# The Effect of Flexible Components on the Durability, Whine, Rattle and Efficiency of an Automotive Transaxle Geartrain System

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**Management Summary**

Gear engineers have long recognized the importance of considering system factors when analyzing a single pair of gears in mesh. These factors include important considerations such as load sharing in multi-mesh geartrains and bearing clearances, in addition to the effects of flexible components such as housings, gear blanks, shafts and carriers for planetary geartrains. However, in recent years, transmission systems have become increasingly complex—with higher numbers of gears and components—while the quality requirements and expectations in terms of durability, gear whine, rattle and efficiency have increased accordingly. With increased complexity and quality requirements, a gear engineer must use advanced system design tools to ensure a robust geartrain is delivered on time, meeting all attribute, cost and weight requirements. As a standard practice, finite element models have traditionally been used for analyzing transmission system deflections, but this modeling environment does not always include provisions for analysis of rattle and efficiency, nor considerations for attribute variation, which often require many runs to be completed in a short timeframe. An advanced software tool is available for the analysis of transmission system durability, whine, rattle and efficiency—all within a single programming environment, including the effects of flexible components such as housings, gear blanks and shafting. An example transaxle case study is examined here in detail.

## Introduction

Throughout the gearing industry, the natural progression of higher consumer expectations requires that gear design engineers be tasked with creating quieter, more durable and efficient designs while at the same time reducing costs and development time. Previous accepted practices of optimizing a gear pair independently of the intended application or "system," performing expensive and time consuming durability and noise/vibration testing of system prototypes, then adjusting the gear designs accordingly before repeating the testing cycle, is quickly becoming impractical and unaffordable. Companies simply do not have the resources, especially during an economic downturn, to rely on prototype testing to drive the geartrain design. Testing should only be utilized as a final verification of a design optimized using various statistical methodologies in conjunction with state-of-the-art geartrain system computer-aided engineering (CAE) analysis tools (Refs. 1–2).

These advanced CAE tools have been shown to allow for prediction of the system gear whine performance of a complex automatic transmission used in an automotive application (Refs. 3–4). The predictions included static transmission error of a planetary gearset accounting for the effects of time-varying factors, such as load sharing and carrier deflections, mode shapes and natural frequencies, absolute levels of vibration due to the gear mesh forces and manufacturing variation due to microgeometry variation.

Additional studies using the advanced geartrain system CAE tools included analysis of the high-mileage gear whine performance of an automatic transmission, as well as microgeometry inspection methods used to accurately represent the actual planetary gearset hardware (Ref. 5). Predictions of high mileage performance are important to several industries for varying reasons: for automotive applications, the residual value of previously owned vehicles can be negatively affected by the presence of passenger compartment gear whine, even if the noise itself is not indicative of an impending gear failure; for aerospace applications, the rate of gear wear due to geartrain system effects can be critical to designing a robust gearset beyond just following basic gear standards.

Further studies using the same geartrain system CAE tools have shown the importance of including representative boundary conditions, such as the driveline downstream inertia and gearbox housing loads, and the resulting effect on noise, vibration and durability predictions (Ref. 6). Clearly, the flexible housing containing the geartrain was a critical component, enabling the correct mesh misalignment to be predicted as part of the total system; therefore, allowing a more robust non-linear gear contact study to be performed. Additional investigations also showed that the downstream effects of the durability rig (inertia, dynamics) can inadvertently affect the outcome of the durability testing itself when compared to how the geartrain would perform in the actual vehicle. The study demonstrated that durability rig testing—without proper analysis—may provide an incorrect indication of actual durability performance, possibly leading to unexpected failures in the field. An issue not clearly demonstrated for geartrain systems such as transmissions and transaxles used in various industrial applications is the need for including flexible components as part of the system analysis, specifically for analysis of performance attributes such as gear durability, whine, rattle and total system efficiency with predictions for individual component efficiency contributions. For transmissions with rigid housings, explicitly designed to not deflect significantly even under high geartrain loads, perhaps the flexibility of the housing is not so critical for making accurate gear mesh misalignment predictions, for instance. However, for applications where the gearbox housing is optimized for weight using materials such as aluminum and magnesium with thin-walled designs, housing flexibility becomes exceedingly important when analyzing geartrain deflections—not only for high loads, but across a wide range of loading conditions as well.

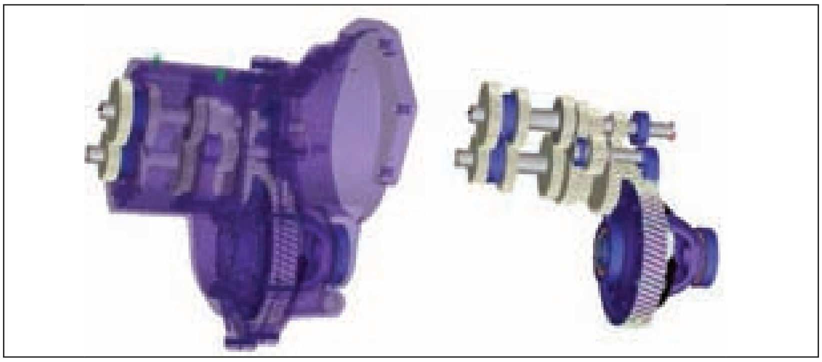

**Figure 1—Advanced CAE transaxle system model, with and without the housing.**

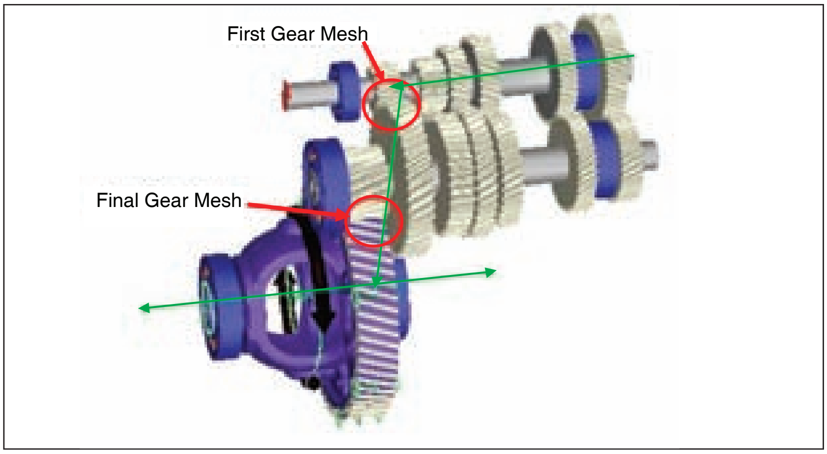


**Figure 2—First gear power flow (green) and mesh locations (red) for transaxle system model.**

This paper will investigate the housing flexibility issue using a generic manual transaxle used in an automotive application as an example. The transaxle was modeled using the advanced CAE tool previously referenced (Refs. 1–6), both with and without the housing, as shown in Figure 1. All gear, bearing and shafting details were the same, except that the outer bearing race connections to the condensed finite element model of the housing were set to ground for the configuration without the housing. Therefore, the differences

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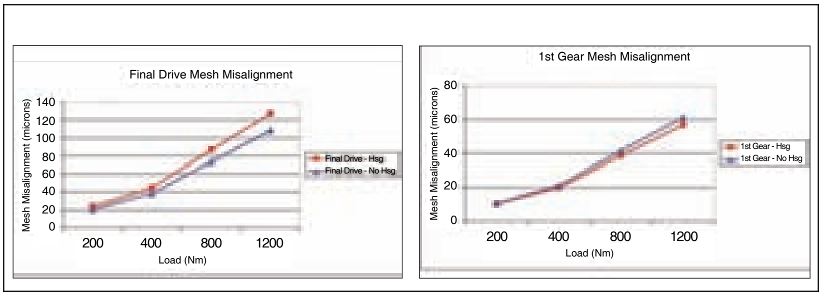


**Figure 3—First gear and final drive mesh misalignment predictions, with and without the housing, various loads.**

**Table 1—Misalignment contribution analysis, with and without the housing, final drive mesh, 1,200 Nm.**

| 1st Speed | Hsg | No Hsg |
|---|---|---|
| Final Drive Pinion → Final Drive Wheel Gear Mesh | **-127.05** | **-108.04** |
| Final Drive Wheel<br>Gear<br>Gear Bearing Outer<br>Gear Bearing Inner<br>Support Shaft<br>Bearing Inner<br>Bearing Outer (Hsg) | **-24.29**<br>0<br>0<br>0<br>8.32<br>0.68373<br>-33.3 | **-7.92**<br>0<br>0<br>0<br>6.25<br>1.68<br>0 |
| | | |
| Final Drive Pinion | **-102.76** | **-115.96** |
| Gear | 0.87111 | 0.82193 |
| Gear Bearing Outer | 0 | 0 |
| Gear Bearing Inner | 0 | 0 |
| Support Shaft | -117.37 | -117.12 |
| Bearing Inner | 0.31704 | 0.33041 |
| Bearing Outer (Hsg) | 13.42 | 0 |

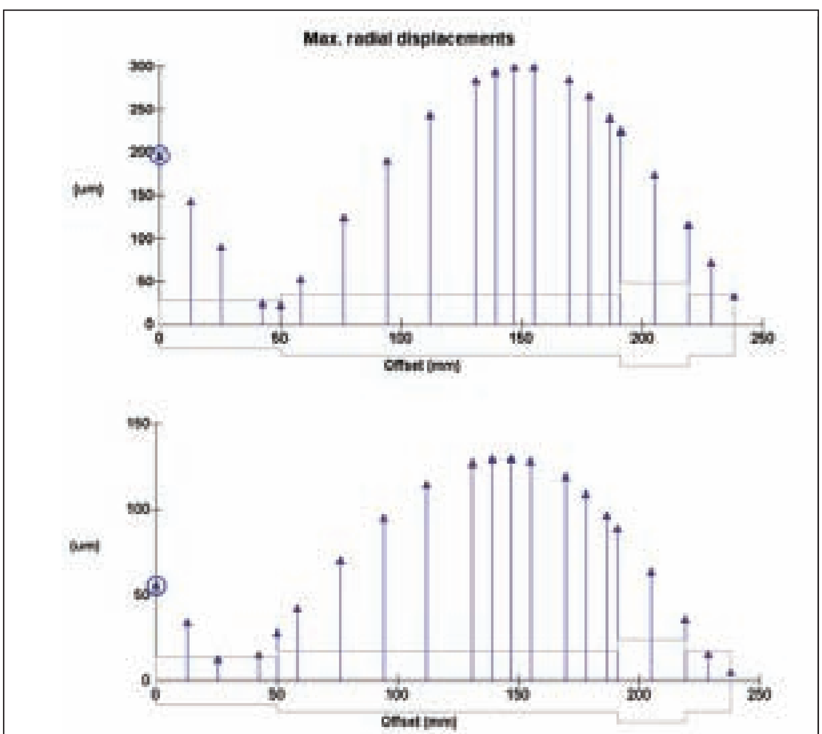


**Figure 4—Lay shaft deflections, 1200 Nm, with (top) and without (bottom) the housing.**

between the performance attributes analyzed and presented below represent the effect of the housing. Additional capabilities inherent to the inclusion of the housing as part of the geartrain system analysis will also be demonstrated.

### Mesh Misalignment

For the purposes of the mesh misalignment investigation, the aforementioned transaxle was analyzed with the power flow of the system set through first gear only, predicting the alignment effects at the first gear and final drive mesh locations, as indicated in Figure 2.

The geartrain was subjected to loading conditions covering light to heavy throttle in an automotive application, both with and without the housing. The resulting mesh misalignments were predicted using calculations encompassing the fully coupled, six-degree-of-freedom system model for each configuration. The mesh misalignment predictions are shown in Figure 3. A more detailed analysis of the 1,200 Nm load case shows the contribution to the mesh misalignment from individual components and the associated clearances and deflections for each configuration as shown in Table 1.

The importance of including the flexible housing as part of a fully coupled transaxle system in the mesh misalignment predictions can therefore be substantiated analytically, providing opportunities to manage undesirable misalignment as a system, rather than immediately assuming options are either microgeometry modifications, such as crowning, or housing stiffness actions, such as adding ribs. Perhaps changing the shaft material properties or dimensions would be a more feasible and effective solution, or perhaps a combination of all approaches. Figure 4 shows the lay-shaft deflections, for example, with and without the housing influence at 1,200 Nm, demonstrating a substantially lower deflection of the shaft with the bearings set to ground. Using statistical methods such as Design of Experiments (Refs. 1–2), the mesh misalignment can be managed objectively.

### Transmission Error and Contact Patterns

The foundation of a successful non-linear gear mesh contact analysis is to fully understand and quantify the relative positions of the two meshing gears (Ref. 7). Determining the housing influence on the misalignment predictions is therefore a prerequisite for accurately predicting static transmission error and the load distribution throughout a tooth mesh cycle. For the theoretical gears used in this investigation, five microns of lead crowning and involute barreling were added to both the first gear and final drive gear pairs in order to avoid some level of edge loading over the wide range of geartrain torques applied. No other significant microgeometry modifications were used in the analysis.

Table 2 lists the peak-peak static transmission predictions, as well as the first three harmonics for the 400 Nm load case of the previous misalignment study, with and without the housing influence. From a system dynamics standpoint, clearly the housing is needed in order to follow any quality function deployment (QFD) process for gear whine. This is accomplished by factoring in the customer requirements cascaded to vibration targets at a system housing location (the QFD process for gear whine is clearly outlined in Ref. 1), then proceeding to cascade to the subsystem, and finally to the component level. A QFD example for gear whine is given in Figure 5.

An example of predicted housing vibration due to the first gear mesh order, the "system" part of the QFD process, exerted to 400 Nm of output load, is shown in Figure 6.

Without the housing, a gear designer will typically attempt to minimize the transmission error without factoring in details of the system influence under all design loads, which includes the "path" between the mesh excitation creating forces and related vibration along the shafting, through the bearings, thus forcing the housing to vibrate at the mesh frequency. However, without the appropriate boundary conditions, including the housing influence, the source optimization process (e.g., static transmission error) cannot be properly implemented without some level of risk. Even the geartrain "subsystem" dynamics cannot be confidently evaluated, either in terms of amplitude or frequency content, without the effects of the gearbox housing influence as evidenced by the dynamic transmission error predictions shown in Figure 7.

Furthermore, including the housing effects in the transaxle system analysis allows examination of various mode shapes that could potentially negatively affect the housing vibration. Presenting in terms of displacement, strain and kinetic energies allows the entire transaxle design team to work together in order to find a solution to desensitize the transaxle to

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**Table 2—Static transmission error: peak-peak, harmonics and percentage difference, final drive—with and without the housing.**

| | Final Drive, Hsg | Final Drive, No Hsg | Final Drive, % Diff |
|---|---|---|---|
| TE (pk-pk) | 2.93 | 2.45 | 20 |
| TE (1st harmonic) | 1.44 | 1.21 | 19 |
| TE (2nd harmonic) | 0.1 | 0.07 | 43 |
| TE (3rd harmonic) | 0.07 | 0.01 | 600 |

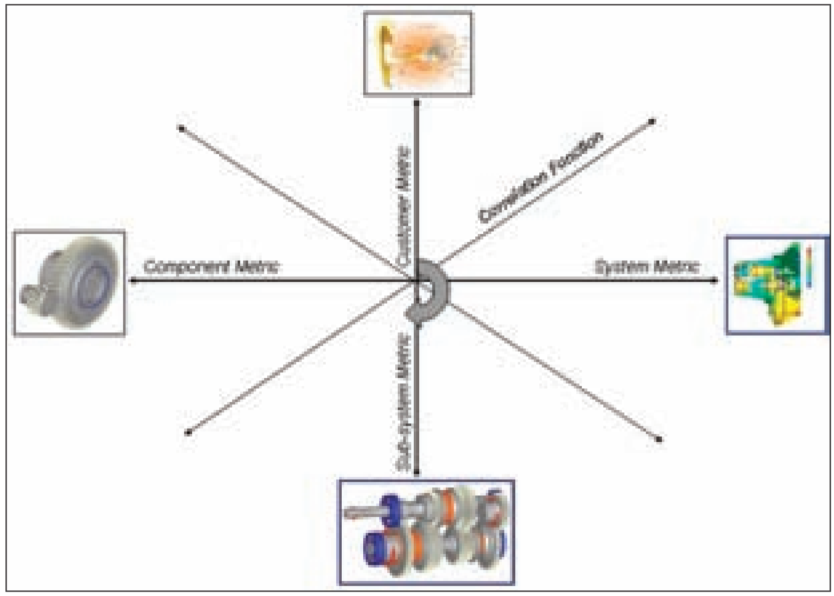


**Figure 5—Quality function deployment (QFD) plot for management of system gear whine.**

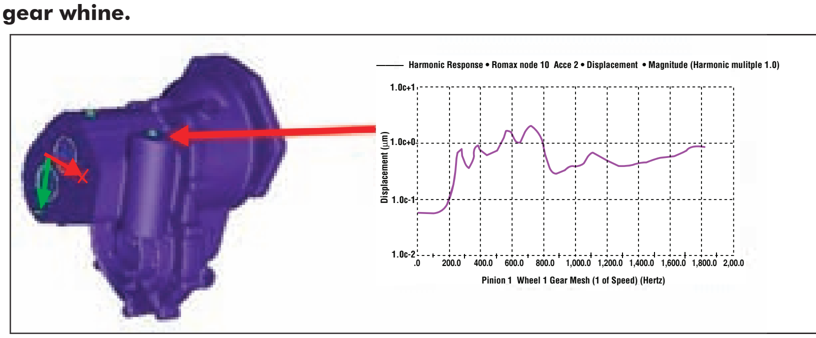


**Figure 6—Predicted housing vibration due to gear mesh vibration, 400 Nm.**

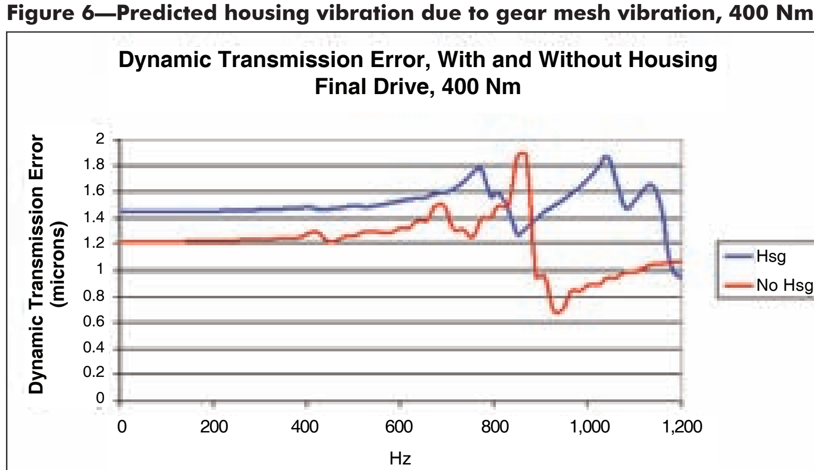


**Figure 7—Dynamic transmission error—first harmonic, final drive, 400 Nm—with and without the housing.**

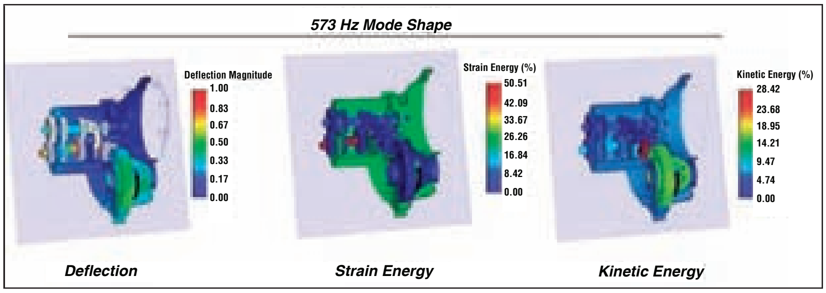


**Figure 8—CAE model of transaxle, 573 Hz mode—displacement, strain energy and kinetic energy—includes housing influence.**

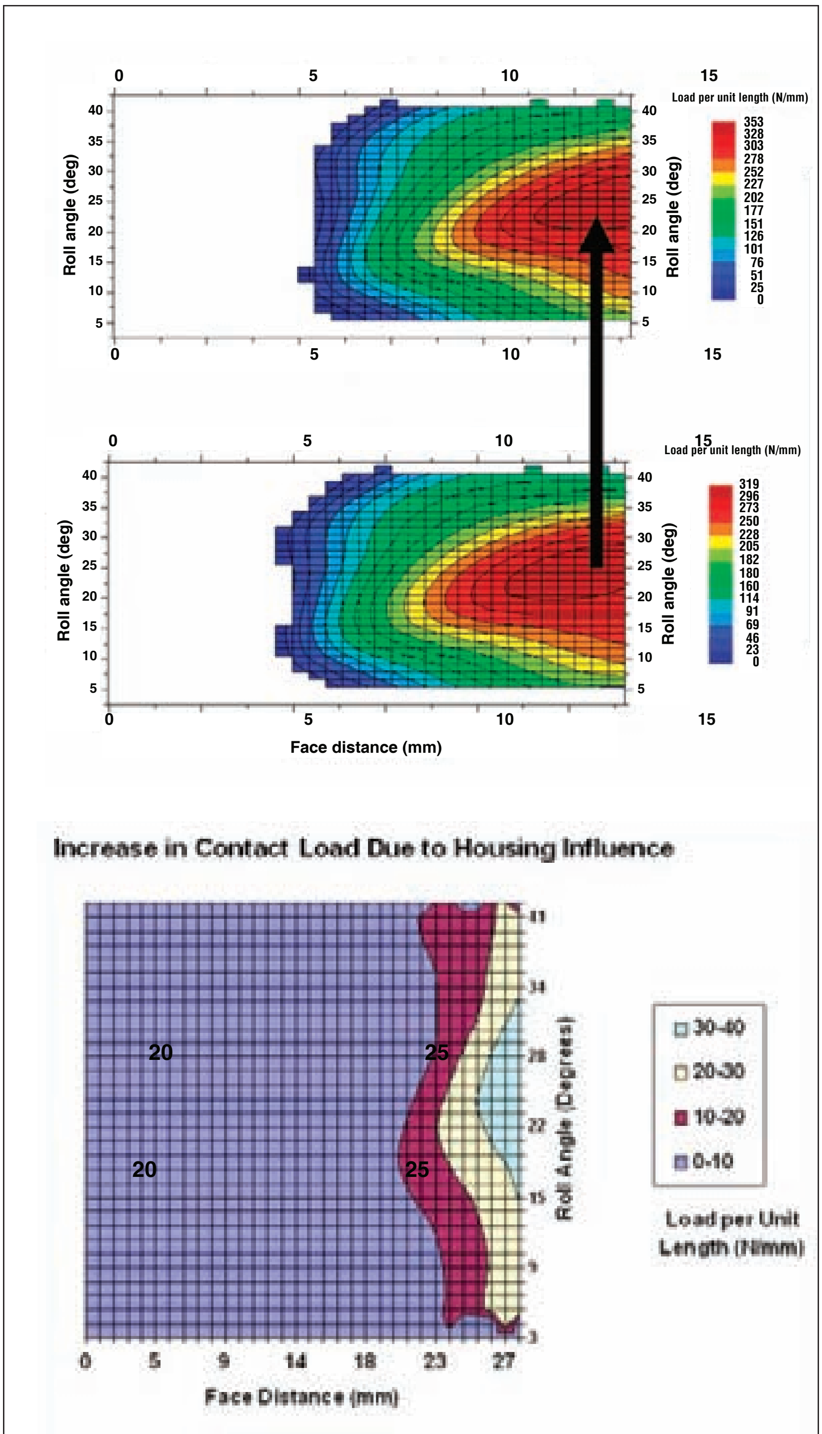


**Figure 9—Contact patterns, final drive gear mesh, 400 Nm, with (top) and without (bottom) housing.**

the inherent static transmission error excitations (the transfer functions in the lower QFD quadrants). An example of such a CAE analysis is shown in Figure 8.

In order to optimize the load distribution, reviewing static transmission error values is of course not sufficient. Standard practice is to review load distribution plots for a complete tooth mesh cycle. Again, the effect of the housing influence is evident by comparison of the plots in Figure 9, showing the load distribution for the final drive gear mesh for both configurations, exerted to 400 Nm half-shaft torque. With the flexibilities of the housing, the final drive gearset is demonstrating slightly more edge loading and a higher load-per-unit length than when considering bearings restrained to ground using fully coupled six-degree-of-freedom calculations for both instances.

The implications of an incorrect contact pattern analysis may result in the specification of unnecessary or overaggressive microgeometry modifications—especially for higher loads—as the difference in mesh misalignment between housing/no-housing configurations increases, as previously shown in Figure 3. As the gears are modified to accommodate higher loads, often the contact at lighter loads is compromised, resulting in increased static transmission error and subsequently higher levels of passenger compartment gear whine.

**Durability**

Traditionally, durability performance is the gear designer's first priority, and since this irrefutable, self-evident requirement has been in place for so many years, with an abundant effort by thousands of engineers and researchers worldwide for more than 100 years, it can be perplexing that gear failures are still all too common of an occurrence. The practical issue facing gearbox design engineers is that the gearbox performance requirements seem to constantly push the design technology. For example, within a few short years, the automatic transmission used in automotive applications has increased from various four-speed combinations to eight speeds and beyond. Since most major transmission OEMs produce durable products, many with warranties up to 100,000 miles, the other performance attributes have become the true differentiators, putting durability design activities in direct competition with noise and efficiency efforts. Using traditional methods of optimizing for durability first, followed by a secondary effort for noise and efficiency, is no longer feasible for some industrial applications, such as automotive. For others, such as aerospace, durability will remain the primary concern, but even in this industry, noise and efficiency are becoming more prevalent.

The cornerstone to any geartrain durability analysis, whether performed using advanced CAE system tools or on a test rig, is the development of representative duty cycles to accelerate the extremes of the wear expected in the field. Duty cycles will vary by application, industry and company, using both experience-based and statistical-based tools to develop the most efficient approach. For this investigation, the duty cycle used on the generic manual transaxle being studied was developed based on previous experience, but it is not intended to be fully correlated to the actual hardware used by the customer, since this transaxle is only a derivative of an actual transaxle. But for illustration purposes, the same model used for the misalignment and vibration studies was used for the durability study in the same programming environment, both with and without the effects of the housing influence. The results are shown in Table 3.

The substantial difference between the two durability life predictions for the final drive mesh can be directly attributed to the flexibilities of the housing as part of the transaxle system, as previously explained. For this reason, the gear design engineers and the transaxle system engineers should work together to ensure any geartrain durability analysis includes provisions for the entire system.

Oftentimes, the complete transaxle design—including the housing—is not finished by the time the geartrain design requirements are needed to satisfy production and manufacturing timelines. With compressed timing on the delivery of new transaxle designs in the automotive environment, for example, it is not unusual for the manufacturing plants to order the gear tooling and determine the final production manufacturing processes before the first prototype has been tested. In situations like this, which are unfortunately becoming more common, the need for a CAE-based transaxle system design tool for durability analysis—and for all attributes for that matter—becomes even more prevalent.

**Efficiency**

Conservation of energy and concern for the environment have become a central area of worldwide focus in recent years. Geartrain technology plays an integral role in helping world communities succeed in the goals being established. For example, gearboxes can contribute to reducing greenhouse emissions through helping make automotive powertrains more fuel-efficient by mechanically coupling wind energy to electric generators—thus reducing the need for new coal burning plants—and by being used in propulsion vehicles used for public transportation. Also, more fuel-efficient powertrains require less fuel, reducing consumer fuel costs and thereby increasing vehicle residual values.

Efficiency calculations were performed using the ISO 14179-2 (US) standard for gear and bearing drag on both geartrain configurations, with and without the housing influence. Options for

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**Table 3—Durability results, first gear and final drive, with and without housing.**

| | Hsg Combined Contact/Bending Life (hrs) | No Hsg Combined Contact/Bending Life, hrs |
|---|---|---|
| Wheel 1 | 3.5 | 4.1 |
| Pinion 1 | 1.0 | 1.2 |
| Final Drive Wheel | 173.6 | 323.1 |
| Final Drive Pinion | 42.7 | 79.5 |

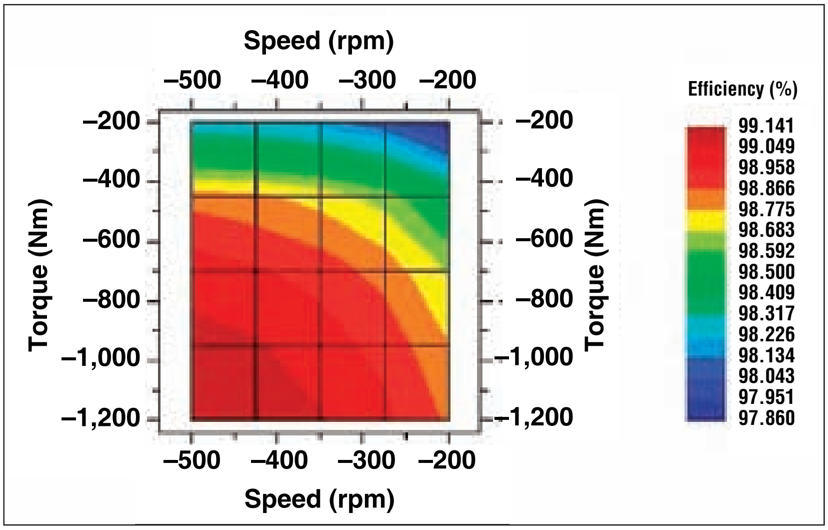


**Figure 10—Efficiency map.**

**Table 4—Efficiency result examples: total system, first gear and final drive. Total efficiency = 98.60%.**

| | Watts |
|---|---|
| Total Gearbox Losses | 183.4 |
| 1st Gear Mesh | 50.7 |
| Final Drive Mesh | 59.5 |
| Input Shaft Left Brg. | 25.4 |

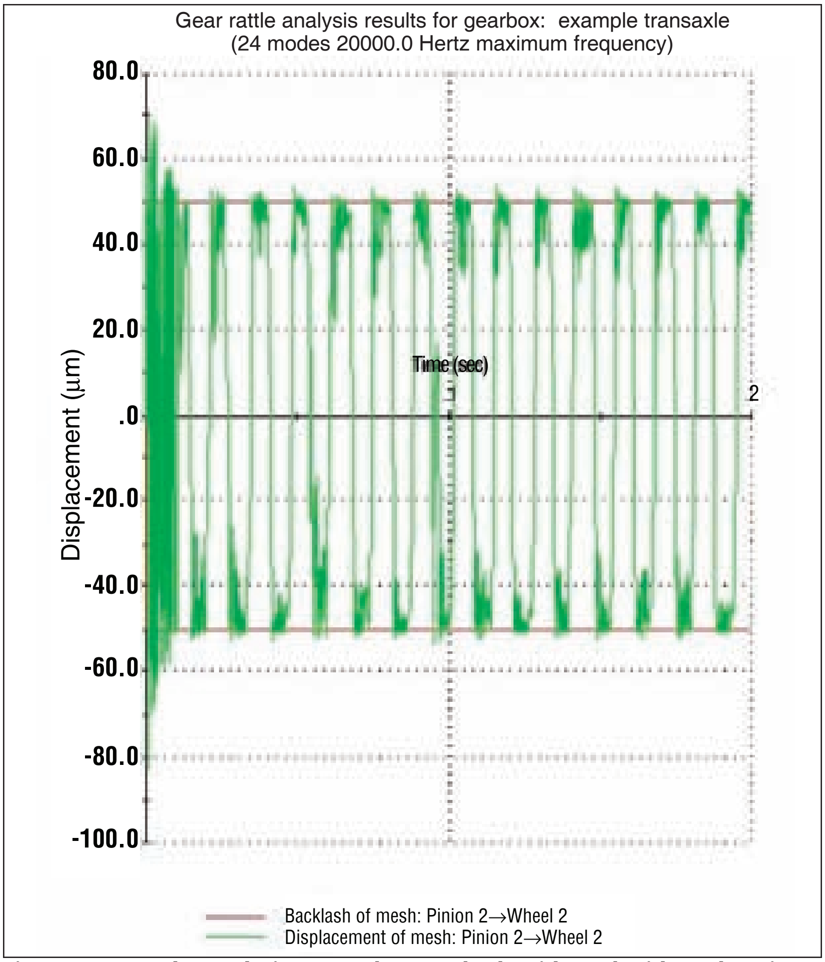


**Figure 11—Rattle analysis, second gear—both with and without housing—demonstrated a similar trend (with housing is shown).**

including the oil fill level were also included, even though this would of course not be possible without the housing in actual hardware evaluations. However, since the standard does not use the microgeometry for the calculation, no change was predicted, as expected. Table 4 shows an example of numerical results for 400 Nm. Figure 10 shows an efficiency speed/torque map.

While the equations do not show any differences between configurations, in reality, slight differences could be attributed again to the mesh misalignments and the bearing loading induced by the effects of the housing flexibility. Essentially, the efficiency can be predicted not just for the entire transaxle system but also for individual component contributions, allowing the transaxle design engineer to objectively quantify design iterations, such as when changing gear designs, oil viscosity, bearing, etc. However, it stands to reason that the housing influence should eventually be factored into the calculations in order to predict results reflective of the actual hardware behavior, which is due to mesh losses caused by misalignment-induced contact variations, compared to conditions with little misalignment (Ref. 7).

### Rattle

Traditional methods for dealing with a geartrain rattle issue were to build prototype transmissions, transaxles and engine gear accessory drives following standard practices for such designs, test the prototypes and then subjectively evaluate various operating conditions for any objectionable rattle. If a rattle condition were discovered, usually at a substantially late date after the geartrain design has long been finalized, and production is fast approaching, the development engineer will resort to swapping parts one at time, hoping to alleviate the rattle issue. For manual transmissions and transaxles, this usually means tuning the damper springs. For engine accessory drives and "live" power take-off units, the challenge can be more substantial, often looking for a combination of effects, such as increasing bearing and gear drag, adding inertia to the system at strategic locations, changing gear backlash values and, if all else fails, adding scissor gears. Quite often, these design actions pose a risk to gear whine, efficiency and durability.

To help estimate the potential effectiveness of any proposed design actions, the CAE team would be asked to build a simple torsional degrees-of-freedom model to investigate the different combinations. For light loads, this is usually sufficient since smaller geartrain deflections are occurring, containing the issue to the geartrain subsystem (gears, shafts and bearings, modeled as lumped masses and inertias) (Ref. 8). For medium-to- heavy loads, where housing deflections may be a possibility, bearing stiffness and drag values change proportionately to the load, and clearances change accordingly; rattle may also occur. For this case, the housing influence may play a role, and it needs to be accounted for.

Figure 11 shows a rattle analysis for the first gear power flow, examining the non-loaded second gear pair for single-sided or double-sided rattle for both the housing and no-housing configurations. While a few subtle differences were noted (rms power, frequency of impacts), both configurations demonstrated similar double-sided impact behavior.

The rattle model was based directly on the same full-system transaxle model used for whine, durability and efficiency, allowing multi-attribute studies to be performed in the same programming environment, reducing development time, allowing the performance of statistical studies such as Monte Carlo for manufacturing variability and Design of Experiments for optimization, thereby improving accuracy.

## Conclusions

As geartrain architectures continue to become more complex, with more stringent requirements for performance attributes, development time and costs, geartrain system CAE tools will also continue to evolve to meet these demands. This investigation has shown that in order to optimize the gear components for durability, efficiency, gear whine and rattle, the geartrain must be analyzed as part of the total transaxle system, and in some cases, including the effects of housing influence. Without the housing flexibility factored into the design process, the gear designer runs the risk of:
1. Incorrectly predicting the geartrain durability performance; 2. Incorrectly predicting static transmission error; 3. Not properly optimizing the efficiency factoring in microgeometry. And for high-load operating conditions: 4. Incorrectly predicting rattle performance. Current state-of-the-art CAE tools and research results are available to help the gear design engineer reduce these potential risks.

## Acknowledgments

*The authors would like to thank their colleagues at Romax Technology for their invaluable advice while developing this paper. Notably, Dr. Jamie Pears and Dr. Michael Platten.*

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