This is the third in a series of articles on Root Cause Failure Analysis that examines some of the available metallurgical and physical evidence. While previous articles in RELIABILITY® Magazine have concentrated on some of the academic aspects, this article examines a specific case history.

A large reciprocating air compressor was experiencing repeated bearing failures. Although the installation was less than two years old, 11 bearing failures had already occurred. As shown in Figure 1, the compressor was driven through a reducer with two elastomeric couplings. It was an unusual installation as the plant had selected a variable speed drive to reduce energy usage. The plant personnel had the capability of conducting vibration analysis, and they also had a vibration consultant look at the installation twice in an effort to find out why the bearings were failing so frequently. There was general agreement that with the overall readings in the range of 0.4 inches per second (10.2 mm/sc), the vibration was excessive, but they had not detected any specific problem that might be causing the failures. The reducer (3.9 to 1) was a double helical unit. The input speed varied between about 2400 and 3600 rpm, although there were a few times when it ran slower. The failures had occurred on the pinion shaft and had been almost evenly distributed between the two roller bearings. One key point in the failure analysis was that the bearing cages were failing, not the race spalling that frequently ends a bearing’s life. Inspection of the bearings which had been run showed the sides of the cages were highly polished, almost to a mirror finish, after only a few weeks operation. Also, several of the failed bearings had cage bars that had failed from fatigue. (See Figure 2.)

Looking at the cages and how they had failed, we searched for the forces that could have caused the failure. The bearing lubrication was excellent, the driven gear bearings were in good condition, and the couplings were in like-new condition. However, close inspection showed that even though they looked to be almost perfect, there had been contact with the back of the teeth on part of the driven gear. Looking at this contact, it was in two arcs of about 60 degrees each on opposite sides, i.e., from 2 o’clock to 4 o’clock and from 8 to 10 o’clock.

One of the basic principles of fracture analysis is that the crack always grows perpendicular to the plane of maximum stress. Looking at the fatigue faces of the fractured bars on the cage means that the stresses were applied in a circular manner. Based on this and the tooth wear pattern, a torsional vibration analysis was made, and it confirmed that the pinion was going through wild variations in angular velocity during each revolution. At certain speeds, and at the most common operating speeds, the elastomeric coupling on the output shaft was operating at a resonant frequency. Instead of transmitting the output power smoothly, it was winding up, then releasing twice during each revolution. This rapid change in velocity on the heavy driven gear then forced a change in the pinion speed that was multiplied by the gear ratio. The roller loads on the bearing cage from these rapid speed changes caused high stresses the cage and the fatigue cracking that resulted in the eleven failures.

Our initial recommendation was that the unit not be run in certain speed ranges, i.e., those where the operating speeds of the compressor were close to the resonant frequencies of the couplings. Eventually, the elastomeric
couplings were exchanged for torsionally stiffer units with a much higher resonant frequency. The problem was solved and there hasn’t been a failure in the last four years.

Comments – Two important points to understand about bearings in this analysis are the effects of loads on the bearing components and the function of the cage.

Bearings are designed for a given number of load cycles on the races. If these fatigue loads and the resultant Hertzian stresses high enough, they will eventually cause cracks and spalling to develop in the races and lead to bearing failure. The life of the average reducer bearing is in the order of 25 years and doubling the load on the roller bearing cuts the fatigue life by a factor of ten, to 2.5 years. Thinking of this relationship, a life as short as a few months means something very unusual has to be acting on the reducer.

Looking at the functions of a bearing cage in a rolling element bearing, we find it is primarily designed to keep the elements evenly distributed around the race. In operation: As the ball passes through the load zone, the loaded ball pushes the cage and drives the unloaded balls around the race.

- As the ball leaves the load zone, it slides back in the cage, and the cage begins to drive the ball in a path around the rest of bearing.

- This transition from "driving to driven" occurs hundreds of times per minute and can put substantial fatigue loads on a bearing cage.

One very important function of the lubricant is to act as a cushion between the cage and the rolling elements. In a ball bearing, without great variation in speed around the race, the loads are low lubrication is usually not very demanding. However, in roller bearings the cage has another function, that of keeping the rollers parallel and eliminating their tendency to skid sideward with changes in loads and geometry. (We like to think the rollers and the races are perfectly round and parallel, but they deform just like any other stressed component.) This increases the loading on the cage, increased the demands on the lubricants, and results in the need for higher viscosity oils and more frequent re-lubrication.

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