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EXHIBIT	
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NATIONAL TRANSPORTATION SAFETY BOARD
Investigative Hearing
Norfolk Southern Railway general merchandise freight train 32N derailment with subsequent hazardous material release and fires, in East Palestine, Ohio, on February 3, 2023

Agency / Organization

American Society of Mechanical Engineers

Title

Exhibit 5- Tarawneh, IJAV, November 2015

ture of all bearings along the same side of the train. If a bearing is running significantly hotter than the average temperature by some predetermined threshold, it is removed from field service and labelled as "trended."[1]

In addition to the HBDs, acoustic measuring devices known as the Trackside Acoustic Detection System (TADS®) have been used in the field to identify defective bearings. The success rate of capturing a defective bearing is heavily based on the severity of the defect. Bearings with large defects known as "growlers" have a much higher rate of being recognized as opposed to bearings with smaller defects.[2] Although nearly five thousand HBDs are currently in service, only fifteen TADS® have been implemented in North America,[3] which means a train can run thousands of miles before encountering an acoustic bearing detector. Furthermore, the majority of trended bearings are found to be defect-free, which means that TADS® will not detect them.

From the above discussion, it is evident that the existing wayside monitoring equipment utilized in field service does not constitute a true continuous bearing health monitoring system, but rather a sporadic check on the bearing condition. Thus, neither temperature nor acoustic measuring devices, currently in use, can accurately characterize the internal condition of a temperature-trended bearing. To this end, the railroad research group at the University of Texas-Pan American (UTPA) conducted a series of experimental and theoretical studies focused at understanding the bearing temperature trending phenomenon and its root causes, with the main objective of finding ways to distinguish healthy (defect-free) bearings undergoing temperature trending from defective bearings nearing catastrophic failure. The latter will aid the railroad industry in minimizing unnecessary train stoppages and false bearing removals.

2. BACKGROUND

Thermal investigations of roller bearings have been carried out for a few decades now.[4–8] These efforts range from purely theoretical works to studies containing a combination of theory and application to purely experimental projects. The theoretical studies examined two abnormal operating conditions: partially or fully jammed roller bearings and a stuck brake. A partially jammed roller bearing is one that rotates with velocities greater than zero but less than the epicyclical speed, while a fully jammed roller bearing is one that has no velocity with respect to the cage. The investigation into the jammed rollers and a stuck brake condition revealed that the maximum temperature within the bearing assembly can reach 268°C (514°F) and 126°C (259°F), respectively, compared to the normal operating condition temperature of 81°C (178°F).[5,6] In a similar study, a dynamic model of the torque and heat generation rate in tapered-roller bearings under excessive sliding conditions was developed using a lumped-parameter approach in the program SHABERTH.[7] This investigation focused on jammed roller bearings, and the model was run with an assumed ambient temperature of 25°C (77°F), a load per row of 80,000 N (18,000 lb), and a rotational speed of 560 rpm, which corresponds to a train speed of 97 km/h (60 mph). The study concluded that the heat generated in the bearing was proportional to the number of jammed rollers and that the heat generation at the roller-end contacts increased with the number of jammed rollers.

In yet another theoretical work, finite element analysis with ABAQUS and FORTRAN was used to model thermally induced failures in railroad class F ($6\frac{1}{2} \times 12$) tapered-roller bearings, which resulted from laboratory experiments conducted at high operating speeds.[8] The experiments were conducted by the Association of American Railroads (AAR), and showed that new (defect-free) bearings failed after 200 to 300 hours of operation at a speed of 161 km/h (100 mph) and none failed at 129 km/h (80 mph).[8] The study concluded that the thermal and mechanical instabilities in a railroad roller bearing are directly related to the heat generated at the roller-end contacts. The increase in the heat generation is a direct consequence of the grease starvation mechanism caused by the high operating speeds, which results in a larger friction coefficient.[8]

Understanding that manufacturing tolerances of bearing assembly components are hard to maintain on a large-scale production basis, in the 1970's Jamison et al. conducted a detailed study of the geometric effects on the rib-roller contact in tapered-roller bearings,[9] as the latter directly influences bearing performance. As anticipated, their findings indicate that out-of-tolerance components can result in abnormal operating conditions leading to excessive friction and rapid wear at the rib-roller contact region, which can cause roller skew and grease starvation that may eventually lead to catastrophic bearing failure. Consequently, a number of other theoretical and experimental studies have been performed to explore the effects of geometrical imperfections and surface irregularities on the vibrational characteristics of tapered-roller bearings under varying axial loading and rotational speeds[10–12] and to detect bearing defects using frequency domain analysis.[13] Gupta investigated the dynamics of a tapered-roller bearing by modelling the general motion of the roller and cage based on the frictional behaviour and cage clearances of the bearing.[14] His study showed that roller skew (misalignment) increases with increasing friction. At relatively high friction and low cage-pocket and guide-land clearances, the roller was found to maintain steady contact with the cage pocket on one side while the contact is cyclic (pivoting) on the other end. Gupta's findings were later confirmed by an experimental study that was conducted by Yang et al. using specialized capacitance probes to measure and examine the effects of speed and lubrication on the degree of the roller skew (misalignment).[15] It was discovered that the friction between the rib-roller end contact causes the roller's large end (pointing toward the grease seal) to run ahead of the smaller end (pointing toward the spacer ring), thus, leading to roller misalignment. The degree of the roller skew may vary between rollers, and it increases with increasing speed and the lubrication of the larger end of the roller.

Despite the progress made in understanding the dynamics of tapered-roller bearings, no methods or techniques have been developed to identify temperature trending events that result from vibration-induced roller misalignment. Tarawneh et al. conducted a series of experimental and theoretical studies aimed at exploring temperature trending in railroad bearings, finding the root cause of this troubling phenomenon and devising ways to identify it using vibration analysis techniques.[16–22] The authors first set out to replicate the discoloration of tapered rollers (evidence of heating) observed in the trended bearings removed from service.[16] Theoretical modelling agreed with the laboratory testing results in proving that extreme roller temperatures can occur without noticeably raising the temperature of the bearing cup outer surface—the part of the bearing

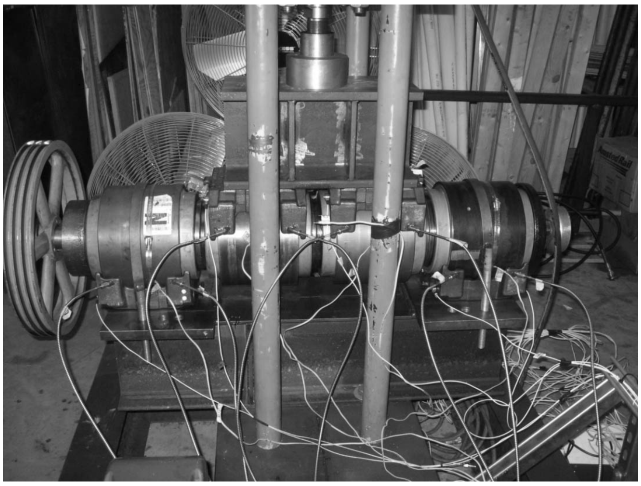

Figure 1. A picture of the four-bearing dynamic tester at UTPA showing the sensor placement.

scanned by the current wayside detectors.[17–19] For example, one of the case studies demonstrated that it is possible for three consecutive rollers within a cone assembly to operate at an abnormal temperature of 232°C (which would cause roller discoloration), yet the bearing cup temperature stays at 88.5°C.

With the new understanding of heat transfer paths of tapered-roller bearings based on experimental and theoretical studies,[17–19] Tarawneh et al. next conducted a series of tests on "trended" bearings pulled from field service along with their mates (the bearing on the opposite end of the axle).[20] The latter work was based on eight laboratory experiments performed on a four-bearing dynamic tester, depicted in Fig. 1, with an axle rotational speed of 536 rpm, which is equivalent to a train traveling at 91.7 km/h (57 mph). One experiment featured a trending event in which the bearing cup temperatures of two bearings experienced a 24°C (43.2°F) increase over a period of 50 minutes.[20] This behaviour occurred simultaneously in a field-trended bearing and its mate, which were separated by a control bearing. The synchronized temperature response ruled out heat transfer as the source for the abrupt change in temperature. The only other explanation for the observed phenomenon is vibration induced heating, where the effects of the oscillations of the rollers in the trended bearing travel along the axle and trigger similar behaviour in the mate bearing. Upon disassembly and inspection, a spall was found on one of the cup raceways of the trended bearing, providing evidence that roller vibration could be the root cause of the temperature trending. Subsequent laboratory tests were effective in achieving temperature trending by using a bearing with a known cup raceway defect to trigger vibration-induced roller misalignment, which is responsible for producing excessive frictional heating in short time periods. The aforementioned laboratory findings were validated in a carefully controlled and executed field test.[21] A bearing in field service, unlike the laboratory setting, is exposed to a variety of vibration excitation sources such as wheel impacts resulting from wheel flats, railcar hunting, and rail track defects. Hence, the performed field test utilized defective wheels as the vibration source to initiate temperature trending in the test bearings. The field test was successful in reproducing several bearing temperature trending events, thus, providing further evidence to verify the laboratory results.[21,22]

The authors used vibration-monitoring techniques to identify dynamics likely associated with roller misalignment and correlate the onset of such dynamics with abrupt changes in bearing temperature. Depending on the relative spacing between the rollers, cage bars and cones, and on the magnitude of the vibration trigger, roller oscillations can be severe and are more likely to lead to rollers being caught misaligned as they enter the loaded region of the bearing. In such cases, the roller's large end will tend to lead due to the friction between the cone rib and the roller end contact point. Consequently, excessive frictional heating will be generated as the severity and number of misaligned rollers increases. The latter will manifest in a sudden rise in bearing-cup temperature. Friction created by sliding misaligned rollers will reduce the epicyclical speed of the cage, which may also vary based on the severity and number of misaligned rollers. In the presence of misalignment, a roller will occupy more space within the cage pocket causing deformations of the cage and essentially creating a tighter fit throughout the cone assembly. Hence, it is expected that the occurrence of roller misalignment will effectively decrease the level of vibration within a bearing as rollers no longer roll or have space to oscillate within the cage pockets.

Roller realignment can occur for one of many reasons including thermal expansion in the cone assembly components, grease circulation, external vibration sources, etc. During the realignment process, a roller will pivot within the cage pocket as the cage and radial clearance reduces forcing rollers to realign. This behaviour is similar to the investigations conducted by Gupta in which one end of the roller maintains steady contact with one side of the cage pocket while the other end pivots.[14] In this scenario, the constant pivoting interaction between the roller and raceway surfaces may explain why defects tend to generally develop on raceways anywhere between the centre of the roller and the smaller end. The pivoting motion, while detrimental to a bearing's life, may promote grease flow, decreasing the temperature to or below normal operating conditions. The cyclic behaviour of the roller will affect the rotational frequency of the cage, momentarily speeding up or slowing down depending on the location and impact of the roller within the cage pocket. Overall, the vibration levels are expected to increase due to the pivoting motion of the roller. Hence, it is essential to quantify the total vibration energy of a bearing in order to garner vital information regarding its operating condition.

It has been experimentally observed that a bearing operating at normal conditions maintains a steady level of vibration energy with minor fluctuations. However, a bearing's vibration energy tends to decrease shortly prior to an increase in temperature and will increase prior to a drop in temperature. This observation proposes a relationship between vibration energy, roller misalignment, and temperature trending events.

The work presented in this paper differs from others in that it focuses on relating the temperature and vibration energy characteristics of a bearing experiencing a temperature trending event in an effort to devise an algorithm that can be used to differentiate between defective bearings nearing the end of their service life and defect-free bearings exhibiting a trending event. To this end, the following methodology was utilized to meet the objectives of this study:

1. The bearing components' fundamental frequencies were determined from a two-dimensional kinematic model.[21,22]

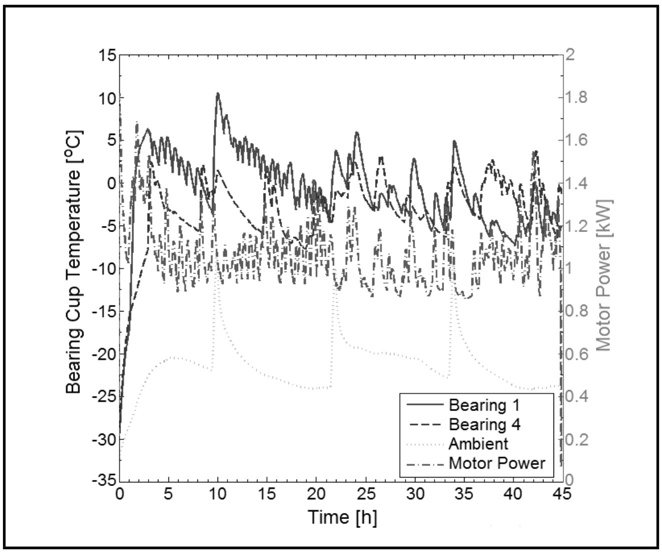

Figure 19. Experiment 5 bearing temperature profiles vs. motor power; defect-free class K bearings; freezing ambient conditions; speed: 40.23 km/h (25 mph), load: 17% (empty railcar).

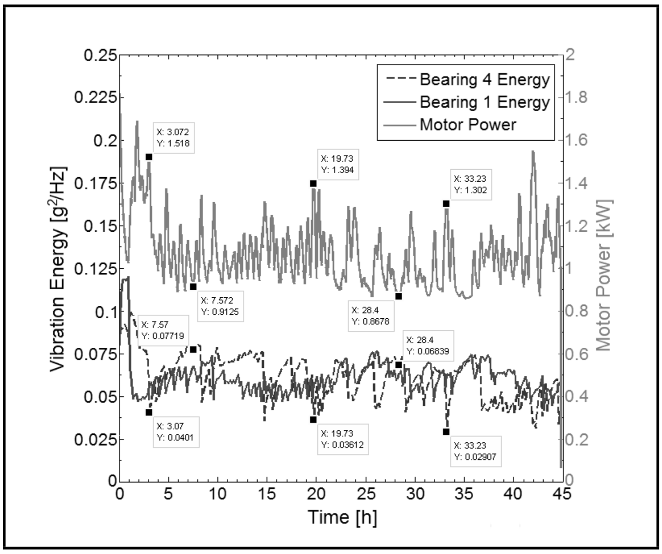

Figure 20. Experiment 5 vibration energy captured by radial accelerometers vs. motor power.

The motor power profile, seen in Fig. 19 and more clearly in Fig. 20, supports the abovementioned observations as the power consumption is found to increase prior to a rise in temperature and drops just before the bearing temperatures start decreasing. The increase in power consumption is expected as the motor (which is managed by a smart controller) is trying to overcome the resistance to the rolling motion caused by the roller-misalignment in an attempt to maintain the axle rotational speed within $\pm 0.5\%$. As soon as the rollers return to normal operating conditions, the motor power drops, which is then followed by a decrease in the bearing temperatures. Further proof is found in Fig. 20, which illustrates that instances of maximum motor power consumption correspond to instances of minimum vibration energy and vice versa. Note that the abrupt changes in ambient temperature, seen in Fig. 19, are caused by the commercial freezer unit defrost cycle which commences every 10 to 15 hours.

6. CONCLUSIONS

The ability to differentiate between defective (faulty) bearings and defect-free (healthy) bearings that are undergoing a temperature trending event is of utmost importance to the railroad industry as this will significantly reduce the number of unnecessary, and very costly, train stoppages associated with *non-verified* bearing removals and/or inspections. This study presents an experimentally validated technique that utilizes the vibration energy approach to identify and differentiate the temperature trending phenomenon exhibited by some defect-free (healthy) bearings in field service.

The technique is built on the hypothesis that, during a bearing temperature-trending event, the vibration energy will decrease prior to a temperature uptrend and will start increasing moments before a temperature downtrend. The aforementioned behaviour is triggered by vibration-induced roller-misalignment. Misaligned roller(s) will slide on the raceways and create a tighter fit between bearing components, thus, slowing the epicyclical speed and producing less vibration than that of pure rolling motion. Meanwhile, the roller-skidding will generate excessive metal-to-metal frictional heating caused by thinning of the elastohydrodynamic (EHD) lubricant film, which accounts for the observed rise in bearing temperature. Once the roller(s) return to normal operation, the vibration energy of the bearing will increase back to its original level and the bearing temperature will start to drop.

This paper provides sufficient experimental proof to validate the proposed hypothesis. The relationship between vibration energy and temperature is further verified by the shift in the ω_{out} frequency, which is dependent on the fundamental frequency of the cage. A shift to a lower ω_{out} frequency indicates that the rotational frequency of the cage has slowed down from its epicyclical motion due to the friction caused by roller misalignment. A decrease in the bearing temperature or return-to-normal operating temperature would be caused by geometrical thermal expansions and roller realignment. During the reduction in cage pocket and radial clearances, the rollers pivot as they attempt to realign, promoting grease flow and momentarily speeding up and causing fluctuations in the rotational speed of the cage.

The abovementioned behaviour is observed in numerous laboratory experiments, and the technique presented here is consistent in identifying the bearing temperature trending phenomenon. The selected tests discussed in this paper varied from setups containing bearings with known defects that would trigger vibration-induced roller misalignment to experiments conducted with all defect-free bearings run at high speeds and loads or low ambient conditions. Note that roller misalignment is not only triggered by vibrations from neighbouring sources, it can be caused by many factors including inadequate lubrication conditions, grease degradation, loose cone assemblies, high operating speeds and loads, or simply low ambient temperatures that significantly increase the grease viscosity and cause grease-induced roller misalignment.

In the case of a defective (faulty) bearing, the vibration energy levels and the bearing temperature will be noticeably higher than those of normal operation—see Fig. 7 and Fig. 8 and compare the defect-free Bearing 1 to the defective Bearing 4, which contains a spalled cup raceway—and the vibration energy of the bearing will continue to increase as the defect worsens with continued operation. Ongoing work in this area is focused on acquiring vibration signatures at different oper-