Better Pumps and How They Differ from Average Pumps

By

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The term “better pumps” describes fluid movers that are designed beyond just soundly engineered hydraulic efficiency and modern metallurgy. Better pumps are ones that avoid risk areas in the mechanical portion commonly called the drive-end. An over-emphasis on cost-cutting by pump manufacturers and purchasers has negatively affected the drive-ends of many thousands of process pumps. Flawed drive-end components are among the main contributors to elusive repeat failures that often plague these simple machines; plain and straightforward in-plant statistics attest to the veracity of this contention. To date, this is an issue that has not been addressed with the urgency it deserves.

In late 2008, the purchasing entity representing a large reliability-focused plant in the United States had thoughtfully and deliberately specified better pumps. Although willing to pay the anticipated incremental cost for better pumps instead of standard products, the customer’s improvement requests were declined by every one of the companies that responded to an invitation to bid. The disappointed owner-user company suggested an article that would get out the message to users and manufacturers alike: *Better pumps are possible. Understand why reliability-focused users need them and realize why, for the value-seeking purchaser, certain “standard products” are no longer good enough.*

This article discusses the attributes of better drive-ends for pumps. A stand-alone appendix and a checklist follow the body of this article. Both elaborate on, or briefly recapitulate, the steps needed to give the value-oriented user community pumps that operate for six or more years before repairs are needed.

Why insist on better pumps

There are a number of reasons why well-versed reliability engineers will be reluctant to accept pumps that incorporate the drive-end shown in Fig. 1. The short overview of reasons is that reliability-focused professionals take seriously their obligation to consider the actual, *lifetime-related* and not just *short-term*, cost of ownership. They have learned long ago that price is what one pays, and value is what one gets. Anyway, while at first glance the reader might see nothing wrong, Fig. 1 contains clues as to why many pumps--next to electric motors the simplest machine used by industry---tend to fail relatively frequently and often randomly.

Just to restate the point, the pump drive-end of Fig.1 does not measure up to the expectations of reliability-focused owner-operators. It shows areas of vulnerability and not eliminating these is a mistake. Allowing these risks to exist will sooner or later hurt the profitability of users and manufacturers alike. Both should pay very close attention to matters relating to better pumps. Owner-operators must specify better pumps and insist on getting better drive-ends than the one shown in Fig.1. A number of important vulnerabilities or risk areas exist in that illustration:

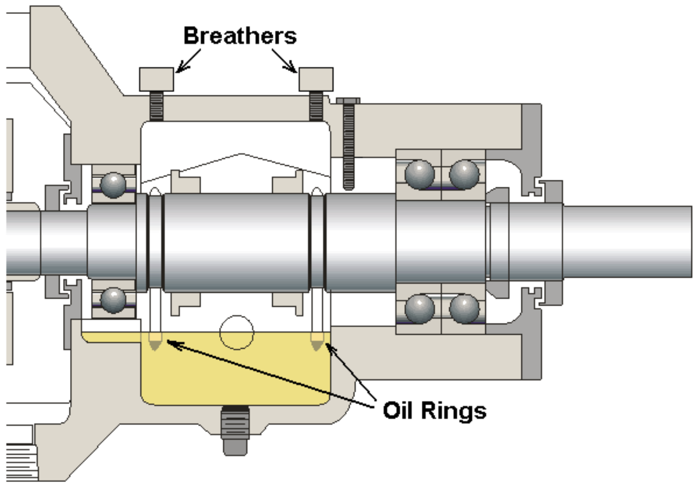

Fig. 1: A typical bearing housing with potentially costly vulnerabilities

- Oil rings are used to lift oil from the sump into the bearings
- The back-to-back oriented thrust bearings are not located in a cartridge
- Bearing housing protector seals are missing from this picture
- Although the bottom of the housing bore (at the radial bearing) shows the desired passage, the same type of oil return or pressure equalization passage is *not* shown near the 6 o'clock position of the thrust bearing
- There is uncertainty as to the type or style of constant level lubricator that will be provided; unless specified, the best one is rarely found on new pumps

Each of these issues merits further explanation and will be discussed in this article. Our considerations are confined to lubrication issues on process pumps with liquid oil-lubricated rolling element bearings. The great majority of process pumps used worldwide fall into that category. Small pumps with grease-lubricated bearings and large pumps with sleeve bearings and circulating pressure-lube systems are not discussed in this article.

Lubricant application via sump level reaching center of lowermost bearing elements vs. lower oil level needed to prevent oil churning and overheating

By way of overview, we note that one of the oldest and simplest methods of oil lubrication consists of an oil bath through which the rolling elements will pass during a portion of each shaft revolution (Fig. 2). However, this "plowing through the oil" may cause the lubricant to heat up significantly and should be avoided on process pumps whenever DN, the product of shaft diameter (inches) times revolutions-per-minute (N) exceeds 6,000. Although cooling is routinely applied in pumps with high speed and heavily loaded bearings, it is questionable if cooling of the oil is needed. Cooling the bearing housings of pumps equipped with rolling element bearings is both unnecessary and harmful (Ref. 1, pp. 249), since cooling often promotes moisture condensation and oil contamination.

In certain high-load services, synthetic lubricants, oil mist application, oil jets and even circulating systems deserve to be considered. Circulating systems are selected for large pumps utilizing *sleeve* bearings. In these systems, the oil can be passed through a heat exchanger before being returned to the bearing. However, irrespective of lube application method on *rolling element* bearings, cooling will not be needed as long as high-grade synthetic lubricants are utilized (Ref. 1, pp. 217).

Because bearing overheating occurs on many pumps operating at 3,000 or 3,600 rpm (where DN values generally exceed 6,000), the oil level must be lowered. Lowering the

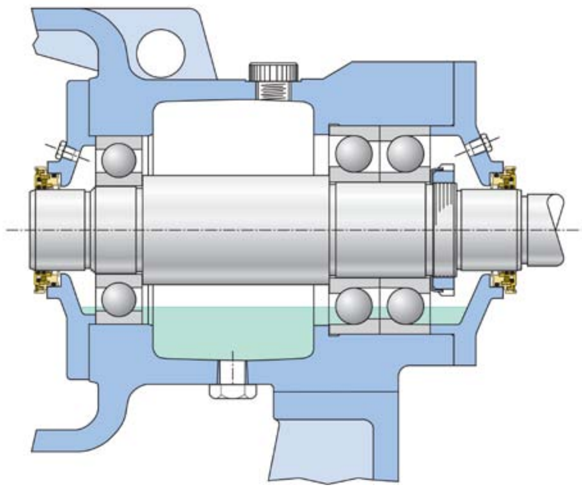

Fig. 2: Typical pump bearing housing with oil level reaching to center of lowermost rolling elements. Here, keeping the DN-values below 6,000 reduces the risk of oil overheating (Ref. 1, pp. 232)

oil level limits the frictional power loss due to the “plowing effect” of rolling elements. With lowered levels two separately derived empirical rules pertain: (1) at a DN > 6,000, or (2), with a shaft velocity in excess of 2,000 fpm, the oil level should be no higher than a horizontal line tangent to the lowermost bearing periphery, meaning there should be no

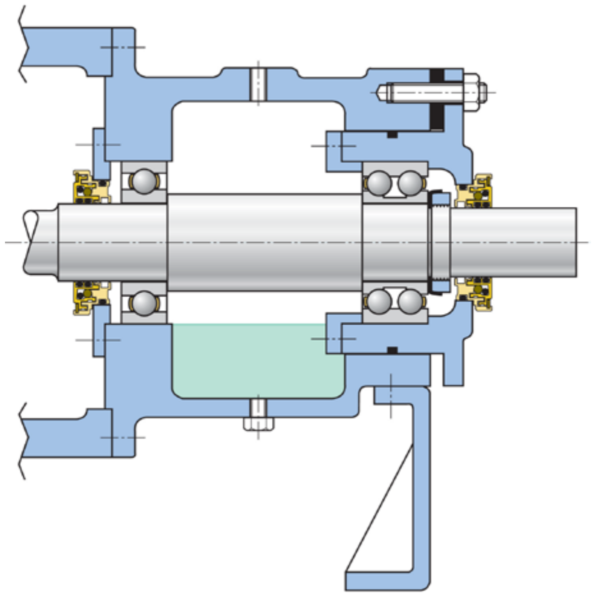

Fig. 3: Bearing housing with oil level lowered to accommodate high DN value. A flinger disc or other means of spraying oil into bearings will be needed (Ref. 3)

contacting of the oil level with even the periphery of a rolling element at the 6-o'clock location of the bearing. Because this lowering of the level was done in Fig. 3, either oil rings (Fig. 1) or shaft-mounted flinger discs (Figs. 4 and 5) will now be required to lift the oil, or to somehow get the oil into the bearing by creating a spray.

To illustrate Rule (1): A 2-inch shaft at 3,600 rpm, with its DN value of 7,200, would operate in the risky or instability-prone zone, whereas equipment with a 3-inch shaft operating at 1,800 rpm (DN = 5,400) might use oil rings without undue risk of ring instability. In another example, using Rule (2): A 3-inch (76 mm) diameter shaft at 3,600 rpm would operate with a shaft peripheral velocity of (πD/12)(3,600) = 2,827 fpm (~14.4 m/s), which would disqualify oil rings from being considered for highly reliable pumps. The fact that a pump manufacturer can point to satisfactory test stand experience at higher peripheral velocities is readily acknowledged, but field situations represent the "real world" where shaft horizontality and oil viscosity, depth of oil ring immersion, bore finish and out-of-roundness are rarely perfect. We can thus opt for using either the *DN < 6,000* or the *Surface Velocity < 2,000 fpm* rules-of-thumb. Either way, the vendor's test stand experience is of academic value at best. Pump manufacturers test under near-ideal conditions of shaft horizontality, oil ring concentricity and immersion, oil level and lubricant viscosity. For the reliability-focused, the wide-ranging field experience that led to these two rules-of-thumb will govern over all else.

The trouble with oil rings

Assuming that flinger discs are technically feasible, they should be given strong preference over oil rings (sometimes called slinger rings, Fig. 6). Serious issues with oil rings have been known for decades. These issues were alluded to in the 1970's when a well-known pump manufacturer saw fit to point out that its (superior to the competition) products incorporated an *"anti-friction oil thrower [i.e., a flinger disc], ensuring positive lubrication to eliminate the problems associated with oil rings"* (Ref. 1, pp. 237).

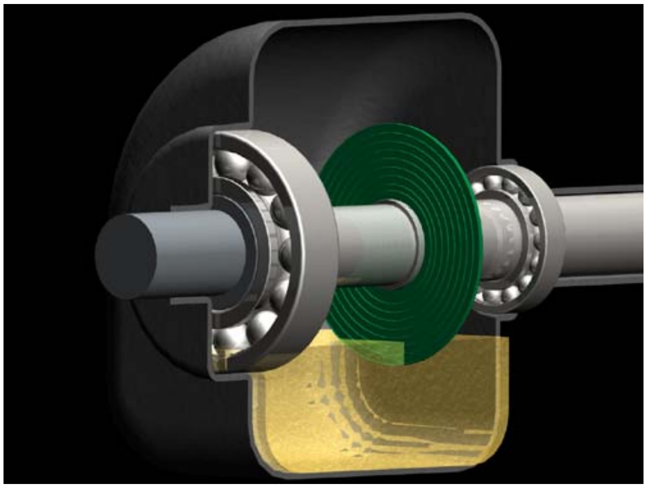

Fig. 4: Shaft-mounted flexible flinger disc splashes lube oil into bearings (Ref. 2)
(Source: also

In essence, oil rings are rarely (if ever) the most reliable choice of lubricant application. They often tend to skip around and even abrade (Fig. 6) unless the shaft system is truly horizontal, unless ring immersion in the lubricant is just right, and unless ring eccentricity, surface finish, and oil viscosity are within tolerance. Taken together, these parameters are rarely found within close limits in actual operating plants (Ref. 1, pp. 176-181).

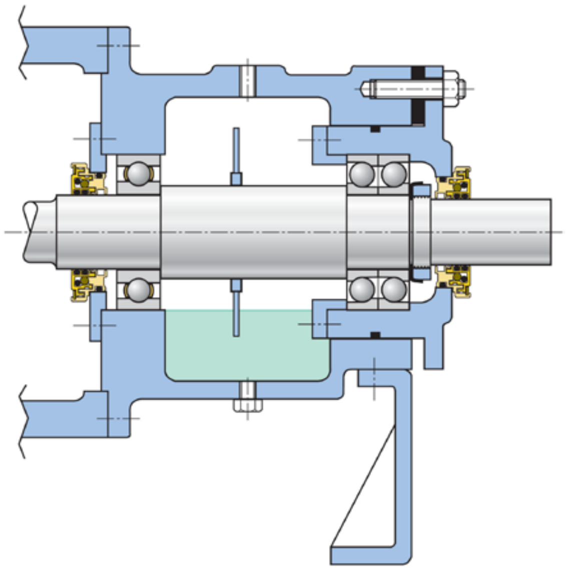

Fig. 5: A bearing housing accommodating a cartridge on the outboard side. The bearing housing bore is slightly larger than the diameter of the steel flinger disc, making assembly possible. The drawing does *not* show the desired oil return passage at the 6 o'clock bearing positions.

Fig. 6: Oil rings in as-new (“wide and chamfered”) condition on left, and abraded (“worn and narrow”) condition on right side (Source: Ref. 2)

In view of the shortcomings of oil rings, serious reliability-focused purchasers specify and select pumps with flinger discs. Although sometimes used to prevent temperature stratification of the oil (Ref. 1, pp. 127 and 232), flinger discs serve as efficient oil spray producers. Of course, the proper flinger disc diameter must be chosen and solid steel flinger discs should be preferred. Insufficient lubrication results if the diameter is too small to dip into the lubricant; conversely, high operating temperatures can be caused if the disc diameter is too large. *Flexible* flinger discs (Fig. 4) are needed to enable insertion in many “reduced cost” designs, i.e., ones where the bearing housing bore diameter is smaller than the flinger disc diameter.

To accommodate the preferred solid steel flinger discs, bearings must be cartridge-mounted (Fig. 5), in which case the effective bearing housing bore (i.e., the cartridge diameter) will be large enough for passage of a steel flinger disc of appropriate diameter.

To reiterate: At DN > 6,000 and to satisfy minimum requirements in a reliability-focused plant environment, a stainless steel flinger disc fastened to the shaft should be supplied. The disc contacts the oil and flings it into the bearing housing (Ref. 1, pp. 127). So as to function consistently and as intended, the flinger disc O.D. must exceed the thrust bearing’s outside diameter. To restate, this dimensional requirement strongly favors placing the outboard (thrust) bearing(s) in a separate cartridge. Providing such a cartridge will add to the cost of a pump. However, in the overwhelming majority of cases, the incremental cost will be much less than what it would cost to repair a pump just once.

Bearings and bearing housing protector seals

For decades, pumps that conform to the stipulations of API-610 (Ref. 4) have incorporated two thrust bearings mounted back-to-back, as depicted in Figures 1, 2, and 5. This allows selecting bearings with either the same or, for technical reasons slightly different, load contact angles (Ref. 1, pp. 157). Moreover, by carefully manufacturing the bearing inner rings a slightly different width compared to the width of the outer rings, either a negative or positive preload can be imparted (Ref. 1, pp. 153). Negative preloads allow for slight thermal growth and internal looseness, positive preloads prevent skidding of the rolling elements. Double-row angular contact bearings (Fig. 3, also Ref. 1, pp. 159) are less expensive, but do not match the thrust load capability of a set of dimensionally equivalent back-to-back angular contact bearings.

But moisture and dust often enter bearing housings at the shaft protrusion through old-style labyrinth seals or lip seals as airborne water vapour, or via a stream of water from hose-down operations. Contaminants can also enter through a breather vent, or from the widely used non-pressure balanced constant level lubricators (Fig. 7 and Ref.1, pp. 232-234). As mentioned earlier, abraded oil ring material is an often-overlooked source of oil contamination. A second possible source of lubricant contamination is the debris coming off so-called dynamic O-rings which, on old-style bearing protector seals, are often contacted by the sharp edges of opposing O-ring grooves (Fig. 8). Furthermore, certain “constant level lubricators” present two surprising but elusive contaminant entry locations, as will be discussed later.

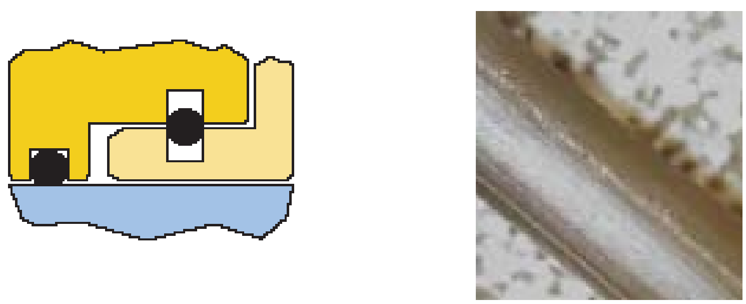

Fig. 8: Old-style bearing protector seals with sharp-edged grooves invite dynamic O-ring to scrape on edges; shavings then often contaminate the lubricant. Also, using only a single O-ring for clamping the rotor to the shaft makes the rotor less stable than if two rings were used for clamping duty

Unless the rotating equipment is provided with suitable bearing housing seals, an interchange of internal and external air (called “breathing”) takes place during alternating periods of operation and shutdown. Bearing housings “breathe” in the sense that rising temperatures during operation cause gas volume expansion, and decreasing temperatures at night or after shutdown cause gas volume contraction. Open or inadequately sealed bearing housings promote this back-and-forth movement of moisture-laden, contaminated air.

To stop this breathing and resulting contamination, there should be little or no interchange between the housing interior air and the surrounding ambient air. Breather vents (Fig. 1) should be removed and plugged. In advanced-style bearing housing seals (Fig. 9) a large cross-section shut-off ring contacts a wide and well-contoured surface and because

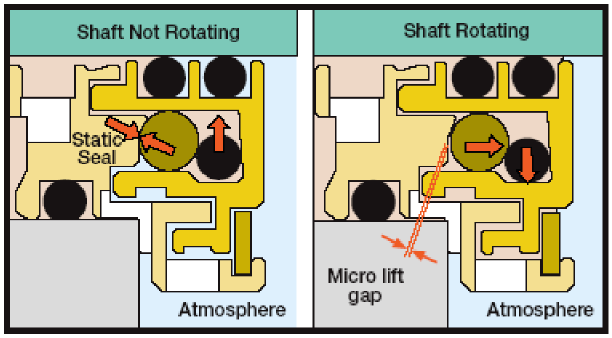

Fig. 9: Advanced non-contacting rotating labyrinth bearing protector seal with two O-rings securely clamped the rotor to the shaft. Dynamic O-rings do not contact sharp edges

pressure = force divided by area ("psi"), there's low pressure at standstill, and virtually no pressure during operation. Thus, at standstill, there is no fluid passage possible between the vapor space inside a bearing housing and the surrounding atmosphere on the outside.

As of mid-2009, none of these highly engineered models has failed in operation and tens of thousands of them are now running. A rigorous Weibull-WeiBayes analysis demonstrated exceedingly long life for this style of protector seal which, incidentally, is field-repairable. There is little (if any) risk of rotor-stator contact because the mass-symmetrical rotor is clamped to the shaft with two O-rings. This clamping method imparts great stability and resists "rotor wobble."

In essence, bearing protector seals can greatly improve both life and reliability of rotating equipment by safeguarding the cleanliness of the lubricating oil. However, bearing protector seals serve no purpose if oil contamination originates with oil ring inadequacies or if used with unbalanced oilers, or if the oil is not kept at the proper oil level. The following segment will emphasize our point.

Issues with oil supplies using "constant level" lubricators

For well over a century, various types of constant level lubricators have been used and their respective vulnerabilities are often overlooked. However, if there are pressure differences between points "A" and "B" in Fig. 10, the two oil levels will differ as well. Also, whenever caulking is used to secure an oiler bulb or transparent bottle to a supporting component, the caulking can "weather" and develop small fissures. Rainwater can enter at these fissure locations via capillary action. Therefore, these constant level lubricators must be part of a precautionary replacement strategy.

Also, not all versions of "constant level" lubricators supplied by pump manufacturers will best serve the reliability-focused user. In the widely used pressure *non-balanced* device of Fig. 10, the oil level below the reservoir bottle is contacted by ambient air and this represents another entry point for contamination. Because, in the *pressure-balanced* device of Fig. 11, the oil is *not* contacted by ambient air, a pressure-compensated, i.e., "pressure-balanced" constant level lubricators should be installed (Ref. 1, pp. 234).

In both pressure non-balanced and pressure-balanced styles, a wing nut adjustment at "A" sets the height of the transparent bottle. However, in the non-balanced style of Fig. 10, the oil level near the tip of this wing nut is contacted by the surrounding (ambient) air. An increasing gas temperature (usually air, or an air-oil mixture) in the bearing housing (Point "B" in Fig. 10) tends to elevate pressure in the bearing housing. This elevated gas pressure drives down the oil level in the housing, as indicated by the arrow at "B". The displaced oil reappears as an increased oil level in the narrow annular space above the tip of the wing nut, and can result in overflow and spillage of oil from the annular space. When this happens, bearings starve for oil and will fail.

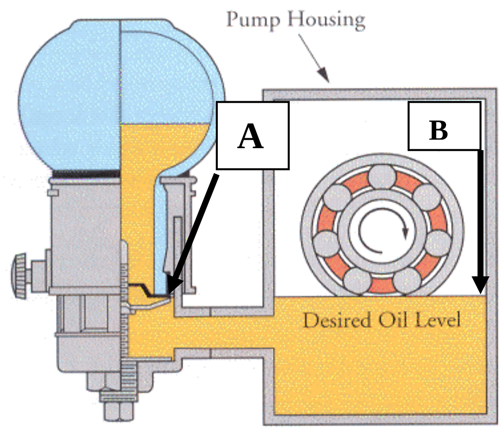

Fig. 10: Constant level (non-pressure-balanced) lubricator (Source: Ref. 2)

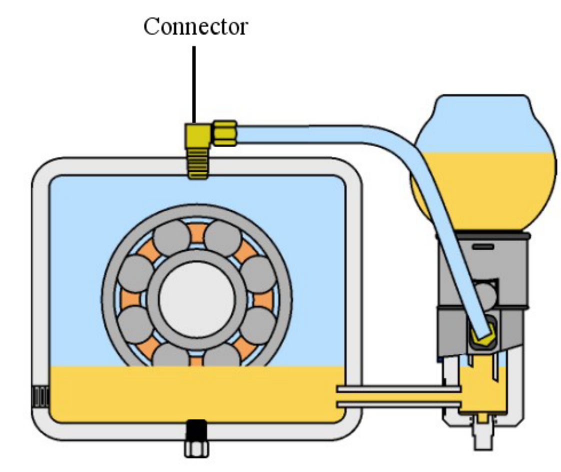

Fig. 11: Pressure-balanced constant level lubricator (Source: Ref. 2)

In contrast, pressure-balanced oilers (Fig. 11) decrease downtime risk. They differ from the non-balanced type by incorporating an external or internal pressure balance pipe or balance line. Pressure-balanced oilers utilize suitable internal O-rings or gasketing, which then ascertains that the pressures inside the bearing housing and the pressures at the tip of the wing nut in the constant level lubricator are always identical. The *same* pressure is pushing downward on the oil at the adjusting wing nut in the constant level lubricator and the oil in the bearing housing; therefore, there is no change in the oil level.

Best lube application practices examined

Finally, and in carrying out best lubrication practices, it will be noted that full sealing of the bearing housing is required. Face-type bearing housing seals may be needed if modern oil mist lubrication is applied. An API-610 compliant (Ref. 4) magnetically-activated dual-face seal used on oil mist-lubricated rolling element bearings is shown in Fig. 12; a similar device is used to seal conventional oil-splash lubricated bearings. A suitable face seal, along with the other recommendations above (plugging the vent and using balanced oilers), will preclude entry of all external contaminants into the housing. Oil mist represents the most technically effective and consistently reliable combination of excluding contaminants and supplying lubricant to rolling element bearings in pumps.

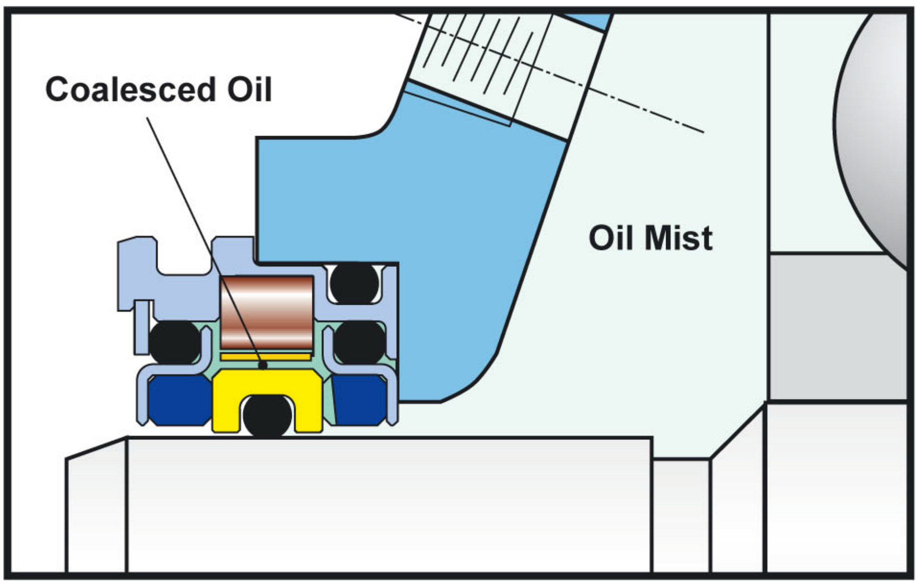

Fig. 12: Dual-face magnetic bearing protector seal in oil mist service
(Source: AESSEALS, Inc., Rotherham, UK and Rockford, Tennessee)

Plant-wide oil mist lubrication systems have proven superior to conventional lubricant application since the late 1960's. Pump bearing failure reductions ranging from 80 to 90% (Refs. 1, 5, 6) have been reported with pure oil mist, applied as shown in Fig.13. The advantages and disadvantages of oil-mist lubrication as compared to conventional oil lubrication can be summarized as follows:

Advantages:

- Reduced bearing failures of 80 to 90%.
- Lower bearing operating temperatures of 10 to 20 F.
- No recirculation of bearing wear or debris particles.
- Slight positive system pressure eliminates contaminant entry.
- Reduced energy costs of 3 to 5%.
- Reduced oil consumption of about 40%.
- No moving parts.

Disadvantages:

- Capital investment
- Cost of compressed air

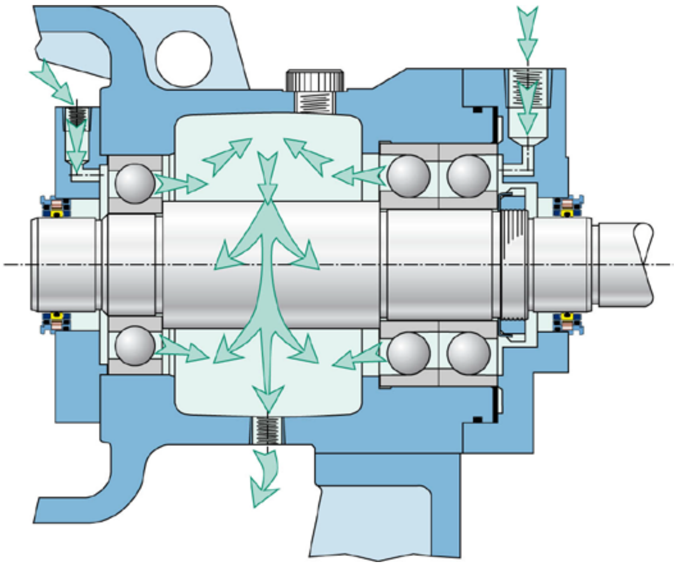

Fig. 13: Oil mist lubrication applied to a pump housing in accordance with API-610, 10th Edition. Note dual mist injection points and dual face bearing housing seals that prevent mist from escaping to atmosphere. With oil mist, there are neither oil rings nor flinger discs

Conclusions

As we perhaps re-read this article, let's resolve to not claim "things generally work passably well." Passably well results in repeat failures and tolerating them is no longer good enough. As reliability professionals and as pump manufacturers we have to do better and should know how to do better. The proposed new minimum requirements for reliability-focused users must include

1. Advanced versions of bearing protector seals for both the inboard and outboard bearings. Lip seals are not good enough, and neither are old-style rotating labyrinth seals. Know what you're buying and see beyond advertising claims.
2. As a matter of routine, the housing or cartridge bore must have a passage at the 6 o'clock position to allow pressure and temperature equalization and oil movement from one side to the other side of the bearing. Note that such a passage was shown in Fig. 1 for the radial bearing, but not for the thrust bearing set.
3. With proper protector seals and the right constant level lubricators, breathers (or vents) are no longer needed on bearing housings. The breathers (or vents) should be removed and one of the openings in Fig. 1 should be plugged.
4. A pressure-balanced constant level lubricator should be supplied and its balance line should be connected to the second of the two breather ports.
5. Bearings should be mounted in suitably designed cartridges and loose slinger rings (oil rings) should be disallowed.

6. Suitably designed flinger discs should be secured to the shaft whenever the oil level was lowered to accommodate the need to maintain acceptable lube oil temperatures (i.e., for pumps operating with DN-values in excess of 6,000).

Of course, we realize pump manufacturers are often concerned about legal issues. In answer to upgrade requests they will rightly point to test stand data certifying that things work well, even if the aforementioned upgrade measures are disregarded. With the same degree of firmness, knowledgeable engineers can prove that things tend to malfunction in the real world and it's in the user's best interest to reduce downtime risk. Plain logic should lead to agreement on this premise: As we get further and further away from solid training and from taking the time needed to do things right, we become ever more vulnerable. One way to counteract this vulnerability is by designing-out maintenance.

Regrettably, the trend to cheapen everything is progressing. When the major user mentioned at the beginning of this article invited several U.S. pump manufacturers to submit bids on 23 pumps that were to incorporate many of the features discussed here, the vendors took exception to some or all of the listed experience-based requirements. We can speculate on the various reasons, but have given up writing about them because honesty and candor are no longer rewarded in an imperfect world. Yet, hopefully, a few pump manufacturers will wake up before the foreign competition gears up to supply us with what we really need: Better pumps for forward-looking users. We know that some overseas vendors are working hard to capitalize on the opportunity.

We also realize that housing redesigns will be needed for many pumps, and that redesigns cost money and that fewer equipment failures will result in fewer replacement parts being sold by their makers. Still, it should not be difficult to demonstrate the value of the upgrading effort. While it would be utopian to expect companies in possession of exact statistics to publicize these, we should acknowledge that a particular risk element might reduce MTBR (mean time between repairs) by 10%. In other words, the continued use of flawed oil rings would reduce MTFR from a multiplier of 1 to an MTBR multiplier of 0.9. If each of six risk-increasing items would reduce MTBR by 10%, then the MTBR for that pump is only 50% of what it would be after upgrading.

Of course, parties refusing to see the light will continue to be at liberty to buy whatever is cheapest. There will always be vendors producing things of lesser value, "stuff" that ultimately puts jobs at risk. Just remember that it doesn't have to be that way and that for many pump drive-ends the alternatives and options were summarized here on just a few pages. That said, the choice is now yours to make.

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