Proceedings of The Institute of Acoustics

THE NOISE AND SOURCE IDENTIFICATION OF AUTOMOBILE GEARBOXES AND FINAL DRIVES

S. A. ANDREWS

AUSTIN MORRIS DIVISION BL CARS LTD

SYNOPSIS Two batches of gearboxes were tested to establish whether changes in manufacturing techniques had improved consistency in geometric toleruncing and its effect on vibration forcing arising from geometric errors. Measurements with digital gratings were used to determine whether a correlation existed between the noise frequencies and rotational geometric errors, the elimination of which may reduce the level of subjective perception of the noise. The paper shows that further work is necessary to determine whether there is a direct relationship between geometric errors and noise at sideband frequencies.

INTRODUCTION

A random selection of ten gearboxes was used to establish typical noise levels caused by gear tooth meshing and its sidebands and the effect adjustment in manufacturing methods had on this noise. Measurement of single flank geometry variation between input and output for each gear in the gearbox was determined and the importance of this relative to the noise emitted was established.

It is important to understand that the human ear is most sensitive to noise in the range 1-4 KHz and is still quite sensitive down to 600-700 Hz. Motor car gearboxes and axles create noise in this frequency range due to the gear teeth in the meshing pairs of gears, their harmonics and sidebands. The excitation frequencies at tooth run out, eccentricity, ovality etc are very much lower and virtually inaudible to the ear.

The following text outlines the steps taken to evaluate the effects of manufacturing changes to the gears, and the correlation between measured noise and the measured geometric deviation between input and output shaft rotation in the various gears.

TEST PROCEDURE

Noise measurements were recorded on magnetic tape while the gearbox was run under load on a test bed, and although this limited the assessment to loading on only one side of the gear teeth, it was sufficient for these initial comparisons of the gearboxes. The same test cell was used for all the gearboxes tested, and the microphone location was as illustrated in Figure 1. *

Measurements were taken over an input shaft speed range 1000-5000 rpm, and a high pass filter was used with a cut-off at 300 Hz to eliminate the unwanted low frequency noise and give a better signal level on the magnetic tape.

From the results obtained in this testing, two gearboxes having known characteristics were then subjected to measurement of single flank error testing (see Reference 1) in each of the gears, and this data was used to compare with the analysis of the noise generated during gearbox rig test.

*It is regretted that it is not possible to reproduce the figures for this paper.

THE SOURCE IDENTIFICATION OF AUTOMOBILE GEARBOX AND FINAL DRIVE NOISE

ANALYSIS

Analysis of the magnetic tape recordings was made using fast fourier transform computer methods (2) from which the typical Campbell's diagram and tracked orders of the predominating noise components of the gearboxes could be reproduced. Because the noise which the ear detects from gearboxes is complex and usually has components within a few Hertz of one another, the ear cannot detect these as separate frequencies or tones, but only as a complex pattern which appears as noise. Hence the need for a sophisticated means of analysis.

RESULTS AND DISCUSSION

Campbell's diagrams of the type shown in Figure 2 were used to identify the predominating frequencies present in the various meshing pairs in the gearbox, a diagram of this type being reproduced in each gear for each gearbox. In order to simplify an assessment as to the relative merits of the two manufacturing methods of producing the gears, histograms of the levels of noise for each gearbox were plotted and used to show the relative levels due to tooth meshing frequencies and their sidebands. Figure 3(a) shows a typical pair of such diagrams and compares the noise measured in first gear. From this data, it was a fairly simple matter to establish mean levels and the standard deviation to give a single figure for comparison between the sets of gears for each gear selected. The table in Figure 3(b) is based on results from each gear in each of the ten gearboxes tested.

It was clear from the histogram plots that the two gearboxes featured in Figure 3(a) warranted close examination to determine the marked difference in noise in first gear. Analysis of all the gearboxes had shown a marked predominance of a frequency corresponding to the constant meshing pair. This pair of gears only carries load when intermediate gears are selected; a typical computer plot of this tracked meshing frequency of the constant gears is shown in Figure 4(b) and can be compared to the overall noise level and with third gear meshing minus one. It also shows the presence of a number of resonances, particularly at 810, 1060, 1150 and 1260 Hz in both graphs on this figure. Figures 5-7 inclusive show the noise levels recorded from frequencies giving the highest levels in the two boxes numbers 388157 and 388583. The measurement technique developed at Cambridge University Engineering Department (1) was used to see what correlation existed between geometric variations in the gears and the noise measurements.

Some typical results for these two gearboxes are shown in Figure 8, and the summary of the measurements is given here.

Comparison of 388583 and 388157 suggests a slightly rougher final drive differential in the former, but a larger wheelspeed error in the latter. Third gear in 388157 has large beating effects on overrun from once per revolution and medium errors at once per tooth but with a beating effect, whereas 388583 has a larger but more regular error from the meshing frequency only; this latter should give a steady whine noise.

On Drive, 388157 gives an irregular pattern with some eccentricity, and once per tooth components; amplitudes are about one minute of arc. Gearbox 388583 gave no clearly repetitive pattern of errors, though the traces suggested a dominant eccentricity at input shaft frequency of about ten minutes of arc. Once per tooth errors of about three minutes of arc were present, and as this is rather large, it could cause noise.

Second gear once per revolution errors are comparable on the two gearboxes, so there would be little to choose between them. The once per tooth errors on 388157 are much more regular than on 388583, and the errors of the latter are beating together and would be expected to give more irritation than that from 388157.

First gear results are dominated by once per tooth effects, though 388583 has a higher harmonic content than 388157, and hence would be expected to be noisier.

Comparing these comments, based on measurements using 'Gratings in Driveline Noise Problems", with the actual noise measured under operating conditions for the gearboxes and shown in Figures 4-7 inclusive, the following comments are relevant.

Final drive differential had been established as a noise source in a previous investigation and corrections to manufacturing techniques based on single flank errors had brought about a marked improvement in noise. Thus the comments based on geometric measurements were correct.

Third gear noise for the two boxes can be seen by comparing Figures 4(a) and 6, but as no overrun measurement was taken, it is only possible to compare the drive conditions.

Figure 6 does indeed show once per tooth and a sideband of once per tooth minus one for 388157, which arises from eccentricity (3) (4) (5). Figure 4(a) for 388583 shows once per tooth minus one and is in keeping with the eccentricity measured; however, once per tooth noise was relatively small and thus is not in keeping with the geometric prediction.

Second gear noise was very low in gearbox 388157, but Figure 5 shows the level for 388583. This shows that tooth meshing frequency is the one giving the higher levels of noise, although the 'beating' suggested by the geometric analysis was not shown to predominate. However, the prediction from the geometric analysis that gearbox 388583 would give higher noise in second gear was correct.

First gear noise was only of concern in gearbox 388157 as shown in Figure 7, and hence the geometric analysis, which predicted a high level of the harmonic was not in agreement with the measured levels.

CONCLUSIONS

Only in first gear tests was there complete disagreement between the geometric light load tests and noise tests under representative power and speed, rig testing. Otherwise, the findings of the two methods were in reasonable agreement, and the measurement of geometric variation using the 'Gratings' measurement technique must be a useful means of determining errors in gears which may give rise to noise.

THE SOURCE IDENTIFICATION OF AUTOMOBILE GEARBOX AND FINAL DRIVE NOISE

Insufficient data is available at this stage to establish a relationship between dimensional geometric errors and the measured sideband noises.

ACKNOWLEDGEMENTS

The Author would like to thank those colleagues who assisted in obtaining the experimental data and analysis of the noise measurements which are contained in this paper.

Thanks are also extended to Dr J D Smith for providing the geometric analysis and his assistance in discussions leading to a fuller understanding of the gear noise problems.

Thanks are also extended to Austin Morris Division, BL Cars Ltd, for their encouragement and permission to publish this work.

REFERENCES

1) K J DALY and J D SMITH 'Using Gratings in Driveline Noise Problems'

I Mech E Conference; Noise and Vibrations of Engine and Transmissions.

2) S A ANDREWS Modern Analysis Techniques Associated with Gearbox and Axle Noise

I Mech E Conference; Noise and Vibrations of Engine and Transmissions.

3) D B WELBOURN 'Gear Errors and Their Resultant Noise Spectra'

I Mech E Conference; Gearing 1970

4) W D MARK 'Analysis of the Vibrating Excitation of Gear Systems'

Journal of Acoustics Society of America

5) KURT G F MOELLER 'Gear Noise'

Handbook of Noise Control by HARRIS