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ACOUSTIC IDENTIFICATION OF A SPUN CONE ROLLER BEARING DEFECT

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Summary

Acoustic emissions and accelerometer vibrations were recorded from a laboratory-induced spun cone roller bearing defect in tests conducted by the Association of American Railroads (AAR), Transportation Technology Center (TTC). The vibration response of the spun cone defect was distinctly different than that of a "good" bearing. The spectral content of the vibration signatures showed its largest amplitudes below three (3) times running speed. Important peaks were detected at one (1) time running speed, sidebands around two (2) times running speed, and near the bearing cage frequency.

Of all the bearing defect types to be detected, the most challenging and highest priority is that of a spun or loose cone spinning on a journal. It is known that this defect type is responsible for the majority of bearing-related derailments. For several years, the AAR has been developing new techniques to detect defective roller bearings as part of their new generation wayside acoustic detector program.

Today there is no reliable method of detecting a loose or spinning cone in its early stages. In its latter stages the bearing generates heat as its internal components degenerate rapidly. At this stage, the bearing may be "caught" by a Hot Bearing Detector (HBD) or it may cause a burn-off derailment between HBD locations due to the rapidity of the event. Early detection would prevent derailments and allow preventative maintenance to be performed on the bearing without costly stoppage of trains on the right-of-way.

The steps in the identification process include producing and identifying the defect in the laboratory and extending the technique to a simulated revenue service environment. The AAR completed laboratory testing of spun cone defects in July 1996 at TTC, Pueblo, Colorado. A simulated revenue service test was recently completed in November 1996. This report covers a description of the technique used to identify the spun cone defect based on laboratory data. The technique will be applied to the field test data in 1997.

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INTRODUCTION AND CONCLUSIONS

An extensive laboratory test program classifying the acoustic and vibration emissions of various bearing defect types was conducted in 1995-96 by the Association of American Railroads (AAR), in conjunction with the Federal Railroad Administration. The spun cone defect was the last to be tested in July of 1996 at the Transportation Technology Center (TTC), Pueblo, Colorado. The following conclusions are presented from the spun cone data analysis:

- The vibration characteristic of a bearing with a spinning cone is different from that of a "good" bearing.
- The distinctive spectral components, when properly demodulated, are below three (3) times running speed.
- The loose cone moves relative to the journal rotation (called slip) at a slow rate proportional to the running speed.
- Loading of the bearing has little effect on the significant spectral components.
- Increased cap screw torque decreases the significant spectral components of vibration.
- A neural network can be trained to distinguish between a spun cone and a good bearing.

The occurrence rate of bearing-related derailments has remained relatively constant over the past ten years, in spite of important changes to the industry-wide bearing reconditioning standards and an extensive network of hot bearing detectors. The goal of the AAR bearing defect detection research program has been to develop techniques for improved wayside defect detection to prevent derailments while managing costly train stoppages due to false detector "hits."

A key to achieving the research goal is the identification of the spun cone defect — a major culprit in bearing-related derailments. Although acoustic wayside detectors have been in service for almost a decade, the rudimentary analysis techniques are insufficient to identify this important defect. Therefore, it has been a major subtask of the AAR to isolate and study this particular bearing defect type, both in the laboratory and in simulated revenue service testing.

TEST SETUP

Exhibit 1 shows a diagram of the spun cone test bearing and the microphone location used to record the acoustic emissions. Two spun cone bearings were tested — one Class E (6"x11") bearing and one Class F (6 1/2"x12") bearing. Each was subjected to a series of runs where the bearing axial bolt clamp load, the radial load, and running speed were varied independently in a systematic manner. Five end-cap bolt clamping torques (axial loads) were used: 0, 40, 60, 80, and 160 ft-lbs. Two radial loads equivalent to an empty car condition (8,000 pounds) and a loaded condition (27,000-33,000 pounds) relative to the bearing capacity were also applied. Bearings were tested at equivalent train operating speeds of 25 mph to 70 mph in 5-mile per hour increments.

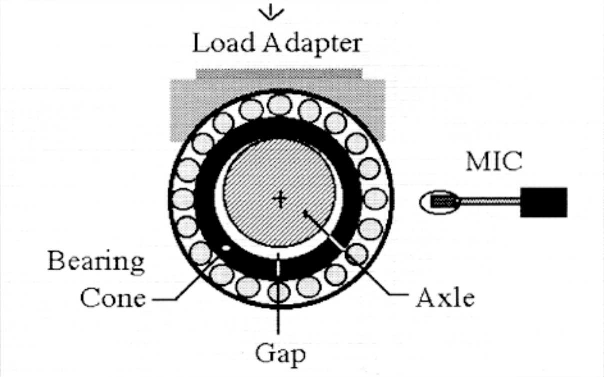

Exhibit 1. Spun Cone Test Bearing and Microphone

Bearing cone rotation rate (about the axle) was measured with tachometers attached to each bearing (inboard and outboard) cone assembly. Applied radial loads and outer cup temperatures were also monitored. A microphone and an accelerometer was used to record acoustic emissions and bearing cup vibrations, with sufficient sampling rates to identify all frequency content for each sensor up to 40 kilohertz (kHz).

Each spun cone test bearing contained no additional component defect other than the groove (gap) between its inboard cone and axle mounting diameter. The groove gap was 0.015 inch for the Class E bearing and 0.065 inch for the Class F bearing. Only the inboard cones spun during testing. The outboard cones did not rotate relative to their mounted axle surface during any test.

RESULTS AND ANALYSIS

A major step in the analysis involves processing all vibration data with demodulation techniques or an envelope detector.[1] Envelope detection extracts the low frequency bearing component operating information from higher frequency carrier vibrations generated by loose components in the test bearings.

Exhibit 2 is a summary scatter plot of the largest demodulated spectral peaks measured in 132 separate tests. This exhibit reveals that dominant peak indicators from loose components are found at frequencies below three times running speed.

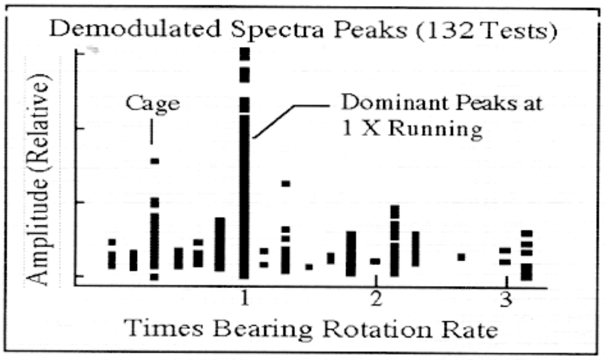

Exhibit 2. Demodulation Spectra Peaks

The largest and most common of the vibration peaks from all tests appeared at cage rotation, one time running, and at sideband frequencies slightly above or below two times running. These results are consistent with observations from the simulated burn-off tests performed in 1995 at TTC.[2]

Cone slip detectors provided direct measures of cone rotation as it lagged in speed relative to the driven axle. Exhibit 3 provides all measured slip rates from testing plotted over the full range of test speeds. The Class E and F size bearings had distinct average slopes when plotted in this manner. The slope of the observed slip is directly related to the gap (depth of the groove) between the cone bore and the mounted axle diameter.

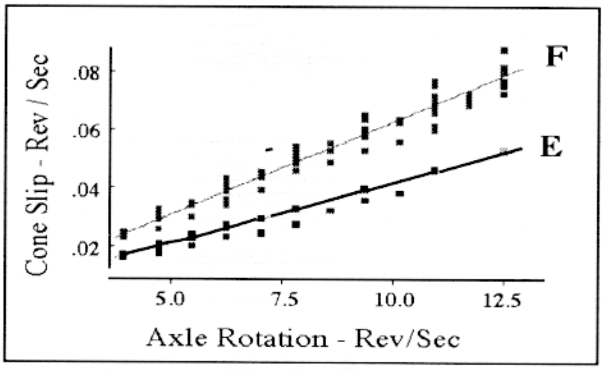

Exhibit 3. Cone Slip Versus Axle Speed

As shown in Exhibit 4, the percent slip is nearly constant with speed for the two bearing sizes (and gaps) tested. If pure rolling contact between the cone bore and the axle were maintained, one could expect the relative percent slip to be defined by the gap-to-axle ratio.

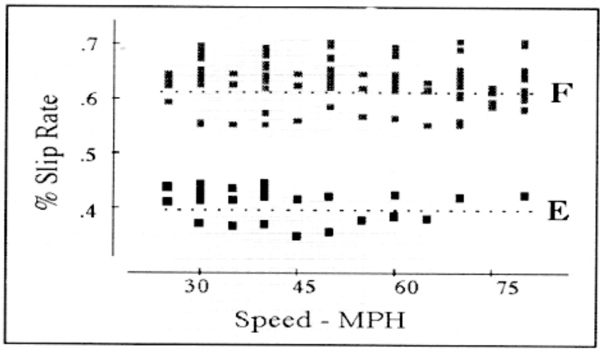

Exhibit 4. Slip Rate as a Percent of Speed

For this test, the ratios were .26 percent for the size E bearing (.015/5.665) and 1.06 percent for the size F bearing (.065/6.124).

These ratios correspond to the average values of .39 percent and .62 percent provided by the cone slip detector (inboard only) shown as dotted lines in Exhibit 4.

Observations showed cone slip and bearing vibration amplitudes were affected by the bearing load and/or cap screw bolting torques. Exhibit 5 shows the dependence of the running speed vibration amplitude on bearing load as measured during testing. Simply stated, higher bearing load yields smaller vibration amplitudes. This implies that bearings with loose cones on empty cars may be easier to identify if vibration amplitudes are being monitored.

Exhibit 6 provides experimental evidence that cap screw bolting torque affects bearing vibration when a spun cone is present. Increased bolt torques lowered the vibration in the present tests. In fact, the exhibited data implies that cap screw torques in excess of 300 ft-lbs would almost eliminate the observable vibration, even if the gap is .065 inch — the maximum test case reviewed.

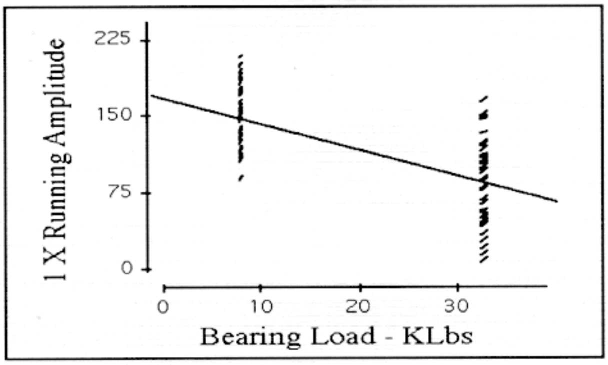

Exhibit 5. Vibration Amplitude Versus Load

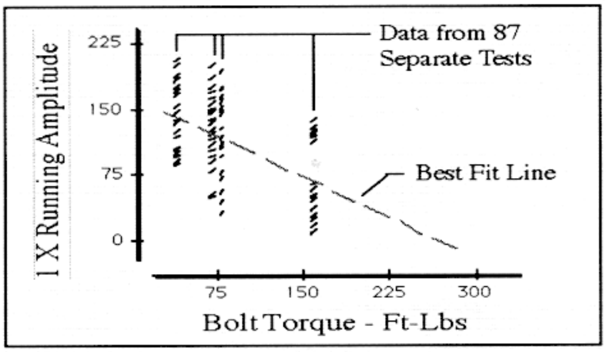

Exhibit 6. Amplitude Versus Cap Screw Torque

FUTURE PLANS

Analysis of recently completed wayside tests should provide answers from field data that represents real rail service conditions with spun cone bearings. Additional laboratory testing with larger grooves might improve the understanding of how these dangerous bearings react under different service conditions as their operating lives come to an end.

REFERENCES

1. Wang, John, Gerald Anderson, Richard Smith, "A New Detection Technique to Identify Defective Railroad Bearings," *Technology Digest* TD 96-004, Association of American Railroads, February 1996.
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3. Sneed, William, Gerald Anderson, Richard Smith, "Freight Car Roller Faults Identified with a Laser Vibrometer at the Conrail Wheel Shop," *Technology Digest* TD 96-019, Association of American Railroads, September 1996.

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