COMPARISON OF MORGOIL® KLX® BEARINGS AND ROLLER BEARINGS TO A TANDEM COLD MILL[1]

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Abstract
Bearing selection in any cold mill is very important for the production and quality of strip or tin plate produced. There are two types of bearings available for the load carrying rolls on cold mills: oil film bearings and rolling element bearings. The purpose of this paper is to evaluate the application of the newest MORGOIL® KLX® oil film bearings as compared to roller bearings in this application. In general, both bearings are suitable for use on backup rolls within the context of design engineering theory. Practically, it has been proven that accuracy requirements for HAGC equipped mills has limited bearing selection to keyless type oil film bearings and cylindrical roller bearings with interference fit inner races. The newest MORGOIL KLX bearing is a keyless bearing designed to reduce mill capital investment and operational costs. This paper will compare the newest MORGOIL KLX to cylindrical roller bearings for a five stand tandem cold mill. Comparisons of gauge quality, bearing life, and cost are made. Also included is an example of on-gauge performance. Both bearing types have their strengths but experience has shown that oil film bearings are superior in highly loaded and/or higher speed applications. The stiffer roll neck resulting from the MORGOIL KLX design is an added advantage for gauge control.
Key words: Oil film bearings; Rolling element bearings; Backup roll bearings

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INTRODUCTION

The two types of bearings involved in this comparison are keyless MORGOIL® KLX® oil film bearings and cylindrical roller bearings. An oil film bearing is simply one cylinder rotating in a second cylinder separated by a pressurized film of oil upon which the internal cylinder rides. The oil film supports all the load, there is no contact of radial load supporting bearing parts. The design is very simple with few moving parts. The KLX design has brought the load carrying ability of the oil film bearing to a new level by optimizing the sleeve geometry without giving up any of the oil film bearing benefits.

A cylindrical roller bearing has an inner race and outer race separated by cylindrical rollers. The design needed for these high load applications has four rows of approximately 42 rollers each. The inner race is permanently fitted to the roll neck by an interference fit and rotates on the rollers which in turn rotate on the outer race. A very thin oil film separates the races from the rollers.

The mill selected for a bearing comparison is a five stand tandem cold mill. It has a maximum separating force of 3000 tonnes and a speed range of from 145 m/min to 1850 m/min. Also, the mill operates in a fully continuous mode, slowing, but not stopping, to weld coils.

The oil film bearing selected is a 44"-76 KLXTRT-HD-HB MORGOIL bearing with a maximum load rating of 1638 tonnes per bearing. The roller bearing selected is a 850x1180x875/850 four row cylindrical roller bearing with a C (dynamic rating) of 2,341 tonnes.

OIL FILM BEARING DESIGN

The left side of Figure 1 shows an oil film bearing cross section. The oil film that supports the load is generated by the sleeve rotating in the bushing. Since the bearing is supported by an oil film, there is no metal to metal contact, therefore no Hertzian stresses and no fatigue, resulting in infinite fatigue life.

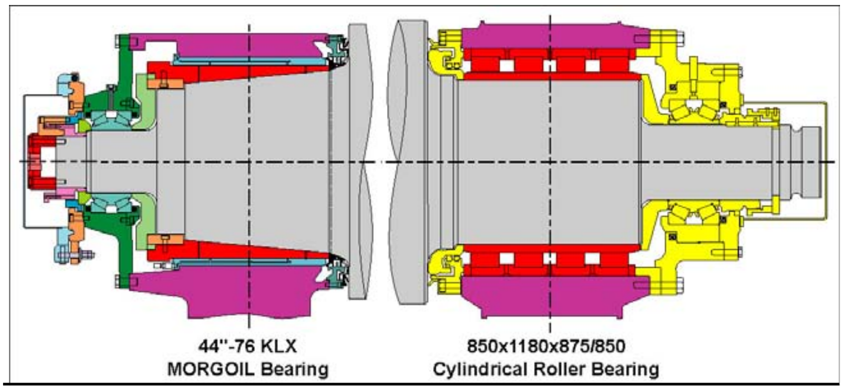


Figure 1. Bearing Design Comparison

The oil film also provides the bearing with other advantages: it is a very stiff component of the bearing. This stiffness, along with its viscous characteristics, allows the bearing to absorb impacts. Additionally, the bearing can help to dampen vibrations and increase the natural frequency through high stiffness.

Oil film bearings are very simple and very compact. For the same load carrying capability, the chock needed for the bearing is smaller than a roller bearing chock and the roll neck diameter is larger.

The lubrication system for oil film bearings is a simple, low pressure system which removes heat from the bearing and roll neck and helps to cool the roll. Further, any contaminates that enter the chock are immediately flushed back to the lubrication system. The lubrication system tank provides settling time for any particles and water contamination to separate from the oil. The oil is then filtered and cooled before returning to the bearing.

The chock/bearing assembly of an oil film bearing is easy to mount and dismount from the roll neck. All parts stay with the chock and the critical surfaces between the sleeve and bushing remain protected. During mounting, the tapered surfaces on the sleeve and roll neck provide a pilot for sliding the chock back onto the roll. When the chock is removed, the neck seal can easily be inspected and/or replaced.

ROLLER BEARING DESIGN

The right side of Figure 1 is a typical cylindrical roller bearing chock assembly cross section. The rollers, located between the inner and outer races, support the rolling load. There is theoretical line contact where these parts contact. As load is applied, the rollers and races deflect and an area of contact is formed. This is commonly referred to as Hertzian contact stress. Since this stress level is far beyond the fatigue endurance limit of the materials used in large diameter roller bearings, fatigue induced failure of the load carrying elements is inevitable.

Mounting the chock/bearing on the roll neck can be very difficult. The only clearance for the operation is the clearance in the bearing and great care must be taken not to scratch or dent any surfaces. In addition, the neck seal can very easily roll under during mounting and be damaged. For these reasons, a chocking machine is virtually a necessity for chocking and dechocking cylindrical roller bearing equipped roll assemblies. Backup roll chocking machines are typically large inflexible devices that burden the customer with a very significant additional investment.

Reasonable care and cleanliness are good maintenance practices with any bearing maintenance program. However, another characteristic of roller bearing maintenance involves the periodic rotation of the outer races. Fatigue life is a function of the number of load cycles. These cycles are evenly distributed on the inner race and the rollers by the rotation of the bearing. In contrast, the load zone of the outer races is fixed at 20-30% of the inner race circumference of the bearing race. To achieve relatively equal cyclic load, the bearing must be periodically disassembled and the outer races rotated 90-180 degrees.

OPERATION - PRODUCT GAUGE VARIATION

Centerline gauge performance is typically measured during two phases of the cold rolling operation. The first and most critical relates to the ability to achieve and maintain accurate centerline product thickness during steady state rolling. Clearly this gauge control phase is primarily a function of control software and advances in this field have been dramatic. The second phase of operation relates to centerline gauge control during acceleration and deceleration. Once again, gauge control software is the critical element in maintaining the required thickness tolerance, but during this transitional phase, the control requirements are complicated by many speed related rolling process elements. Tension control, regulation of motor speeds, rate of deformation of the rolled product, process heat input, etc., all change during this acceleration and deceleration phase.

Lubricant film thickness in the backup bearings also changes with speed. The fact that the film thickness change in oil film bearings is significantly greater than in rolling element bearings has lead to speculation about whether or not roller bearings hold a gauge control advantage during the acceleration and deceleration phases of cold rolling. Experience in many modern cold mills equipped with oil film bearings shows this is not true. In fact, tension control problems are usually the limiting gauge control factors in cold mills. Oil film thickness change in an oil film bearing occurs at a slow rate and is highly predictable and repeatable when compared with other speed change variables. It is very easy for modern gauge control systems to adjust for this film thickness variation.

Figure 2 demonstrates the tension verses gauge control relationships. Shown is an oil film bearing equipped mill which decelerates from 1797 m/m to 279 m/m, during the time that the head and tail are welded together. High frequency tension variation of almost 500 kg starts as soon as the mill decelerates. This high frequency variation continues through 9 seconds of deceleration and 6 seconds of steady state rolling during which the mill is rolling outside the extremely tight target control range of +/-0.5%. It is noteworthy that during this time period, the worst gauge is produced during constant speed, not during acceleration or deceleration. Since the oil film thickness is not changing during constant speed operation the gauge variation cannot be attributed to the backup roll bearings. Likewise, when the mill starts accelerating, the high frequency tension variation declines to approximately 250 kg and the gauge variation falls to +/-0.5% shortly after acceleration begins.

Oil film thickness variation during rolling speed changes do not translate into product gauge variation. Modern gauge control systems easily compensate for this variable. Other speed related variables are much less predictable and often affect both tension and gauge control at much higher frequencies than can be related to backup rolls or bearings.

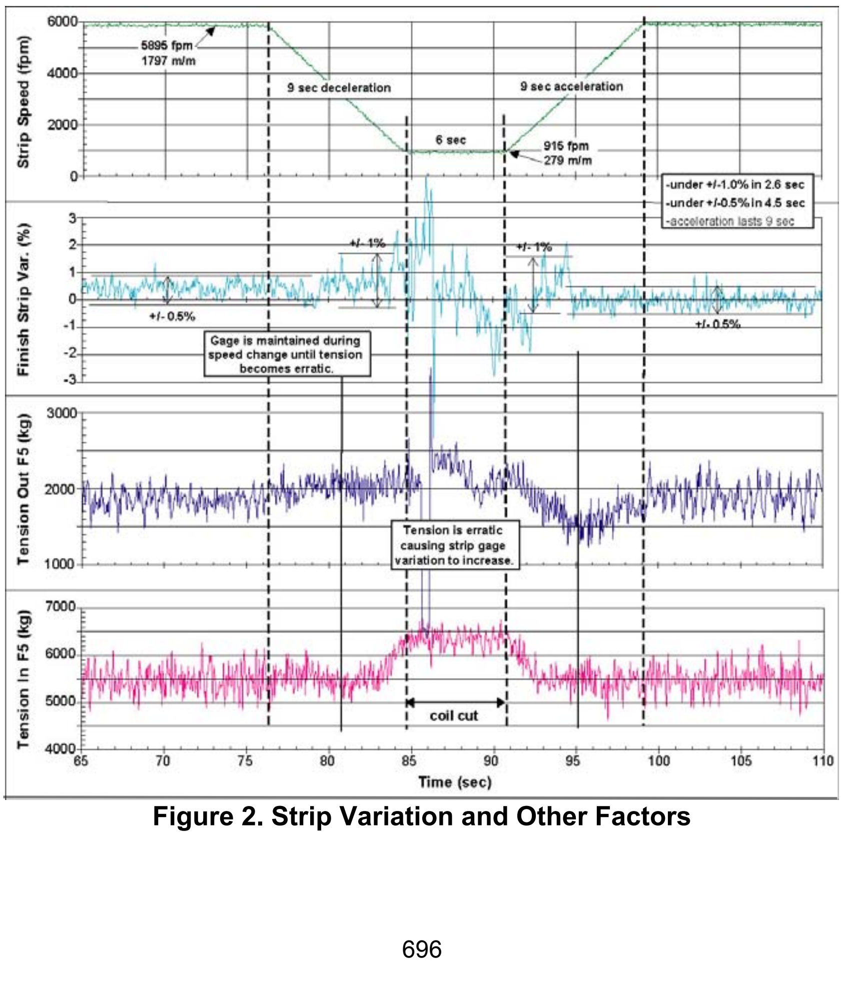


Figure 2. Strip Variation and Other Factors

Size

For similar load carrying capacity, an oil film bearing requires less space than a roller bearing. In this example, the chock will be much more robust and the roll neck shorter and stronger with the oil film bearing. Figure 3 is a comparison of the outside diameters required for the two bearings. The oil film bearing uses a chock bore 18% smaller than the roller bearing making the oil film bearing assembly stiffer. Also on the figure are the minimum required chock sizes, the housing size could be reduced if desired.

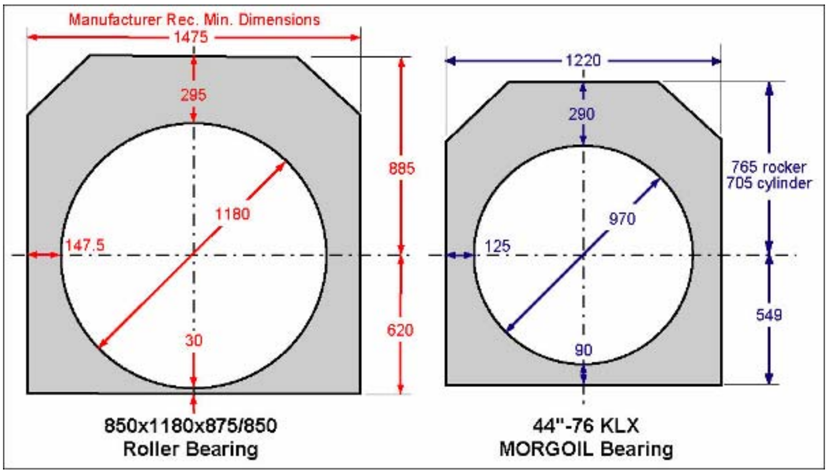


Figure 3. Chock Size Comparison

A chock that deforms has a negative impact on roller bearing life. Figure 4 shows how the load is applied to a chock. The load causes the chock to deform into a cardoid shape. The result of this deformation is that the rollers directly under the load carry more of the force than if the chock did not deform. This is less than an ideal condition, with the result, a reduction in theoretical bearing life.

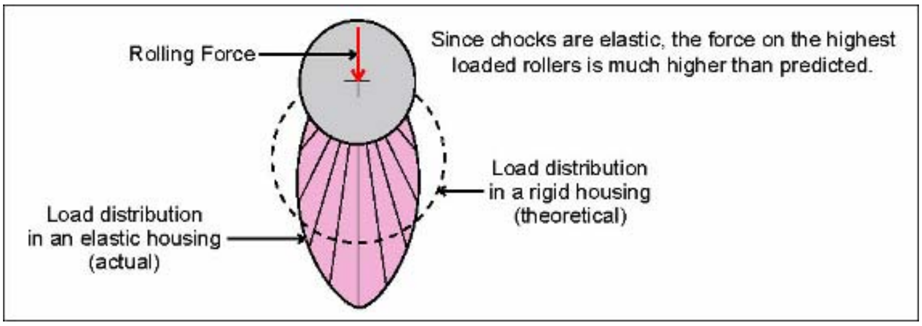


Figure 4. Roller Bearing Chock Bearing Support

Figure 5 shows roll neck dimensions for the two bearings selected. Particularly note the distance from the end of the roll barrel to the center line of the bearing. This is important for both deflection and stress. The oil film bearing distance is 25% shorter due to the shorter bearing needed when using an oil film bearing. The roller bearing is 875 mm long whereas the oil film bearing is 630 mm long.

The left side of Figure 6 is a summary of deflections for the roll necks and bearings. The deflections were calculated at two loads; one 20% greater than the other, and the change in deflections displayed. Also, results were calculated at two different speeds so the effect of oil film thickness variation could be seen. The first set of three bars in the chart shows only bearing deflection. The roller bearing has a greater deflection due to the deformation of the rollers and races. The difference in oil film thickness is also shown. The center set of bars is the roll neck deflections, again the oil film bearing is less than the roller bearing due to its inherent better geometry. The last set of bars is combined deflection of the bearing and roll neck. The roller bearing deflects 26% more than the oil film bearing and the oil film thickness variation is only 5% of the total roller bearing deflection.

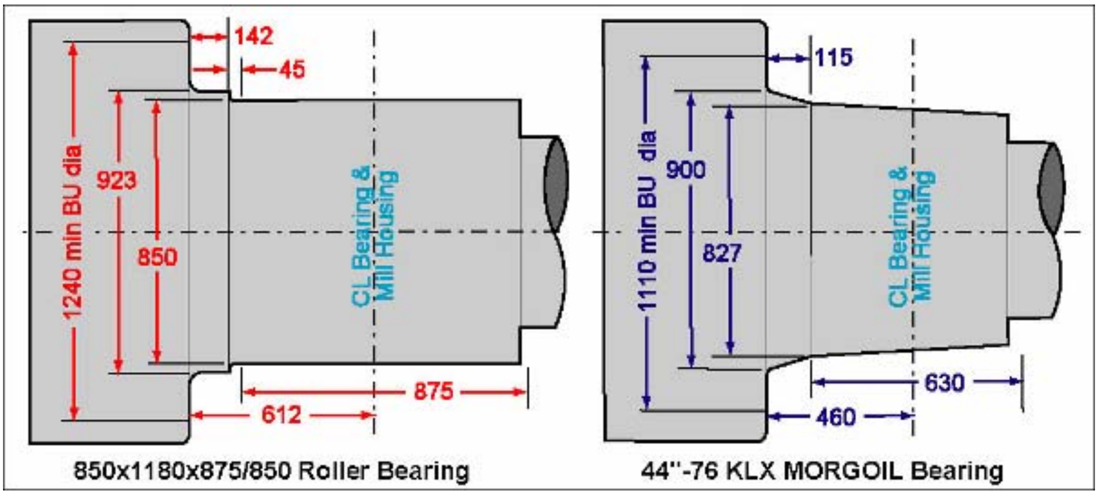


Figure 5. Roll Neck Comparison

The right side of Figure 6 is a bar chart comparing roll neck bending stresses. Due to the shorter distance from the roll barrel end to the centerline of the bearing, the stresses are less on the oil film bearing roll neck by approximately 12% with stress concentration factors.

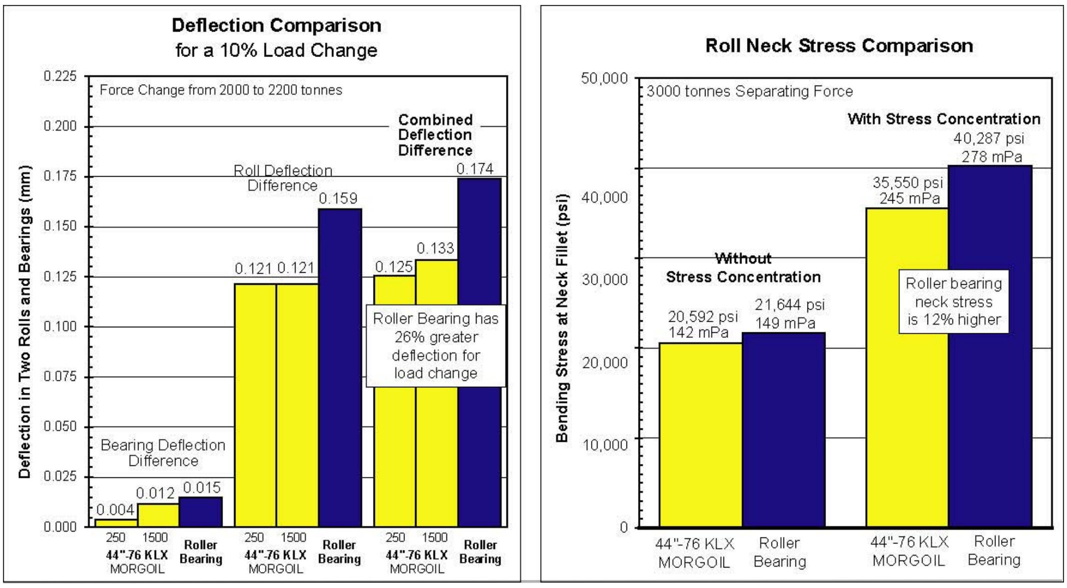


Figure 6. Roll Neck and Bearing Deflection and Stress Comparison

Bearing Life

As discussed previously, an oil film bearing has infinite fatigue life. Many bearings are running over 20 years and some over 50 years.

Roller bearings by comparison have a finite life because the Hertzian contact stress is above the fatigue endurance limit of the bearing material. Roller bearing life is further reduced by factors such as speed and load, water in the oil, particulates in the oil, bearing support, and others. Also, to get maximum life from the bearing, preventative maintenance is high. Outer races must be rotated and rollers interchanged. Following are descriptions of the effects of the various factors.

Speed and Load

Figure 7 is a set of theoretical life curves for this cylindrical roller bearing. These curves show pure fatigue life under laboratory test stand conditions. A factor can be applied to this life that accounts for exceptionally good bearing materials and quality of lubrication, the a_{23} factor. This factor can vary from a life multiplier of 3 to a life reducer of 30%. Many bearing manufacturers catalogs recommend leaving this number at 1, so for this paper the value will be set to 1.

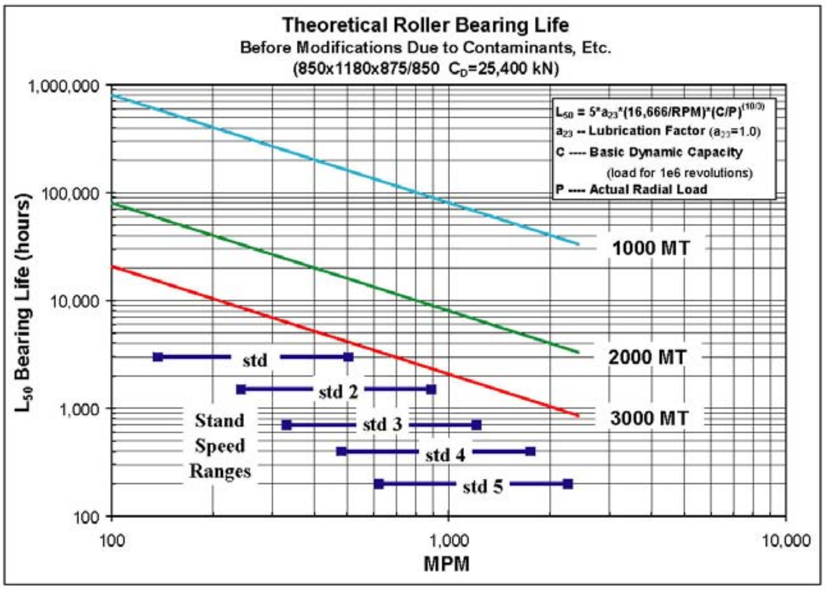


Figure 7. Roller Bearing Life

Bearing life decreases dramatically with increases in both load and speed. The difference between operating at 1000 tonnes and 3000 tonnes is a reduction in life of 97%, and the difference between operating at low speed and high speed is a life reduction of 90%. These life reductions compound for combined high speed and high load. The desired operating point is located on this chart and the life determined, then all the following life reductions start with this theoretical best case life prediction and further reduce it.

Water

Figure 8 is a compilation of data from articles in various technical journals[1]. It is a chart that shows life reduction verses percent of water in oil. A water percentage of

[1] "The Effects of Water in Lubricating Oil in Bearing Fatigue Life", Richard Cantley, ASLE Transactions, vol 20, #3, pages 244-248
"Evaluation of Water Accelerated Bearing Fatigue in Oil Lubricated Ball Bearings", E.L. Armstrong, etc., Lubrication Engineer, vol 34, #1, pages 15-21
"Effect of Seawater on the Fatigue Life and Failure Distribution of Flood Lubricated Angular Contact Ball Bearings", I.M. Felsen, etc., ASLE Transactions, vol 15, #1, pages 8-17

0.01% is considered to be normal at ambient conditions; at this level roller bearing life is not affected. An oil film bearing has a recommended limit of 2% water content without affect on the bearing, and many have operated at much higher levels. At these levels, roller bearing life is only 4.2% of rated life, even at only 0.1% water content bearing life is reduced to 15% of rated life.
It is extremely hard to keep water out of bearings in a mill environment. The lubricant flow of an oil film bearing flushes any water that enters the bearing out its drains, before it contacts any bearing surfaces.

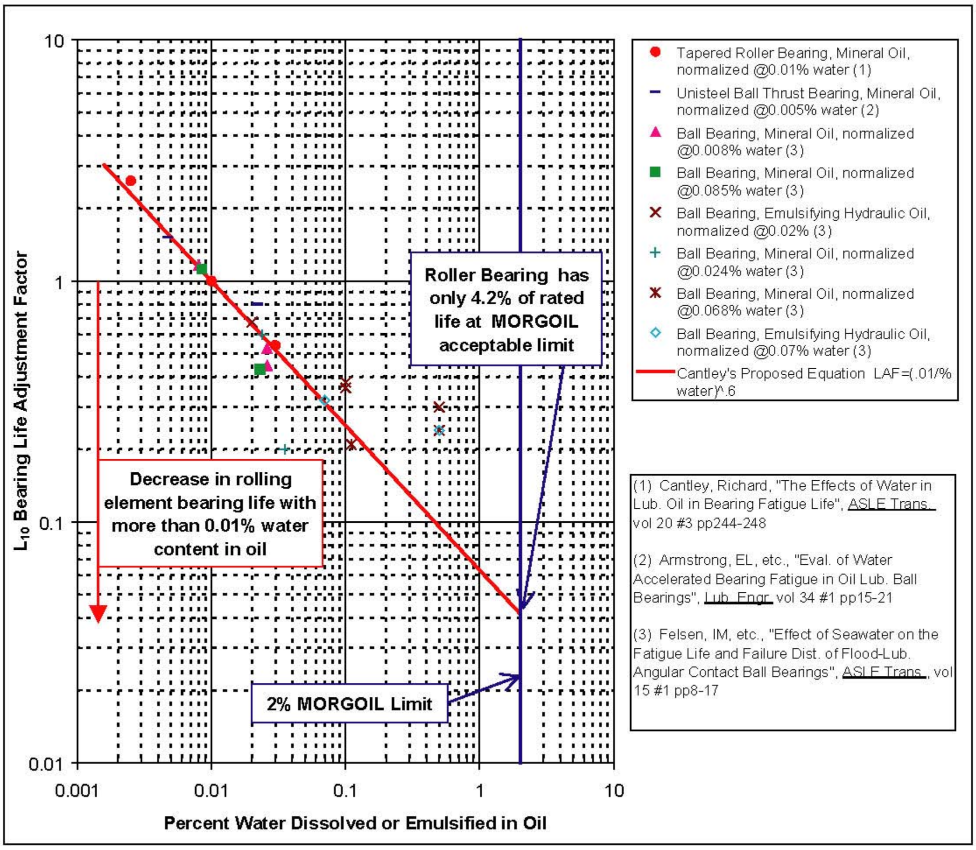


Figure 8. Roller Bearing Life verses Water Content

Particulates

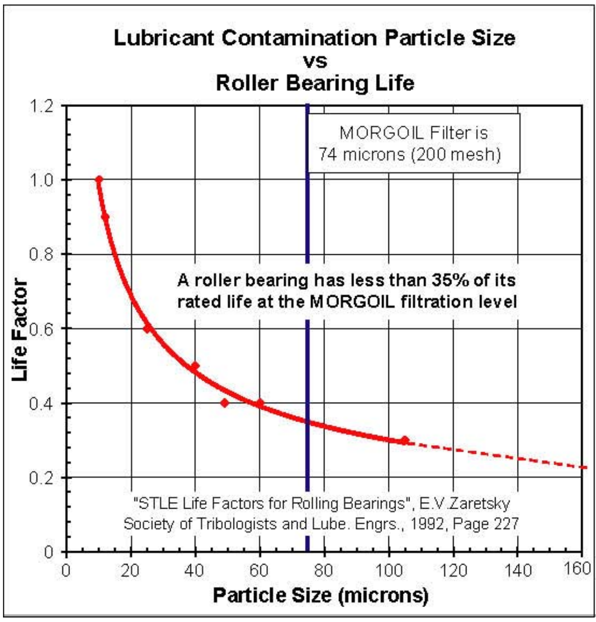
Particulates can also cause severe life reductions, the left side of Figure 9. Particles in the oil must be below 10 microns in size to not reduce life[2]. Particles of 50 microns reduce life by 55%. An oil film bearing uses a 200 mesh filter which removes particles over 74 microns. Also, generally roller bearings flush particles through the rollers before they can exit through the drains, whereas with oil film bearings, the particles are flushed down the drains before they can get to the bearing surfaces.

Bearing Support

To achieve predicted theoretical bearing life, a roller bearing must be supported in an absolutely rigid chock. This is difficult to achieve under ideal situations, let alone when trying to keep a chock as small as possible. The right side of Figure 9 is a

[2] STLE Life Factors for Roller Bearings, E.V. Zaretsky, Society of Tribologists and Lube Engineers, 1992, Page 227

chart of experimental work that was done on chock deflection and bearing life[3]. The rolling load was applied at the center of the chock and then expanded towards the sides. In the best case, life is only 68% of theoretical life, then life deteriorates to about 10% of theoretical life. Also, note there is an optimum location, not in the center and not at the edges, but somewhere in between.



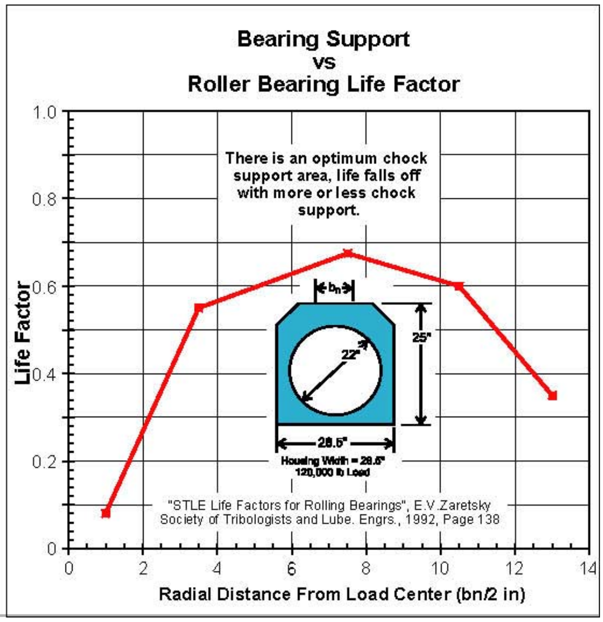


Figure 9. Roller Bearing Life verses Particle Size and Bearing Support

Cost

Typically an oil film bearing costs much less than a roller bearing. This cost advantage is offset, however, by the recirculating lubrication system required for oil film bearings. In general, the installed cost of mills with either bearing type will be approximately the same when only the initial mill fill of chocks and bearings is considered. Adding a second standby compliment of chocks and bearings plus spare parts shifts the cost evaluation dramatically in favor of oil film bearings.
The operating cost of oil film bearing equipped mills is always less than a similar mill equipped with roller bearings. Once the roller bearings start to reach their life limit, replacement becomes a routine event. The high cost of replacement roller bearings plus the high level of maintenance required often leads to a high annual cost.

CONCLUSION

In conclusion, oil film bearings have many advantages over roller bearings for backup roll applications. They are simple to mount, easy to maintain, can operate under high loads and high speeds, tolerate oil contamination, and are less expensive to repair or replace. Oil film thickness variation associated with oil film bearings is easily compensated by mill controls, so is not a determining factor in bearing selection. Roller bearings use a smaller lubrication system, which is not necessarily an advantage. The lubrication system used with oil film bearings is a benefit since it

[3] STLE Life Factors for Roller Bearings, E.V. Zaretsky, Society of Tribologists and Lube Engineers, 1992, Page 138

removes large amounts of heat from the bearings and rolls and purges any contaminants from the chock before they can get into the bearing.

Considering both the total initial cost and the long term operating cost, the evidence clearly points to keyless oil film bearings as the best choice for high precision backup roll bearing applications.