Minneapolis, Minnesota
NOISE-CON 2005
2005 October 17-19

Vibro-Acoustic Prediction of Low-Range Planetary Gear Noise of an Automotive Transfer Case

Ravi Narayanaswamy
Charles D. Glynn
BorgWarner Torq Transfer Systems
3800 Automation Avenue
Auburn Hills
MI 48326
rnarayan@tts.bwauto.com
cglynn@tts.bwauto.com

ABSTRACT

This paper discusses a practical approach to predict low-range planetary gear noise in an automotive transfer case using commercial numerical codes. Dynamic responses of the planetary gear set are calculated using a 2D finite element/contact mechanics model. These responses are used as inputs to calculate surface velocities on the transfer case housing using modal frequency response analysis. Subsequently, the surface velocities are used in a vibro-acoustic model to predict acoustic responses of the transfer case. The predicted acoustic responses are compared to experimental measurements conducted in a hemi-anechoic chamber. It is shown that the predictions are in reasonable agreement with the experiments and the approach enables designers to obtain required information on acoustic responses of the transfer case in a timely and cost-effective manner.

1. INTRODUCTION

Transfer Cases are used in Sports-Utility, Off-Road, Light Trucks and other 4x4 and AWD vehicles to split the transmission output torque between the front and rear axles. In addition to splitting the torque, they can provide speed reduction or torque multiplication, in low-range mode. Planetary gear systems are typically used to perform this speed reduction due to several advantages over conventional parallel shaft gear systems. The advantages include compactness, high torque to weight ratios, reduced radial bending loads etc. Despite their advantages, the noise induced by the vibration of the planetary systems is a concern, particularly in the automotive industry where the vehicle interior noise is a key quality metric. The primary sources of noise are the dynamic mesh forces between various planetary gear components. These dynamic mesh interactions are transmitted to the supporting structure through the sun bearing and directly through the splined ring gear. Figure 1 shows a cut-away of an automotive transfer. Dynamic characterization of planetary gear response is more complicated than parallel axis gears due to multiple meshes, limited accessibility for instrumentation and multiple moving bodies. Due to this difficulty, past studies have opted for an analytical approach where the gear bodies are modeled as lumped masses connected with springs with equivalent mesh stiffness. Lin and Parker[2] presented this approach to characterize vibration modes in a general planetary gear system. Another approach to study the dynamic behavior of planetary gears is to use multi-body analysis software like ADAMS or DADS where the gears are modeled as rigid bodies. This approach relies on an accurate CAD model to represent the gear geometry and a good approximation of dynamic stiffness variation of the contacting surfaces. Parker, Agashe and Vijayakar[1] presented yet another approach where finite element software called Planetary2D was used to study the dynamic response of the planetary gears. They showed good agreement with both analytical and experimental studies. The significance of this approach compared to the multi-body simulation approach is that the user need not provide the dynamic mesh stiffness or a detailed CAD geometry. The Planetary2D software can generate detailed surface geometry, run out errors, misalignment, pitch errors and involute errors based on user inputs. The dynamic mesh stiffness between the contact surfaces is calculated at each time step.

Vibro-acoustic characterization of planetary gear system in assemblies such as transmissions and transfer cases has received considerably less research attention. This study develops a practical approach to study the acoustic response of the planetary gear system in an assembly. The Planetary2D software is used to calculate dynamic responses of a planetary gear system in a transfer case and the responses are then used as boundary conditions to calculate the acoustic responses.

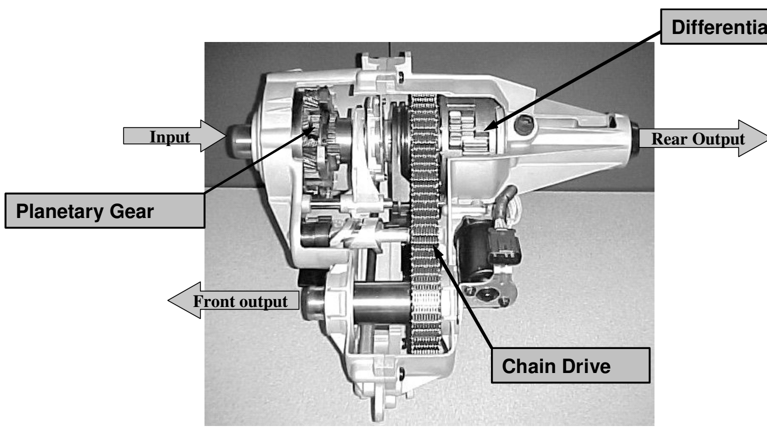

Figure 1: Cut-away of an automotive transfer case

2. ANALYSIS SCHEME

In this study, a three-step process is used to calculate the acoustic response of the transfer case. First, the dynamic responses of the planetary gears are calculated using 2D finite element/contact mechanics software called Planetary2D. In the second step, the force responses calculated from step 1 are used for conducting a modal frequency response analysis on the transfer case housing assembly using MSC.Nastran. The output from MSC.Nastran is the surface velocities on the transfer case housing. Finally in step 3, an indirect boundary element analysis is conducted using Sysnoise to calculate the sound pressure levels (SPL) and sound power levels (L_w). Figure 2 shows the analysis scheme used.

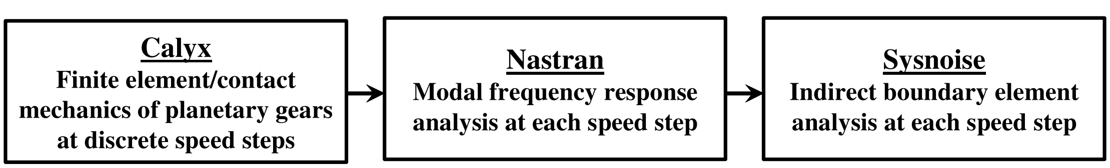

Figure 2: Analysis flow

A. 2D Finite Element Analysis of Planetary Gears

The input to the low-range planetary gear system in the transfer case is through the sun gear, which is integral with the input shaft. The output of the planetary gear is through the carrier, which is floating. The input shaft is supported on a bearing in the housing and the ring gear is supported in the housing through 18 splines. The four pinions are equally spaced and supported in the carrier through pinion pins and needle bearings. Table 1 has the details of the planetary gear system used for this study.

	Sun	Ring	Pinions
No. of Teeth	50	82	17
Base Circle Dia(mm)	88.616	145.33	30.130
Major Dia (mm)	97.05	163.01	35.96
Minor Dia (mm)	88.94	155.00	27.52
Helix Angle (deg)	23.380	23.380	23.380
Normal Pressure Angle (deg)	20.00	20.00	20.00
Normal Module (mm)	1.75	1.75	1.75
Face Width (mm)	28.4	28.4	28.4

Table 1: Planetary gear details

An idealized 2D finite element model of this planetary gear set was generated using the gear prints. Nominal gear geometry was used for modeling and manufacturing tolerances, axis misalignments and run-out errors were disregarded to simplify the study. Bearing stiffness files were used to simulate the input bearing at the sun gear and the needle bearings at the pinion gear locations. The carrier was modeled as a lumped mass. The outer diameter of the ring gear was supported through splines on its outer diameter to a rigid housing. Contact constraints were enforced between the two sides of each spline tooth and the housing. A clearance was modeled on one side of the spline based on the amount of radial movement allowed to the actual ring gear. Figure 3 shows the finite element model of the planetary gear bodies.

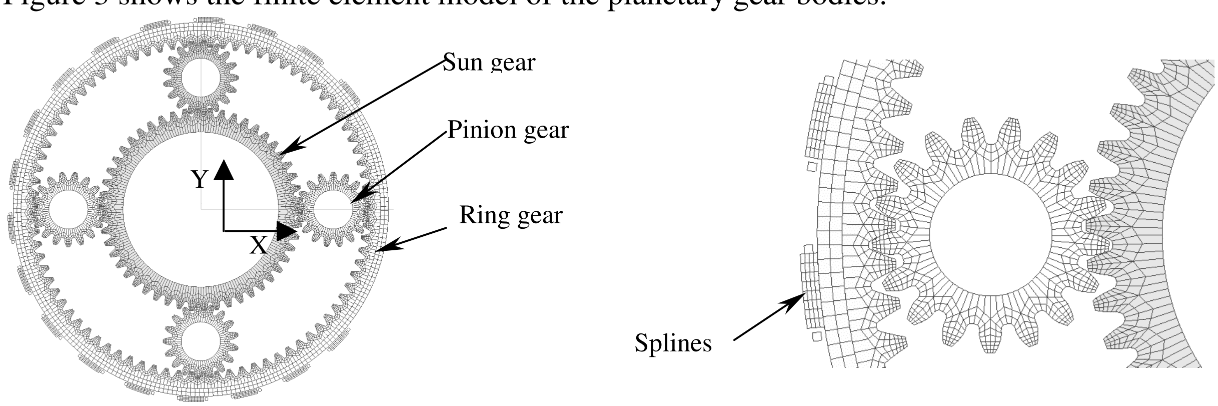

Figure 3: Finite element mesh of the planetary gear system

The dynamic analysis was conducted at steady state speed of 1000 RPM of the output shaft by applying an input torque of 356 N-m at the sun gear. The time step required for the analysis was calculated using expression 1,

$$\Delta T = \frac{\left(\dfrac{60}{RPM_c * Z_r}\right)}{N_{steps}} \ldots\ldots 1$$

where, RPM_c is the RPM of the carrier, Z_r is the number of teeth on the ring gear and N_{steps} is the number of steps per tooth cycle, which was chosen to give at least 256 shaft orders. The total number of time steps was chosen to capture at least three rotations of the carrier. The analysis was conducted for a total of 10,600 time steps. Responses were monitored at the sun bearing and at all the splines of the ring gear interfacing with the housing. Figure 4 shows a typical force time series response at the sun bearing in the X-direction and the force response normal to the face of splines 1 and 2 at 1000 RPM.

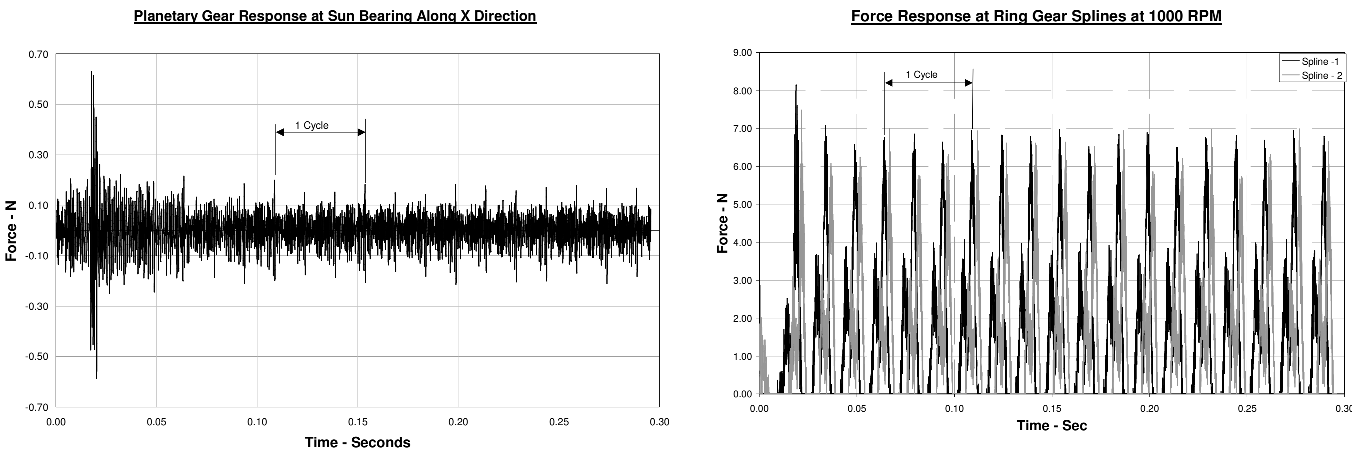

Figure 4: Force response at the sun bearing and ring gear splines 1 and 2 at 1000-RPM carrier speed

B. Modal Frequency Response Analysis

The finite element model of the transfer case assembly was generated using the CAD geometry, the housing was modeled as a combination of plate and solid elements and the shafts were modeled as bar elements. The model consisted of 91411 elements and 97768 nodes. Force time series obtained from the planetary gear analysis was converted into frequency domain and was applied at the input bearing and the splines modeled in the housing. During the conversion, care was taken to remove the transient effects. A modal frequency response analysis was performed over a frequency range of 200-3200 Hz at a frequency resolution of 10 Hz using MSC.Nastran. In addition, analysis steps at all gear tooth mesh frequencies and natural frequencies of the transfer case were considered. Figure 5 depicts the finite element mesh of the transfer case assembly.

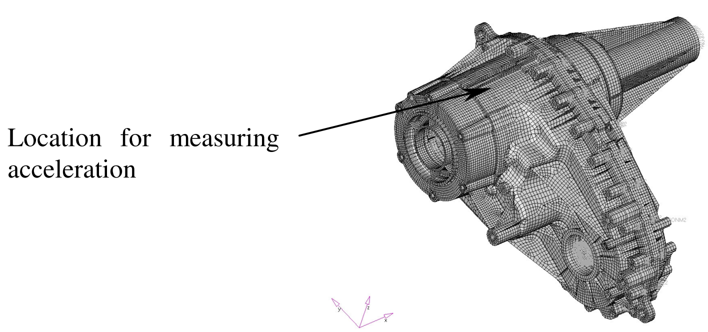

Figure 5: Finite element mesh of transfer case assembly

C. Indirect Boundary Element Analysis

The boundary element mesh was generated from the structural mesh used in the frequency response analysis. The boundary element mesh consisted of 3297 elements and an indirect boundary element analysis was conducted using the surface velocities generated from the modal frequency response analysis. The analysis was carried out over a frequency range of 200-3200 Hz with a frequency resolution of 10 Hz. In addition, care was taken to have solutions at all tooth mesh frequencies and natural frequencies of the transfer case. The sound pressure levels (SPL) around the transfer case were calculated using ISO 3744-1994 field point mesh and the SPL at field point 10 representing the microphone at one (1) meter above the transfer case was used to compare with the experimental measurements taken in a hemi-anechoic chamber. In addition, a separate field point mesh as shown in Figure 6 was used to calculate the sound power levels.

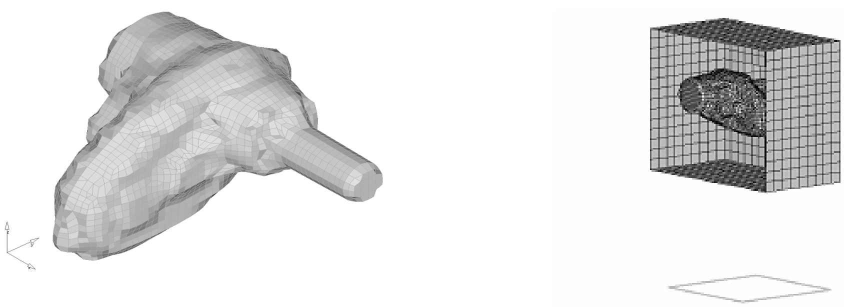

Figure 6: Boundary element mesh and field point mesh for the calculation of L_w

3. EXPERIMENTAL SETUP

In order to validate the numerical models, measurements were performed in a hemi-anechoic chamber. Acceleration on the housing near the ring gear, sound pressure levels and sound power levels were measured. The transfer case was mounted onto a rigid weldment and the input shaft was connected to the drive motor outside of the hemi-anechoic chamber. The rear output shaft was connected to an output motor through a prop-shaft and load cell. The chain drive (see Figure 1) was removed from the transfer case in order remove the effect of the chain system.

Figure 7: Experimental setup for measuring SPL and L_w

A constant load of 135 N-m was applied at the rear output, which translates to 356 N-m input torque at the sun, which was used for the planetary gear analysis. A constant input speed of 2640 RPM was maintained at the input also consistent with the analysis. Figure 7 shows the setup used for these measurements. Five (5) microphones were utilized for calculating the sound power levels and a sixth microphone was placed at one meter above the transfer case for SPL data consistent with the simulations. The SPL and acceleration time histories were sampled at a rate 48,000 samples/sec for a total of 15 seconds in duration. The microphone method was used for calculating the L_w using five (5) microphones.

4. RESULTS AND DISCUSSION

In a planetary gear system the tooth mesh forces are periodic, and the gear set used for this study has equally spaced pinions, hence the responses at the ring gear spline will also be periodic, which is evident from Figure 4. The excitation is predominantly concentrated at the mesh frequency. The mesh frequency is proportional to the output speed and can be calculated for a fixed ring gear by $F_m = Z_r(RPM_c/60)$, where Z_r is the number of teeth on the ring and RPM_c is carrier speed. Figure 8 shows the frequency domain force responses at the sun bearing and the ring gear spline 1 at 1000 RPM. For the planetary gear set used for this study, the tooth mesh frequency will be 1366.7 Hz at 1000 RPM and the second tooth mesh harmonic will be 2733.3 Hz. However, the first tooth mesh frequency is missing from the response, instead there are responses at 1333 and 1400Hz. This is due to the so-called "asymmetric side-banding" phenomena as discussed by McFadden[4].

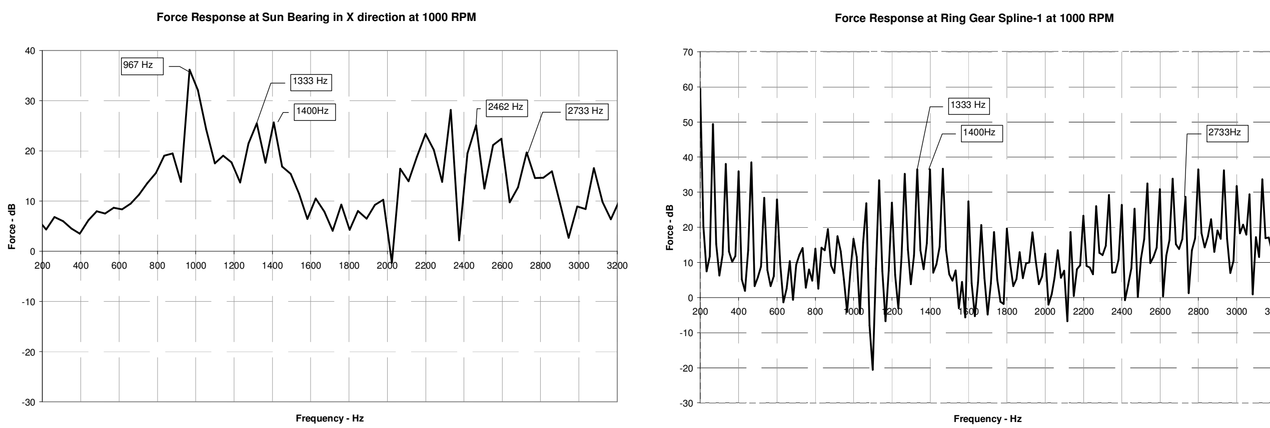

Figure 8: Force response in the frequency domain at the sun bearing along X-direction and normal to spline 1

Other unique phenomena related to planetary gear systems can be observed in the spectra, at frequencies 967 Hz and 2462 Hz there are sun gear responses, which correspond to first and third translation modes, and at 3000 Hz there is a first rotational mode of the planetary gear system. This behavior is consistent with Lin and Parker's[2] studies and can be verified analytically.

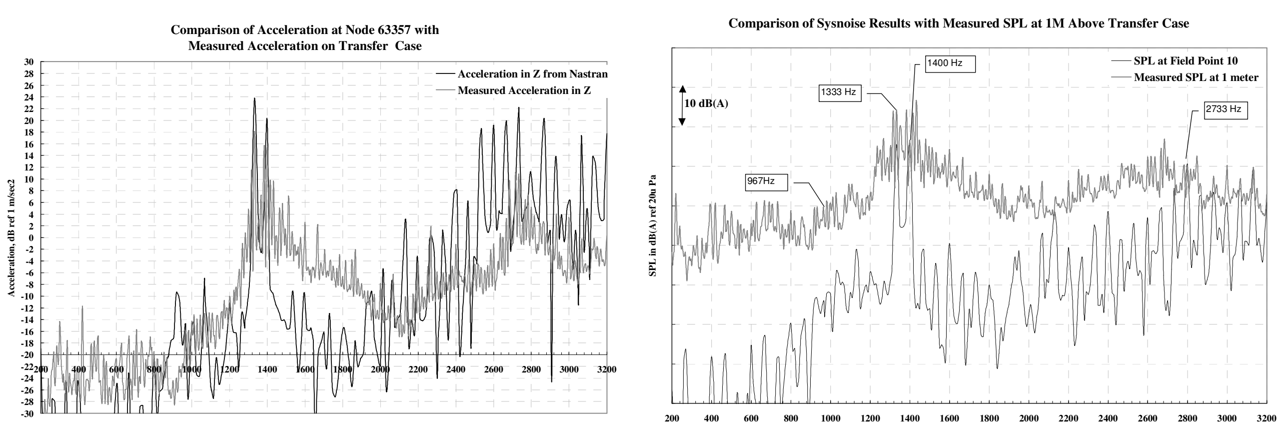

Figure 9: Acceleration at node 63357 on the housing near ring gear at 1000RPM

Figure 9 compares the acceleration and SPL measured with the results of the simulations. These results are consistent with earlier studies predicting important frequencies that are of interest to gear design engineers. The higher spectral content of the test data contains the effects of manufacturing and assembly tolerances, which were not considered in the simulation. There is a good agreement in terms of frequencies, however the magnitudes are lower compared to the test data particularly at the lower frequencies from 200-1000Hz. These differences may be due the boundary condition assumptions in both the 2D planetary gear analysis where the ring gear was assumed to be supported on a rigid housing, and the modal frequency analysis where the housing was modeled to be on a on a rigid support. At this time, the authors are investigating possible refinements to their assumptions to improve the correlation.

Figure 10 shows a comparison of L_w from the simulations with experimental data. The same trend is observed as with the acceleration results, where there is good agreement in terms of frequencies. Also, there is good agreement at the mesh harmonics in both magnitude and frequencies. The asymmetric side bands from the first and second mesh are clearly present and correlates well with test data. In addition, spline pass harmonics and ring gear blank resonance can be seen in the acceleration, SPL and Lw predictions, which is consistent with studies conducted by Tanna and Lim[5]. At the lower frequency range, the magnitudes deviate from test data, which may be due to differences in boundary conditions and other extraneous noises present during the measurements in the hemi-anechoic chamber. The authors are conducting additional studies to understand this deviation. The peaks at the higher frequencies (2870, 2930, 3005 and 3065 Hz) correspond to housing resonance whose magnitudes are affected by modal damping assumed in the FEA model. Using modal damping from test data will improve correlation at these frequencies.

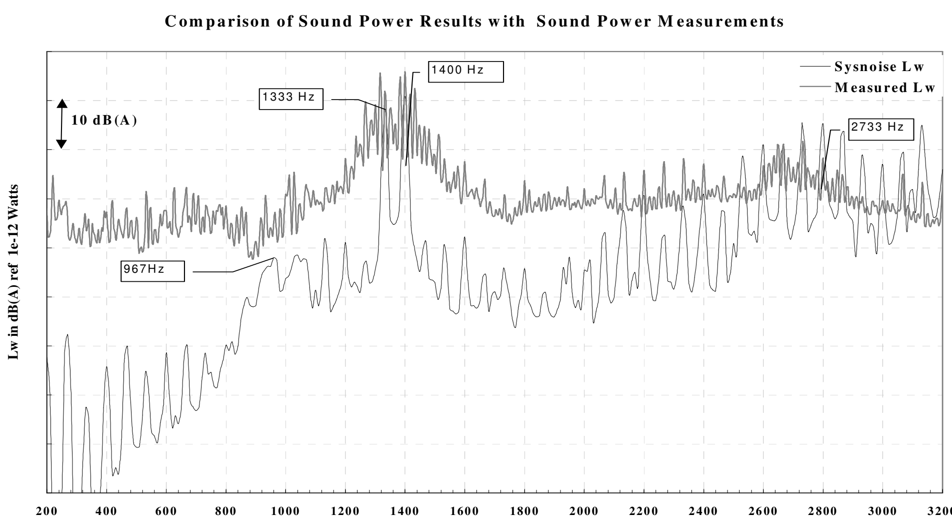

Figure 10: Comparison of simulated sound power simulations compared with test data

The study has shown that it is possible to predict planetary gear noise of a transfer case with reasonable agreement with experiments. This three-step process for calculating the acoustic responses can be an important design tool to study the effects of gear variables, different housing structures, and for bearing selection.

5. CONCLUSIONS

This paper described a three-step process to predict the low-range planetary gear noise in an automotive transfer case using vibro-acoustic analysis. First, the dynamic responses of the planetary gears are calculated using 2D finite element/contact mechanics software called Planetary2D. In the second step, the force responses calculated from step 1 are used for conducting a modal frequency response analysis on the transfer case housing assembly using MSC.Nastran. The output at this step is the surface velocities on the transfer case housing. Finally in step 3, an indirect boundary element analysis is conducted using Sysnoise in order to calculate the sound pressure and sound power levels.

The results from the planetary gear set analysis illustrate sun gear translation response, the ring gear blank response, and the asymmetric side-band phenomena unique to planetary gear systems, which is consistent with earlier studies[1-5]. The acceleration response on the housing near the ring gear was compared to measurements, and showed good agreement in predicting frequencies which was verified both analytically and by measurements. The A-weighted sound pressure level (SPL) and sound power level (L_w) results from Sysnoise simulation showed good agreement with the measurements made in the hemi-anechoic chamber at frequencies of interest. There were some deviations in acceleration, SPL and Lw simulations, which were more pronounced at lower frequencies, which may be due to,

1. Modeling idealizations
2. Boundary condition assumptions
3. Effects of extraneous noises in the hemi-anechoic chamber
4. Manufacturing and assembly errors

Further studies are being conducted to understand these factors. However, this study has shown that this method can be used to study the effect of various gear and housing parameters. This knowledge can be beneficial during the design stage in order to anticipate any NVH issues. The authors feel that this methodology will become a standard engineering procedure within the automotive industry.

ACKNOWLEDGEMENTS

The authors would like to acknowledge Dr. Rajendra Gunda and Dr. Sandeep Vijayakar of Advanced Numerical Solutions for their technical advice in this work. The authors would also like to thank Dr. Sathya Vaidyanathan and Sue Stroope of BorgWarner TorqTransfer Systems for their support of this work.

REFERENCES

[1] Parker, R.G., Agashe, V., and Vijayakar, S. M., "Dynamic Response of a Planetary Gear System using a Finite Element/Contact Mechanics Model", *Transactions of ASME vol. 122 September 2000.*

[2] Lin, Jian, Parker, R.G., "Analytical Characterization of the Unique Properties of Planetary Gear Free Vibration", *Transactions of the ASME, Volume 121, July 1999*

[3] Parker, R.G, "A Physical Explanation for the Effectiveness of Planetary Phasing to Suppress Planetary Gear Vibration", *Journal of Sound and Vibration (2000)* ***236****(4), Pgs 561-573.*

[4] McFadden, P.D., Smith, J.D., "An Explanation for the Asymmetry of the Modulation Sidebands about the Tooth Meshing Frequency in Epicyclic Gear Vibration", *Proc Instn Mech Engrs, Vol 199 No C1, IMechE 1985.*

[5] Tanna, P.R, Lim, C.T., "A Parametric Study on the Vibration Transmissibility Characteristics of Transmission Ring Gear Structure", *2003 Proceedings of Society of Automotive Engineers, Noise and Vibration Conference, Paper #2003-01-1660.*

[6] Drago, J.R., "How to Design Quiet Transmissions", *Machine Design, December 11, 1980.*

[7] Advanced Numerical Solutions, "Planetary2D User's Manual", Chapter 9.