Experimental Investigation of the Effect of Lead Errors on Helical Gear and Bearing Vibration Transmission Characteristics

Chan Il Park*
Kangnung National University, Kangwon 210-702, Korea
Jang Moo Lee
Seoul National University, Seoul 151-742, Korea

The characteristics of gear meshing vibration undesgo change as the vibration is transmitted from the gear to the housing. Therefore, vibration transmission characteristics of helical gear systems must be understood before the effective methods of reducing gear noise can be found. In this work, using a helical gear with different lead errors, the gear vibration in the rotational direction and the bearing vibration are measured. The frequency characteristics of gear and bearing vibration are investigated and a comparson is also provided.

Key Words : Helical Gear, Lead Error, Bearing, Vibration, Mesh Frequency, Harmonics

Nomenclature

α : Angular acceleration
ω : Angular velocity
k : Combined sensitivity of accelerometer and amplifier
α_y : Cross axis sensitivity
$\hat{e}_r$, $\hat{e}_\theta$: The unit vector
e_A, e_B : Output voltage from accelerometer A and B

1. Introduction

Since the vibration due to the meshing of gear teeth is transmitted to the shaft, bearings, and housing, it is necessary to know the vibration transmission characteristics in helical gear systems with lead errors from the gear excitation to the housing. Although the gears do not have manufacturing errors, the contact pattern in a loaded gear due to the twisting moment and the bending moment is different from that in an unloaded gear. Actually, the gear comes into partial contact because of the shaft misalignment, bearing clearance, and deformation of the shaft and housing. Therefore, the original lead in the teeth should be modified so that the gear does not come into partial contact, thus reducing noise and vibration. In previous research works by Kiyono on the vibration in a helical gear system, the three dimensional acceleration of the helical gear and dynamic strain were measured in a power absorbing experimental set-up. He reported that increasing the helix angle rather than the face width could significantly reduce the vibration (Kiyono et al., 1978). Tanaka measured the root strain to find the mesh force using the strain gages and the bearing load using the load cell. He investigated the dynamic load factor, defined by the ratio of the maximum strain to the average dynamic strain (Tanaka, 1989). Maruyama et al. investigated the effect of the lead error on the transmission error and bending moment of the helical gear. He measured the root strain using a strain gauge and the transmission error using a rotary encoder (Maruyama, 1989). Experimental studies on topics such as the effect of the misalignment on the gear vibration (Umezawa et al., 1986), the influence of contact ratios on the gear noise and vibration (Esaki and Furukawa, 1991), the gear noise and vibration on lead errors (Park and Lee,

* Corresponding Author,
E-mail : pci@kangnung.ac.kr
TEL : +82-33-640-2392; **FAX** : +82-33-640-2244
Kangnung National University, Kangwon 210-702, Korea. (Manuscript **Received** January 5, 2002; **Revised** August 30, 2002)

1996a; 1996b; 2001), the transmission error (Houser and Blankenship, 1989), and dynamic force on the gear teeth and bearing (Rauter and Kollmann, 1989), were conducted. However, the paper on the relation between gear vibration and bearing vibration according to the direction of lead error has not been published. Therefore, this paper investigates the vibration transmission characteristics of gears with lead errors from the gear meshing to the bearing. For this study, accelerometers were mounted directly on the gears and bearing. The gear vibration and bearing vibration are measured, and several parameters in terms of the gear vibration and bearing were investigated.

2. Experiment

2.1 Experimental apparatus

For this experiment, power absorbing test rigs are specially designed, as shown in Fig. 1. The rig is powered by a DC motor, and the power is transmitted to the test gear through a toothed belt and pulley. The test gear is connected with the dynamometer through the output shaft. Special lubrication setup is designed, and oil is fed during the test.

2.2 Test gears and test condition

The specifications of the test gears are shown in Table 1. The test gear is machined by grinding and hardened by carburizing. To investigate the effect of tooth lead error, driving gear with a linear tip relief of $5\mu m$ was used and driven gears with three kinds of tooth error were used: (I) driven gear with a linear tip relief $5\mu m$ and linearly increasing lead error of $20\mu m$ from the bottom to the top (II) driven gear with a linear tip relief $5\mu m$ and linearly decreasing lead error from the bottom to the top, and (III) driven gear with linear tip relief $5\mu m$ and no lead error. The experiment is performed by varying the rotational

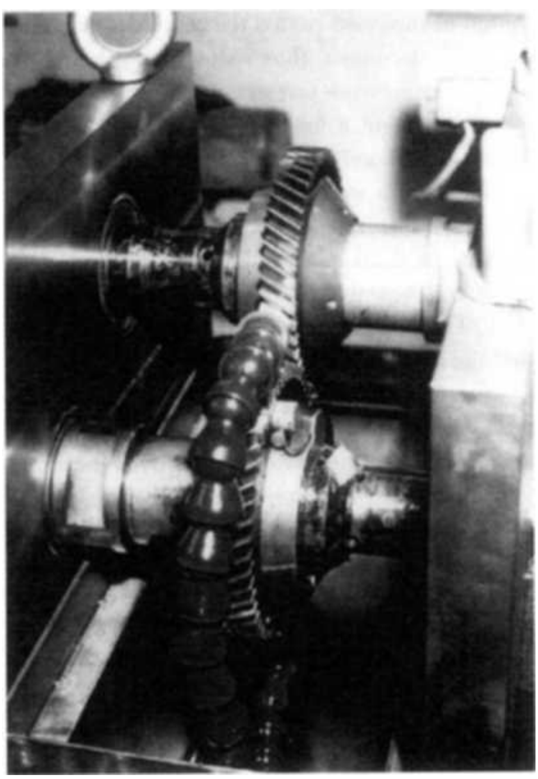

Photo. 1 Accelerometers mounted for gear vibration measurement

Table 1 Specifications of the test gear

	Driving	Driven
Normal module (mm)	2.25	
Normal pressure angle (deg)	17.5°	
Helix angle (deg)	28°	
Face width (mm)	18	
Whole depth (mm)	6.6	
Center distance (mm)	127	
Outside diameter (mm)	130.36	135.26
Pitch diameter (mm)	122.32	127.41
Root diameter (mm)	117.16	122.06
Amount of add. mod. (mm)	1.17	1.07
Number of teeth	48	50
Finishing	Grinding	

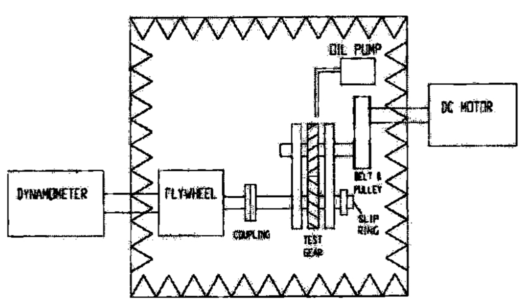

Fig. 1 Experimental apparatus layout

speed of the gear from 500 to 1400 rpm, at two input torques 51 Nm and 74 Nm.

2.3 Measurement method

To measure the gear vibration, two aluminum blocks, each bolted with an accelerometer, are attached to the driven gear, as shown in Fig. 2. The accelerometer line is connected to a slip ring through a hole in the shaft. The accelerometer signal is amplified in the pre-amplifier and the noise signal is removed by a low pass filter.

The output signal from each accelerometer is composed of the rotational component and the radial component of the gear vibration. The output signal from accelerometer A, one of the two accelerometers mounted equidistant from the axis of rotation as shown in Fig. 2, is given by

$$e_A = k_A[(ra - g\sin\theta)\hat{e}_\theta - a_{yA}(r\omega^2 + g\sin\theta)\hat{e}_r] \quad (1)$$

Similarly, the output signal from accelerometer B is given by

$$e_B = k_B[(-ra - g\sin\theta)\hat{e}_\theta - a_{yB}(r\omega^2 - g\sin\theta)\hat{e}_r] \quad (2)$$

If the combined sensitivities of both linear accelerometers and amplifiers are adjusted such that $k_A = k_B = k$ and if the accelerometers are selected so that the cross axis sensitivities in the y direction are $a_{yA} = a_{yB} = a_y$, then subtracting signal B from A will result in the following equation.

$$e_A - e_B = k[2ra\hat{e}_\theta - 2a_y g\sin\theta\hat{e}_r] \quad (3)$$

If cross sensitivities are neglected after comparison with the main axis sensitivities, Eq. (3) will be composed of only the rotational vibration component as given by

$$e_A - e_B = 2rak\hat{e}_\theta \quad (4)$$

The result of Eq. (4) is stored in the tape recorder. In order to measure the bearing vibration, an accelerometer is mounted on the outer ring of the bearing supporting the test gear, and the signal obtained is also recorded in the tape recorder. Using a 2-channel FFT analyzer, the frequency spectrum of the signal is calculated.

Impact testing is conducted to determine system resonance. The transfer functions of the gear test system, excited at the driving part and driven part, respectively, are obtained.

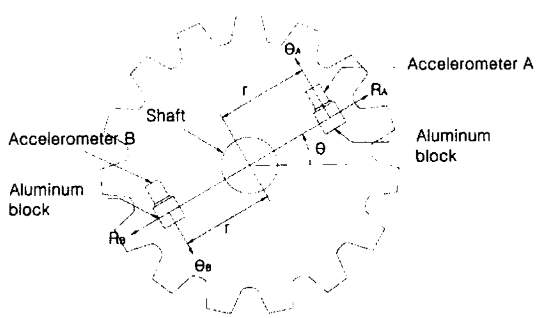

Fig. 2 Mounting position of the accelerometers

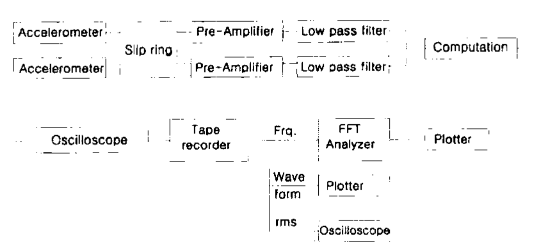

Fig. 3 Block diagram of experiment

3. Experimental Results and Discussion

The vibration of a helical gear has rotational, radial and axial components. These types of vibration are coupled, but the rotational vibration is considered the most important among the three types. Gear vibration characteristics in the rotational direction and bearing vibration characteristics arising from gear vibration are investigated.

3.1 Gear vibration

3.1.1 Contribution to the mesh frequency and its harmonics

Figure 4 compares the amplitudes of the mesh frequency and its harmonics with tooth error I subjected to the torque of 74 Nm. In the frequency range considered, the amplitude of the fundamental mesh frequency is generally the largest. However, of the nineteen data collected in the measured rotational velocities, four data near 1000 and 1300 rpm showed that the amplitude of the second harmonic of the mesh frequency is higher than that of the fundamental mesh fre-

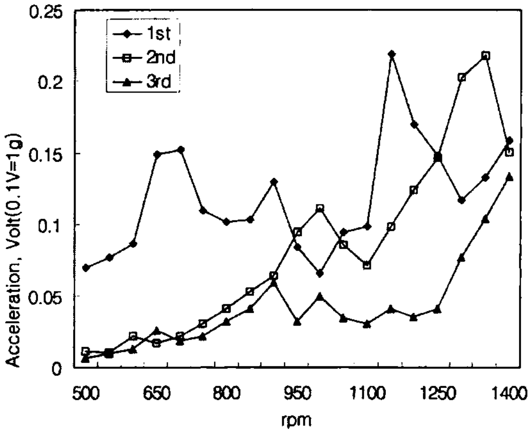

Fig. 4 Harmonics of mesh frequency of vibration (tooth error I), 74 Nm

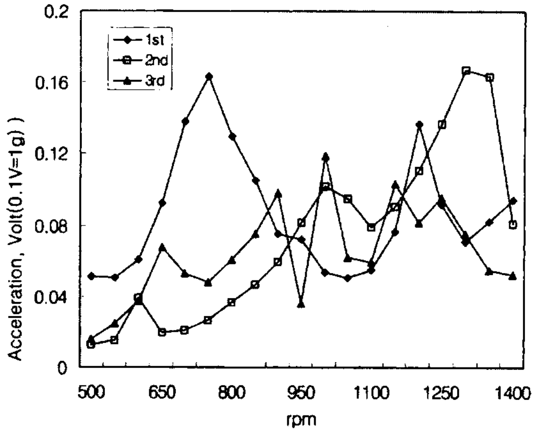

Fig. 6 Harmonics of mesh frequency of vibration (tooth error II), 51 Nm

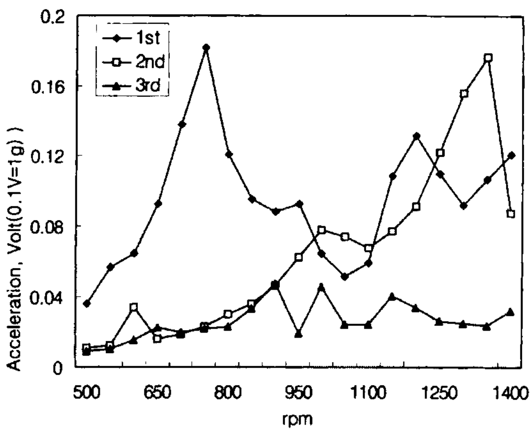

Fig. 5 Harmonics of mesh frequency of vibration (tooth error II), 74 Nm

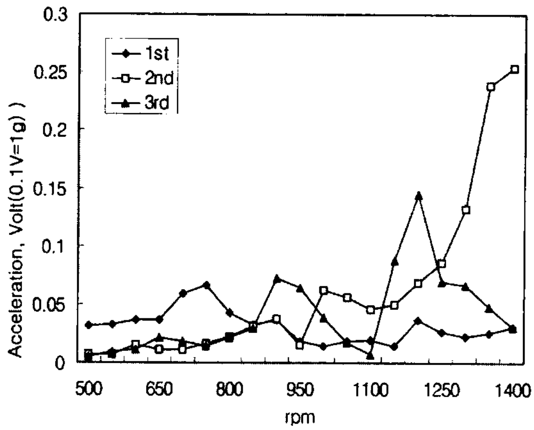

Fig. 7 Harmonics of mesh frequency of vibration (tooth error III), 74 Nm

quency. The third harmonic has a considerable contribution near 1400 rpm. However, it generally does not show large amplitudes. In the case of tooth error II at torque 74 Nm, the amplitude of the second harmonic of the mesh frequency is higher than that of the fundamental mesh frequency in six data points near 1000 and 1300 rpm, as shown in Fig. 5. The third harmonic has little contribution. The amplitude of the second harmonic of the mesh frequency is higher than that of the fundamental mesh frequency near 1000 and 1300 rpm at torque 51Nm, as shown in Fig. 6 and the third harmonic component has the maximum amplitude near 1000 and 1200 rpm. The second harmonic response is shown to be similar to Fig. 4.

In the case of tooth error III at 74 Nm, the second harmonic shows large amplitudes in some of the data, as shown in Fig. 7. Therefore, the fundamental frequency, and its second and third harmonics should be considered together for accurate vibration analysis.

3.1.2 The effect on the torque and rotational velocity

Figure 8 compares the fundamental mesh frequency of gears with tooth error I and II at two torque levels, 51 Nm and 74 Nm. In the cases with tooth error I, the amplitude of the fundamental mesh frequency at torque 74 Nm is higher than

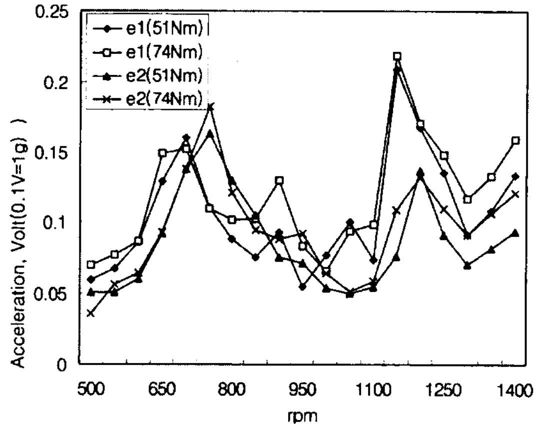

Fig. 8 Fundamental mesh frequency of vibration (tooth error I and II)

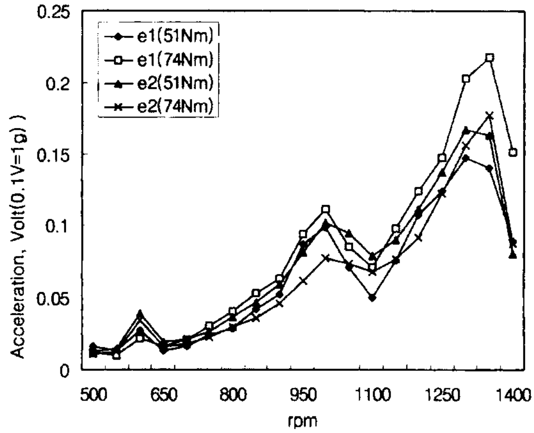

Fig. 9 Second harmonic of vibration (tooth error I and II)

that at 51 Nm, except for data point 3 of the total of 19 data points. In the case with tooth error II, the fundamental mesh frequencies are similar to those of tooth error I at both torques. That is, the amplitude of the mesh frequency at torque 74 Nm is higher than that at torque 51 Nm. Even a 45% increase in the torque only slightly increases the gear rotational vibration.

Figure 9 compares the second harmonic components of cases with tooth error I and II at torques 51 Nm and 74 Nm. Although the types of tooth error and torque levels differ, the form of the harmonic components is similar. In the case with tooth error I, the amplitudes of most components at torque 74 Nm are higher than those at torque 51 Nm. However, although the differences are small, the amplitudes of components at torque 51 Nm are higher than those at 74 Nm in the case of tooth error II.

Figure 10 shows the third harmonic components of the cases with tooth error I and II. The magnitudes of the peaks are smaller than those of the fundamental frequency and second harmonic components. In the case of tooth error I, the higher amplitudes of the components are shown below 1050 rpm at 51 Nm but above 1050 rpm at 74 Nm. In the case of tooth error II, the amplitudes of third harmonic components are significantly higher at 51 Nm than at 74 Nm. From these results, we can see that the increase in torque does not always cause concomitant in-

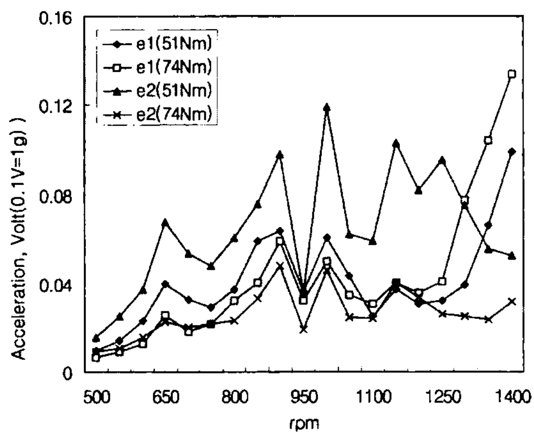

Fig. 10 Third harmonic of vibration (tooth error I and II)

crease in vibration. Gear vibration depends more on speed than on torque.

3.1.3 The effect on the tooth error

Figure 11 illustrates the amplitude of the -undamental mesh frequency of test gears with different types of lead error. As the figure shows, the gear without lead error has the least vibration. Gears with tooth error I show the most severe vibration except for the four data points. This is especially true at approximately 1200 rpm. Therefore, gear vibration shows significant dependence on the direction of the lead error. If the operation region is fixed, proper lead modification can result in a significant reduction in gear vibration.

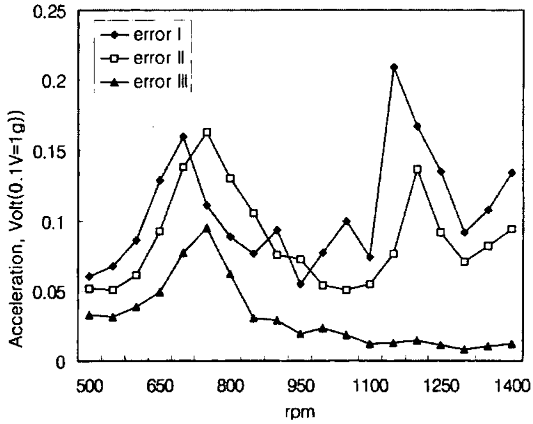

Fig. 11 The comparison of vibration (tooth error I, II and III), 51Nm

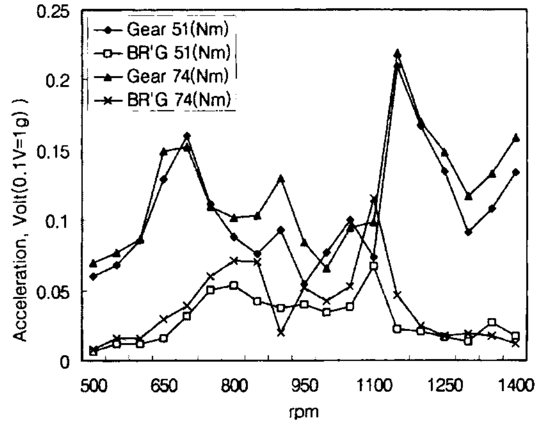

Fig. 12 Comparison between gear vibration and bearing vibration with tooth error I

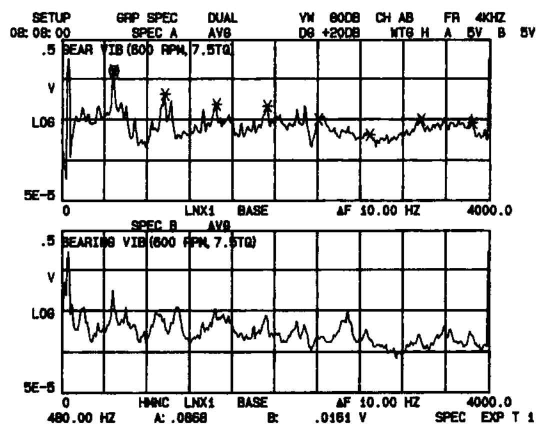

Fig. 13 Gear and bearing frequency spectrum with tooth error I (600 rpm, 74 Nm)

3.2 The relation between gear vibration and bearing vibration

Figure 12 compares the magnitude of the fundamental mesh frequency of the rotational gear vibration with that of the bearing vibration in the case of tooth error I. We can see that the bearing vibration is smaller than the gear vibration. The gear vibration shows large amplitudes at 700 rpm, 900 rpm, and 1150 rpm, and the bearing vibration shows the peak value at 1100 rpm. That is, the rotational velocities of the peaks in the gear vibration differ from those in the bearing vibration. This trend is extended to the case with tooth error II. The bearing vibration has large amplitudes at 800 rpm and 1150 rpm, and the gear at 750 rpm and 1200 rpm. The vibration characteristics of the case with tooth error III are similar to those with tooth errors I and II.

Figure 13 shows the frequency spectrum of gear vibration and bearing vibration for tooth error I measured at 600 rpm and 74 Nm. The peak fundamental mesh frequency is shown in both spectra. The second and third harmonic components are greatly reduced in the bearing spectrum. The peaks existing in only the bearing spectrum are shown near 1080 Hz and 2700 Hz. These peaks exist in the entire measured rotational velocity range even though their magnitudes change as the rotational velocity is vairied. As the torque increases, the peak at 1080 Hz is prominent at 550 rpm and 650 rpm. This frequency does not correspond to the rotational velocity and appears in the bearing spectrum only. Therefore, it seems to be one of the natural frequencies. Here a big spike at 60 Hz is due to electrical noise.

Figure 14 shows the frequency spectrum of the gear vibration and bearing vibration with tooth error I measured at 1250 rpm and 74 Nm. While the amplitudes of the mesh frequency and its harmonic components are significant in the gear spectrum, the amplitude of the fundamental mesh frequency is dramatically reduced in the bearing spectrum.

Figure 15 illustrates the frequency spectrum of the gear vibration and bearing vibration with tooth error II measured at 700 rpm and 51 Nm.

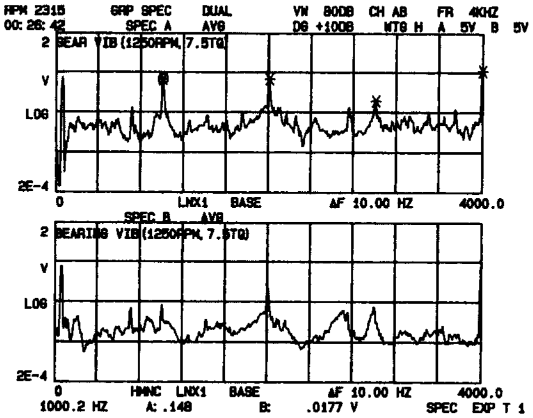

Fig. 14 Gear and bearing frequency spectrum with tooth error I(1250 rpm, 74 Nm)

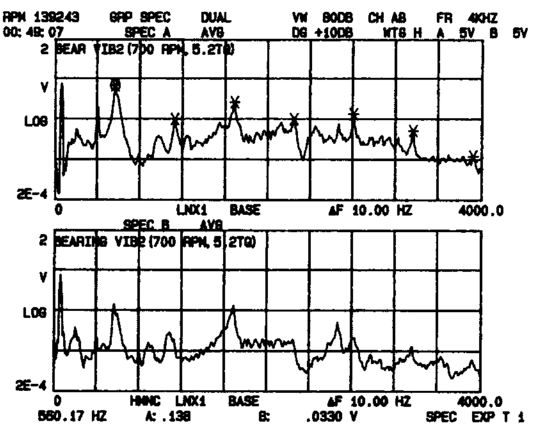

Fig. 15 Gear and bearing frequency spectrum with tooth error II(700 rpm, 51 Nm)

In this figure, the fundamental mesh frequency and its second harmonic are quite pronounced in the gear and bearing spectra. A significant peak at 400 Hz exists in the gear spectrum and is shown in all other measured data. Increasing the torque resulted in the highest peak at 400 Hz. This peak moves in the frequency dominan as the rotational velocity is vairied. Therefore, it may be caused by the bearing excitation or another periodic excitation from outside.

Figure 16 shows the frequency spectrum of gear vibration and bearing vibration with tooth error III measured at 1100 rpm and 74 Nm. A large peak near 650 Hz, though not a mesh frequency, exists and a second harmonic is also obvious. The

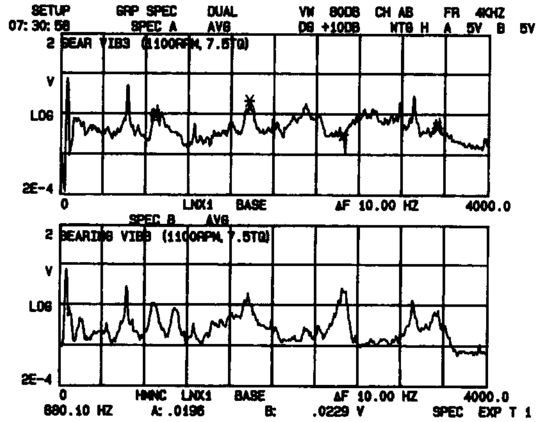

Fig. 16 Gear and bearing frequency spectrum with tooth error III(1100 rpm, 74 Nm)

large peak at 2700 Hz, unrelated to the fundamental mesh frequency, is shown in the bearing spectrum. Since the shaft rotational frequency is 18 Hz, it is not related.

The peaks corresponding to the fundamental mesh frequency and its harmonics in the gear are also shown in the bearing spectrum. But their reduced values indicate damped out response. In the bearing spectrum, other peaks unrelated to mesh frequencies exist. In most cases, the peaks are resonances because they do not shift with the rotational velocity. In order to calculate the natural frequencies of the system, a three-degree-of freedom torsional-transverse model is used (Kahraman and Singh, 1991). This is a simple model that accounts for the most important rotational motion in the helical gear. The mass moment of inertia of the driving gear of 5027.154 kg·mm^2, that of the driven gear of 4764 kg·mm^2, bearing stiffness of 1.1×10^8 N/m and mesh stiffness of 1.3×10^8 N/m are used as input data. Calculated natural frequencies are 1085Hz, 1404 Hz, and 2790 Hz, which are close to experimental results. The mode characteristics of the resonance also determine the magnitude of peaks in the gear and bearing spectrum.

The gear vibration is dependent on the dynamic effect of the mesh spring. The magnitude of the resonance depends on the relative motion between the two gears. That is to say, the resonance will be fairly small if the gears move in phase with each other. Small resonance amplitudes result in small peaks of the gear spectrum.

The bearings are the main transmission path for gear noise and therefore the forces generated in the bearings have the direct correlation with the structure-borne sound (Townsend, 1992). The bearing force and vibration are directly related to the amount of translational motion of the gears. If the participating resonant modes contain translational components, the natural frequencies will result in significant peaks in the bearing spectrum (Bolze, 1996). The bearing stiffness can greatly affect the dynamics of geared systems and the dynamic tooth load depending upon the relative stiffness of the other elements in the system. Decreasing the values of bearing stiffness beyond a certain value lowers the natural frequency governed by the gear mesh considerably and the amplitudes of the peak responses and dynamic loads (Kahraman et al., 1992). For this reason, it leads to the difference of peaks between the gear spectrum and bearing spectrum.

Gear mesh frequencies are more significant than those of the bearing and shaft at normal conditions. Therefore, the general gear vibration analysis should consider the fundamental mesh frequency, second harmonics and third harmonics simultaneously. However, the frequencies related with bearing and shaft should be also investigated in conjunction with the gear mesh frequencies for more accurate vibration analysis.

4. Conclusions and Summary

This study investigates the vibration transmission characteristics of a helical gear system with different types of lead error. For this purpose, an experimental test rig is designed and accelerations are measured in the gear and bearing. Based on the results, the following conclusions were obtained.

(1) An increase in torque does not always lead to the increase of vibration. Gear vibration is more sensitive to the rotational velocity than to the torque.

(2) Gear vibration shows important peaks in the mesh frequencies and up to its third harmonics.

(3) Gear vibration is significantly influenced by the direction of the lead error. The gears without lead error has less vibration than gears with lead error. Therefore, proper lead modification can reduce gear vibration.

(4) Gear vibration is transmitted to the bearing with reduced amplitude at the mesh frequency and its harmonics. Most dominant peaks of the bearing vibration are related to those of the gear vibration at the mesh frequencies.

(5) Bearing spectrum has other significant peaks in addition to the mesh frequencies. They are the resonance related to the participating modes.

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