Optimization of Gear Microgeometry for Durability and NVH

M. Platten, **C. Blockley**, **B. James**, **S. Prabahakaran**, and **D. Scott**
Romax Technology Ltd., Nottingham, United Kingdom

Microgeometry design is a fundamental part of gear design affecting both durability and NVH quality, but it is often left until late in the process. A lack of suitable and accurate analysis tools leads to low confidence in predictive design that in turn leads to over-engineered gears, which are necessarily biased toward durability. As a result, NVH quality tends to come off second best and may not even be considered at all.

In automotive applications, it is common for a gearbox to be designed to meet some duty cycle. The gear designer is then presented with a condensed version of the duty cycle. Using this "worst-case" approach to gear design may be quicker, but it has the disadvantage in that the variation of durability and NVH performance with load is not taken into consideration.

A "typical traditional approach" to microgeometry deign would be as follows:

- Design gear macrogeometry for maximum load (possibly with some consideration for NVH, but often not).
- Determine gear durability based on maximum load and duration only.
- Apply some default microgeometry in the hope of avoiding tooth contact durability problems.
- Hope it is not too noisy!
- Refine microgeometry by experience/experiment and repeated prototyping.

Introducing an analysis step in this naïve process enables calculating deflections of the system under load, giving accurate predictions of gear misalignments. This has two main benefits.

- It gives a more accurate estimate of the durability of the gears, allowing safety factors to be reduced with greater confidence.[1]
- It also offers clues to the microgeometry modifications that may be required to correct for misalignment.

However, it still does not take into consideration variations in load and misalignment and can only give a vague qualitative indication of tooth contact behavior of the gears. This must subsequently be validated by a gear tooth marking test on a prototype gearbox.

Further enhancements to the gear design process – namely detailed analysis of the gear tooth contact behavior across the whole range of operating loads – would provide the engineer with valuable information that could be used to design better microgeometry as well as give a much more accurate prediction of gear durability.

However, there is a potential problem with this information – there is too much of it. The use of automated optimization tools can help the engineer cope with this glut of data. Optimization requires multiple calculations of the gear contact behavior, and the analysis must therefore be fast and accurate.

Transmission Model

An image of the model used for this study is shown in Figure 1. This represents a five-speed, front-wheel-drive, manual-transaxle gearbox. The model is constructed in RomaxDesigner software.[2] All subsequent analyses are also performed with this software.

The RomaxDesigner model is a fully detailed static and dynamic model of the gearbox and includes shafts, bearings, gears, synchronizers, differential and housing. The analysis that follows is concerned solely with the microgeometry design of the third-speed gear pair.

A duty cycle histogram for the third gear has been calculated from road load data and is shown in Figure 2. The nominal torque capacity for this transmission is 170 Nm. The duty cycle includes both drive and coast conditions and a number of shock loads that exceed the nominal capacity for short periods of time. A condensed design load duty cycle that comprises a single-load case (at maximum nominal load) is derived. This attempts to represent the equivalent damage of the full duty cycle (170 Nm for 17 hours).

Analysis Method

Basic Rating. A basic ISO 6336 rating of the gear that may be carried out by a gear designer takes into account the basic macrogeometry properties of the gear and the load applied to it. No misalignment of the gear is taken into consideration and the peak load factor K_H is assumed to be 1. The condensed duty cycle yields the same result as the full duty cycle when the most basic rating is used.

Misalignment. To calculate a more accurate and reliable rating for the gears, the misalignment of the gear pair must be taken into consideration. To calculate the misalignment of the gear mesh, the deflection of the gears under load must be calculated. The most accurate way to do this is to apply the torque load to the RomaxDesigner model and solve iteratively to determine the deflections of the entire system. The resulting displacements of the gear mesh can then be resolved into the line of action of the gear pair to obtain the misalignment.

This static analysis step must be performed for each loading condition separately. Solving the entire system is necessary, since the flexibility of all components, including the housing, have an influence on the displacement at the gear mesh.

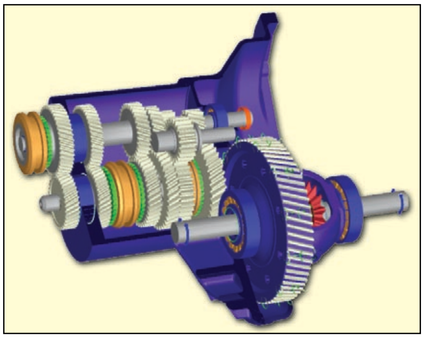

Figure 1. Model of five-speed manual transmission with housing cut away.

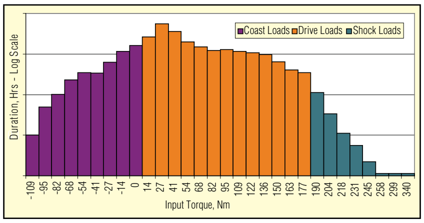

Figure 2. Duty cycle from road load data.

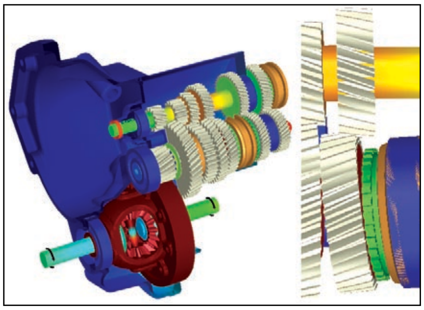

Figure 3. Static deflection of the model at 170 Nm.

The system deflections at the design load of 170 Nm are shown in an exaggerated form in Figure 3. Also shown is a detail highlighting the misalignment at the third-speed gear mesh due to the flexibility of the entire system.

The misalignments of third gear for the whole detailed duty cycle are shown in Figure 4. There is an approximate linear relationship between torque and misalignment for the drive loading conditions.

The ISO 6336 rating including the effects of misalignment can be calculated by assuming a simplified function for the gear mesh stiffness. For the condensed duty cycle, the damage is 51% and 40% for the pinion and wheel respectively. When the complete duty cycle with the variation in misalignment is used to calculate contact durability, the damage increases still further to 68% and 53% for the pinion and wheel respectively. Clearly, the effect of including misalignment has a major impact on the durability of the gears. In addition, the difference between the results for the condensed and full duty cycles shows that the effect of misalignment variation also serves to ensure that the condensed duty cycle is no longer representative of the full duty cycle from a damage point of view. These effects are summarized in Table 1

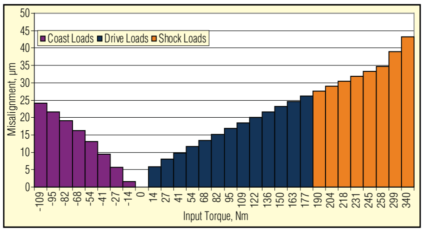

Figure 4. Third-gear misalignments for complete duty cycle.

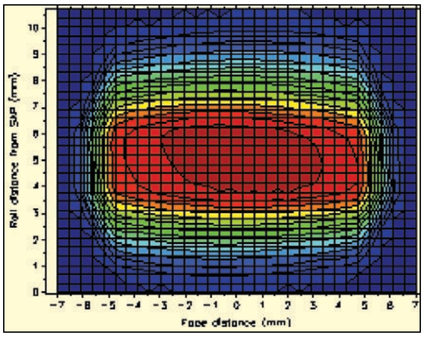

Figure 5. Contact load distribution at design load for manually designed microgeometry.

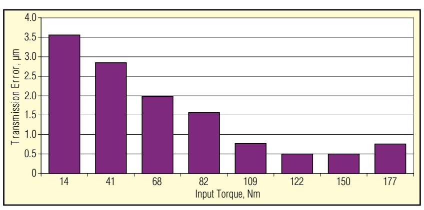

Figure 6. Variation of TE with load for manually designed microgeometry.

Optimization

The aim of microgeometry optimization is to ensure good distribution of load across the tooth face while simultaneously keeping the transmission error (TE) low across as much of the operating range as possible. TE is the source of gear whine.[3] It is caused by the non-conjugacy of motion in the gear pair due to a combination of misalignment and the deflection of gear teeth under load. Unfortunately, the modifications that are often required to maintain a good contact for durability are not always conducive to low TE (good NVH performance) and some sort of balance must be struck.

As the misalignment varies with load, there is an additional problem that this balance must be maintained across a range of misalignments. Effectively the gear must be designed so that it is insensitive to misalignment variation.

Manual Optimization. Returning to the "traditional approach" to gear design described previously, a process for designing microgeometry can be defined (remembering that the engineer is only considering what is happening at the design load for the condensed duty cycle):

- Apply lead correction to counteract misalignment at the design load (assuming misalignment is known).
- Apply some crowning to account for the unknown variation in misalignment.
- Apply some tip/edge relief to ensure tip/edge contact does not occur.

Table 1. Summary of gear damage.

Duty Cycle	Misalignment	Contact Damage Wheel	Contact Damage Pinon
Condensed	No	6%	5%
Full	No	7%	6%
Condensed	Yes	51%	40%
Full	Yes	68%	53%

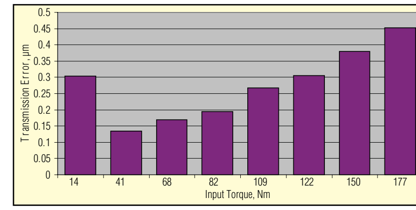

Figure 7. Variation of TE with load for automated microgeometry design.

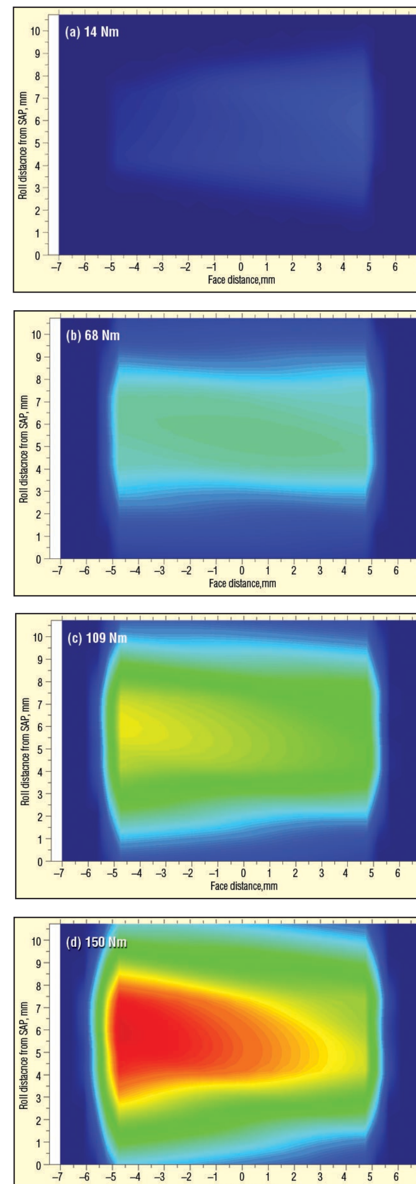

Figure 8. Contact load distribution across load range for automated microgeometry design.

This approach has been used to design microgeometry for the third-gear pair in the transaxle model. Using the RomaxDesigner software, the effect of the new microgeometry design on the durability and TE can be assessed.

Because microgeometry is now defined, further improvements in the accuracy of the gear rating can be made by analyzing the detailed contact behavior of the gear teeth. This gives accurate information about the load distribution and tooth stiffness and also allows the TE to be calculated.

The contact damage for the condensed duty cycle is now 56% and 44% for the pinion and wheel respectively. This does not necessarily imply that the microgeometry design is worse than the design with no microgeometry even though the contact damage has increased. Rather, it is an indication that the more detailed analysis allows a more accurate rating to be calculated.

The contact load distribution for the manually designed microgeometry is shown in Figure 5. It shows that the design has successfully centred the contact and that tip and edge contact have been avoided. However, the maximum load on the tooth is quite high due to the reduced area of contact. The TE calculated for the rated load is 0.7 µm, which is relatively low.

Of course the apparent "success" of this design is only valid for the rated load. The effects of the new design when the full duty cycle is considered can be examined. This reveals that the design is not "successful" at all. The contact damages are calculated to be 82% and 64% for the pinion and wheel respectively, and the TE (Figure 6) at lower torques is unacceptably high.

Automated Optimization. The RomaxDesigner software includes automated optimization tools to assist in microgeometry design. The automated optimization needs to combine the potentially conflicting demands of different targets (durability, NVH etc.) and this is done by a cost function that can be controlled by the user.

For a first attempt at automated optimization of the third- gear pair, the cost function was defined as follows

- TE – low TE increases score.
- Load weighting – TE in higher load conditions given slightly lower weighting (gear whine noise is often perceived to be less of a problem at high load).
- Edge/tip/root contact – maximum contact load close to tooth perimeter greatly reduces score.

The results of this optimization are shown in Figures 7 and 8. The TE is very low across the whole torque range, indicating that the NVH quality of the design is good. The contact load distributions show that the optimizer has avoided edge contact by introducing end relief and the load is distributed over a wide area of the tooth, reducing overall peak load. As a result, the contact damage for the full duty cycle has reduced considerably (pinion 33%, wheel 26%).

Refined Optimization. Further refinements can be made to improve the design. The cost function can be adjusted by

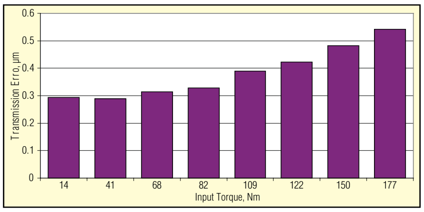

Figure 9. Variation of TE with load for refined microgeometry design.

including a factor that biases the result so that maximum contact load is close to the center of the tooth. This is likely to force the optimizer to increase the amount of crowning, which may in turn have a detrimental effect on the TE. But since the TE is already very low for a transmission of this type, it is acceptable to sacrifice a little NVH performance for a design that is better overall.

The results of this refined optimization are shown in Figures 9 and 10. The contact load distribution shows an improvement in the sense that the contact is more centralized across the load range, and the peak load has been reduced at the highest torques. But there is a slight increase in the peak load at lower torques due to the reduction in contact area caused by the additional crowning. The effect of this can be seen in the full duty cycle contact damage, which has increased to 48% and 38% for the pinion and wheel respectively.

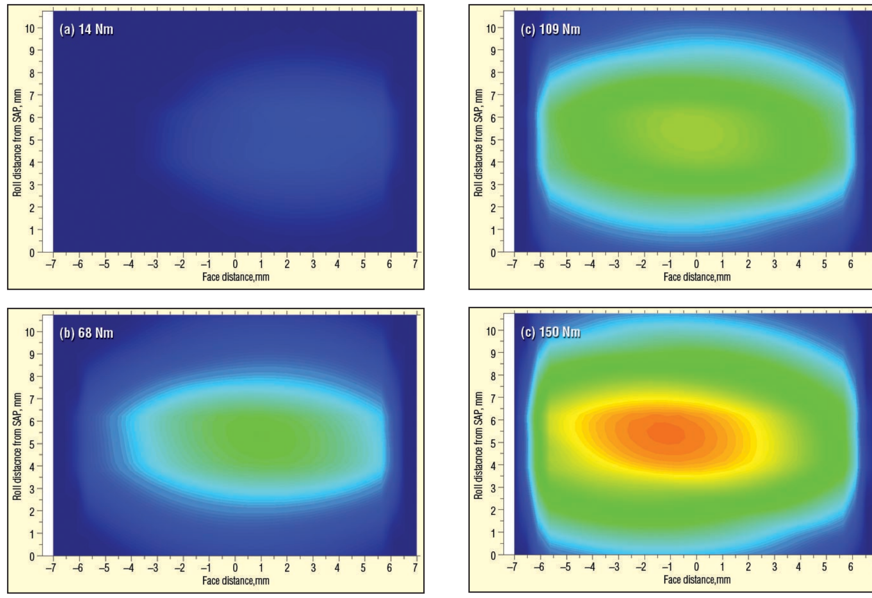

Figure 10. Contact load distribution across load range for refined microgeometry design.

A secondary effect of crowning is that it tends to increase TE at low loads due to the deviation of the gear surface from the ideal involute. This is confirmed by the TE results, which show a proportionally larger increase in TE below 100 Nm than at higher torques when compared with the previous optimization results. However, the TE is still low and falls well within what would be considered the "quiet" range. SV

The author can be contacted at: romax@ignitionpr.co.uk.