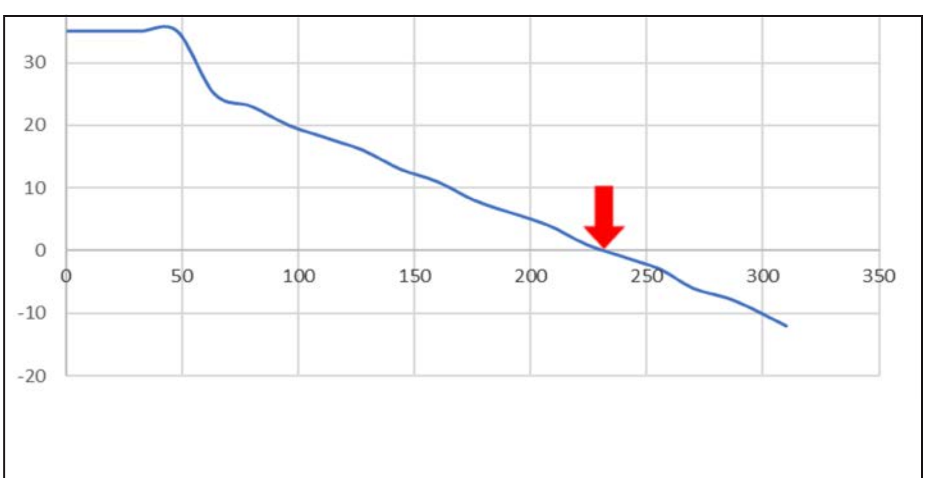


Figure 8 Backlash vs Torque; the value is calculated taking in account all deformation already considered in the LTCA.

lead to the optimal solution, the two research loops should be nested. In this phase we have merely conducted a visual check of the contact trace calculated with the LTCA in the various load conditions present in the spectrum.

Subsequent calculations. Once the macro-geometry and micro-geometry of the gears was defined, transmission noise and efficiency were assessed. Although these should be design values, in this case we identified them subsequently.

FFT of the TE. The graph of the Fourier spectrum for transmission error was generated for 10 levels from 1% to 100% of the maximum torque (Fig. 7).

Efficiency map. In order to evaluate the overall efficiency of the transmission, an efficiency map like the one in Table 3 was used. The table is blank, it's only an example. It was filled for the project but not shared in this paper.

To complete this, the transmission calculation model implemented the power losses due to the bearings, sealing and meshing, with an approach similar to the one described in (Ref. 19). Without going into detail, we can simply recall that the losses on the bearings follow the formulas of the catalogue (Ref. 20), sealing losses (Ref. 21) and meshing losses (Ref. 22). This is the same calculation model as the thermal capacity according to ISO/TR 14179 (Ref. 23). In this case the system temperature was set at 40°C and 80°C and the contribution of oil churning was not considered, since this would have required a CFD analysis to obtain realistic results.

Backlash. The LTCA is the calculation of the tooth deformation: it's a good opportunity to calculate the backlash, that is a function of the load.

It is not possible to avoid checking the meshing of the macrogeometry concentrating exclusively on optimization the microgeometry. A wrong choice of the tolerances of the tooth thickness in the first step of the design could generate interference and wear under load.

Figure 8 shows the decrease of the backlash as the input torque increase. If the input torque is bigger than the marked value, there will be wear by interference on the teeth.

Compound Epicyclic for Hybrid Transmission

Description of kinematics. The experience acquired in the activities above described was then applied on the hybrid transmission (Fig. 9). In this case too, the kinematic layout was applied to different products.

In general, the core of the hybrid transmission is the need to mix in the same axle (the one to the wheels) the torques coming from two completely different engines, the electric one and the IC one, which have very different speeds. Also, as seen in the previous case, the electric engine has the double operation mode motor/generator.

In this case, the bevel gear differential mixes the torque coming from the IC engine (upstream from the parallel stage) and the electric engine (connected to the epicyclic gearset, which provides two different speeds).

There follows a description of the design activity regarding only the compound epicyclic gearset (Fig. 10).

The input is on the first sun z_{s1}, connected with a double planet gear $z_{p1}|z_{p2}$. There is only one ring z_r which is fixed. The outputs can be the second sun z_{s2} or the planet carrier.

If the second sun is neutral, the ratio n1/nc is from Equation 2 as stated in Figure 3 and Table 2B of (Ref. 24). An example of the Ravigneaux graphic method for this configuration is in Figure 11, as explained in (Ref. 25).

In the other case, the transmission ratio is calculated solving the Equation 3, where the Willis formula is applied as usual for simple epicyclic, when the input is the carrier, the output is the sun and the ring is fixed.

$$\frac{n1}{n_c} = 1 + \frac{z_r \cdot z_{p1}}{z_{s1} \cdot z_{p2}} \tag{2}$$

$$\frac{n2}{n_c} = 1 + \frac{z_r}{z_{s2}} \tag{3}$$

where

n_1, n_2 and n_c are the speed of sun1, sun2 and carrier

z_{s1}, z_{p1}, z_{p2}, z_r, z_{s2} are the number of teeth of the gears (Fig. 10)

Design requirements. The design objectives are two well defined transmission ratios i_1 and i_2.

The constraints are both the geometry (radial and axial max dimensions) and the load capacity, with 5 load spectra

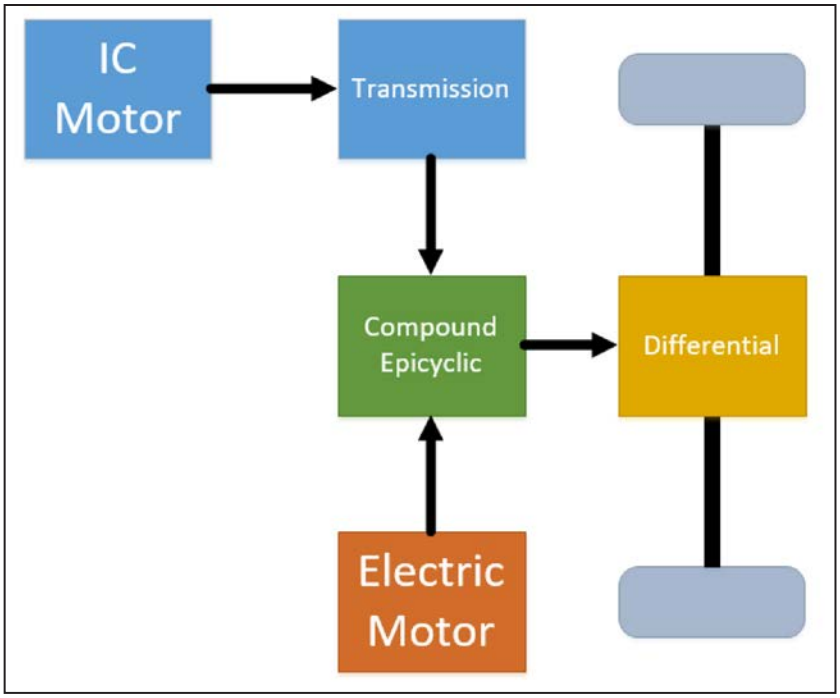


Figure 9 Parallel transmission for hybrid vehicle with IC and E motors.

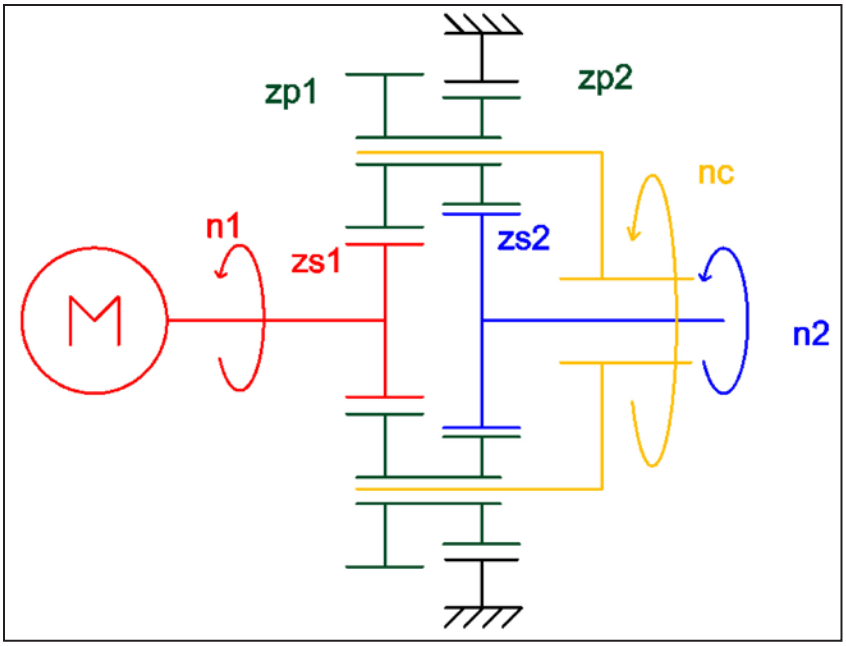


Figure 10 Compound epicyclic; it is used in hybrid passenger cars but also in bikes (Ref. 26).

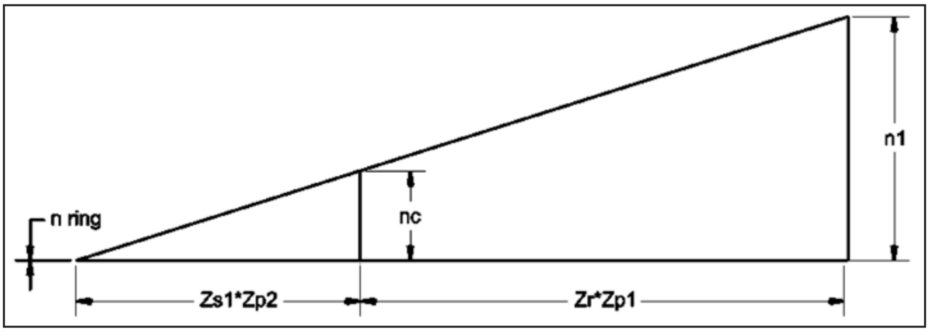


Figure 11 Ravigneaux graphic method for the calculation of the ratio n_c / n_1 of the compound epicyclic of the previous figure.

```
Sub CercaDenti()
    ' DATI
    i1ObjMin = 11.1
    i1ObjMax = 12
    i2ObjMin = 4.1
    i2ObjMax = 5
    Y = 123 ' il massimo è 125, mi tengo largo
    beta = 21
    nsat = 3
    hap = 1.3 ' fattore di addendum (1 standard, >1 dentatura alta)
    zmin = 15
    beta = beta * 3.1415 / 180
    For z5 = zmin To 100 ' sol2
        For z4 = zmin To 100 ' sat2
            For z2 = (z5 + 2 * z4 - 4) To (z5 + 2 * z4 + 4) ' cor (dcor = dsol+2*dsat circa)
                For z1 = zmin To 100 ' sol1
                    For z3 = ((z5 + z4) - z1) - 4 To ((z5 + z4) - z1) + 4 'sat1 (z3 e z4 hanno lo stesso centro, (
                        i1 = 1 + z2 * z3 / z1 / z4
                        i2 = i1 / (1 + z2 / z5)
                        ' rapporti
                        If i1 >= i1ObjMin And i1 <= i1ObjMax And i2 >= i2ObjMin And i2 <= i2ObjMax Then
                            ' fattorizzabile
                            If z1 Mod nsat = 0 And z2 Mod nsat = 0 And z5 Mod nsat = 0 Then
                                ' integrabile
                                If max(z3, z4) Mod min(z3, z4) = 0 Then
                                    ' satelliti equispaziati
                                    If (z5 + z2) Mod nsat = 0 Then
                                        ' calcolo approssimato del modulo dalla relazione Y= a54 + da3/2
                                        m = Y / ((z5 + z4) / 2 / Cos(beta) + (z3 / Cos(beta) + 2 * hap) / 2)
                                        a54 = (z5 + z4) / 2 / Cos(beta) * m
                                        ratio_c = z3 / z1
                                        ratio_e = z2 / z5 + 1
                                        Debug.Print z1, z3, z5, z4, z2, i1, i2, ratio_c, ratio_e, m, a54
                                    End If
                                End If
                            End If
                        End If
                    Next
                Next
            Next
        Next
    Next
End Sub
```

Figure 12 ***Excel*** **macro sample with nested loop for generation of variants. It involves only some geometrical variables, leaving the strength calculation to a next step. A different approach is used by (Ref. 27) to design of minimum volume spur and helical gearsets, considering pitting, bending and scuffing resistance.**

being set in the various operating conditions:
1. Electric drive speed 1
2. Electric drive speed 2
3. Electric coast speed 1
4. Electric coast speed 2
5. IC drive

Out of all the possible solutions, the least noisy one should be chosen.

Design steps. The first phase involved seeking all the combinations of teeth leading to the two transmission ratios.

As in the previous case, a VBA macro was developed on *Excel* to generate and filter variants. This is a series of nested loop on the number of teeth z_{s1}, z_{p1}, z_{p2}, z_r, z_{s2} filtered on assembly requirements: a factorizing planetary geartrain (as described in (Ref. 24)), with planet gears mounted with equidistant spacing. An indication of the module respecting spatial criteria was made by setting the addendum factor and using it to calculate the tip diameters of the various gears (except for the x factor). This enabled to draw up an initial list (Fig. 12).

The choice of a high addendum (HCR) is typical in the automotive industry. It ensures quiet operation without adverse effects on resistance (Ref. 28).

The variants found on *Excel* were then developed in greater detail on a gear analysis and optimization software, to define the profile shift coefficient, balancing the specific sliding and thus to calculate their resistance and the noise level.

In order to rapidly evaluate the noise level of a pair of gears, at least for purposes of comparison with other similar solutions, the Sato formula (Eq. 4) (Ref. 29) was used. It comprises the variables linked to the load (power and speed), the macro-geometry (contact ratio and transmission ratio) and accuracy (dynamic factor). More in-depth calculation would require LTCA (Masuda formula, Eq. 5 from the same reference), but also a timespan too long to compare hundreds of solutions.

$$L = \frac{20\cdot(1-\tan(\beta/2))\cdot\sqrt[8]{u}}{f_v\sqrt[4]{\varepsilon_\alpha}} + 20\cdot\log W \tag{4}$$

$$L = \frac{20\cdot(1-\tan(\beta/2))\cdot\sqrt[8]{u}}{\sqrt[4]{\varepsilon_\alpha}}\sqrt{\frac{5.56+\sqrt{v}}{5.56}} + 20\cdot\log W + 20\cdot\log\tilde{X} + 20 \tag{5}$$

where
- L is the overall noise level at 1 meter from a gearbox in dB(A)
- β is the helix angle
- u is the gear ratio
- ε_α is the transverse contact ratio
- W is the transmitted power in hp
- f_v is the speed factor, analogous to the AGMA dynamic factor v.
- v is the pitch line speed in m/s
- $\tilde{X}$ is the vibration displacement amplitude normalized by static deflection and it can be calculated by TCA

Once the optimal solution (the one respecting geometric requirements, transmitting the two set ratios, maximizing resistance and low noise level) is found, the next step was to define the micro-geometry as in the case of the electric transmission, while avoiding modifications of the teeth on the internal crown.

In the case of the hybrid transmission, the efficiency map was newly compiled for 2 temperatures (40°C and 80°C) but in all the 5 operational cases indicated previously.

Further development

The analyses required did not include the optimization of contact for the bevel gears. In any case, the same approach can be used as for cylindrical gears: exchange of data between the calculation of deformation of shaft and gears from the software (Ref. 10) to (Ref. 30) for LTCA and definition of the microgeometry, as described in (Ref. 31). The bevel planet of the differential gears should be calculated with a static (fixed) torque, independent from the motor: the slip torque of the wheels.

A next step for a faster optimization could be the definition of an object metric to evaluate the contact pattern, i.e. the percent of full contact or the position of the max pressure in the grid of the tooth flank.

Conclusions

The paper is not the proof of a discovery, but it is the description of a method: the optimization of the microgeometry for cylindrical gears. The method has been applied and described on some transmissions with helical gears and compound epicyclic, used on different hybrid vehicles. However, the method is also valid for industrial gearboxes.

Since the objective is "micro," it has been seen that it is necessary to pay attention to the "smallest" causes of deformation of the geometry to optimize, in particular the deformation of shafts, bearings, housing as well as of tooth as a cantilever on flexible gear body.

It has been seen that the objectives are manifold and of different types: on the one hand the safeties against bending and pitting, which are indicators summarizing the load history, and on the other hand the trend of the transmission error, of the maximum pressure, of the noise and of the efficiency over the load.

It has also been pointed out that the deformations taken into consideration also influence the backlash between the teeth in operation, which must therefore be verified and guaranteed at the maximum torque, the condition that further reduces the backlash.

All these calculations are possible with commercial software already known and widespread.

Finally, in the case of a planetary compound, a simple algorithm for optimizing the macrogeometry was proposed, introducing the existence of multi-objective optimization software.

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