ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

PROFILE MODIFICARTION OF HYPOID GEAR TO REDUCE A GEAR WHINE NOISE OF THE AXLE SYSTEM IN A PASSENGER VAN

Sung-Jong Kim[1] and Sang-Kwon Lee*[1]

[1]Acoustic Noise Signal Processing Laboratory, Department of Mechanical Engineering,
Inha Unicersity, Inchon, 402-751, Korea
sangkwon@inha.ac.kr

Abstract

This paper presents practical work on the reduction of gear whine noise. In order to identify the source of the gear whine noise, transfer paths are searched and analyzed by operational deflection shape analysis and experimental modal analysis. It was found that gear whine noise has an air-borne noise path instead of structure-borne noise path. The main sources of air-borne noise were the two global modes caused by the resonance of an axle system. These modes created a vibro-acoustic noise problem. Vibro-acoustic noise can be reduced by controlling the vibration of the noise source. The vibration of noise source is controlled by the modification of structure to avoid the resonance or to reduce the excitation force. In the study, the excitation force of the axle system is attenuated by changing the tooth profile of the hypoid gear. The modification of the tooth profile yields a reduction of transmission error, which is correlated to the gear whine noise. Finally, whine noise is reduced up to 10dBA.

INTRODUCTION

In today's highly competitive automotive industry, the vehicle sound quality becomes more and more important [1-6]. There is a constant pressure to achieve a low vehicle interior noise level in system design, especially for a passenger van. For an axle system, gear whine is one of the major factors in achieving the vehicle interior sound quality. This paper presents the results of the practical research to reduce the whine noise of an axle system of a passenger van. The problem occurs in the compartment of a passenger van. In general, gear whine noise has two paths: airborne and structure-borne [7-8]. In order to identify the transfer path of whine noise, the accelerations on all transfer paths from the axle to the car body are measured together with interior noise of the passenger

Eds.: J. Eberhardsteiner, H.A. Mang, H. Waubke

van simultaneously. From vibration tests, it is found that the isolator between the body and axle system can reduce enough the vibration transfer energy from the axle system to the car body. Therefore, the structure-borne noise is not significantly important for whine noise. As the next process, operational shape analysis and experimental modal analysis are used to identify the source of airborne noise due to the structural behavior of the axle system during acceleration of the passenger van. It is found that the major reason for airborne noise is the vibro-acoustic noise of the axle system. This vibro-acoustic noise is controlled by reducing the excitation force due to gear meshing. The excitation force is reduced by modifying the tooth profile, which is related to the transmission error. Transmission error is very much correlated with gear whine noise [9-13]. In the paper, the transmission error of the ring gear has been reduced by modification of the tooth profile of the ring gear under test. Finally, whine noise in the compartment of the passenger van is reduced to 10dBA. The sound quality due to whine noise is also improved.

MEASUREMENT OF INTERIOR NOISE

An axle system with a gear ratio of 3.616 is developed. The number of teeth of a pinion gear is 13 and the number of teeth of a ring gear is 47 in an axle system. A passenger van loaded with this axle system is accelerated in order to have a subjective evaluation of gear whine noise. During acceleration, the gear whine noise in the compartment of the passenger van is especially heard at 60km/h and 120km/h. To identify the source of this whine noise objectively, first, the interior noise is measured at the front seat. Fig. 1 shows the waterfall map for the noise signal. On this waterfall map, two peaks are clearly identified at 1800 rpm and 2300rpm. These two peaks occur at the frequency band, which is the meshing frequency of the hypoid gear of the axle system. These peaks well correspond with the previous subjective evaluation. Meshing frequency of the hypoid gear is 13th times the rotation speed of driveshaft since the number of teeth of a pinion gear is 13. Therefore, it infers that gear whine noise is related to the harmonics of the meshing frequency of hypoid gear in an axle system.

IDENTIFICATION OF TRANSFER PATH

Gear whine noise originates from the excitation force by the meshing of a hypoid gear as shown in Fig. 2. There are two paths for whine noise: structure-borne noise path and airborne path. The transfer of the vibration energy from the axle system to car body induces structure-borne noise. The direct transfer of vibro-acoustic sound due to the shell vibration of an axle system from axle system to compartment of a car induces air-borne noise.

Structure-borne path

The structural vibration of an axle system is measured at several points as shown in Fig. 3. Three accelerations are measured on the housing of axle and eight accelerations are measured on the bracket of isolation system during acceleration from 1800 rpm to 3500rpm of driveshaft speed. Fig. 4 shows the photo information about the attachment of three accelerometers attached on the axle housing.

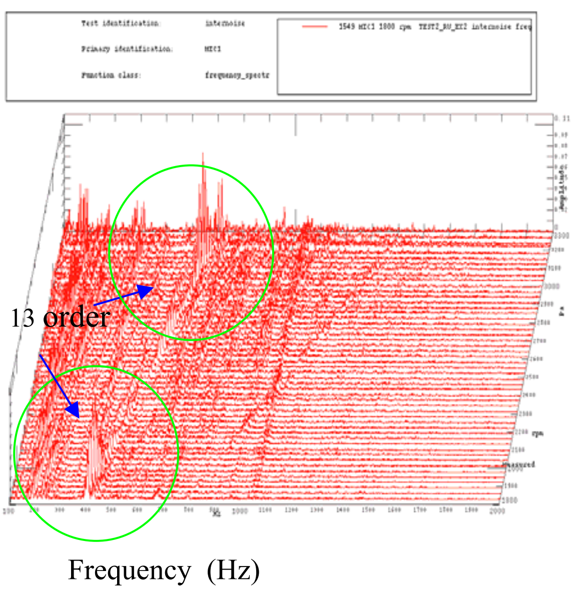


Figure-1 Waterfall map for the interior noise data

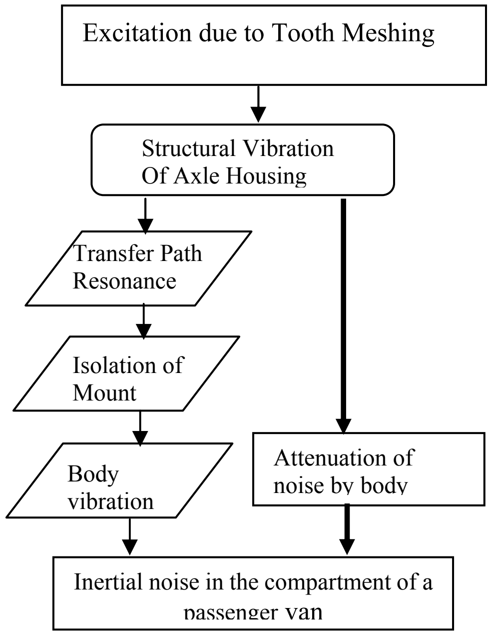


Figure-2 Transfer paths of air-borne noise and structure-borne for gear

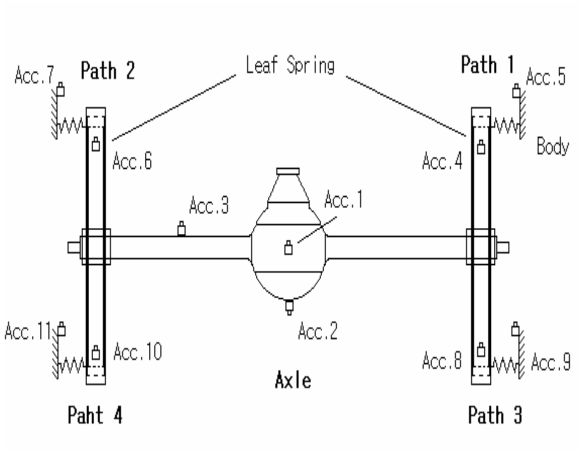


Figure-3 Measurement point of the structure vibration of an axle

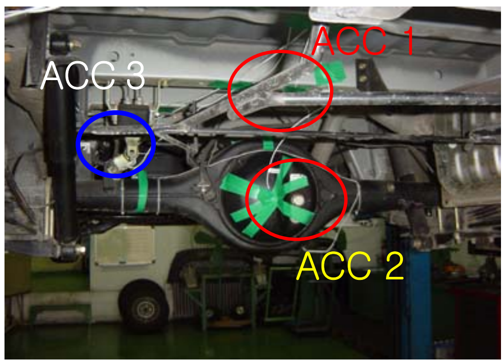


Figure-4 Photo explanation for measurement point of the structure

Throughout the measurement of the acceleration the point 1, 2, and 3 of axle system respectively it is identified that there is one structure resonance around 3100rpm. Resonance frequency is about 670Hz (3100rpm x 13 / 60). The other resonance of house structure occurs at 2600 rpm. Resonance frequency is about 1100Hz (2600rpm x 26/ 60). The resonance caused by 13th order of driveshaft speed is global mode of axle system at 670Hz. The resonance caused by 26th order of driveshaft speed is the local mode of axle housing itself at 1100Hz. This local mode does not contribute to the interior since there is no gear whine noise at 1100Hz as shown in Fig. 1. The global mode excites the whole axle system. Therefore, the axle system has high vibration level at all other four points (point 4,6,8 and 10 in Fig. 3) before the isolator shown in Fig. 5. It is confirmed with vibration result measured at point 4. The isolator, which is installed between the axle system and the car body, attenuates this horrible vibration due to global mode of the axle system. Fig. 6 shows the waterfall map for the vibration data measured at point 5. This point is the place after the isolator rubber; there is no resonance peak. Although there are a few resonances in the axle system, all vibration energy is dissipated through damping of the rubber isolator, which is installed between the axle system and the car body. Therefore, the gear whine noise as shown in Fig.1 is not caused by structure-borne noise.

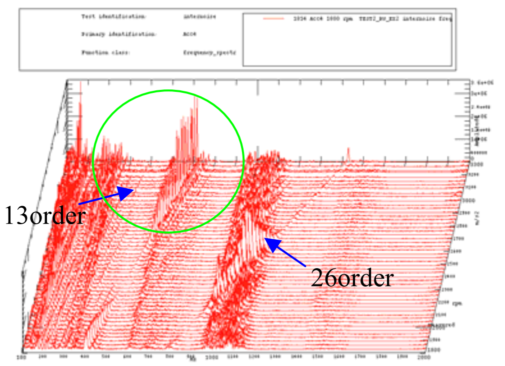


Figure-5 Waterfall map for the vibration data measured at the point 4 before isolator

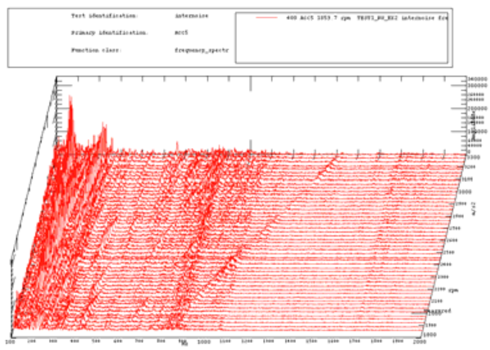


Figure-6 Waterfall map for the vibration data measured at the point 5 after isolator

MODIFICATION OF TOOTH PROFILE

In the previous section it was found that structure-borne noise does not contribute to gear whine noise because all vibrational energy caused by the resonance of the axle system is damped out through the isolator between the body and the axle system. Therefore, it is inferred that part of the vibration energy caused by resonance of the axle system is radiated as vibro-acoustic energy. This vibro-acoustic energy is air-borne noise and a major source for gear whine noise in the compartment of a passenger van. There are three methods for reducing this air-borne noise. The first is a design modification of the car body with high acoustic transmission loss required. The second is a structure modification of the axle system to avoid resonance. The last one is to attenuate the excitation force by modifying the profile of the teeth in the hypoid gear. For economical efficiency, in the paper, the last method is adapted for the reduction of whine noise.

***Table 1** The profile of tooth in the ring gear*

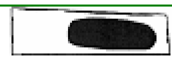
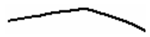
	Cutter Dia.	Tooth Profile		Contact Ares
Original	9 Inch		Slow	
Modification	7.5 Inch		Steep	

***Table 2** The design details of the ring gear*

	CUTTING BY 9 INCH CUTTER	CUTTING BY 7,5 INCH CUTTER
Tooth Profile	Helix Form	Helix Form
Cutting Method	Gleason	Gleason
Cutter Diameter	9 inch	7.5 inch
Module	5.268	4.907
Number of Teeth	47	47
Face Width	31.75	31.75
Shaft Angle	90°	90°
Pinion Offset	38.10	38.10
Spiral Angle	26° 49’	22° 5’
Whole Depth	9.83	10.57
Addendum	1.13	1.78
Dedendum	8.70	8.79
Face Angle	76° 12’	76° 12’
Root Angle	74° 9’	69° 53’
Pitch Angle	79° 21’	75°
Gear Ratio	4.89R (44X9)	4.89R (44X9)
Mating Gear	3009 4580	3009 4580
Backlash	0.15~0.20	0.13~0.20
Transmission Error	0.104	0.056

In order to modify the tooth profile, the original ring gear is processed by using a 9-inch cutter, but the modified ring gear is processed by using a 7.5- inch cutter. In the contact area, the black area is the initial contact area of teeth. The outline is the contact area of teeth under full load condition. The design details of the ring gear under test are listed in Table.2.

Transmission error

Gleason 500Ht is used for the measurement of transmission error of two-ring gears. The total transmission error is reduced from 0.104 to 0.056 as listed in Table.2. According to Table 1, the initial contact area of two gears has the same area. However, the contact area of the ring gear with high transmission error goes out of the boundary of tooth face under full-load condition while the contact area of the ring gear with low transmission error is inside of the boundary of the tooth face. Fig. 7 shows a photo of two ring gears. The upper one shows the ring gear with high transmission error; the bottom one shows the ring gear with low transmission error.

Figure-7 Photo of two ring gears of axle system

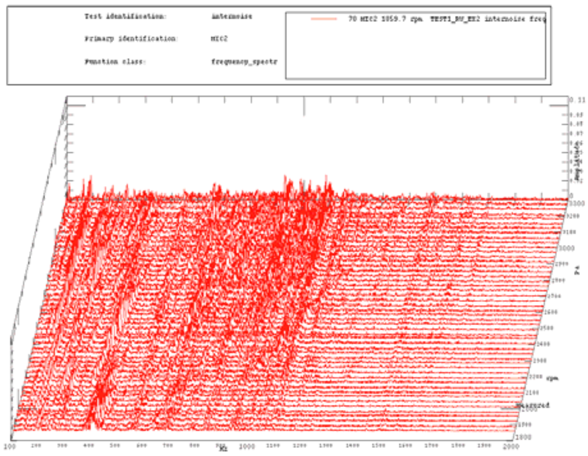


Figure-8 Waterfall map for structure vibration of a axle housing measured on the positing 2 in modified axle system

Gear whine noise reduction

The modified axle system with new profile tooth is replaced of noisy axle system and installed on the passenger van under test. Interior noise for a passenger van is measured and its results are plotted in Fig. 8. From these results, high vibration around 1800rpm and 3200rpm of driveshaft speed is removed since the excitation force due to tooth meshing is significantly attenuated by reducing transmission error although structure resonance exists. Therefore, Interior noise also reduced up to 10dBA at the same speed of driveshaft speed as shown in Fig. 8 comparing with Fig. 1. Around 2600rpm of driveshaft speed, the 26th order component of vibration is still high level since it is a local mode of axle housing and it does not significantly contribute to the gear whine noise.

CONCLUSION

1. Basic noise and vibration tests for a passenger van are assessed for the identification of problems. Gear whine noise occurs around 1800rpm and 3200 rpm, at which vibration level is also high.

2. Experimental modal analysis and operational deflection shape are used to identify the vibration transfer path. Vibrations caused by these modes are attenuated through an isolator at the mount point of the axle system. Therefore, it is inferred that the structure-borne noise is not contributing to gear whine noise. However these modes contribute to the air–borne noise for gear whine noise.

3. In order to control the air-borne noise, the structure should be modified to avoid resonance or the excitation force should be reduced. For gear whine noise, the modification of structure mode requires very high stiffness in structure. In the paper, instead of modification of the axle system, the excitation force is attenuated by reducing the transmission error by optimizing the profile of teeth of the axle gear.

4. By reinstalling the new axle system with optimized tooth profile on the same passenger van with gear whine noise problem, interior noise is reduced up to 10dBA.

ACKNOWLEDGMENTS

This work was supported by Automotive Fundamental Technology Devolvement Project in Korea. This work was also partly supported by DYMOS Company in Korea.

REFERENCES

[1]. Lee, S. K., Yeo, S. D., and Choi, B. U., "Identification of the Relation Between Crankshaft Bending and Interior Noise of A/T Vehicle in Idle State," SAE930618, 1993

[2]. Lee, S. K. "Weight Reduction and Noise Refinement of Hyundai 1.5 Liter Powertrain", SAE940995, 1995

[3]. Lee, S. K. Vibrational Power Flow and Its Application to a Passenger Car for Identification of Vibration Transmission Path, Traverse City, Michigan, USA. SAE Noise & Vibration Conference & Exposition, SAE 2001-01-1451, 2001, 2001

[4]. Lee S. K., Chae, H. C, Park, D. C and Jung, S. G. "Booming Index Development for Sound Quality Evaluation of a Passenger Car," SAE 2003-01-1497, 2003

[5]. Becker S. B. and Yu. S "Objective Noise Rating of Gear Whine" SAE1999-01-1720

[6]. Becker S. B. and Yu. S "Gear Noise Rating Prediction Based on Objective

Measurement” SAE1999-01-1721

[7]. Sun Z., Voight M and Steyer G. “Driveshaft Design Guidelines for Optimized Axle Gear Mesh NVH Perfprmance”, FISITA2004, F20004V287

[8]. Sun Z., Steyer G. Meinhardt G. and Ranek R. “NVH Robustness Design of Axle System” SAE2003-01-1492

[9]. M.G. Donley, T.C. Lim and G.C. Steyer “Dynamic Analysis of Automotive Gearing Systems,” Journal of Passenger Cars: Mechanical Systems, 101(6), pp. 958–968, 1992.

[10]. Glover R. and Rauen D. “Gear Transmission Error for Use with Ger Inspection Machine,” SAE2003-01-1663

[11]. Tarutanl I. and Maki H. “A New Tooth Flank Form to Reduce Transmission Error of Helical Gear,” SAE2000-01-1153

[12]. Athavale S., Krishnaswami R. and Kuo E. “Estimation of statistical Distribution of Composite Manufactured Transmission Error, A precursor to Gear Whine for A Helical Planetary Gear System,” SAE2001-01-1507

[13]. Houser R. and Harianto J. ”Manufacturing Robustness Analysis of Noise Excitation and Design of Alternative Gear Sets,” SDAE2001-01-1417

[14]. Wyckaert, K and Van der Auweraer, H., "Opernational Analysis, Transfer path Analysis, Modal analysis: Tools to Understand Road Noise Problems in Cars," SAE Paper 951752.