BREAKING THE CYCLE OF PUMP REPAIRS

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INTRODUCTION

Of the numerous process centrifugal pumps undergoing repair right this very minute, an estimated 90% have failed randomly before. Some have run just fine until the very first time we fixed them, and were never quite the same since. That begs answers to a few interesting questions: Could it be that we don't really know why so many of these pumps are failing? Could it be that we just don't give pumps the attention they deserve? Is it because everybody's priorities are elsewhere?

One of us has worked in industry for 62 year and graduated from a great engineering college 50 years ago. The younger of us also has a solid background in the field of Reliability Engineering, although it's fair to point out that he wasn't even in elementary school when my supervisor at then _Esso Research and Engineering_ in New Jersey first sent me out to monitor a pump problem with an old vibrograph. Rather than presenting a theoretical treatise, consider this tutorial a practical guide, a rule of thumb, and a call for common sense engineering evaluation of certain improvement opportunities that were actually implemented. Anyway, we ask you to please assume that we speak from experience and have no hidden agenda.

Allow me to draw the right conclusions from over five decades of careful observation. We will skip talking about the consequences of "business as usual" attitudes. Instead of adding to the laments, let us explain proven steps to improve pump life.

Improvement is both possible and valuable. Its cost-effectiveness can be quantified without much difficulty. The key is to talk "management speak," which includes "monetizing" the repairs, quantifying the process losses, and doing B-over-C (benefit-to-cost) and life cycle analyses.

The cost of failures

One way of exploring the value of extending pump mean-time-between-failures (MTBF) is to examine the likely savings and 1000/5.5 = 182 repairs after understanding the savings of, say, improving MTBF from previously 4.5 years to now 5.5 years. We would wish to understand and then solve the various problems. Avoiding 40 repairs at $ 6,000 = $240,000. Manpower would be freed-up for other tasks: At 20 man-hours times 40 incidents times $100 per hour, reassigning these professionals to other repair avoidance tasks would be worth at least $80,000. That's just the tip of the iceberg.

There is also one ~$3,000,000 fire per 1,000 pump failures. That's high in the estimation of some competent engineers. An engineer whom we respect advised it might more likely be 1 fire per 1000 pump failures, then out of 10 fires he figures seven are less than $50K, two are $50-500K and 1 is >$500k. He asked us to provide a source on our numbers, but he should know that U.S. employers or companies would not be pleased if we divulged sources. Virtually all consulting done today by highly qualified independent professional engineers is linked to legal non-disclosure agreements. The client must file reports with governmental and civil entities. Clients are fearful their explanations might differ from the findings of consulting engineers who understand the true root causes of failures, or whose sense of priorities is tuned to a different standard. Diverging statements or findings might feed a bureaucratic machine that will busy itself with issues of that type. For now, we will stick by our story. Our story simply means that

avoiding 40 repairs would be worth 40/1,000 x $ 3,000,000 -- $120,000. The three items ($240k, $80k and $120k) add up to $440,000. Much more is at risk if the standby pump malfunctions upon being started up.

We could examine other ways to calculate as well. It would be reasonable to assume that implementing a component upgrade (generally the elimination of a weak link) extends pump uptime

by 10%. Implementing 5 upgrade items yields 1.1^5 = 1.61--- a 61% mean-time-between-repair (MTBR)
increase. Or, say, we gave up 10% each by not implementing 6 reasonable improvement items. In that instance, 0.9^5 = 0.53, meaning that the MTBR is only 53% of what it might otherwise be.
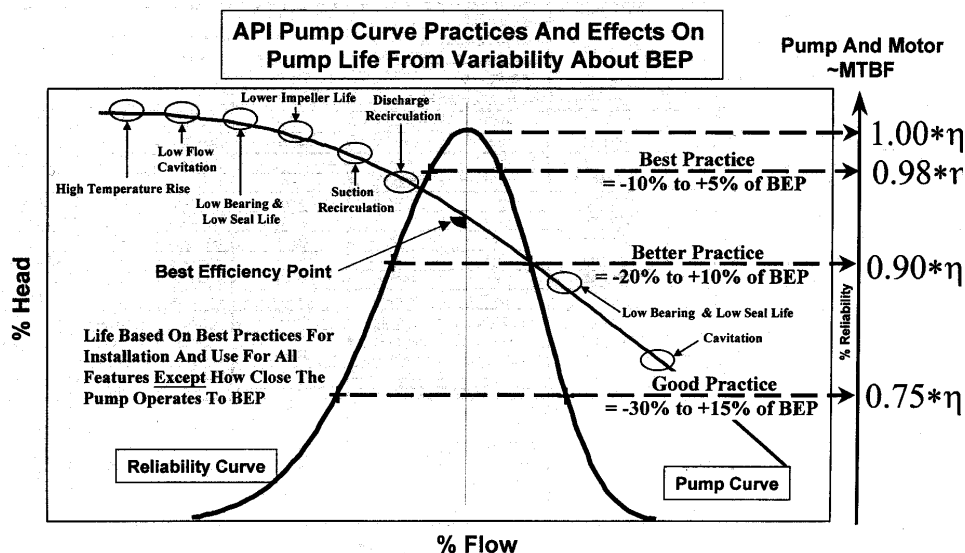

Figure 1. Barringer-Nelson curves show reliability impact of operation away from BEP
(www.barringer1.com)

That might well explain industry's widely diverging MTBRs. The MTBR-gap is quite conservatively assumed to range from 3.6 years to 9.0 years in U.S. oil refineries and, as of today, no informed pump professional has disagreed with these numbers.

Pumps have a defined operating range

So, let's start with Figure 1, the typical HQ curve with the eight traditional non-BEP problem areas plotted on it. It amplifies the notion that centrifugal pump reliability can approach zero as one operates farther away from the best efficiency point or BEP. The curve depicted in Figure 1 has been widely copied and certainly reflects the expert opinions of Paul Barringer and Ed Nelson.

Let us put a frame around the implications of Figure 1 and call it our Point 1. Just because centrifugal pumps are able to operate at low flow doesn't mean that it's smart to do so. Compare it to a car that's capable of going 12 mph in 6th gear, or 47 mph in 1st gear: It can be done, but there will be a big price to pay! Talking about it would make us

drift into pump hydraulics and internal recirculation issues that have been discussed or many decades. Since we have been asked to present a tutorial on _Breaking the Cycle of Pump Repairs_, we will try not to dwell too much on what many of you are already doing, or what has not made much of a dent in repeat failure events. Still, we must take time to acknowledge the pioneers whom some of us had the pleasure of meeting.

> _Point 1: Stay well Inside the Defined Operating Range. Safe Operating Margins are the Key to Failure Avoidance_

The hand sketch, Figure 2, was originally done by Irving Taylor in 1977 (Taylor, 1977). His work is worth mentioning because he explained in one illustration what others have tried to convey in complex words and many mathematical formulas. Taylor deserves much credit because he managed to keep things simple.

To facilitate making a copy on black-and-white copiers (and we encourage you to make those

copies) we have used dashes and dots to label Taylor's work in Figure 2.

Although more precise calculations are available, Taylor's trend curves of probable NPSHr for minimum recirculation and zero cavitation-erosion in water, Figure 2, are sufficiently accurate to warrant the attention of reliability professionals who wish to work within safe margins. Hundreds of references exist on the subjects of cavitation and internal recirculation; stable operation is always the central aim (Bloch, 2011). The actual NPSHr needed for *zero* damage to impellers and other pump components may be many times that published in the manufacturer's literature. The manufacturers' NPSHr plot (lowermost curve in Figure 2) is based on observing a 3% drop in discharge head or pressure. Taylor's plot places the Q = 100% intersect at an NPSHr = 100% of the manufacturer's stated value. Unfortunately, whenever this 3% fluctuation occurs, a measure of

damage may already be in progress. It is prudent to assume a more realistic NPSHr and to provide an NPSHa in excess of this likely NPSHr. This serves to build a certain margin of safety into the pump and greatly reduces the risk of catastrophic failure events.

Stating a bottom line for what the safe margins may be will invite arguments, and understandably so. We have seen hydrocarbon services where an NPSHa surplus of just 1 ft over NPSHr was sufficient to avoid cavitation. However, there are also services, such as Carbamate solution, where a 25 ft surplus is not nearly enough. Remember Taylor: His trend curves are approximations for the prudent user. He indicated that we should enlist help from competent pump manufacturers and above-average design contractors to define the NPSH multipliers or bracket experience-based margins for a particular pumpage or service.
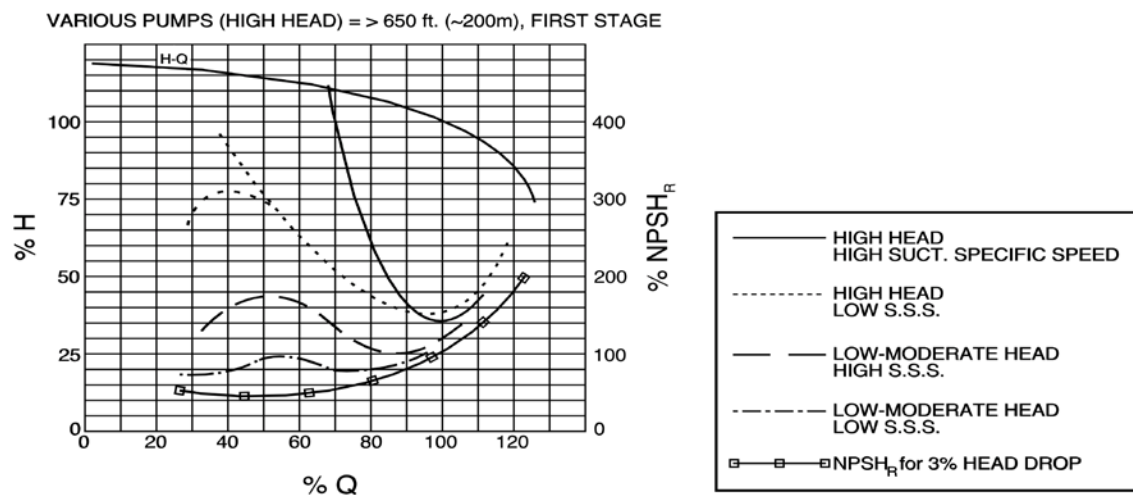

Figure 2. Pump manufacturers usually plot only the NPSHr trend associated with the lowermost curve. At that time, a head drop or pressure fluctuation of 3% exists at BEP flow (Taylor, 1977). BEP stands for "Best Efficiency Point"

Again, Irving Taylor acknowledged that his curves were not totally accurate; he then mentioned the demarcation line between low and high suction specific speeds (N**ss**) was probably somewhere between 8,000 and 12,000. Many surveys taken after 1980 point to 8,500 or 9,000 as numbers of concern. If pumps with N**ss** numbers in excess of about 9,000 are being operated at flow rates much higher or lower than BEP (Best Efficiency Point), the life expectancy or repair-free operating time of these pumps will be reduced. Most reliability engineers are faced with root-cause failure analysis (RCFA) of pumps that have already been purchased and are in operation. If the hydraulic fit

of the pump does not match the process, options are available to retrofit pump internal components to redefine the pump BEP closer to the process requirements.

Also, the vulnerability of operating certain pumps in parallel is not always emphasized by pump vendors. There are problems with short elbows near the suction nozzle of certain pumps and flow stratification and friction losses are sometimes overlooked. Some sources advocate a minimum of 5 diameters, others advocate a 10-diameters equivalent of straight pipe run at the pump suction. Collective piping issues make up our Point 2.

The tie-in between the lack of conservatism in piping and issues of less-than-adequate pump reliability is tenuous; still, the multi-point trouble illustration (our earlier Figure 1) is of interest here. Suffice it to say that tight-radius elbows and their incorrect orientation can quickly wreck double-flow pumps. Neglecting piping issues can be a costly mistake, but ours is not a piping tutorial. Moreover, by listing Point 2 we simply wish to make you aware of hydraulic and flow separation issues. The flow velocity at the small-radius wall of an elbow will differ from that at the large-radius wall. And again, because these facts are generally well known and many symposia have been devoted to them, we will direct our discussion to pump mechanical or drive end (power end) issues. Our topic is failure avoidance. Every so often, we must remind ourselves of the basic question: Why do we have repeat pump failures and how can we avoid them?

Why insist on better pumps

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. Using different words and so as to avoid confusion, note that the drive end of a process pump is often called the mechanical assembly (SKF, 1995). The mechanical assembly comprises shaft seal, shaft, bearings, bearing housing or frame, and drive coupling or sheave. The same reference (SKF, 1995) distinguishes the hydraulic assembly from the mechanical assembly. SKF includes in the hydraulic assembly the impeller or propeller, the suction inlet, the volute, and the seal rings.

So, it's not just hydraulics, metallurgy and driver selection. Start with a proper specification and put vendors on notice that you have good reasons to ask for and pay for better pumps. Don't overlook the merits of testing, foundation, mechanicals (reasonable L^3/D^4 ratios), reliable bearings and lube application methods, low-risk couplings and, finally, soundly selected pump drivers. An over-emphasis on (initial) cost-cutting by pump manufacturers and purchasers has negatively affected the drive ends of many thousands of process pumps. Each party blames things on the other and it's a fruitless argument to say any more. Still, flawed drive end (power end) components contribute to elusive repeat failures that often plague these simple machines. Pump drive end

failures represent an issue that has not been addressed with the urgency it deserves. Always

remember that repeat failures can only happen if the true root cause of failure has remained hidden or, if the true root cause is known, someone decided not to do anything about it. Either of these two possibilities runs counter to the avowed goals of asset preservation and operational excellence. As to our nomenclature, please refer back to SKF's definition. They label as "drive end" the mechanical assembly; a bearing housing with its bearings and bearing protector seals is a major component of the mechanical assembly. SKF includes mechanical seals in the mechanical assembly, but excludes what it calls "seal rings". We believe SKF's "seal rings" are really the throat bushing(s)—a component closest to the impeller. For the sake of our tutorial, the driver is either an electric motor or a steam turbine. While located near the "drive end," drivers are not included in our tutorial definition of "drive end" or "mechanical assembly." This tutorial deals with process pumps---notably API-style centrifugal pumps.

Deviations from best available technology are risky

User plants will usually get away with one or two small deviations from best available technology. But when three or more deviations occur, we very often increase failure risk exponentially. That said there are a number of reasons why a few well-versed reliability engineers are reluctant to accept pumps that incorporate the drive end shown in Figure 3. The short overview of reasons is that reliability-focused pros 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 viewer might see nothing wrong, Figure 3 contains a few clues as to why many pumps tend to fail relatively frequently and quite often randomly. The illustration shows areas of vulnerability, and not eliminating these vulnerabilities can be a costly mistake. Allowing these risk increasers to exist will sooner or later hurt the profitability of users and vendors alike. It will also discredit, or cast aspersions at, an entire profession. All involved parties should pay very close attention to lube application matters related to process pumps, which is our Point 3.
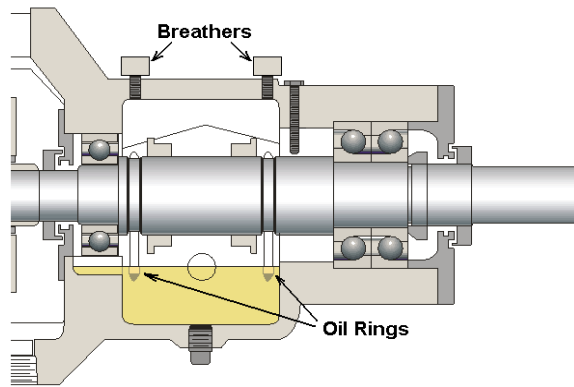
Figure 3. A bearing housing with several, potentially costly, vulnerabilities

So, as we examine Figure 3, we can be certain of five facts:

- In Figure 3, oil rings are used to lift oil from the sump into the bearings. These oil rings tend to skip and jump at progressively higher shaft surface speeds, or if not near-perfectly round, or if not operating in an almost perfectly horizontal shaft system.
- The back-to-back oriented thrust bearings of Figure 3 are not located in a cartridge, which limits flinger disc dimensions-- assuming such retrofits would be desired-- to no more than the housing bore diameter
- Bearing housing protector seals are missing from Figure 3
- Although the bottom of the housing bore (at the radial bearing) shows the needed oil return passage, the same type of oil return or pressure equalizing passage is *not* shown near the 6 o'clock position of the thrust bearing. That introduces risk
- There is uncertainty as to the type or style of constant level lubricator that will be provided. Unless specified, the best one is rarely supplied on new pumps

As discussed, our considerations are confined to lubrication issues on process pumps with liquid oil-lubricated rolling element bearings. Bearing housings with liquid oil reaching to the center of rolling elements at the lowermost part of the bearing will be involved, as will housings with oil levels purposely maintained at much lower levels. Each method has its purpose and its limits. For now we will exclude sliding bearings, although some principles do, in fact, apply also to pumps with sleeve bearings.

It should be noted that the angular contact thrust bearings shown in Figure 3 will usually incorporate

cages (ball separators) that are angularly inclined. This means the cages are oriented at a slant. These cages then often act as small impellers, and impellers promote flow from the smaller towards the larger of the two diameters. This is more readily evident from Figure 4, and particular attention should be given to windage created by the impeller-like air flow action of an inclined cage. In many cases, the pump manufacturer places an oil ring to the left of this bearing; the obvious design intent is for oil to flow from left to right. Unfortunately, that's where the lubricant is often opposed by windage effects that act from right to left. So, whatever oil application method is chosen, it will be necessary for the lubricant to overcome this "windage."

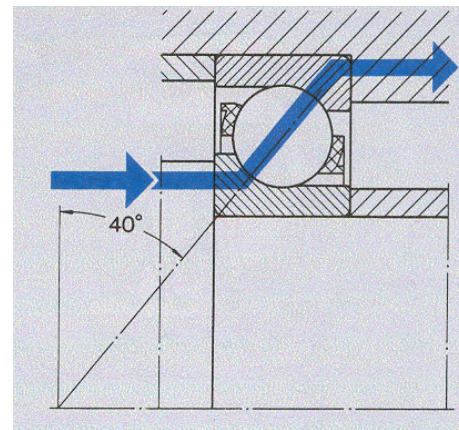*Point 4: Windage in AC Bearings can Oppose Oil Flow*

Figure 4. Attempts to apply lubricant in the direction of the arrow meet with windage (air flow) from an inclined cage.
The two directions often oppose each other.

Therefore, windage is our Point 4, and we must ask: How does one alleviate windage and/or its effects? The fact that windage may be generated by some of these bearings and is more likely in certain bearing housing configurations requires thoughtful--and sometimes purely precautionary--abatement of unequal pressures inside a bearing housing.

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

Before we now progress further into the topic, note how carefully the long-defunct Worthington Pump

Company ascertained that pressures surrounding all bearings were equalized. They went through the trouble of drilling balance holes right above the bearings, Figure 5. Do you have balance holes in your pump bearing housings? If not, then why not? Perhaps you don't need them, but then again --- maybe you do. The game is all about risk reduction.

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 (Figure 5). However, this "plowing through the oil" may cause the lubricant to heat up significantly and should be avoided on susceptible process pumps.
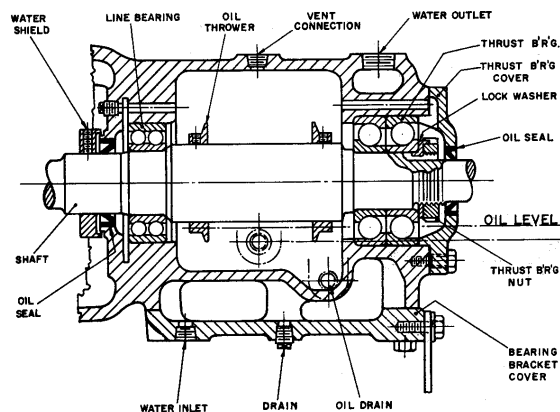

Figure 5. The oil level in this 1960s-vintage housing was set for low-to-moderate speed pumps. Oil throwers create a spray that overcomes windage; the two throwers also prevent oil stratification. Important pressure equalization passages are drilled near the top of all bearings (Bloch, 2011)

There's excessive heat generation risk whenever dn, the mean distance from diametrically opposite rolling element centers, as expressed in [mm] x [rpm], exceeds a particular number. That 6-digit number ranges from 150,000 to perhaps 300,000. It is predetermined by bearing manufacturers who estimate at what point churning and heat buildup will exceed desired safe limits. The manufacturers then advocate lowering the oil level so that oil no longer contacts the rolling elements directly. In essence, as a certain dn threshold is exceeded, some other means of lifting oil into the bearing must be chosen.

Fortunately, and aiming to stay within the inch-system preferred by pump users in the United States, a number of bearing manufacturers and users found that the ratios of bearing outside diameters (OD) to bearing inside diameters (ID) are similar in different bearing sizes typically used in process pumps. This allowed these users to focus on a simplified approximation, DN, the product of

shaft diameter (D, inches) times revolutions-per-minute (where N = rpm). Whenever DN exceeds 6,000 and so as to avoid risking excessive heat buildup, oil levels reaching the ball center or the lower third of the lowermost rolling element are considered a churning risk. In that case, some other means of lifting oil into the bearing are chosen.

Note also the cooling water jacket in Figure 5. Bearing housing cooling is not needed on process pumps which incorporate rolling element bearings. Cooling is harmful if it promotes moisture condensation or restricts thermal expansion of the bearing outer ring (Bloch, 1982/2011). When, in 1967, these concerns were seen to influence pump reliability, the jacketed cooling water passages in Figure 5 were left open; the passageways were from then on flooded by the ambient air environment. This "no more cooling water" decision was reached and implemented in 1967 at an oil refinery in Sicily. The owner's engineers had recorded bearing lube oil in four identical pumps reaching an average of 176 degrees F with cooling water in the jacket passages. Without cooling water, the lube oil averaged 158 degrees F, which is 18 degrees cooler. The cooler bearings now lasted much longer and we shared these findings with all those that were willing to read, or willing to listen. They were included in many books and presentations (Bloch, 1982).

Today, 45 years later, not everybody has acted on the message. That is why cooling water issues are listed here as Point 5.

> ***Point 5: Cooling Water can Cause Bearings to Run Hot***

Note also that Figure 5 depicts the traditional oil sump with the lubricant reaching to about the center of the lowermost bearing elements. This arrangement works well at low shaft surface velocities. To gain reliability advantages, synthetic lubricants, oil mist application (called "oil fog" in some languages), oil jets --also known as "oil spray" and well documented (Bloch, 2011; SKF, 1995; Eschmann et al, 1985; Bloch/Budris, 2010; Bloch, 2009; MRC, 1982). Even circulating systems deserve to be considered in certain high-load or very large pumping services. Generally speaking, 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 (SKF, 1995). The lubricant viscosity required is a function of bearing diameter and shaft speed. It is described in numerous books and articles (among

them Bloch, 2011; SKF, 1995; Eschmann et al, 1985; Bloch/Budris, 2010; Bloch, 2009; MRC, 1982 and dozens of others). Most process pump bearings will reach long operating lives if the oil viscosity (at a particular operating temperature) is maintained in the range from 13 to 20 cSt (SKF, 1995). It should be noted that whenever oil rings are used to lift the oil from sump to bearing, maintaining a narrow range of viscosities takes on added importance. If a bearing housing accommodates both rolling element and sliding bearings, it will be prudent to understand and address the implications of (some) oil rings not being able to function optimally in the higher viscosity (ISO Grade 68) lubricant that's often chosen for rolling element bearings. The oil ring may have been designed to cater to the needs of a lower viscosity (ISO Grade 32) lubricant, but VG 32 _mineral_ oils are rarely a best choice for rolling element bearings. A high performance _synthetic_ VG 32 will often succeed as the most suitable selection for different bearing styles sharing the same housing.

To restate the above: Overheating occurs on many pumps operating at 3,000 or 3,600 rpm. Because the "plowing effect" of rolling elements produces frictional power loss and heat, an oil level below that indicated in Figure 5 is then chosen. A widely accepted empirical rule calls for lower oil levels whenever DN > 6,000. Another, separately derived empirical rule, allows shaft peripheral velocities no higher than 2,000 fpm in bearing housings where the oil sump level is set to reach the center of the lowermost rolling element. It is generally agreed that with shaft surface velocities in excess of 2000 fpm, windage effects are opposing the flow of oil mist. This is being observed more and more often, and some rather uninformed users have, in some cases, reverted back to wet sump oil lubrication. In sharp contrast, reliability-focused users have, for many decades, installed "directed" oil mist reclassifiers to overcome windage at >2,000 fpm. The mist dispensing opening in these reclassifiers is located 0.2-0.4 inches from the rolling elements. Thousands of these have been supplied and used with total success. This information is again available from dozens of texts and articles (among them Bloch, 1987; Bloch, 2009; Bloch, 2011; Bloch/Budris, 2010; Bloch/Shamim, 1998; MRC, 1982; Towne, 1983).
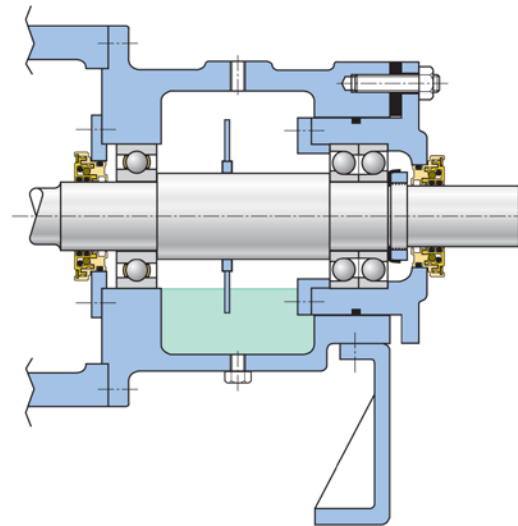

Figure 6. A bearing housing with a cartridge containing the thrust bearing set.

In Figure 6, the bearing housing bore is slightly larger than the diameter of the steel flinger disc, making assembly possible. Note that the drawing does _not_ show the required (and needed for failure risk reduction!) oil return passage at the 6 o'clock bearing positions (Bloch, 2011; Bloch, 2009).

Again, once the shaft peripheral velocity exceeds 2,000 fpm, the oil level should be no higher than a horizontal line tangent to the lowermost bearing periphery. This means there should be no contacting of the oil level with any part of a rolling element. Assume that Figures 3 and 6 represent situations where DN > 6,000. Therefore, and because initial cost was to be minimized, either oil rings (Figure 3) or shaft-mounted flinger discs (Figure 6) were chosen. Both arrangements are available to lift the oil, or to somehow get the oil into the bearing by creating a spray. Point 6 serves as a reminder.

> **_Point 6: Understand where to set oil levels_**

Two different DN-rules explained

When determining oil level settings, either of the two rules could be applied. To illustrate----

Rule (1): A 2-inch bore bearing at 3,600 rpm, with its DN value of 7,200, would operate in the risky or ring instability-prone zone, whereas equipment with a 3-inch bore bearing operating at 1,800 rpm (DN = 5,400) might use oil rings without undue risk of ring instability. In another example using, as an illustration,

Rule (2): A 3-inch (76 mm) diameter bearing bore at 3,600 rpm would operate with a shaft peripheral velocity of

$$(\Pi D/12)(3,600) = 2,827 \text{ fpm } (\sim 14.4 \text{ m/s}),$$

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 interest at best. Pump manufacturers test under near-ideal conditions of shaft horizontality, oil ring concentricity and immersion, oil level and lubricant viscosity. As users we might ask ourselves how often we have seen non-round oil rings, or rings that have shaft radius wear mark on one side of the ring, never having turned. If the answer is "never," perhaps another look will be warranted. We might also ask what prompted at least some pump manufacturers to point out certain shortcomings of oil rings (Figure 8; also Bloch, 2011). For the reliability-focused, the wide-ranging field experience that led to these two rules-of-thumb will outrank all theories.

The cartridge approach shown in Figure 6 has been in use for about 50 years on thousands of open-impeller ANSI pumps because it facilitates shaft position adjustment in the axial direction. Of course, cartridge-mounted bearings are a cost adder and you will hear claims that the benefit-to-cost-ratio will not justify upgrading to cartridges. We beg to differ. With the average API pump repair costing slightly over $10,200 at a Texas refinery and $11,000 at a refinery in Mississippi, some might be surprised at the high payback multiplier. Avoiding a single failure over the 30-year total life of a pump will pay for it.

We stand by these numbers because we know what needs to be considered in a pump repair cost calculation: Direct labor, direct material, employee benefits at roughly 50% of direct labor, refinery administration and services costs at 10% of direct labor, mechanical group overhead costs amounting to 115% of direct labor, and materials procurement costs of 7.4% of materials outlay (Bloch/Budris, 2010). By disregarding the true costs of failures, some professed experts are inadvertently impeding progress towards better pumps. In particular, progress towards obtaining pumps that are not subject to frequent repeat failures, or unexplained random failures, is sluggish when the true costs are not brought to management's attention.

The trouble with oil rings and constant level lubricators

In 2011, an industry source mentioned that "ring lubrication is an accepted practice and it would take user consensus to damn it." This statement neither solves problems nor will it acknowledge the work of Heshmat (1985), Wilcock & Booser (1957), or Leonardo Urbiola (Texas A&M Masters Thesis, ~2000/2001). Each writer observed issues with oil ring components. We just ask you to keep in mind that this tutorial is for the reliability-focused. Nothing will convince those who accept without questioning dozens of repeat failures of centrifugal pumps at their plants. The co-authors of this tutorial have submitted and included illustrations of failed oil rings and have demonstrated that rings are unreliable under typically less-than-perfect field conditions. One of the authors has performed field measurements (Bloch, 2009/2011) that proved that some oil rings exceeded the maximum out-of-roundness tolerances given in one of our cited references by a factor of 30. In their widely used handbook, Wilcock & Booser (1957) asked for oil rings to be concentric within 0.002 inches. Copies of Urbiola's master's thesis can be obtained from Texas A&M University. That these and other writings are clearly available, yet are not commonly known, is patently clear to us as co-authors. On the other hand, the fact that industry suffers from repeat failures is not being questioned by reasonable people. And that, of course, is the reason for this tutorial---sharing contributing issues that many may have overlooked.

Those of us who have studied the issue and have collected data believe that, in essence, oil rings are rarely (if ever) the most dependable means of lubricant application. They tend to skip around and even abrade (Figure 7) 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. Hence, we make our Point 7. Anyway, taken together, these parameters are rarely found within close limits in actual operating plants (Bloch, 2011). We have many photos in support of these findings.

> *Point 7: Flinger Disc can Outperform Oil Rings. Oil Rings Must be Concentric Within 0.002 in.-- which Mandates a Stress-Relieving Step in the Ring Manufacturing Sequence*

Serious reliability-focused purchasers often specify and select pumps with flinger discs. Although

sometimes used in slow speed equipment to merely prevent temperature stratification of the oil (see Figure 5), larger diameter flinger discs serve as efficient (non-pressurized) oil spray producers at moderate speeds. Of course, the proper flinger disc diameter must be chosen and solid steel flinger discs should be preferred over plastic disc materials. Insufficient lubrication results if the disc diameter is too small to dip into the lubricant. Conversely, high operating temperatures can be caused if the disc diameter is excessive, or if no thought was given to its overall geometry.

Flexible flinger discs have been used to enable insertion in some "reduced cost" designs, i.e., configurations 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 (Figure 6), 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. We know of many attempts to get around the use of oil rings. There have been roll pins inserted transversely in pump shafts (Bloch, 2010, pp. 251) and flexible (plastic) flinger discs. All brought mixed results and marginal improvement at best. Cheap discs pushed on the shaft became a source of failure and were disallowed by API-610 about 10 years ago. Cheap plastics and disc configurations chosen without the benefit of sound engineering practices have not been sufficiently reliable. All in all, we should never lose sight of the charter and mission of reliability professionals. We believe their goal should be to work in harmony with basic science. We must achieve high reliability and availability; trial and error solutions should be left to others.

We estimate the incremental cost (comprising material, labor, CNC production machining processes) of an average-size (30 hp) process pump with cartridge-mounted bearings at $300. The value of even a single avoided failure was earlier shown to be $10,000 and the benefit-to-cost ratio would thus be 33-to-1. As to our estimate of the percentage breakdown of different lubrication methods for process pumps in refineries in the industrial countries: Oil rings—30%; oil mist—30% overall, but 90% in the top-tier most profitable refineries; ball immersion—30%; flinger discs: 10%. According to Houston-based LSC, there are

70,000 pumps and motors operating on dry sump oil mist lubrication.

Point 8: Oil Rings Can Become Unstable; Skip, Scrape, Abrade

The shortcomings of oil rings were known in the 1970's. A then well-known pump manufacturer claimed superior-to-the-competition products. This manufacturer's literature pointed to an "anti-friction oil thrower [i.e., a flinger disc], ensuring positive lubrication to eliminate the problems associated with oil rings" (Figure 8; also Bloch 2011; Bloch, 2010; Bloch, 2009).

And about two decades later, in 1999, at least one major pump manufacturer saw fit to examine the situation more closely. In a comprehensive paper the manufacturer described remedial actions that included more closely controlled oil viscosity and oil rings made of high performance polymers (Bradshaw, 2000).

Figure 7. Oil rings in as-new (wide and chamfered) condition on left, and abraded (worn narrow and now without chamfer) condition on right side (Bloch, 2011; Bloch, 2010; Bloch, 2009).

This prompts us to refer to our Point 8 and to re-state, again, that oil rings work fine in a perfect universe. However, our work environment is rarely perfect. Skilled labor is increasingly hard to find. Alignment procedures concentrate on shaft center congruence, not system horizontality.

Were problems known 40 years ago?

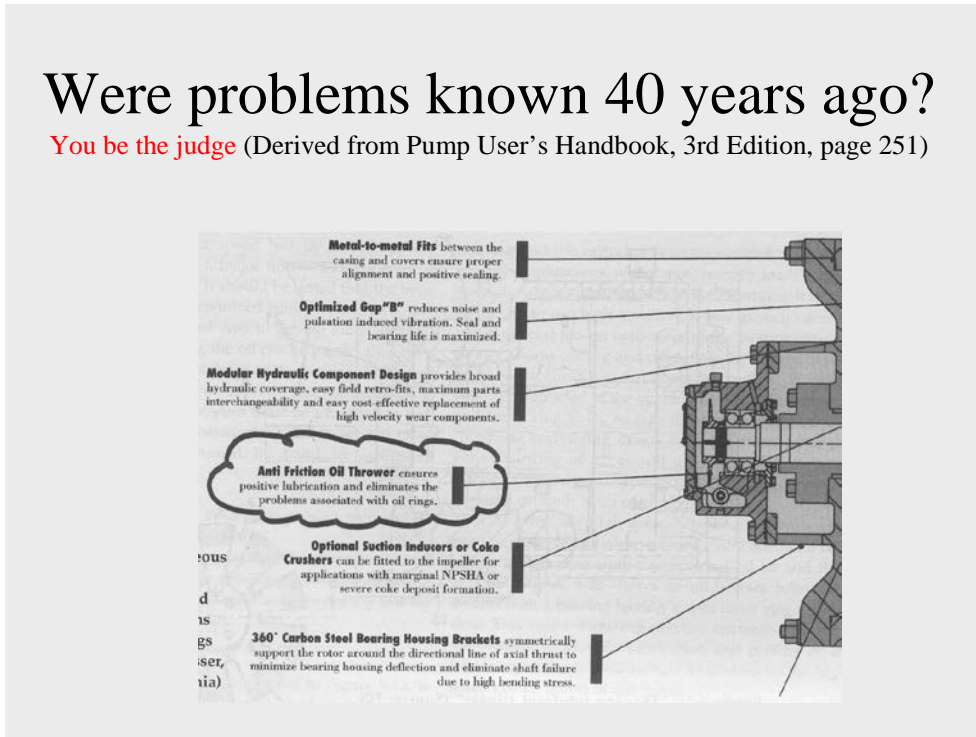You be the judge (Derived from Pump User's Handbook, 3rd Edition, page 251)

Metal-to-metal Fits between the casing and covers ensure proper alignment and positive sealing.

Optimized Gap "B" reduces noise and pulsation induced vibration. Seal and bearing life is maximized.

Modular Hydraulic Component Design provides broad hydraulic coverage, easy field retro-fits, maximum parts interchangeability and easy cost-effective replacement of high velocity wear components.

Anti Friction Oil Thrower ensures positive lubrication and eliminates the problems associated with oil rings.

Optional Suction Inducers or Coke Crushers can be fitted to the impeller for applications with marginal NPSHA or severe coke deposit formation.

360° Carbon Steel Bearing Housing Brackets symmetrically support the rotor around the directional line of axial thrust to minimize bearing housing deflection and eliminate shaft failure due to high bending stress.

Figure 8. A 1970s advertisement mentions "problems with oil rings"

Because the anti-friction oil thrower(s) [i.e., flinger discs] mentioned in Figure 8 were found to ensure positive lubrication and eliminated the problems associated with oil rings, many European-made pumps incorporate flinger discs ("oil throwers"). So does at least one U.S. manufacturer. And all of this certainly re-enforces our Point 9.

> *Point 9: Measure Oil Rings New and After Use. The* □*Width Diff*
> *Abrasive Particles Which Limit Bearing Life*

Unlike oil rings, potential malfunction risks with constant level lubricators are more widely known; a number of makes, models and brands are in common use; Figure 9 is rather typical. The uni-directionality of constant level lubricators is described in manufacturers' literature, but not many users know that the caulking at the transparent bottle-to-metal joint will, over time, develop stress cracks (fissures). These cracks allow rain water to reach the oil via capillary action. Note that bottle-type constant level lubricators are a preventive maintenance replacement item. Every 4 or 5-years, they should be replaced with (hopefully) a best-available-technology ("BAT") component. Buy only the best. Review available technology and specify it for risk-free operation and for future maintenance avoidance. View every maintenance event as an opportunity to upgrade.
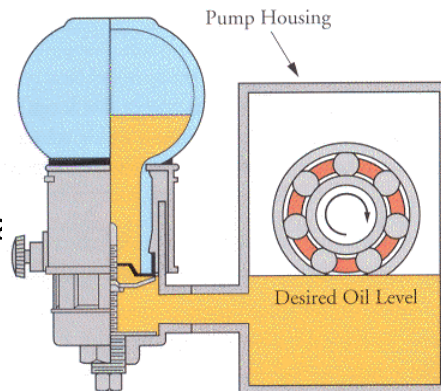
Pump Housing

Desired Oil Level

Figure 9. A typical constant level lubricator. Lowering the oil level may deprive this bearing of lubricant. Note directional arrow and observe our Point 10.

That said the oil ring problem did not go away after the reported research. Users in Canada reported that the problem persisted even after adopting non-metallic oil rings. Black oil and other problems returned. Black oil can have only two origins and a simple oil analysis will point to either overheated oil or slivers of an elastomeric O-ring coming off a particular bearing housing protector seal.
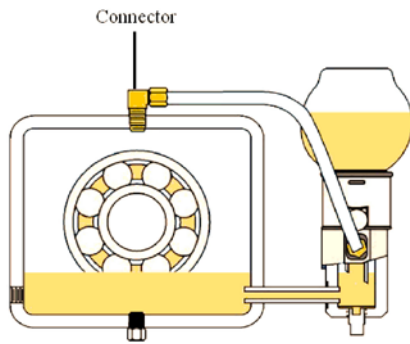
Figure 10. Pressure-balanced constant level lubricator. Be sure to use a large diameter balance line.

Note how, in Figure 9, the oil level in the bearing housing is no longer reaching the rolling elements. This constant level lubricator lacks pressure balance. The lubricator in Figure 10 is configured for a balance line which ensures that the oil levels at the lubricator support and in the pump bearing housing are always at the same pressure (TRICO, 2003). We have seen undersized balance lines; it will be wise to ascertain that either a generous diameter hard pipe or a stainless steel hydraulic balance line is installed.

Of course, bearing failures will occur if a constant level lubricator does not maintain the desired constant oil level. An incorrect level setting can be caused by a number of factors. It will be clear from Fig. 9 that even extremely small increases in the bearing housing-internal pressure will increase the failure risk. Suppose there is heat generation _and_ because of the addition of bearing protector seals the air no longer escapes _and_ there's a lack of housing-internal pressure balance. The result may well be that the housing-internal pressure goes up. As the housing-internal pressure rises ever so slightly, it will exceed the ambient pressure to which the oil level at the wing nut in the lower portion of this pressure-unbalanced constant level lubricator (Figure 9) is exposed. According to the most basic laws of physics, a pressure increase in the bearing housing causes the oil level near the bottom of the bearing inner ring shoulder to be pushed down. Lubricant will no longer reach the bearing rolling elements, oil turns black, and the bearing will fail quickly and seemingly randomly.

> **Point 10: Constant Level Lubricators Must be Installed on the Up-Arrow Side. If the Shaft Rotation is Clockwise, the Up-Arrow is on the Left.**

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 will perform better and be far less prone to cause unforeseen outages than many other presently favored methods. Remember that traditional oil rings will abrade and slow down if they contact a housing-internal surface. They are sensitive to oil viscosity and depth of immersion, concentricity and RMS surface roughness.

Ask if the other methods are used because they cost less money initially, but will cost much more in the long run. If you opt for flinger discs, you are at least paying heed to the legacy manufacturer whose advertisement is illustrated in Figure 8. We believe this manufacturer's findings were factual. Still, it must be ascertained that flinger discs are used within their applicable peripheral velocity so as to contact the oil and fling it into the bearing housing (Bloch, 2011). Of course, the flinger disc OD must exceed the outside diameter (OD) of the thrust bearing. 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, as will the cost of a well-designed flinger disc. 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. If you believe it merely costs $2,000 to repair an API pump, you will argue about the matter. For our part, we would then prefer to simply move on and focus on tutoring the teachable ones in our audience or readership.

Bearing housing protector seals

Lubricant contamination originates from a number of possible sources and can also be a factor in "unexplained" repeat failures. 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 air volume expansion, and decreasing temperatures at night or after shutdown cause air volume contraction. Open or inadequately sealed bearing housings promote this back-and-forth movement of moisture-laden and dust-containing ambient air. But, simply adding bearing protector seals could change windage or housing-internal pressure patterns in unforeseen ways. This, too, we must recognize as a potential source of "unexplained" failures.

Back to housings that breathe, and thus ingest contaminants. Aim for little or no interchange between the housing interior air and the surrounding ambient air. The breather vents shown

earlier in Figure 2 can often be plugged. Think of the hundreds of millions of refrigerators and automotive air conditioning systems that operate with neither vents nor breathers!
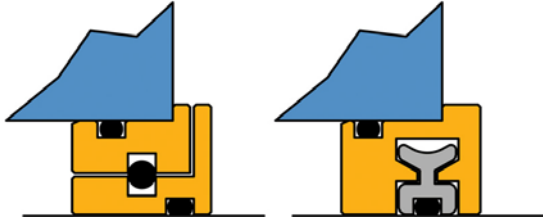

Figure 11. Illustration of principles; neither replicates an actual product but highlights what can happen with products that incorporate sharp-edged grooves, or cause excessive drag

Next, become familiar with some old-style bearing housing seals that allow an O-ring to contact an O-ring groove, as depicted in Figure 11, left side. Contact with sharp-edged grooves surely invites dynamic O-ring to scrape on edges. Or what happens when assembly pressure is applied? Will the rotating ring move relative to the stationary and will it score the dynamic O-ring?

Recall that force applied to a surface equals pressure; force on a small area equals high pressure. None of us would think that sliding our fingers over a knife is without risk. A pinched O-ring can liberate shavings. Elastomer shavings will contaminate the lubricant—there's one reason for "black oil." An O-ring contacting a wide area stands the best chance of surviving.

So then, one design in Figure 11 invites dynamic O-rings to scrape on sharp edges; the shavings have been known to show up in the lubricant. That's Point 11. Also, designs using only a single O-ring for clamping the rotor to the shaft impart less rotor stability than designs using two rings for clamping duty. Visualize this by mentally removing the stationary component in this picture. At any speed, the rotor would pivot around the clamping O-ring and destructive rotor vibration would result. We could study the rotor dynamics of such a situation and spend a nice sum on such an academic research project. Or, we might reach the same common-sense conclusion by giving it some thought: Two clamping O-rings provide more stability than one clamping O-ring.

Point 11: O-Rings in Contact With Sharp Corners Will Fail Prematurely

In essence, the correct bearing protector seals can greatly improve both life and reliability of rotating equipment by safeguarding the cleanliness of the lubricating oil. However, if pressure-unbalanced constant level lubricators are used that allow air and moisture to intrude, bearing protector seals add little value if oil contamination originates with oil ring wear, or if the oil is not kept at the proper oil level, or if the bearing housing design disregards windage concerns.

We are not mentioning see-through containers at the bottom of the pump bearing housing. By the time water becomes visible in such a "sludge cup container", the saturation limits of oil-in-water will have been exceeded and much damage will have been done to the bearings. That matter was presented at TAMU over two decades ago and a few dozen references were provided at that time. We know that free water is a symptom of not having the right bearing protection. Our reliability focus should be on treating the root cause, not the symptom. We must prevent water from reaching the bearings in the first place. These proactive and precautionary thought processes are at the core of this tutorial on failure prevention.

Ranking the different lube application practices

Although oil ring lubrication is widely used, it is relatively maintenance-intensive and ranks last from the authors' reliability improvement and risk reduction perspective. Flinger discs have been used for many decades and allow operation at higher DN values than oil rings. Because they are clamped to the shaft, there is far less sensitivity to installation and maintenance-related deviations. On the other hand, non-clamped flinger discs were tried a few decades ago, with very disappointing results. We were asked to provide data to back up this contention, but can only point to API-610. This widely accepted industry standard saw fit to disallow push-on discs and flogging pins --- "the paper clip thingie", as one reviewer called it. We certainly agree with API-610 on that one.

Point 12: Pure Oil Mist Represents Many Decades of Fully Proven Technology

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% (Bloch, 2011; SKF, 1995) have been reported by Charles Towne of Shell Oil, and many others. Charles Towne performed these tests on identical process units at Shell Oil in Deer Park, Texas, and deserves much credit for seminal work on the subject (Towne, 1983).

The highly beneficial in-plant, real-life results reported by Towne refer to pure oil mist, not purge mist. Pure oil mist is applied in modern plants as shown in Figure 12. The advantages and disadvantages of oil-mist lubrication as compared to conventionally applied oil lubrication are usually summarized as follows:

Advantages:
- Reduced bearing failures of 80 to 90%
- Lower bearing operating temperatures of 10 to 20 degrees F
- No recirculation of bearing wear or debris particles
- Slight positive system pressure eliminates contaminant entry and fully protects standby equipment
- Electric motor drivers are included in plant-wide oil mist systems at true best-of-class performers. The incremental cost of including motors in a plant-wide oil mist system is quite minimal and has been estimated to add 10% to the system's cost. All types of motor leads are acceptable as long as the correct potting compound is used in the junction box (Bloch/Shamim, 1998). Oil intrusion into motors is acceptable and excess oil will drain at weep holes. If needed, oil mist flow through bearings is usually accomplished during the first bearing replacement. However, flow-through passage is only needed in thrust-loaded motor bearings. Hundreds of vertical pump motors at several Gulf Coast chemical plants have been in uninterrupted use for well over 10 years. Every one of these vertical motors is using through-flow oil mist. A plant in the Houston area commissioned 132 vertical pump motors in 1978 and there had been no failures---zero—when one of us retired from his position as a Senior Engineering Associate in 1986. This author also know that Charles Miannay applied oil mist on a 1,250 hp horizontally arranged motor in 1970 and wrote about it in Hydrocarbon Processing (Miannay, 1974). Over 30 years ago, Siemens described horizontal electric motors in sizes up to 3,000 kW lubricated with oil mist (Bloch, 2009; Bloch/Shamim, 1998). In May of 2012, we received feedback from two plants that affirmed that oil mist lubricated vertical motors had been in service for 30 years without a single bearing failure. That's a key ingredient of our Point 12, dealing with oil mist. Those "in the know" rightly add to oil mist summaries:
- Reduced energy costs of 3 to 5%
- Reduced oil consumption of about 40% for open and 95% for closed plant-wide systems
- No moving parts

- Reduced maintenance; no constant level lubricators; no oil rings
- Simplified lube oil management. (With closed oil mist systems, disposal and collection of stray mist is no longer an issue. Closed systems are preferred by users who are serious about conservation of resources and take action to do something about it.)

Disadvantages:
- Higher initial investment
- Must consider cost of low dew point (dry) compressed air. (Of course, remote installations lacking utilities may--at first--not be able to justify oil mist, although a different picture often emerges when the preservation of standby bearings is fully factored into their cost justification equations).

> *Point 13: __All__ bearing manufacturers rank spray-lube above __every other__ available lubrication method*

Spray-lubricated pump bearings

The world's premier bearing manufacturers are unanimous in ranking _spraying_ oil into the bearing cage area as superior to all other lube application methods, including oil mist (Eschmann et al, 1985). To effectively spray oil into the bearing areas the lubricant has to be pressurized. Numerous ways of pressurization are known; they exist in millions of shaft-driven governors and in auxiliary lube skids and similar lubricating packages where oil is typically pressurized, filtered, cooled and supplied to sleeve bearings. Similar means of providing pressurization and filtration are difficult to incorporate in the standard process pump bearing housing. Therefore, new bearing housing designs will be needed. No cooling will be required with high-grade synthetics. (Note, again, that oil mist and oil spray are completely different application methods. Each is described in the reference texts (SKF, 1995; MRC, 1892; and others).

Keeping these facts in mind, Figure 13 incorporates a number of very important recommendations for the truly reliability-focused:

- It establishes that pump bearing housings need not be symmetrically configured. (Asymmetry is visualized by looking into the pump shaft. The distance to the right edge of the bearing housing is not the same as the distance to the left edge of the bearing housing. The additional volume thus gained will accommodate a small pump, arranged internally to the bearing housing).

- A box-like geometry with a flat cover and ample space to incorporate a wide range of oil pumps is feasible. Box-like bearing housings for process pumps would open up a host of new and inventive solutions. These might incorporate shaft-driven or other reliable self-contained means of oil application pumps. (And the fact that someone's last shaft driven oil pump excited the bearing housing natural frequency to the tune of 2.5 ips is hereby acknowledged. The same excitation has not been experienced on many tens of thousands of shaft-driven hydro-mechanical governors during the past six decades).
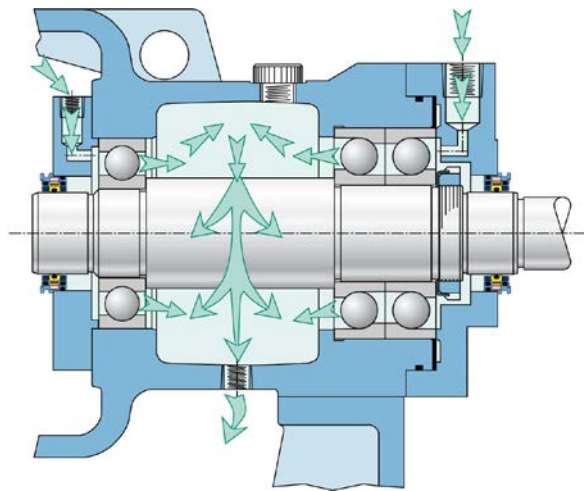

Figure 12. Oil _mist_ lubrication applied to a pump bearing housing in accordance with API-610, 10th Edition (Bloch, 2009). With oil _spray_ lubrication, liquid oil would enter at the nozzles. Note dual _mist_ (or, for spray lube pplication, dual _liquid oil_) injection points. Observe dual-face bearing housing seals that prevent oil _mist_ (or oil _spray_) from escaping to the atmosphere.

(We can conclude that not all engineering skills are on the same plateau. There is no device ever invented that humans cannot defy, and we are particularly vulnerable if we deal with vendors who cut corners. This may have happened in the 2.5 ips resonance event mentioned by a reviewer).

- The oil application pump would take suction from an increased-size oil sump.
- The main process pump shaft need not be in the geometric center of the box.
- Flat surfaces would invite clamp-on, screw-in or flange-on oil pumps.
- Oil pressurized by the oil application pump would be routed through a filter and hydraulic tubing to spray nozzles incorporated in the end caps. Therefore, the cross-section view of a

bearing housing with oil spray would be identical to the one shown for oil mist in Figure 12 (Bloch, 2009).
- Internal pressure equalization and windage issues would never again be a concern.
- The incremental cost of superior bearing housings would be more than matched by the value of avoided failures.

In Figure 12 and with either oil mist or oil spray there would be neither oil rings, nor flinger discs, nor constant-level lubricators. Because the mist (or spray) application nozzles shown here are relatively close to the bearings, oil mist flow will definitely overcome windage. While this seems like a bold idea, the approach is extensively documented by MRC and SKF, also in at least 7 of our many reference texts (among them Bloch, 2001). This lubrication method is very often used in military aircraft, so we really cannot take credit for coming up with it.

That said the duty imposed on self-contained oil spray pumps would be quite benign compared to other known, reliable, shaft-driven oil pump technology. Oil filtration would be easy. The elimination of oil rings and constant-level lubricators would be a very positive reliability improvement step. Part of the energy requirement of an oil application pump would be re-gained in the form of reduced bearing frictional losses.

With spray lubrication, much needed oil application innovation would benefit the drive end and thousands of repeat failures of pumps would no longer occur. However, as of today, little interest has been shown by manufacturers and users to redesign pump bearing housings. Discussions with a major pump manufacturer disclosed they were not interested in devices that cannot be patented.

Still, the market drives new developments. If the buyers are happy with repeat failures and the manufacturers benefit from the sale of spare parts, it will be business as usual. Yes, even at the risk of stubbornly bucking complacency trends: As responsible engineers, we advocate changes in mindsets. We are under no illusions as to where some users and manufacturers will be when the dust settles: We will never reach some of them. All we wanted to do is explain things to those whose reliability focus extends beyond "business as usual" and who are interested in asking for lower risk oil application alternatives. That's Point 14 and serves as the explanation for Figure13.

Point 14: Advocate for risk reduction-self–contained pump bearing lubrication is feasible

To think *out of* the box, you must learn to *think of* a box. The housing could be non-cylindrical

- Shaft not in center is OK
- Shaft in center also OK
- Flat top and sides allow flanged-on or screw-in spray pumps
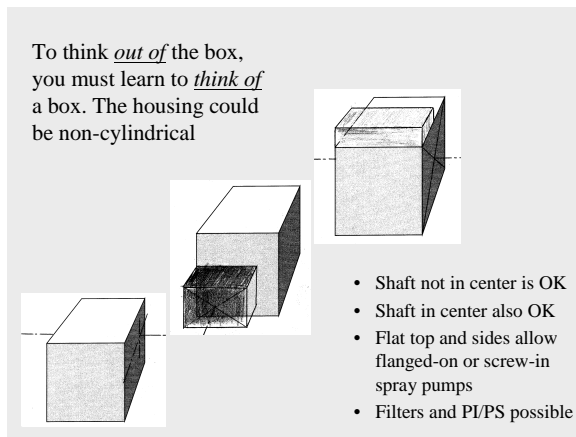- Filters and PI/PS possible

Figure 13. Redesigned pump bearing housings would accommodate a housing-internal lube oil pump

Figure 13 proposes a new generation of bearing housings. The intent is to eliminate oil rings and constant level lubricators. The process pump bearing housing should incorporate an oil pump that will create the pressurized spray deemed most advantageous by all world-scale rolling element bearing manufacturers

CONCLUSIONS

As of today, process pumps experience many repeat failures. As reliability professionals and informed users we can do better and should know how to do better. The proposed new minimum requirements for reliability-focused pump users must aim for:

1. Upgrading and getting away from maintenance-intensive oil rings and, if possible, constant level lubricators.
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 of the bearing to the other. Note that such a passage was shown in Fig. 3 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 Figure 3 can often be plugged.
4. As a minimum, a pressure-balanced constant level lubricator should be supplied and its balance line should be connected to the breather port that's closest to it.

5. Bearings should be mounted in suitably designed cartridges and loose slinger rings (oil rings) should be avoided or, in some cases, 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).
7. Modern and technically advantageous versions of bearing protector seals should be used for both the inboard and outboard bearings. Lip seals are not good enough, and neither are outdated rotating labyrinth seal designs.
8. Understand that the implementation of true reliability-thinking must strongly support moves away from traditional bearing housings. These moves should push for exploration of the alternatives alluded to in Figure 13.

Knowledgeable engineers can prove that things tend to malfunction in the real world and it's in the user's best interest to reduce failure incidents and downtime risk. Plain logic should lead to full 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.

Designing-out maintenance starts with simple upgrade measures that some users implemented 30 years ago, although others are disregarding these measures to this day. Designing-out maintenance culminates in re-thinking the entire bearing housing configuration and upgrading the way oil is applied to rolling element bearings (American Petroleum Institute, 2009; Bloch/Shamim, 1998). Trends that cheapen process equipment are not at all healthy. Together, true reliability professionals can break the cycle of avoidable pump repairs.

Better pumps may not be the cheapest pumps

In late 2008, the purchasing entity representing a large reliability-focused plant in the United States had thoughtfully and deliberately specified better pumps. The user wanted better pumps and was willing to pay for the better product. But the buyer's improvement requests were declined by every one of the vendor companies that responded to an invitation to bid. The disappointed owner-user company suggested an article or presentation 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. Well, we hope to have honored their request in presenting this tutorial.

We have focused on getting extended and trouble-free operation from pump drive ends and want to again point out that stand-alone appendices and checklists have been published in many articles and books (Eschmann et al, 1985; Bloch/Budris, 2010). These and many others elaborate on the steps needed to give the value-oriented user community pumps that operate for six or more years before repairs are needed.

In this presentation, we have tried not to repeat too much of the previously published material. We have mentioned neither manufacturers' names nor certain references that would cause embarrassment. On the other hand, we have tried not to bore you with valueless sales-speak. Instead, we have given you tangible steps that can help break the never-ending cycle of pump repairs.

A final thought. As professionals, we owe it to pump users to steer clear of the usual consultant-conceived generalities or marketing-driven hype. Some would like you to believe that the laws of physics do not apply to their products and others will not even allow common sense to be mentioned. For our part, we believe that users benefit from access to facts, not exposure to figurative noise. The code of ethics of professional engineers is not an on-off mindset to be invoked only when it is expedient or profitable. For some of us, it's a way of life. For others, it never will be. Still we will continue in our efforts not to attack their integrity, even if they never wake up or scheme to attack the messenger. Chances are they just don't know any better.

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