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Signal Analysis Techniques to Identify Axle Bearing Defects

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ABSTRACT

Vehicle NVH (Noise Vibration & Harshness) is of continued concern to customers in this increasingly competitive market and driveline NVH performance is a key factor in overall vehicle quality. A typical way to increase this quality is the use of axle end of line test stands that utilize NVH signal analysis methods to offer pass/fail criteria. In the manufacturing environment there are high costs associated with axle assemblies that are rejected due to the amount of time for NVH analysis to determine root cause for the failure. Of more interest to both product development and manufacturing activities is the ability to understand the root cause of the failures from the axle end of line test stand. This information can improve the manufacturing process by eliminating errors, streamlining re-build activities, aiding in product design improvements, and in turn reducing cost.

This paper describes the activities used to identify specific bearing defects for rear axle application through signal analysis of end of line test stand data. Measurements of both nominal and intentionally mis-built (head, tail, & differential) bearings, with known defect characteristics, were used to develop the analysis techniques. Various bearing fault frequencies and their harmonics of the NVH signatures were used to identify the specific failure mode for each of the mis-built (head, tail, & differential) bearings, which included rolling element pass defects, and cage spin defects.

INTRODUCTION

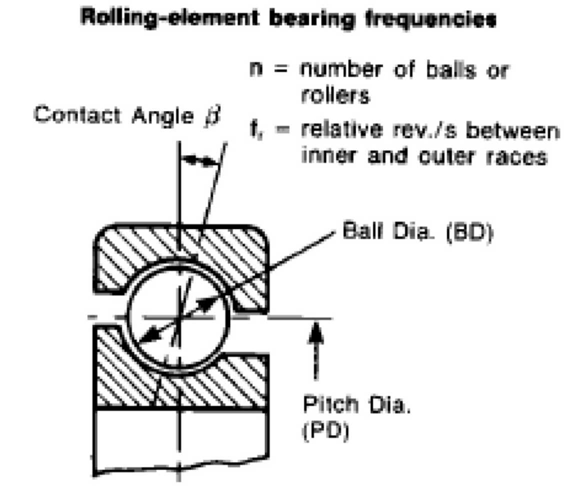
Noise and vibration measurements conducted on final assemblies at manufacturing facilities can be used in great detail to extract information regarding the product, including the quality of the components and the quality of the assembly process. The retrieval of this information is often times difficult due to improper measurements or more often due to incomplete data analysis. The noise and vibration signature generated by a product contains countless pieces of information that can be used to characterize the assembly. This is of great importance, particularly in manufacturing environments to further understand the effectiveness of the processes used to develop and assemble the final products. This holds particularly true for axle development. End of line testing is often used to characterize the noise and vibration performance of axle assemblies prior to shipment to the end user. The data is typically analyzed to understand the quality of the final assembly, but can also be used to further understand the manufacturing processes. If an assembly does not pass the end of line test stand it is extremely useful to understand the reason for the failure and to have the knowledge of which portion of the assembly process was responsible for the failure.

IDENTIFICATION OF APPLICATION

A manufacturer of automotive axles set out to determine if it was possible to identify specific bearing defects from existing vibration data from an end of line vibration test stand. At the time, the vibration signatures being collected were not analyzed specifically to identify bearing defects. The purpose of this investigation was to identify the proper analysis techniques to identify and separate assemblies with bearing defects from the rest of the assembly population. To begin the investigation, several axles were assembled with intentional bearing defects. These axles, along with axles that incorporated nominally built bearings were run on the end of line test stand and the standard vibration measurements were conducted. The specific mis-built bearings included a flat on one of the rolling elements and bending of the bearing cage to induce rubbing. In total ten assemblies were utilized for this study, including six assemblies that had a combination of two out of three (pinion head, pinion tail, and differential) bearings with defects, and four assemblies with nominally built bearings. Of the mis-built assemblies each had a combination of one rolling element flat and one cage rubbing.

ANALYSIS TECHNIQUES - FOURIER (FFT)

The test events of the end of line test stand used for this investigation included a steady state condition of 1500rpm (input speed) at a constant drive torque, followed by a constant coast torque. Vibrations were measured using a linear accelerometer located at the rear of the axle carrier and a telemetry system measuring torsional acceleration at the axle pinion nose. Initial data analysis used Fourier (FFT) analysis of the bearing fault frequencies to determine if the assemblies of mis-built bearings could be identified from the assemblies of nominal bearings. The bearing fault frequencies can easily be calculated from the bearing geometry using the following formulas (see Figure 1).[1] Note that the relationships assume pure rolling motion while in reality there exist some sliding. Therefore, the equations should be regarded as approximate.[1]



Impact Rates f(Hz) (assuming pure rolling motion)

For an outer race defect: $$f\,(\text{Hz}) = \frac{n}{2} f_r \left(1 - \frac{BD}{PD}\cos\beta\right)$$

For an inner race defect: $$f\,(\text{Hz}) = \frac{n}{2} f_r \left(1 + \frac{BD}{PD}\cos\beta\right)$$

For a ball defect: $$f\,(\text{Hz}) = \frac{PD}{BD} f_r \left[1 - \left(\frac{BD}{PD}\cos\beta\right)^2\right]$$

For a cage defect: $$f\,(\text{Hz}) = \frac{1}{2} f_r \left(1 - \frac{BD}{PD}\cos\beta\right)$$

Figure 1. Formulas for calculating bearing fault frequencies

Data was analyzed for both the drive torque and coast torque. It was determined that a pass/fail criteria could not be set solely on picking associated bearing fault frequencies (as those calculated from the formulas given in Figure 1) from the FFT data because not all of the fault frequencies calculated were distinguishable within the defective parts compared to the nominal parts. Because the vibration signals of a faulted bearing are small compared to shaft order and gear mesh excitations, detection of bearing fault frequencies using Fourier analysis can sometimes be difficult. The conclusion was reached that additional analysis techniques were necessary to identify the mis-built bearing conditions.

ANALYSIS TECHNIQUES - ENVELOPE

The next analysis technique investigated concentrated on envelope analysis. This is a technique of extracting an amplitude-modulated signal typically found in roller bearing defects. Most often the bearing fault frequencies are of low energy compared to the overall energy in the vibration signal. To extract the bearing fault frequencies from the rest of the vibration spectrum, a band-pass filter is applied where structural resonances of the system exist. If a rolling element has a defect there will be an impact created every time the rolling element passes over either of its bearing races. These shock impulses spread through the bearings causing the bearings to "ring" at its natural frequency or resonance. The resulting impulses repeat periodically at a rate determined by the location of the defect and by the bearing geometry as described above (Figure 1). Typically, the higher harmonics of the repetitive impulsive events are amplified at the structural resonances of the test part. Envelope analysis centered at a structural resonance reveals the occurrences of the impulsive events.[3] When using the approach of applying a band-pass filter centered at a structural resonance there may be some drawbacks.[4] First, even though the bearing fault frequencies will be correct, the vibration level measured becomes very sensitive to accelerometer placement and these values may represent the structural properties rather than the bearing condition. Second, the resonance cannot be very lightly damped, because this will cause a broadening of the envelope and may compromise the envelope spectrum. The center frequency chosen for the envelope analysis in this study was 3 kHz as seen in Figure 2. Though other frequencies were explored, the 3 kHz envelope showed the most promising results.

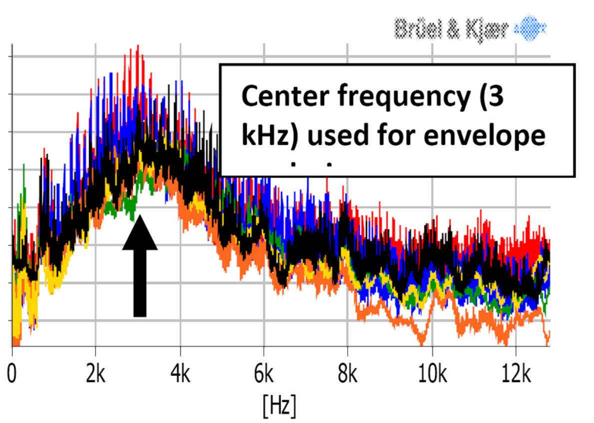


Figure 2. FFT of assemblies with mis-built bearings to determine structural resonance.

By applying an envelope centered at 3 kHz, it can be shown that the amplitude-modulated signal is demodulated so that the result is a greater increase in signal to noise ratio of the fault frequencies of interest. This is due to the fact that the structure of which the bearing is a part of has some natural frequencies or resonances that are very sensitive to excitation and these resonances will amplify the bearing fault frequencies, giving a better signal to noise ratio. For example, when trying to distinguish the differential bearing roller defect (54 Hz), the FFT plot in Figure 3 does not show the 54 Hz fault frequency at all, and there is no distinguishable difference between a nominally built differential bearing and the two mis-built bearings tested. However, when using the envelope analysis method you can readily see that the 54 Hz fault frequency stand out for both mis-built bearings compared to the nominally built bearing.

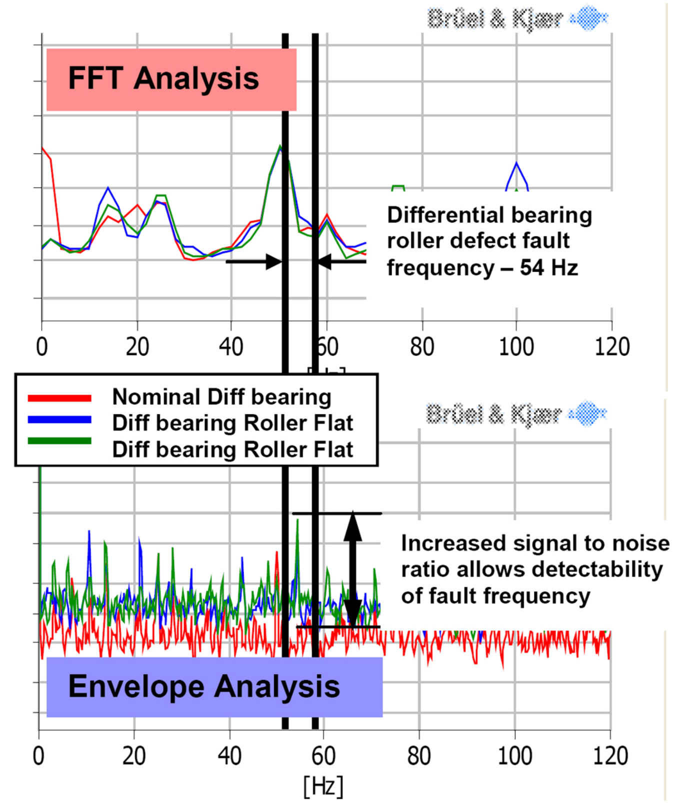


Figure 3. Difference of signal to noise ratio when locating differential bearing roller defect fault frequency (54 Hz) between FFT analysis and envelope analysis.

ENVELOPE ANALYSIS - SEPARATION ISSUES

The envelope analysis technique described above does provide a method to ensure that there is sufficient separation between all of the mis-built bearings compared to the nominally built bearings as tested by the end of line test stand. However, it is desired to be able to separate failure modes between each of the three bearings (pinion head, pinion tail, and differential) associated to the axle assembly. This information would allow the manufacturer to further understand the variability in the manufacturing processes and help to focus efforts to improve the overall process. Additionally the information could be used to streamline re-build activities for assemblies that initially fail the end of line test stand. Continued data analysis efforts focusing on each of the fault frequencies shows (as seen in Figure 4) that the calculated fault frequency can sometimes shift in frequency. Consequently allowing for a range to be selected around the calculated fault frequency is necessary to catch this variation. This frequency shift can possibly be attributed to slip effects considering that the bearing fault frequency relationships assume pure rolling motion while in reality there exists some sliding. Therefore, the bearing equations should be regarded as approximate.[1]

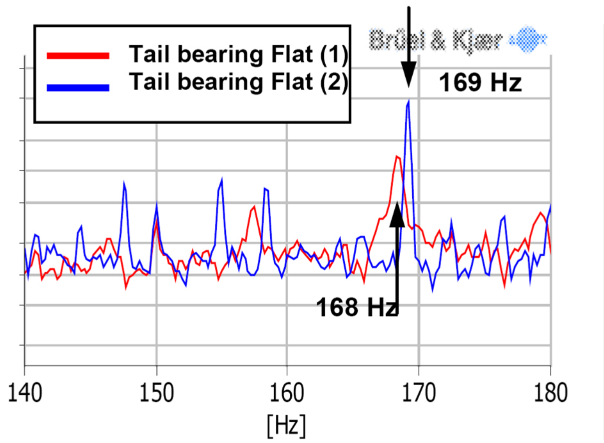


Figure 4. Slipping effects show that both curves with the same (tail bearing) roller fault frequency can produce slightly different fault frequencies.

Inherently when dealing with multiple bearings with similar geometries, it is difficult enough to set pass/fail criteria from selecting frequency peaks to be able to separate dominant fault frequencies with all of the associated sidebands and harmonics let alone having to widen the frequency peak selection range due to slipping. This was a problem encountered for a number of the bearing defects within the scope of this study. In particular, all of the (pinion head, pinion tail, & differential) cage defects were not separable from a mis-built bearing standpoint as shown in Figure 5. Meaning when there was a failure for cage defect, there was not enough separation to be able to root cause which bearing was actually failing. Also, only one of the three roller defects (differential) could not be separated from a mis-built bearing standpoint. Here, the differential roller defect fundamental fault frequency (54.25 Hz) is too close to the fifth harmonic head cage frequency (53.75 Hz), which can be seen in Figure 6. However, it should be emphasized that there was enough separation to be able to distinguish between mis-built bearings and nominally built bearings.

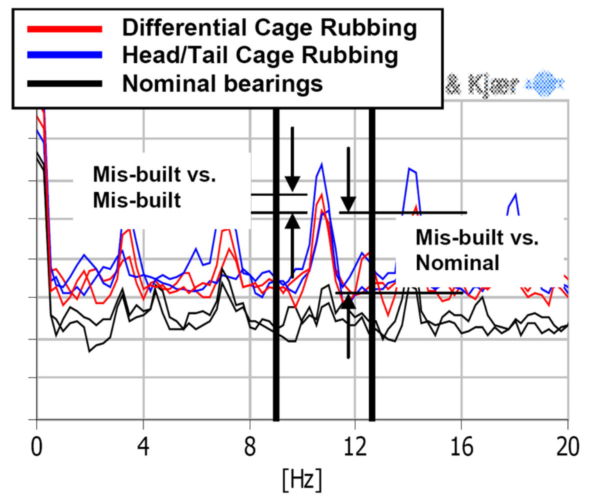


Figure 5. Separation difference of mis-built bearings and nominally built bearings and also separation of different mis-built bearings.

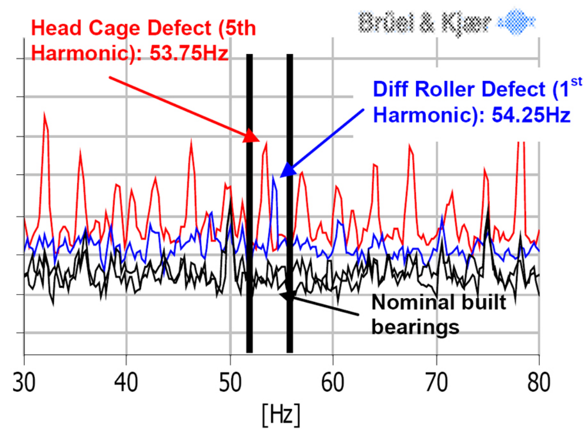


Figure 6. Separation issue regarding differential rolling defect and head cage defect.

PROCESS IMPLEMENTATION

With the analysis method defined to identify each of the bearing fault frequencies for this study, the next step is to implement this method into the production process. This process will provide additional checks to ensure that assemblies with bearing defects are contained within the production process and not released to the end customer. Implementation of this process provides manufacturing engineers with additional data to continue to improve the assembly processes. Obtaining the information to identify the mis-built conditions is particularly significant because it does not require any additional testing or manufacturing cycle time, it simply requires analysis of data that can be easily obtained from the end of line test stand. The analysis technique developed for this particular application can very easily be carried over to additional product lines to provide similar benefits. Specific pass/fail criteria will likely be unique for each application; however the analysis method will likely be similar.

SUMMARY/CONCLUSIONS

Based on the defects evaluated for this study (roller flat and cage rubbing) on each of the bearings (pinion head, pinion tail, & differential) within an axle assembly there was always an ability to distinguish mis-built bearings from nominal bearings tested on the end of line test stand utilizing envelope analysis with a properly placed envelope window. However, even though it is advantageous to be able to separate failure modes, this was not always achievable due to similar dominant fault frequencies with all of the associated sidebands, harmonics, and accounting for slip effects.

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DEFINITIONS/ABBREVIATIONS

SA
: sample abbreviations

UBT
: use borderless table ≤ 3.5 inches wide

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