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Spindle Lubrication Discovery on No. 2 RCM at Nucor Steel-Berkeley

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Ameridrives Couplings

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Spindle Lubrication Discovery on No. 2 RCM at Nucor Steel-Berkeley

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Abstract

Problems with gear spindles on rolling mills are a common occurrence. This paper describes various design and lubrication trials that aided in troubleshooting the spindle problems experienced at Nucor Steel-Berkeley's cold mill.

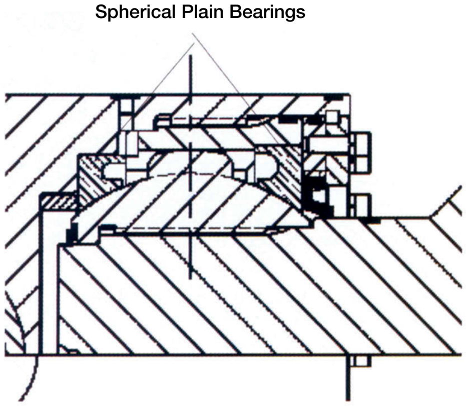
Gear spindles are a common method of torque transmission for rolling mill drives. Several spindle designs throughout the industry are determined by the design features of a particular mill. Each spindle application generates its own unique challenges. This paper illustrates the challenges faced with the Nucor Steel-Berkeley No. 2 reversing cold mill (RCM) gear spindles. This paper also explains the different experimental trials and the results.

Mill Features

The No. 2 RCM at Nucor Steel-Berkeley is a 4-high reversing cold mill with axial work roll shifting. The main mill drive is a tandem AC motor arrangement totaling 10,000 horsepower. Strip width is 32-66 inches, incoming gauge is 0.050-0.240 inch, finish gauge is 0.010-0.110 inch, maximum speed strip 4,000 feet/minute, and maximum strip tension is 40,000 pounds. The spindle is 16.41 inches in diameter, with the center section spline ±4 inches from center to allow for axial work roll shifting; the rated torque capacity is 1,225,200 in-lbs at ±2.

Series of Events, Trials and Results

The mill began operation on April 26, 2001. The spindle design had some unique design features due to the axial work roll shifting feature (Figure 1). in that it required a spherical plain bearing on each side of the gear hub to keep the gear ring centered on the gear hub.

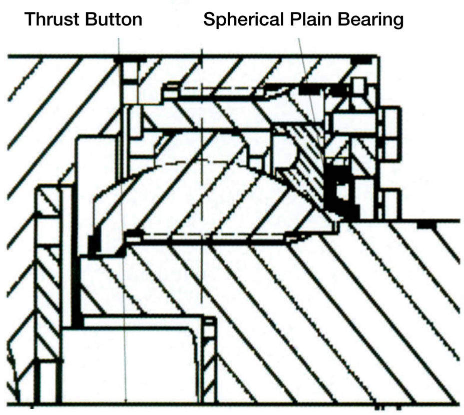


Typical gear arrangement at each end

Figure 1

On Dec. 31, 2001, the first gear assembly failed on the drive end of the top spindle. Severe discoloration of the drive was noticed due to extreme overheating. On April 2, 2002, the drive-end gear assembly on the top spindle failed for the second time. Nucor decided to install cooling water sprays on the drive end of the top and bottom spindles. Ameridrives also recommended changing the grease that was being used. The water sprays helped the heat situation but made a mess on the drive side of the mill, but this was determined to be a necessary evil until the root cause of the problem could be resolved.

On April 19, 2002, the first gearset failure occurred on the roll end of the bottom spindle. This was 12 calendar months after the mill started up.



Gear assemblies with thrust button.

Figure 2

Gear hub and gear ring after synthetic oil trial.

Figure 3

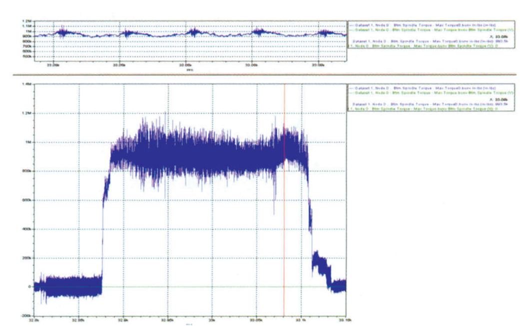
However, the actual run time of this spindle was much shorter due to decisions made by Nucor based on market conditions.

In the past, it had been noticed that the pods or gear assemblies, seemed "stiff." It was observed during the assembly of a spindle that neither the drive-end pod, nor the roll-end pod, would move or "sag" once it was not being supported. It was determined that a source of the heat being generated could be friction caused by insufficient clearances between radii of the spherical bearing and gear hub. The design specification was changed to increase the clearance of the radii. Also, grooves were machined into the spherical bearing radius faces to provide better lubricant paths to the surface of the mating parts. This spindle was installed in September 2002. Temperatures were lower than normal upon start-up, but soon elevated, and the water sprays were reinstalled.

It was still believed that the spherical plain bearings were a problem. Ameridrive's recommendation was to removed one of the spherical plain bearings in each assembly and replace the bearings with a thrust button arrangement (Figure 2). In order to get rid of the second spherical plain bearing, a spring pack would need to be installed in the splined, or center, section of the spindle to keep the thrust buttons seated and to keep the gear assemblies centered. There were concerns with "spring loading" the spindle as a result of the axial forces that would be transmitted to the pinion stand gear assembly, due to the triple-shaft double-helical design of the pinion stand. The spring pack option was decided against because of the risk of pinion stand gear assembly damage.

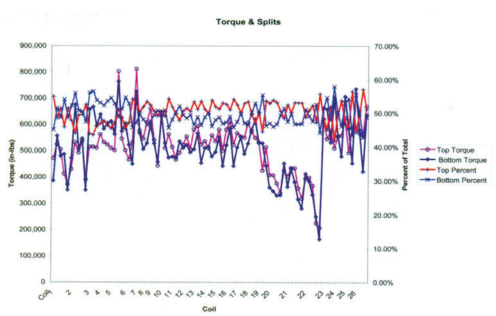
In June 2003, the first top spindle failed since the cooling water practice was begun 14 months earlier. This is sufficient life by some standards, but was still unacceptable considering the problems caused by the cooling water. The spindle with the thrust button modification was installed in the top position with no cooling water. Temperatures were reasonable for approximately one week, and then they began to elevate. The cooling water was turned back on, and Nucor went back to the drawing board. The mill loads were then lowered to 80%. As a result, the spindle lasted 22 months.

From September 2002 to October 2003, two more roll-end gear assemblies failed in the bottom position. During this time, analysis indicated lack of lubrication at the gear mesh due to signs of spalling or "worm tracking." It was decided to run a trial with heavy synthetic gear oil.



Trend showing maximum torque on bottom spindle.

Figure 4

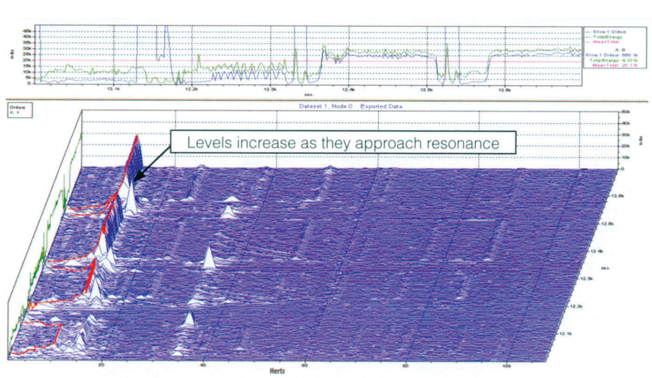


Maximum load splits per coil.

Figure 5

The oil that was chosen has a viscosity of 46,000 cSt at 40˚C, 1,250 cSt at 100˚C or roughly 230,000–6,000 SUS and was supplied by a major oil company. The trail was installed on Dec. 26, 2003. The water was removed, and the mill loads were increased to 100%. Temperatures were very good initially, but started to elevate after running for two weeks. The cooling water was turned back on, and the mill load was reduced back to 80%. The gearset failed after one month of operation (Figure 3). The seals again became a topic of discussion due to the lack of oil remaining in the assemblies when they were removed.

A trend was noticed that raised the question, "Why is the bottom spindle failing at a 3-to-1 ratio compared to the top spindle?" Industrial Vibration Consultants (IVC) was asked to check the actual operation conditions of the spindles. In March 2004, IVC installed strain gauges on the spindles and accelerometers at various locations on the mill to monitor any vibration conditions. The mill load was returned to 100% for the tests. The spindle was designed for 1,225,200 in-lbs. of torque. The highest level of torque seen during the testing was approximately 1,000,000 in-lbs. (Figure 4).



Resonance frequency.

Figure 6

The most extreme load split observed was 42% of the load on the top spindle with 58% of the load on the top spindle (Figure 5). The most disturbing evidence was the discovery of torsional resonance at a frequency that was common based on run conditions (Figure 6).

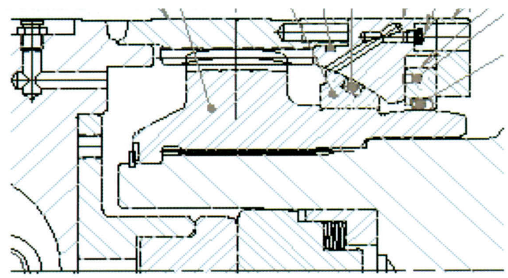
After reviewing the IVC data, it was determined that the peak torque was within design parameters and the load splits were within common parameters of the mill design, but the torsional resonance should be a concern. Nucor Steel-Berkeley agreed to investigate U-joints based on the successful conversion of the No. 1 RCM from gear spindles to U-joints.

In the meantime, a trail was chosen to apply a Teflon coating on the teeth of both the gear hub and gear ring in cooperation with Surface Engineering Associates (SEA). The Teflon lubricity properties were discovered by accident when used in a different application. The spindle with the Teflon-coated gearing was installed in April 2004 with no cooling water and 100% mill load. The temperature elevated within a few shifts, which seemed strange due to the lack of spalling witnessed at disassembly (Figure 7). The cooling water was turned back on and the mill load reduced to 80%, and the spindle was removed after two months for another trial.

In May 2005, Nucor Steel–Berkeley decided that it was appropriate to begin discussions with other gear spindle companies while maintaining open lines of communications and trials with Ameridrives. During the discussions with other companies, the topic of the flow or "pumping" action of the grease at the gear mesh came up again. Nucor was asked to check the angle of the spindles at the static (roll change) position and dynamic (run) position. The results were eye-opening. At the static position, the spindles were at 0.9˚. In the run condition, a 0.6˚ offset was measured with the largest work roll diameter, while there was only a 0.4˚ offset with the minimum-diameter work rolls. These parameters were four to five times less than what the original spindle was designed around. Nucor selected two companies to build a spindle, with one company's design on the roll end and the other company's design on the drive end – a hybrid spindle. The roll-end design company proposed a new gear assembly, making changes to the tooth profile geometry, the sealing arrangement, and the heat treating of the teeth to carburized and ground (Figure 8). The drive-

Teflon-coated gear hub.

Figure 7



Trial roll-end gear assembly.

Figure 8

end design company stayed with the Ameridrives assembly design and elected to change the materials and heat treat process.

Other issues came up while waiting for the "hybrid" spindle to arrive. In August 2004, delivery became an issue with the grease that was being used. It was decided to switch to the standard coupling grease provided by the oil company that was supplying Nucor at the time. No adverse performance changes were noticed with the grease changes because of the conservative practices that were being employed at the time.

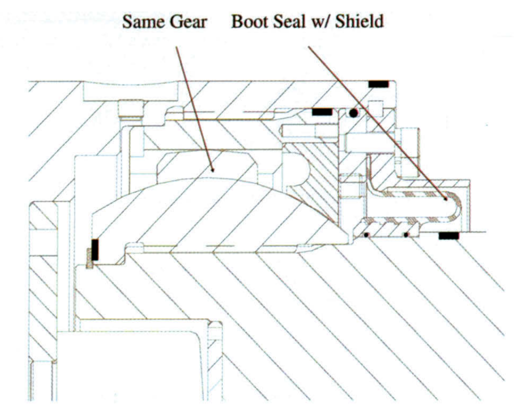
In January 2005, Surface Engineering proposed the use of the Lyon grease product to Nucor. Nucor decided to use it as an additive with the grease that was already being used because of concerns that the current seal arrangement would not be able to contain it alone. On Jan. 10. 2005, the two greases were mixed on-site and purged through both gear sets of the bottom spindle. The cooling water was shut off and the mill load was returned to 100%. The results were the best to date from a temperature standpoint. The temperatures were very low on both ends for approximately one week before they began to elevate. However, temperatures continued to elevate, so the spindle was removed (Figure 9). Due to the promising initial results with the two different oils that had been experimented with, Nucor and Ameridrives decided to aggressively pursue an alternative seal design in this application. Ameridrives had just finished testing a "boot seal" in another

Gear hub after Lyon additive blended with grease.

Figure 9

Results of tooth profile changes on roll end of hybrid spindle

Figure 10



Boot seal with shield.

Figure 11

application and was pleased with the results. Nucor Steel-Berkeley agreed to try it and placed the order on Jan. 28, 2005.

The "hybrid" spindle was installed on Feb. 1, 2005. The drive end quickly overheated, and the cooling water was resumed, but the mill load was left at 100%. On March 24, 2005, the drive end of the hybrid spindle failed, so the mill load was reduced to 80% and a normal spindle was installed. While the spindle was out, the roll-end gearing of the hybrid spindle was inspected and looked to be in good condition (Figure 10). It appears that the tooth profile change had succeeded as indicated by using more surface area of the tooth and spreading out the load.

The spindle was reassembled with an Ameridrives gear assembly on the drive end and installed back in the bottom position in April 2005. It was moved to the top position in July so that the boot seal spindle could be installed. It was removed on Feb. 13, 2006, due to cracking in the roll-end housing. Nucor Steel-Berkeley chose not to pursue this design because of ongoing, more economical trials. It appears to be a promising option if needed.

While waiting for the boot seal trial to arrive, Nucor decided to expand the conservative approach in operating practices and began to grease the spindles at every roll change, approximately every 12 hours, to ensure that no other spindles failed. The plan basically went into survival mode. This was in addition in the cooling water and reduced mill load. It was instituted April 23, 2005.

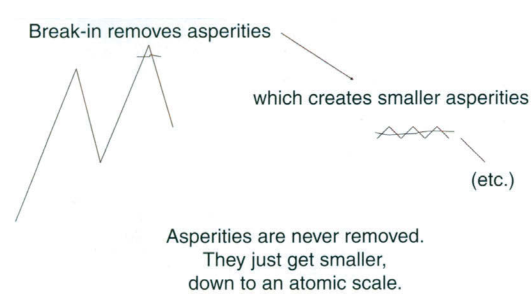
In April 2005, SEA approached Nucor with the idea of running the Lyon product as oil in the boot seal versus the Lyon grease that had been tried previously.

Lubricants

A complete explanation of how the Lyon product works and the chemistry behind it is beyond the scope of this paper, but this brief description will compare the fundamental differences between the old and the new technology. The new patents will explain the specifics at a later time.

Several factors need to be considered with lubricants. For the purpose of this paper, the focus will remain on the fractal nature and geometry of manufactured surfaces; boundary layer lubrication as a function of surface properties and sliding velocity; film thickness characteristics of conventional lubes; and the Lyon boundary layer thickness.

Both natural and manufactured surfaces are not flat, clean or smooth. For instance, the surface of any steel part is covered by a layer of oxides and other contaminants from the atmosphere and surroundings—such as moisture, absorbed gases such as CO_2, oils, etc—and the surface itself is quite rough under close inspection. This roughness is a result of the machining/finishing practices that are used, and consists of a series of peaks and valleys created by the combination of tool properties and material fracture properties. The surface peaks do most of the work of contact, and they are called "asperities" in the field of tribology. Tribology is the study of the friction, lubrication



The fractal nature of surface wear.

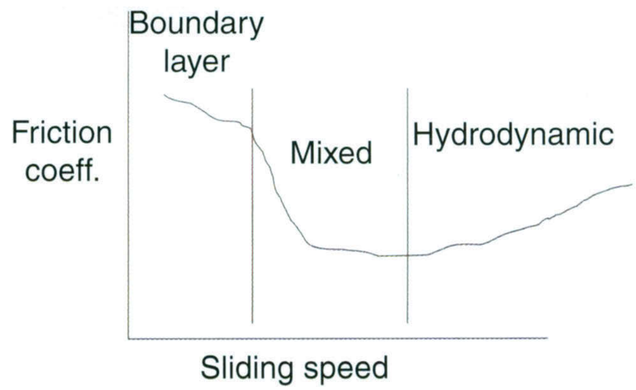
Figure 12

and wear of contacting surfaces. In this case, we are looking at a "tribosystem," which consists of the two gear surfaces and everything in between. The asperities in any tribosystem are actually a very small part of the total working surface. It is commonly understood that, during the "break-in" phase of wear, the perceived contact area is much bigger than the actual contact area until the asperities are removed and a new, "smoother" surface is created. This takes sliding work, releases energy, and explains heating of the surfaces during start-up. In some cases this overheating goes away, but in some cases the trapped debris from start-up continues to be part of the wear equation. Some publications suggest that this debris can be bad for two reasons. The first is that it can become an abrasive wear agent. The second reason has to do with the elastohydrodynamic (EHD) theory of lubrication, which requires that very high pressures be generated to produce the semi-solid EHD films. When the debris carries some of the load, the fluid pressures do not get high enough to trigger the EHD effect. As wear debris increases, the fluid film collapses even further.

To get a better understanding of the surfaces under these films, the use of "fractal geometry" can be quite helpful. A fractal surface is one that consists of repeating shapes—in this case, peaks and valleys—which look about the same at many scales of magnification. Fractal geometry uses an iterated power law (x^y repeated over and over) to generate shapes that are quite similar to manufactured surface profiles, and they can be altered by the computer to replicate rough, smooth or worn surfaces.

The use of fractals allows one to make shapes of any desired smoothness and at any magnification, which is quite useful in the modeling of surface contacts and in the prediction of how they might be altered. Fractal math is all around. When a digital camera takes a picture, one pixel is photographed and the pixels around it are predicted by a fractal formula. This is also how satellite photos achieve their resolution. It is likely that, when a comprehensive predictive model for lubricated friction is achieved, it will use fractal math to define the working surfaces.

The fractal nature of surface wear is illustrated in Figure 12. It is seen that continued magnification does not really change the asperity shape very much, it just gets smaller. Hydrodynamics lubrication hopes to get fluid in the valleys, which creates a pressure to lift the surfaces apart, creating a boundary film. The Lyon technology provides a film that sticks to both the peaks and valleys. This polar bonding mechanism is independent of viscosity, so even if there is a hydrodynamic "lift," the asperities still carry



Stribeck curve.

Figure 13

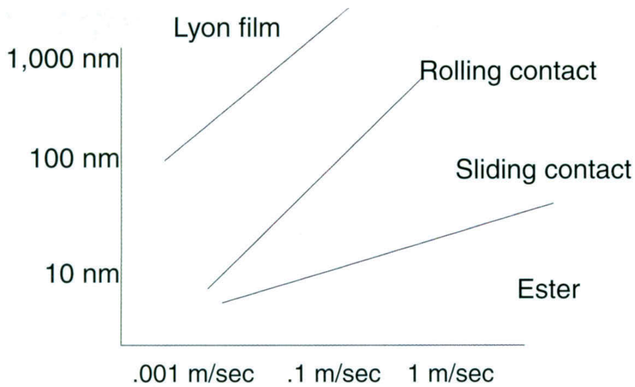
their own boundary layer. The lift function is enhanced by the addition of nanophase layered dichalcogenides (solid lubricants), which are suspended in the polar structure.

The fractal peak phenomenon goes on and on, down to the molecular level, where the term "fluid" really has no meaning, but where the Lyon product begins to do its work.

The Stribeck curve (Figure 13) is commonly used to show the transition from full lubrication, or hydrodynamic state, through mixed lube, and finally into the boundary lubrication condition. Mathematically, this condition can be predicted by the lambda ratio, which compares combined roughness of the two surfaces to the lube film that separates them. This calculation is an approximation because the surface characteristics are an average. Also, the lube film thickness says nothing about the other properties of the lube.

Boundary lubrication occurs when the relative sliding motion is so small that it does not entrain enough lube to keep the surfaces apart. This spindle gearing was in a state of boundary lubrication, which can be proven by applying the lambda ratio to this application with a minimum, or "squeeze film," thickness of a conventional lubricant. It is a compounded problem when the energy density occuring in the gear mesh of these spindles is considered, along with the minimal angular misalignment.

Table 1 shows the Ra roughness for typical machined finishes, which is determined by measuring the average dimension from the peaks to the midline of the surface, as well as the total peak-to-valley dimension. Ra is typically referred to in micro-inches, but nanometers are another unit



Film thickness versus rolling speed.

Figure 14

of measure for these dimensions and are appropriate when dealing with lubricant squeeze films. It is both intuitive and true, to a certain extent, that smoother surfaces are easier to lube because the asperities are lower, so the lube film can be thinner, provided there is a dynamic replenishing effect. However, if the surface is too smooth, the lubricant can be squeezed out under stationary loading. This is why machine tool ways are "scraped," to gain some lube retention along with added precision.

Another way to create a boundary layer film of superior tenacity is to use a mechanically composite surface.

Using a visual surface comparator, observations on the gearing from the boot seal and Lyon trial indicate a gear tooth roughness of about 16-20 micro-inches on the unworn surfaces and about a 4 micro-inch finish on the worn section. By using Table 1, one can see what this means in terms of asperity height: about 200 nm combined in the

Table 1. Using Micro-inches (µin), Nanometers (nm) and Inches (in.)

Ra finish (µin)	Avg. µin	Avg. nm	Peak/valley µin	Peak/valley µnm
4	0.000004	101	0.000016	406
8	0.000008	203	0.000032	812
16	0.000016	406	0.000064	1,024
32	0.000032	812	0.000128	3,248
63	0.000063	1,600	0.000252	6,400
125	0.000125	—	0.0005	—
250	0.0025	—	0.001	—

contact patch and 800 nm on the unworn surfaces. It is certain that the small amount of articulation and thrusting present helped to transfer some lube from the "coarse" low-pressure zones to the much smoother "squeeze" zone.

Figure 14 gives an idea of how squeeze film thickness changes with speed. This test was done at about 150,000 psi with surface finishes on the order of 10 nm. With sliding contact, much rougher surfaces and higher contact stress, the conditions in the spindle were much more severe.

A world-famous expert on boundary layer behavior is Hugh Spikes, from England. He and others have devised a way to measure thin lubricant films using microscopic optical interferometry between a steel ball and a sapphire disc. The tribosystem is flooded, but the point of contact carries a high enough stress to evacuate the lube down to a molecular level. A light shining through the film is then used to measure the actual film thickness. The light travels two paths, one of which is through the film. When the light beams are recombined, there is an interference pattern that is used to calculate the film thickness. For a more complete explanation of this work, see Reference 1.

The thickness of the Lyon boundary layer film is superimposed on Figure 14, along with a representation of how the film would be made thinner by sliding contact.

Just as a rough estimate for this gearset, if one uses a 6-inch pitch-line radius, 1,200 rpm speed, and one degree of articulation, the result is a sliding speed of about 0.0025 m/second, which is on the low end of any film thickness building mechanism. This is, of course, why this application is so tough. From Figure 13, one can see that, with an 8 micro-inch finish, there is a combined asperity height of about 400 nanometers, which would require a sliding speed of well over 1 m/second in order to trigger any viscosity-based hydrodynamic effects. The sliding speed that was encountered was more like a fretting condition.

The Lyon material is used as an additive, so in that sense it belongs in the same family as the other types of additives that contain such surface-active components as sulfur, chlorine or phosphorus. These elements form chemical compounds on the steel surface that can act as boundary layers, but the chemical reactions are corrosive to the steel, like a "rusting" without oxygen. These traditional additives can do an excellent job of reducing friction and wear, but they do have some drawbacks, because they require replenishment

Magnetic plug with metallic shavings.

Figure 15

and they create byproducts. In severe cases, such as this spindle application, the rate of attrition is greater than the rate of replenishment. The proof of how well this type of technology works is found in its energy efficiency. Partial boundary film failures are indicated when heat is being generated, as seen with this spindle application.

The Lyon additive stops the frictional heating by using "matched molecular chemistry," which is "the dissolution of large polymeric materials in the oil which utilize the enormous molecular forces inherent in their chemistry to form near-covalent bonds with the steel surface, providing at least four molecular diameters of polymer protection instead of the one molecular diameter observed with conventional lubricants."

In layman's terms, this means that the Lyon fluid sticks to the steel surfaces by a chemical attraction to the surface, not by a chemical reaction with the surface, and what is left is a multiple molecule boundary layer. Proof of the Lyon technology's abilities to reduce the frictional healing of equipment requires only an ammeter and thermocouple.

Nucor and Ameridrives discussed the situation and decided that a good head-to-head trial would be to use the Lyon in the drive end, and the heavy synthetic oil that had been trialed in the past on the roll end.

On July 1, 2005, the boot seal spindle (Figure 11) was installed. The spindle employs the same gear design, magnetic plugs in the fill ports to collect wear particles, and of course the boot seal. The mill was put back to 100%

Gear hub dripping with oil.

Figure 16

Gear hub with major oil company's product.

Figure 17

Gear ring on Lyon product.

Figure 18

Gear hub on Lyon products

Figure 19

and the water was turned off. Temperatures were monitored closely and remained low on the drive end (less than 100°F) and moderate on the roll end (about 140°F, close to roll coolant temperature).

After approximately four hours of running, the mill was stopped to check the oil level in the pods. The level was good, but a lot of metallic shavings were seen on the magnetic plugs in the roll-end pod (Figure 15), but none were seen on the plugs for the drive end.

Ameridrives stated that some wear-in was expected and recommended that close monitoring continue. The metallic shavings continued, so on July 3, Nucor decided to drain the oil in the roll-end pod and refill it with the same type of oil. The temperatures were still very low on the drive end, but the roll-end temperatures kept creeping up. The oil level was being checked twice per shift, and the wear was still evident on the roll end, but not the drive end. Everyone speculated that the wear was occuring in both ends, but the Lyon product was so viscous that it was keeping the wear particles in suspension, not allowing them to get to the metal plugs. On July 5, 2005, the oil was drained out of both ends of the spindles and examined. There is no evidence of wear particles in the Lyon product. It was decided to put Lyon back in both ends of the spindle. The mill ran for 12 hours and was stopped to check the oil level. There were no signs of wear particles on the magnetic plugs on either end; the temperatures on both ends were very good with no cooling water and at 100% mill load. The levels were checked in each pod daily during a roll change. On July 31, 2005, the frequency of this check was changed to weekly. This is significant because from April 23 to July 31, the mill had 108 unscheduled delays, totaling 3,401 minutes just for spindles. From July 31, 2005, to the writing of this paper there was not a single unscheduled delay for spindles!

The spindle was removed on Nov. 10, 2005, to install a U-joint spindle for trial. The results were exactly what was expected. The boot seals worked very well because both gear assemblies had plenty of lubricant (Figure 16). The roll-end gear hub was severely spalled (Figure 17), but it is believed that the damage happened early and did not progress once the lubricant was changed. The drive-end gear ring and hub (Figures 18-19) were in excellent condition. The contact area was polished, but there was no measurable wear. The results of the boot seal and Lyon oil combination were good and were considered a success.

Spindle Design Review

The original spindles provided by Ameridrives were of a design specified by the mill OEM. The specified spindle design incorporated a splined travel section to provide length compensation during roll shifting. The design further required a pair of spherical thrust bearings on either side of the gear hubs. These spherical thrust bearings transferred the axial force, both positive and negative, created during roll shifting and maintained the gear mesh on center. While most spindles incorporate some type of thrust bearings or buttons, they generally function in a single direction and resist compressive thrust only.

The use of spherical thrust bearings limited the volume of lubricant immediately surrounding the gearing. The spherical thrust bearings created a closed lubrication system with no possibility for circulation or replenishment of lube during normal operation. This generally starved the gearset of fresh lubricant and severely limited the ability of the lubrication/coupling system to transfer heat away from the gear mesh.

The OEM had previous success with this type of design on tandem cold mills. The previous mill applications utilizing this design were one-way mills and, as such, were less challenging for the design. These applications were likely successful in that they could develop and maintain boundary lubrication within the gear mesh. In this particular application, however, the repetitive reversing of the mill constantly broke down the boundary lubrication, creating metal-on-metal contact during each start-up cycle.

The initial spindle design improvements centered on increasing the available volume of lubricant and enhancing the ability of the lubricant to circulate and transfer heat. This was first accomplished by removing the out-board spherical thrust bearings and replacing them with thrust buttons. The thrust buttons still maintained the gear mesh on center, but because they acted at the centerline of the shaft, they increased the available volume of lubricant. Calculation of the lubricant was likely not improved greatly as a result of this change. The operational results — though significantly better than the original design — were still less than the design life goal and still required water cooling.

Brief operation of the spindle with oil in place of grease revealed that oil might reduce the operating temperatures of the spindle. The oil test also revealed that the lip seals were unsuitable for use with oil. It was thought that oil might work better than grease, as it should circulate and cool the gearing better and not channel, as grease tended to do. If oil were to be used, however, a more positive seal would be required.

The original seal design was a high-eccentricity lip seal. Lip seals are suitable for retaining grease, but they proved inadequate for oil retention. When used with oil, the lip seals would continuously leak. Lip seals used for spindles are not intended to be positive seals, as they typically must allow the spindle to purge lubricant or pull in outside air when the volume within the spindle changes. This typically occurs during roll change or changes in the roll opening, which results in slight to moderate changes in the length of the spindle. This spindle was unique in that no volume changes within the spindle were permitted due to the paired thrust bearings. For oil to be used, a positive seal was needed, without the requirement for compensation of length or volumetric changes.

Ameridrives, having had recent experience with boot seals, had determined that they were a positive sealing method with the capability for compensating for angular misalignment. While boot seals can tolerate some axial length changes, they cannot be pressurized. Boot seals for this application were determined to be uniquely suited.

Magnetic lube plugs commonly used on engine, gearbox and transmission oil systems have not been used on gear spindles, as they will not work efficiently with grease. With the change to oil, magnetic lube plugs were suggested by Ameridrives as a means of monitoring the health of the gear spindle. Some initial wear or break-in of the gearing was expected. Without experience with magnetic plugs in a spindle, it would be difficult to know what would be a normal amount of metal entrapment and what would be excessive.

When the spindle with the new boot seals and magnetic plugs was first installed, two types of lubricants were tested. The lubricants had different EP additives and viscosities. The lower-viscosity oil immediately began showing metal particles on the magnetic plugs, where as the high-viscosity (Lyon) oil was clean. It was speculated that the higher-viscosity oil might be retaining the metal filings as grease would. As operation continued, the lower-viscosity oil continued to produce metal filings and the Lyon oil remained clean. When the lower-viscosity oil was finally replaced with the Lyon product, all production of metal filings ceased. The decision to change to the Lyon product eventually proved to be a good one, as during the teardown inspection no damage was found on the gearing that had the Lyon oil from start-up. The use of magnetic plugs proved to be a good indicator of early wear of the gearing, even with heavier-viscosity oil.

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